Shell & Tube Heat Exchanger (STHX)

Dr. Md. Zahurul Haq, *Ph.D., CEA, FBSME, FIEB*

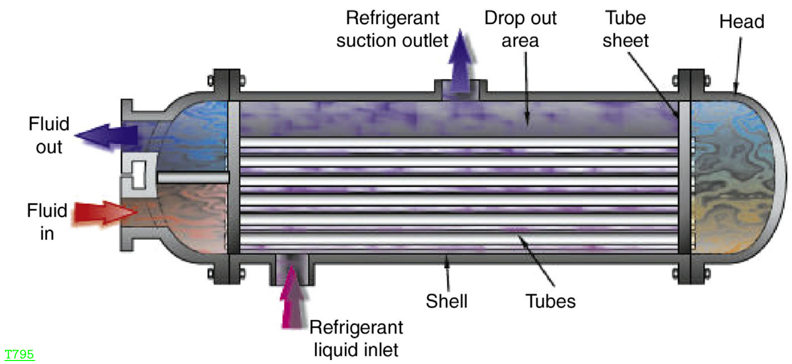
Professor
Department of Mechanical Engineering
Bangladesh University of Engineering & Technology (BUET)
Dhaka-1000, Bangladesh

zahurul@me.buet.ac.bd
http://zahurul.buet.ac.bd/

ME 307: Heat Transfer Equipment Design

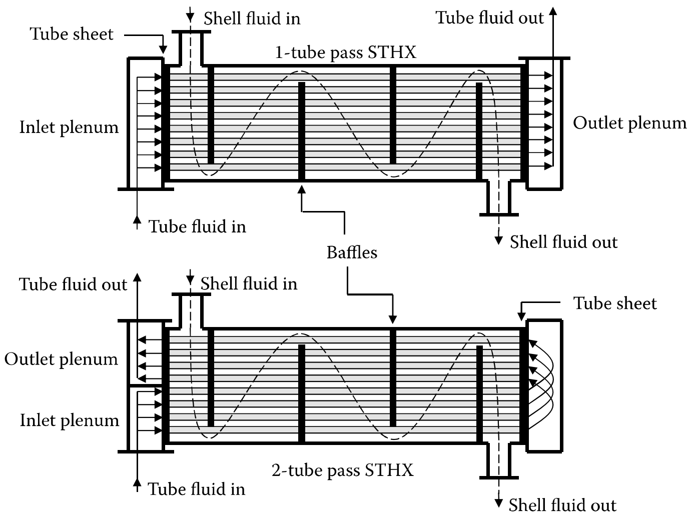
`http://zahurul.buet.ac.bd/ME307/`

Shell & Tube Heat Exchanger (STHX)

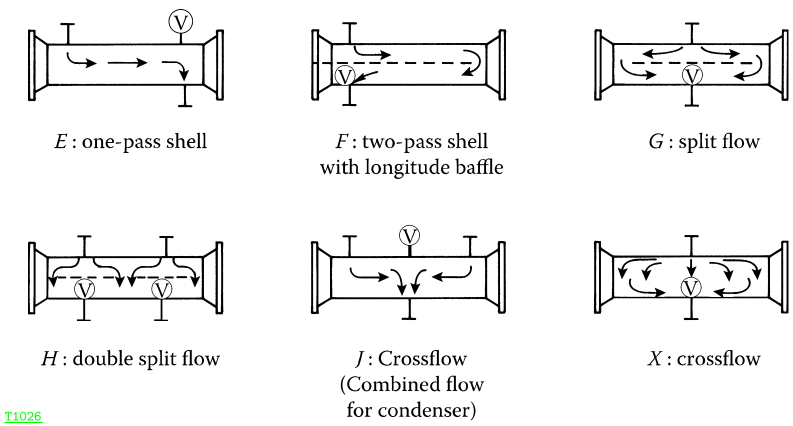


T795

Shell-and-tube evaporator, flooded.

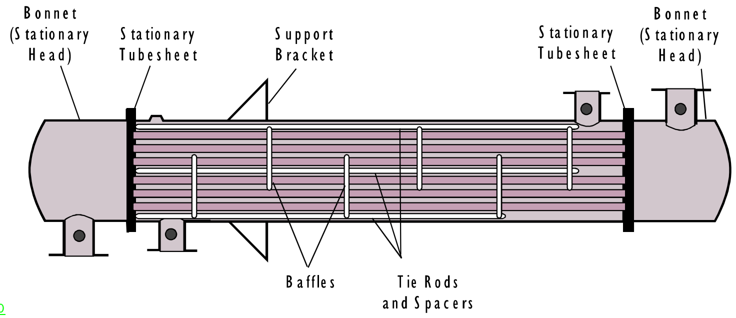


T735



T1026

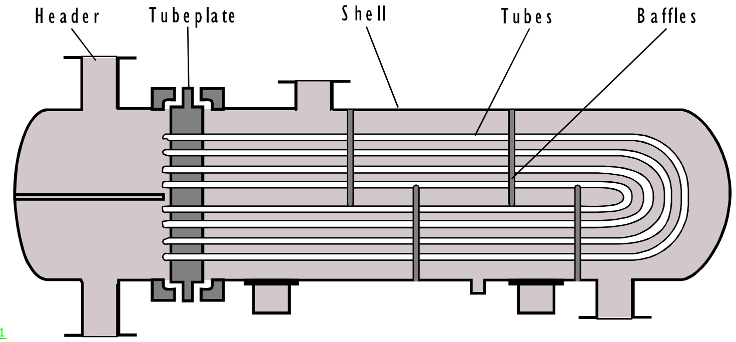
Most common TEMA shell types



T1020

Fixed-tubesheet heat exchanger

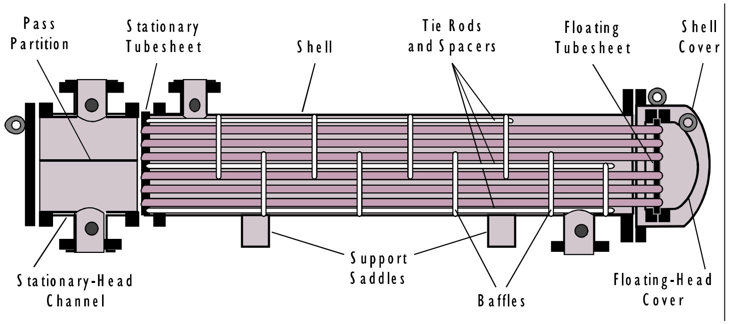
- Low cost because of simple construction.
- Inside of the tubes can be mechanically cleaned easily, but not the outside of the tubes.
- Application of limited to clean services on the shell side.



T1021

U-tube heat exchanger

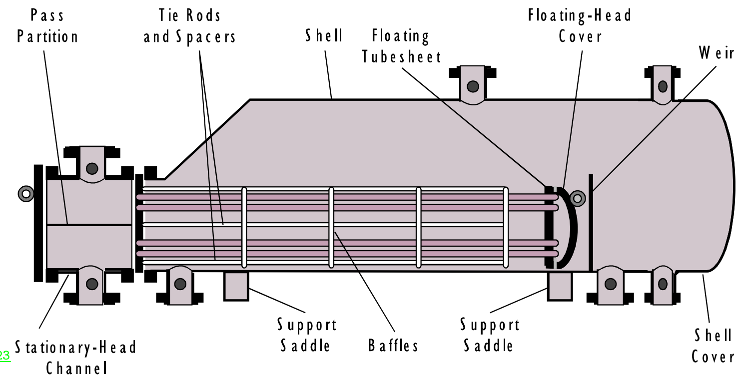
- Tubes can expand or contract in repose to stress differentials.
- Outside of the tubes can be mechanically cleaned, but not the inside.
- Not suitable for services with dirty fluids inside tubes.



T1022

Pull-through floating-head exchanger with backing device (TEMA S)

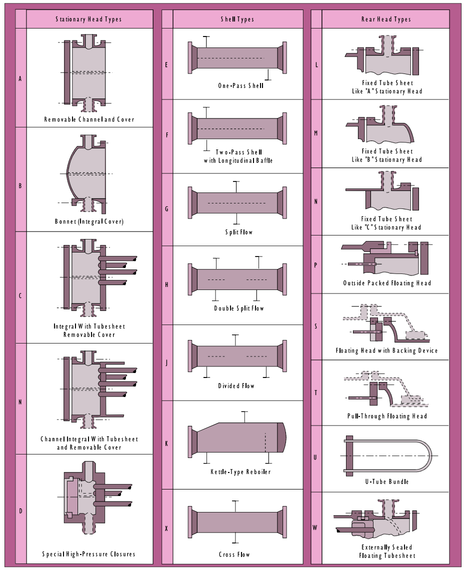
- Permits free expansion/contraction, and inside/outside of the tubes can be mechanically cleaned easily.
- Suitable for services where both fluids are dirty. Most common type of HX in chemical process industries.



T1023

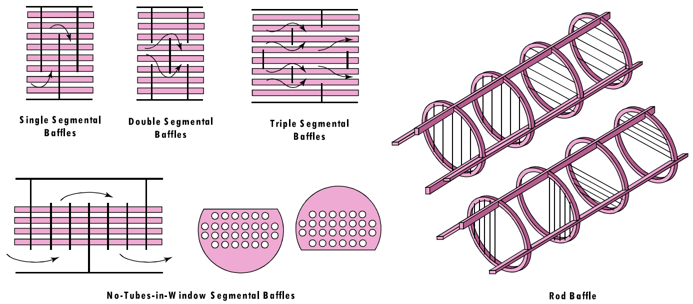
Pull-through floating-head exchanger (TEMA T)

- Permits free expansion/contraction, and inside/outside of the tubes can be mechanically cleaned easily.
- Suitable for kettle reboilers having a dirty heating medium.
- Cost is high.

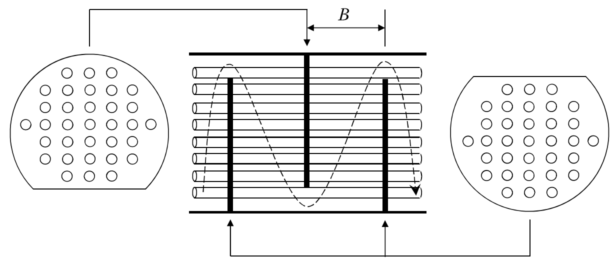


T794

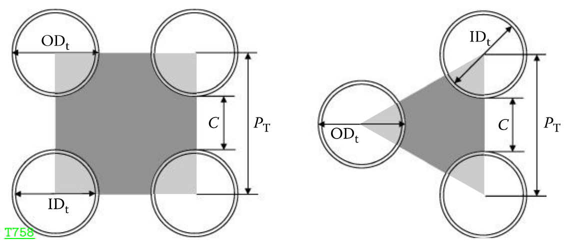
TEMA designations for shell-and-tube heat exchangers.



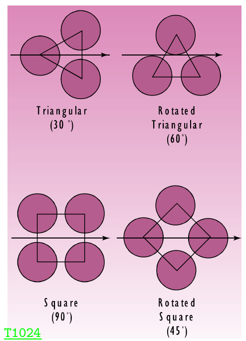
T1025



T760



T758



T1024

- A triangular pattern permits more tubes than square pattern. It produces more turbulence and higher heat transfer rates. However, these are difficult to clean mechanically.
- For dirty shell side fluids $\longrightarrow$ square layout.
- Tube clearance, $C = P_T - OD_t$

- Tube pitch ratio, $PR \equiv P_T/OD_t$, $1.2 \leqslant PR \leqslant 1.5$
- **Preferred tube length, L:**
  6 ft (1.83 m), 8 ft (2.44 m), 12 ft (3.66 m), 16 ft (4.88 m), 20 ft (6.1 m), 24 ft (7.32 m).
- Total outside tube area, $A_o = N_t(\pi OD_t L)$
- $$CL = \begin{cases} 1.0 & \text{for } 90^o, 45^o \\ 0.87 & \text{for } 30^o, 60^o \end{cases}$$
- $$CTP = \begin{cases} 0.93 & \text{for 1 tube pass} \\ 0.90 & \text{for 2 tube passes} \\ 0.85 & \text{for 3 tube passes} \end{cases}$$
- Shell inside diameter, $D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o(PR)^2 OD_t}{L}\right]^{1/2}$

Tube-side Thermo-hydraulics

- Total tube x-section area, $A_t = \left(\frac{N_t}{N_p}\right)\frac{\pi}{4}ID_t^2 \quad N_p =$ no. of pass
- Laminar: $Nu = 1.86(Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}; \quad Gz = \frac{RePr}{L/OD_t}$
- Turbulent: $Nu = 0.023Re^{0.8}Pr^n; \quad n = \begin{cases} 0.4 & : heating \\ 0.3 & : cooling \end{cases}$
- For smooth pipes, $f_t = \begin{cases} 0.316Re^{-0.25} & : Re \leqslant 2\times10^4 \\ 0.184Re^{-0.20} & : 2\times10^4 \leqslant Re \leqslant 3\times10^5 \end{cases}$
- Pressure drop, $\Delta P_t = N_p\left(f_t\frac{L}{ID_t} + 4\right)\left(\frac{1}{2}\rho_t V_t^2\right)$
- $h_i \propto V_t^{0.8}; \ \Delta P_t \propto V_t^2 L: \mapsto \frac{h_i(8\ pass)}{h_i(2\ pass)} \simeq 3, \ \frac{\Delta P_t(8\ pass)}{\Delta P_t(2\ pass)} \simeq 64.$
  2 pass to 8 pass: h increases 3 times but ΔP_t increases 64 times.

Shell-side Thermo-hydraulics

- $$Dh_s = \frac{4A}{P} = \begin{cases} \frac{4P_T^2}{\pi OD_t} - OD_t & : square\ pitch \\ \frac{2\sqrt{3}P_T^2}{\pi OD_t} - OD_t & : triangular\ pitch \end{cases}$$
- $0.4D_s \leqslant B \leqslant 0.6D_s, \quad B =$ baffle spacing
- Number of baffles, $N_b = \frac{L}{B} - 1$
- Bundle cross-flow area, $A_s = \frac{D_s CB}{P_T}$
- $Nu = 0.36Re_s^{0.55}Pr^{1/3}, \quad Re_s = V_s Dh_s/\nu_s$
- $f_s = \exp\left[0.576 - 0.19\ln(Re_s)\right]$
- $\Delta P_s = f_s(N_b+1)\frac{D_s}{D_e}\left(\frac{1}{2}\rho_s V_s^2\right)$

Approximate Overall Heat Transfer Coefficient (U_o)

Fluids	U_o (W /m^2 K)
Water to water	1300-2500
Gases to water	10-250
Water to lub. oil	110-300
Steam to water	2200-3500
Steam to gases	25-240
Evaporators: steam/water	1500-6000
Evaporators: steam/other fluids	300-2000
Evaporators of refrigerants	300-1000
Condenser: steam/water	1000-4000
Condenser: steam/other fluid	300-1000
Plate heat exchanger: water to water	3000-4000
Gas boiler	10-50
Oil bath for heater	30-550

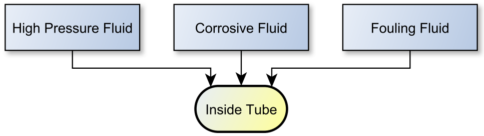
BWG Specification for Tubes in a STHX

OD (inch)	BWG	ID (inch)	ID (centimeter)	OD (inch)	BWG	ID (inch)	ID (centimeter)
¾	10	0.482	1.224	1¼	7	0.890	2.261
	11	0.510	1.295		8	0.920	2.337
	12	0.532	1.351		10	0.982	2.494
	13	0.560	1.422		11	1.010	2.565
	14	0.584	1.483		12	1.032	2.621
	15	0.606	1.539		13	1.060	2.692
	16	0.620	1.575		14	1.084	2.753
	17	0.634	1.610		16	1.120	2.845
	18	0.652	1.656		18	1.152	2.926
	20	0.680	1.727		20	1.180	2.997
1	8	0.670	1.702	1½	10	1.232	3.129
	10	0.732	1.859		12	1.282	3.256
	11	0.760	1.930		14	1.334	3.388
	12	0.782	1.986		16	1.370	3.480
	13	0.810	2.057				
	14	0.834	2.118				
	15	0.856	2.174				
	16	0.870	2.210				
	18	0.902	2.291				
	20	0.930	2.362				

T1027

STHX Design Considerations

- Pressure drop on each side of HX should be less than 70 kPa.
- Fluid placement should be based on either the hydraulic criterion (minimizing the pressure drop) or the fouling criterion (easy mechanical cleaning of the heat exchanger).
- Place fluid with higher fouling through the tubes. For same fouling, place the higher mass flow rate fluid through the larger area.
- STHX should deliver required heat transfer rate when completely fouled. Fouling factors represent fouling after 1 year of service.



T1028

STHX Rating ▷ 20.0 kg/s water at 50°C is to be cooled using 20.0 kg/s water available at 25 °C. $N_t = 200$, $N_p = 2$, $L = 4.88$ m, $N_b = 15$, $B = 304.8$ mm, $D_s = 438.2$ mm, $ID_t = 16.55$ mm, $OD_t = 19.07$ mm. Assume 25.4 mm triangular pitch.

STHX Sizing ▷ 20.0 kg/s water at 50°C is to be cooled to 38.1°C using 20.0 kg/s water available at 25 °C. ~~$N_t = 200$, $N_p = 2$, $L = 5.0$, $N_b = 15$, $B = 304.8$ mm, $D_s = 438.2$ mm, $ID_t = 16.55$ mm, $OD_t = 19.07$ mm. Assume 25.4 mm triangular pitch. Ignore fouling in design.~~