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Design and Development of Parallel - Counter Flow Heat Exchanger

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***Abstract:** Objective of this review paper reviews the literatures related to the parallel and counter flow of different types of heat exchangers and modifications made to improve the performance. Various papers were reviewed from those papers what are the developments in parallel and counter flow heat exchanger are summarized. The development of any system is need because it helps to optimize, improve performance or reduce the cost of system. The heat exchanger development did by many researchers using software's, design methods, changing the designs, changing shape of tubes, applying second law of thermodynamics etc. The fluid velocity, Reynolds number, overall heat transfer coefficient, baffle spacing, number of baffles, pressure drop, LMTD, helix angle of tubes plays very important role in heat exchanger performance.*

Keywords: Heat exchanger, Design, Development, parallel-counter flow, kern method

I. INTRODUCTION

Heat transportation is occurs due to a spatial temperature difference. Whenever a temperature difference exists in a medium or between media, heat transfer must take place. Types of heat transfer devices are heat exchanger (HE), heat pipe, fins, radiators, condensers, etc. [14].A heat exchanger is a device in the habit of transfer heat between a solid objective and fluid, and vice versa. These are broadly used in refrigeration, air conditioning, space heating, power stations, petrochemical plants, petroleum refineries, chemical plants and natural gas processing etc. Energy preservation is critical for the improvement of world's financial system. To use energy more professionally is one of significant way for saving energy. So the study of its optimization design is of immense significance for conserve energy in heat exchange process. HE classification is shown in following figure [16].

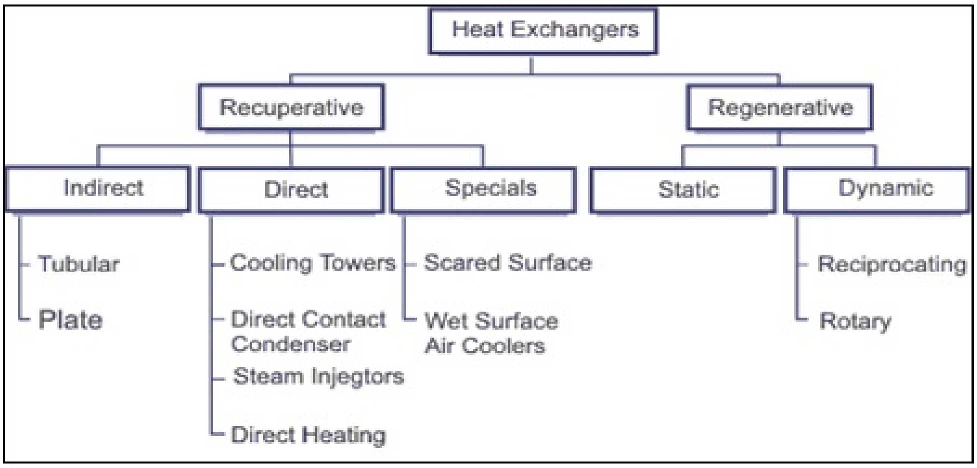

Fig. 1: Classification of Heat Exchanger

RECUPERATOR: - In this type, heat transfers continuously from the hot fluid to the cold fluid through a dividing wall. Although a simultaneous flow of two (or more) fluids is required in the exchanger, there is no direct mixing of the two (or more) fluids because each fluid flows in separate fluid passages.

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In general, there are no moving parts in most such heat exchangers [17]. Flow arrangements, basically there are three flow arrangements are available. In parallel flow HE, the two fluids which are enter into HE in a same direction and travel in same direction. In counter flow HE, the fluids which are enter HE in opposite directions (ends). In cross flow HE, the fluids travel approximately at right angles to each other [17]. Indirect-contact HE, the fluid stream stay separate and heat transfer continuously through an impervious separating wall or else into and out of a wall in a transient way. Thus, preferably there is no direct contact between thermally interacting fluids. This kind of HE also referred to as a surface HE. Direct-contact HE, two fluid streams approaches direct contact, exchanges heat, and are after that separates. Common applications of a direct-contact exchanger engage mass transfer in addition to heat transfer, such as in evaporative cooling and improvement. Tubular Heat Exchangers are in general built of circular tubes, although elliptical, rectangular, or Round or flat twisted tubes have also been used in some applications they are further classified as Shell and tube HE, Spiral tube heat exchanger and Double pipe (tube) HE [17]. Plate-Fin HE, This type of exchanger has corrugated fins or spacers sandwiched between parallel plates. A plate-fin exchanger with 600 fins/m (15.2 fins/inch) provides about 1300m^2 (400 ft^2/ft^3) of heat transfer surface area per cubic meter of volume occupied by the fins. Tube-Fin HE may be classified as conventional and specialized tube-fin exchangers. These are used extensively as condensers and evaporators in air conditioning and refrigeration applications. Regenerator type HE, both fluids flow alternatively through the same flow passages, and hence heat transfer is intermittent. The heat transfer surface (or flow passages) is generally cellular in structure and is referred to as a matrix or it is a permeable (porous) solid material, referred to as a packed bed [17].

II. LITERATURE REVIEW

Akshaykumar Magadum et. al. [1] experimentally investigated tube in tube HE with parallel and counter flow arrangements. Heat transfer rate found more nearly by 30% in counter flow compared to parallel flow. The results plotted on graph for LMTD v/s discharge and LMTD v/s efficiency for both parallel and counter flow. Conclusion of their work is as LMTD increases discharge and efficiency increases. **Swapnil Ahire et. al.** [2] constructed and analyzed counter flow helical coil HE. Helical coil shape is given to tubes. To determine overall heat transfer coefficient, Wilson plot technique was used. They found that centrifugal force due to curvature of tubes results in secondary flow development, this secondary flow enhances heat transfer rate. For low Reynolds number graph of Nu v/s Re and hi v/s Re is steeper than that of high Reynolds number. Conclusion of their work is helical coils are efficient in low Reynolds number. **Alok Vyas et. al.** [3] conducted a variety of experimental analysis to predict performances of HE's characteristics like temperature difference and pressure drop. In this design some factors to be considered like diameter of tube, number of tubes and baffles. Factors which are affecting performances were selected. CFD software is used for analysis; through this they tried to improve performance of tubular HE. **Jian-Fei Zhang et. al**. [4] For developing a shell and tube HE with middle overlapped baffles, a comprehensive simulation model by using commercial code FLUENT and grid generation program GAMBIT. The validation of 40 degree helix angle is performed and results which got those shown a reasonable agreement with available experiment data. The simulation is conducted of a whole HE to predict shell-side fluid flow and heat transfer rate. **Durgesh Bhatt et. al**. [5] in this review paper, involved various conditions where different constructional parameters at various conditions are changed to get performance review. For ease of calculation and obtaining result after varying different parameters a excel program developed. The baffle spacing and tube metallurgy are parameters considered to be changed. From his review paper it is cleared that overall heat transfer coefficient and pressure drop decreases due to baffle spacing increases. **Balaram Kundu** [6] designed Un-baffled shell and tube HE with attachments of longitudinal fins having trapezoidal profile; a parametric variation studied by Kern's method of design of extended surface HE has been made for an un-baffled shell and tube HE. For analysis purpose rectangular and trapezoidal fin shapes attached to tube and separately are considered. While keeping the outer shell diameter is a constant along with all other constraints, the heat transfer rate in trapezoidal fins was lesser than rectangular cross section of a HE. **Simin Wang et. al.** [7] performed an experimental investigation of shell and tube HE for heat transfer enhancement, the configuration of a shell and tube HE improved through installation of sealers at shell side. The gap is blocked between baffle plates and shell by sealers. Shell side heat transfer coefficient by 18.2–25.5%, overall coefficient of heat transfer by 15.6–19.7%, and efficiency by 12.9–14.1% these all results increased. **D. A. Maheshwari and K. M. Trivedi** [8] they did experimental investigation of U tube HE using plain and corrugated structure of tube. Corrugated tubes are prepared from straight tubes bending them into spiral shape. Corrugated tube HE gives more heat transfer coefficient and heat transfer rate than straight tube HE. **Logeshan K. L. et. al.** [9] studied how to increase fluid velocities which results in large heat transfer coefficient and consequently less heat transfer area. On the other hand by increasing fluid velocity causes an increase in pressure drop. **Jingfeng Guo et. al.** [10] developed a new HE based on second law of thermodynamics. In this approach, modified entropy generation amount which can avoid entropy creation is taken as the objective function. **H. S. Dizaji et. al.** [11] Heat transfer, pressure drop and effectiveness of a double pipe HE prepared from of corrugated outer and inner tubes experimentally investigated. Convex and concave corrugated tube was investigated for heat transfer coefficient determination using Wilson plots. Concave corrugated outer tube and convex corrugated inner tube this arrangement obtained maximum effectiveness.

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Resat Selbas et. al. [12] in this study, for the optimal design a genetic algorithm (GA) has been successfully applied to shell and tube HE by varying the design variables like outer diameter of tube, layout of tubes, number of tubes and passes, outer diameter of shell, spacing of baffle and baffle cut. To determine the heat transfer area for a given design configuration LMTD method is used. **K. P. Venkatesh et. al.** [13] Spiral flow HE is known as excellent HE because of far compact and high heat transfer efficiency. SFHE is a unique design where it consists of single fluid as working fluid for heat exchange. Heat transfer characteristics of SFHE are observed at various Reynolds number and base temperature. SFHE is designed and fabricated with new arrangement for measuring the heat transfer is employed for obtaining the experimental results. CAD model is developed and computations are performed using commercially available CFD package ANSYS-CFX for the same boundary conditions, input conditions as that considered for experimental purpose. Obtained CFD results are compared with that of experimental results for validation.

III. DESIGN PROCEDURE

Design of segmental baffled shell and tube HE [16]

PRELIMINARY STAGE

1. Heat transfer rate
   $Q = \dot{m}C_p\Delta T$
2. Provisional area
   $$A_o = \frac{Q}{U_o F \Delta T_{lm}}$$
3. Calculate log mean temperature difference
   $$\Delta T_{lm} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\left(\frac{T_{hi} - T_{ci}}{T_{ho} - T_{ci}}\right)}$$
4. Calculate over all heat transfer coefficient
   Assume over all heat transfer coefficient from the table given below [16]
5. Calculate correction factor for log mean temperature difference

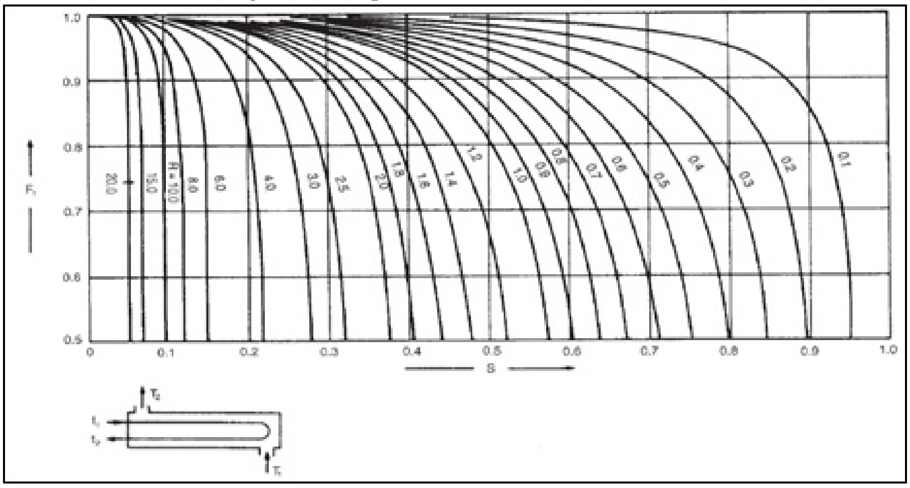

Fig. 2: Correction factor [16]

$$R = \left(\frac{T_1 - T_2}{t_2 - t_1}\right)$$

$$S = \left(\frac{t_2 - t_1}{T_1 - t_1}\right)$$

6. Calculate dimensions of HE

Assuming the length of HE : To calculate shell diameter and no. of tubes for single pass take CTP=0.93 this value is for single tube pass, CL=1 this value is for 45^o and 90^o degree tube layout

$$PR = \frac{P_T}{d_o}$$

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$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left(\frac{A_o (PR)^2 d_o}{L}\right)$$

$$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_s^2}{(PR)^2 d_o^2}$$

$$B_s = 0.6 D_S$$

Now rate the HE according to TEMA standard and recalculate the length of HE, from Table number 8.3, page number 293, [16]

SECONDARY STAGE

1. Calculate heat transfer coefficient
   Shell side heat transfer coefficient

$$D_e = \frac{4\left(P_T^{\,2} - \pi d_o^{\,2}/4\right)}{\pi d_o}$$

$$C = P_T - d_o$$

$$A_s = \frac{D_s C B_s}{P_T}$$

$$v = \frac{\dot{m}_t}{\rho_t A_{tp}}$$

$$R_e = \frac{\rho v d_i}{\mu}$$

$$N_u = 0.36\left(\frac{D_e G_s}{\mu}\right)^{0.55}\left(\frac{C_p \mu}{k}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad \text{...... for} \quad 2\times 10^3 < R_e < 10^6$$

Take μ_w from table at approximate wall temperature T_w

$$T_w = \frac{1}{2}\left(\frac{T_{ci}+T_{co}}{2} + \frac{T_{hi}+T_{ho}}{2}\right)$$

$$h_o = \frac{N_u K}{d_e}$$

Tube side heat transfer coefficient, Take C_p, ρ, μ, P_r at cold fluid mean temperature from appendix Table B.2 [16]

$$A_{tp} = \frac{\pi d_i^{\,2}}{4}\frac{N_t}{2}$$

$$v = \frac{\dot{m}_t}{\rho_t A_{tp}}$$

$$R_e = \frac{\rho v d_i}{\mu}$$

$$N_{u_b} = \frac{(f/2)(R_e - 1000)\mathrm{P}_r}{1 + 12.7(f/2)^{0.5}\left(\mathrm{P}_r^{\,2/3} - 1\right)}$$

$$f = (1.58\ln R_e - 3.28)^{-2}$$

$$h_i = \frac{N_u K}{d_i}$$

Over all heat transfer coefficient for cleaned surface

$$U_c = \frac{1}{\dfrac{d_i}{d_i h_o} + \dfrac{d_o \ln(d_o/d_i)}{2K} + \dfrac{1}{h_o}}$$

Over all heat transfer coefficient for fouled surface

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$$U_f = \frac{1}{\frac{d_i}{d_i h_o} + \frac{d_o R_{ft}}{d_i} + \frac{d_o \ln(d_o / d_i)}{2K} + R_{fo} + \frac{1}{h_o}}$$

Take R_{ft} and R_{fo} from table of fouling factor

To calculate tube length

$$A_{of} = \frac{Q}{U_{of} \Delta T_{lm} F} = \pi d_o L N_t$$

$$L = \frac{A_{of}}{\pi d_o N_t}$$

Calculating pressure drop for shell side

$$\Delta P_s = \frac{f G_S^2 (N_t + 1) D_s}{2 \rho D_e \phi_s}$$

$$f = \exp(0.576 - 0.19 \ln R_e)$$

$$\varphi_s = \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

Calculating number of baffles

$$N_b = \frac{L}{B_s} - 1$$

Calculate total pressure drop.

$$\Delta p_t = \left(\frac{4 f L N_p}{d_i} + 4 N_p\right) \frac{\rho v^2}{2}$$

IV. CONCLUSION

A primary objective of this review paper is to understand types of heat exchangers and developments in heat exchangers. From the literature review of parallel & counter flow heat exchanger we came to the conclusion those are as follows,

1. *In the heat exchanger design, the estimation of the minimum heat transfer area is very important for a given heat transfer duty, as it governs the overall cost, size, and weight of the heat exchanger.*
2. *By increasing mass flow rate of cold fluid, heat transfer coefficient increases which results in increase in the Reynolds number.*
3. *Increasing baffle spacing reduces overall heat transfer coefficient (U) and also the Reynolds number decreases and hence the pressure drop.*
4. *Shell and tube heat exchanger gives better performance using helical baffle instead of straight baffles.*
5. *Using 25 degree helix angle for baffles in shell and tube heat exchanger gives better performance compared to segmental baffles.*
6. *There is increase in pressure drop with increase in fluid flow rate in shell and tube heat exchanger which increases pumping power.*
7. *As LMTD increases discharge increases as well efficiency of heat exchanger increases.*
8. *Helical coils are efficient in low Reynolds number.*
9. *Kern's method of design of extended surface heat exchanger*
10. *Corrugated tube heat exchanger gives more heat transfer coefficient and heat transfer rate than straight tube heat exchanger.*
11. *Increased fluid velocities results in large heat transfer coefficients and consequently less heat transfer area and exchanger cost for given rate of heat transfer.*
12. *A new heat exchanger optimization approach based on second law of thermodynamics can be used for development.*
13. *To investigate heat transfer coefficient of convex and concave corrugated tube Wilson plots can be used.*
14. *In comparison of parallel and counter flow, overall heat transfer rate and log mean temperature difference is higher in counter flow.*
15. *For ease of calculation and obtaining result after varying different parameters a excel program can be develop.*

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TABLE 1: NOMENCLATURES

NOTATION	MEANING	NOTATION	MEANING
A_o	Heat transfer area based on outside surface area of tube, m^2	U_{of}	Overall heat transfer coefficient for fouled surface based on outer side surface area of tube, $W/m^2.K$
A_{pt}	Heat transfer area based on outside surface area of tube per number of tube passes, m^2	U_c	Overall heat transfer coefficient for cleaned surface based on outer side surface area of tube, $W/m^2.K$
A_{of}	Heat transfer area with fouling based on outside surface area of tube, m^2	R_{fo}	Shell side fouling resistance referring to the outside tube surface, $m^2.K/W$
B	Baffle spacing, m	A_s	cross flow area at or near shell centre line, m^2
C	Clearance between the tubes, m	F	Correction factor
CTP	Tube count calculation constant	v	Average velocity inside tubes, m/s
CL	Tube layout constant	R_{ft}	Total fouling, $m^2.K/W$
C_p	Specific heat, J/kgK	L	Tube length, m
D_s	Shell diameter, m	Q	Heat transfer rate, W
D_e	Equivalent Shell diameter, m	$\dot{m}$	Mass flow rate, m/s
d_o	Tube outer diameter, m	N_u	Nusselt number
d_i	Tube inside diameter, m	N_t	Number of tubes
ΔT_{lm}	Log mean temperature difference, ^oC, or K	R_e	Reynolds number
T_{hi}	Hot fluid inlet temperature, ^oC, K	h_i	Tube side heat transfer coefficient, $W/m^2.K$

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T_{ho}	Hot fluid outlet temperature, ^oC, K	h_o	Shell side heat transfer coefficient, W/m^2.K
P_r	Prandtl number	U_o	Overall heat transfer coefficient, W/m^2.K
PR	Pitch ratio	T_2	Shell side outlet temperature, ^oC, K
P_T	Pitch size, m	t_1	Tube side inlet temperature, ^oC, K
G_s	Shell side mass velocity, Kg/m^{2}K	t_2	Tube side outlet temperature, ^oC, K
f	Friction factor	N_b	Number of baffles
φ_s	Viscosity correction factor for shell side fluids	N_p	Number of tube passes
μ	Dynamic viscosity N.s/m^2	$\dot{m}_s$	Shell side mass flow rate, m/s
K	Thermal conductivity, W/m.K	$\dot{m}_t$	Tube side mass flow rate, m/s
ρ	Density, Kg/m^3	T_1	Shell side inlet temperature, ^oC, K
Δp_t	Tube side pressure drop, P_a	ΔP_s	Shell side pressure drop, P_a
T_{ci}	Cold fluid inlet temperature, ^oC, K	ΔT	Temperature difference, ^oC, K
T_{co}	Cold fluid outlet temperature, ^oC, K	T_w	Wall temperature, ^oC, K

TABLE 2: APPROXIMATE RANGE VALUES OF OVERALL HEAT TRANSFER COEFFICIENT FOR PRELIMINARY STAGE [16]

Fluids	**U W/m^2.K**	**Fluids**	**U W/m^2.K**
Water to water	1300-2500	Steam to heavy organics	30-300
Ammonia to water	1000-2500	Light organics to light organics	200-350
Gas to water	10-250	Medium organics to medium organics	100-300
water to compressed air	50-170	Heavy organics to heavy organics	50-200
Water to lubricating oil	110-340	Light organics to heavy organics	50-200
Light organics to water	370-750	Heavy organics to light organics	150-300
Medium organics to water	240-650	Crude oil to gas oil	130-320
Heavy organics to water	25-400	Plate heat exchanger: water to water	3000-4000
Steam to water	2200-3500	Evaporator :steam/water	1500-6000
Steam to ammonia	1000-3400	Evaporator :steam/other fluids	300-2000
Water to condensing ammonia	850-1500	Evaporator of refrigeration	300-1000
Water to Freon-12	280-1000	Condenser : steam/water	1000-4000
Steam to gas	25-240	Condenser : steam/other fluids	300-1000
Steam to light organics	490-1000	Gas boiler	10-50
Steam to medium organics	250-500	Oil bath for heating	30-550

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