HTFS – General Introduction to the Thermal Design of Shell & Tube type Heat Exchangers

HTFS Computer Based Training

- This Computer Base Training for the HTFS is divided into the following sessions:
  - **General Heat Exchanger Design Introduction** – overview of the design of shell and tube type heat exchangers
  - **Thermal Design Process** – Determination of heat exchanger size, arrangement, and configuration
  - **Condensers** – Theory and design of condenser applications
  - **Design Optimization** – How to better optimize an exchanger application to the lowest cost solution
  - **Vibration** – How to interpret and solve potential vibration indications

General Heat Exchanger Design Introduction

HTFS – TEMA Shell Types Applications

- **E-type** – Is the most common of the shell types. The one shell pass with the entrance and exit nozzles at opposite ends is the ideal arrangement for excellent performance. Used with a single tube pass temperature crosses can be avoided.
- **F-type** – The F-shell is typically used when a temperature cross exists that would otherwise force the design into multiple shells in series. They are not recommended with removable bundles. A tight seal is required at the long baffle as the unit will not perform as designed should there be fluid leakage across the longitudinal baffle. The amount of heat transferred is greater than for an E-shell, but the shell side pressure drop is also higher and there is some thermal leakage across the long plate.
- **G-type**- The shell side fluid splits into two with the two halves flowing in opposite directions around a partial longitudinal baffle. The G-shell is used when the available pressure drop is limited. It is a hybrid 2-pass shell.
- **H-type** – The arrangement is quite often used for shell side thermosiphon applications or shell side condensers with low allowable pressure drop.
- **J-type** – It is used when there is limited available pressure drop on the shell side. It can not be used when the tube side temperature crossed the temperature on the shell side. The J type is quite often used for shell side condensers.

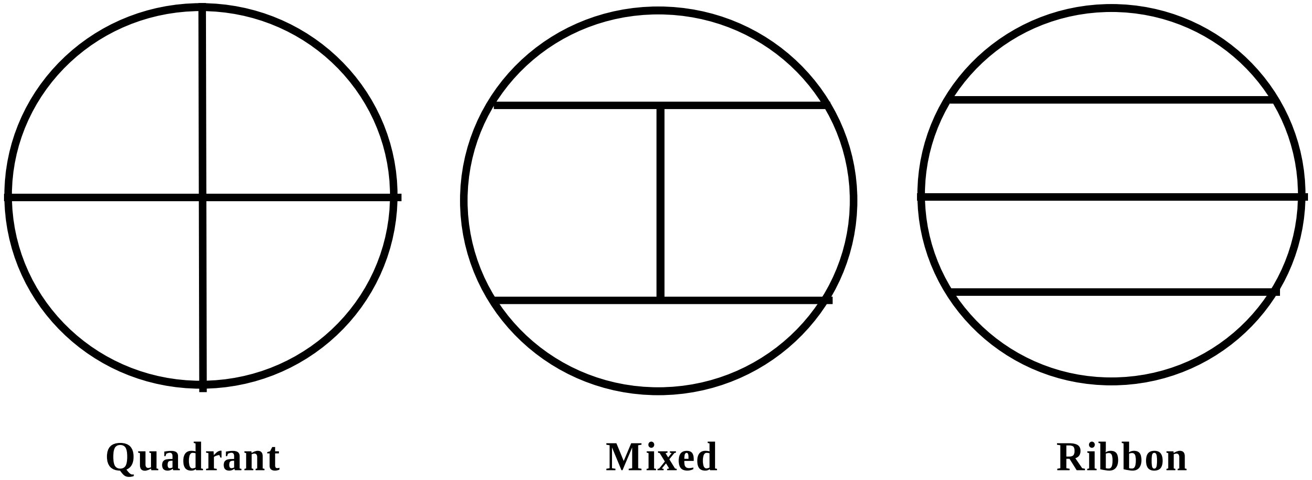
HTFS – TEMA Shell Types Applications

- **K-type** – Kettles are frequently used under distillation columns to provide vapor reflux and energy back to the column for distillation.
- **X-type** – Flow is distributed along the entire length of the bundle and flows across the bundle perpendicular to the tubes. The distribution is accomplished by multiple nozzles along the shell or via open areas at the top & bottom of the shell. Support plate type baffle are used to support tubes. With this shell arrangement, the shell side pressure drop is minimized.

HTFS – Tube Layout

- There are several possible ways to layout tubes for four or more passes. The primary effect on the thermal design is due to the different number of tubes, which are possible for each type.
- The **Quadrant** layout has the advantage of usually (but certainly not always) giving the <u>highest tube count</u>. It is the required layout for all U-tube designs of four or more passes. The tube side nozzles must be offset from the centerline when using quadrant layout. The program will automatically avoid quadrant layout for shells with longitudinal baffles and 6, 10, or 14 passes, in order to avoid having the longitudinal baffle bisect a pass.
- The **Mixed** layout has the advantage of keeping the tube side nozzles on the centerline. It often gives a tube count close to quadrant and sometimes exceeds it. The program will automatically avoid mixed layout for shells with longitudinal baffles and 4, 8, 12, or 16 passes.
- The **Ribbon** layout nearly always gives a <u>layout with fewer tubes</u> than quadrant or mixed layout. It is the layout the program always uses for an odd number of tube passes. It is also the layout preferred by the program for X-type shells. The primary advantage of ribbon layout is the more gradual change in operating temperature of adjacent tubes from top to bottom of the tubesheet. This can be especially important when there is a large change in temperature on the tube side, which might cause significant thermal stresses in mixed and especially quadrant layouts.

HTFS – Tube Layout types



HTFS – Deviation in number of tubes per tube side pass

- For multipass layouts, it is desirable to have close to the same number of tubes in each pass when there is no change of phase on the tube side. However, for most layouts of more than two passes, this would require removing tubes which would otherwise fit within the outer tube limit. Since it is preferable to maximize the surface area within a given shell and minimize the possible shell side bypassing, a reasonable deviation in tubes per pass is usually acceptable (5% or less deviation). It is recommended that you avoid large deviations since this gives significantly different velocities in some passes and wastefully increases the pressure drop due to additional expansion and contraction losses.

- Some heat exchanger technical specifications set limits on tube side minimum velocity which is mainly due to minimize the corrosive effect of process fluid. For such cases it is desirable to intentionally decrease the last tube side passes tube count in total condensation applications to meet the min. velocity limit.

Note: Hetran (B-Jac) program bases the tube side calculations on an average number of tubes per pass, so such deviation are not reflected in the thermal design.

HTFS – Tie-rods and Sealing Strips

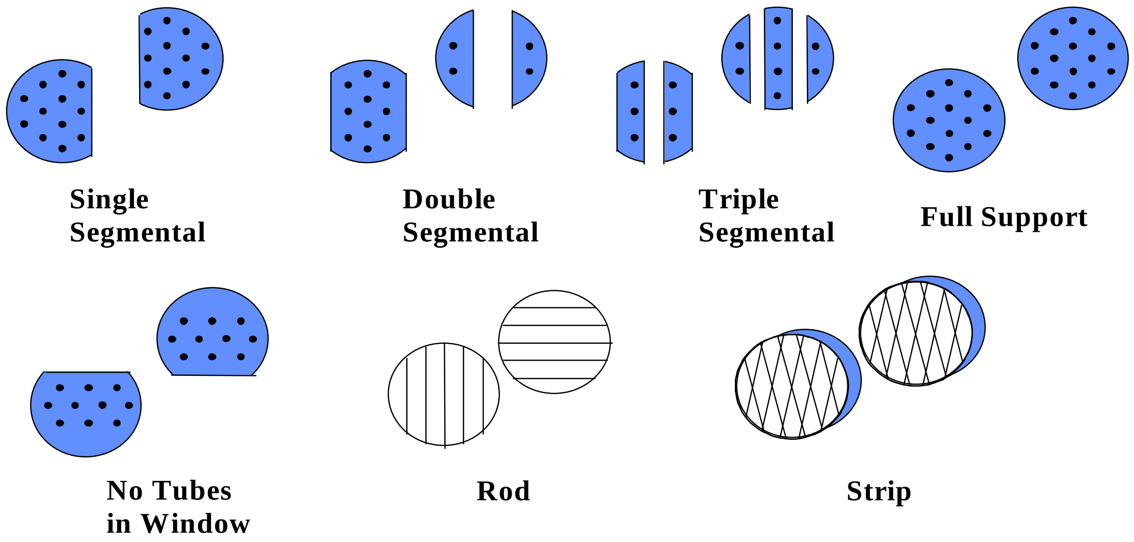
- The tie-rods and spacers are necessary to keep the baffles and tube support plates at their appropriate locations. Their number and diameter vary with the shell diameter. The number of tubes in an exchangers will be reduced due to the space occupied by the tie-rods and spacers. Usually each set of a tie-rod and spacer requires the removal of four to six tubes around it. The TEMA standards provide guidance as to the minimum number required. TASC adheres to the TEMA recommendation.
- Sealing strips are used to reduce bypassing of the shell side flow around the bundle between the shell ID and the outer most tubes. They are installed in pairs on the baffles usually by welding. In fixed tubesheet (L, M, & N rear heads) and U-tube heat exchangers the clearance between shell ID and the outer tube limit is comparatively small. Therefore sealing strips are seldom needed for these types. In inside floating head (S & T rear heads), outside packed floating head (P rear head), and floating tubesheet (W rear head) heat exchangers, the potential for bypassing is much greater. In these cases sealing strips are generally required. The thermal design calculations in HTFS assume that sealing strips are always present in P, S, T, & W type heat exchangers. Generally, <u>one pair of seals strips is used for every 6 rows</u> of tubes in cross-flow.

HTFS – Baffles types

– Baffles are used to direct the shell side flows so that the fluid velocity is increased to a point to maintain a <u>high heat transfer coefficient</u> as well as <u>to minimize fouling</u>. In horizontal exchangers the baffles also aid in supporting the tube bundle <u>to prevent the tubes from sagging or vibrating</u>.

HTFS supplies the following baffles:

1. Single segmental provides the best thermal performance but also the highest pressure drop.
2. The multi-segmental types decrease pressure drop significantly with a corresponding reduction in heat transfer coefficient.
3. The Rod and Strip types will provide the lowest pressure drop but with a significant reduction in performance.

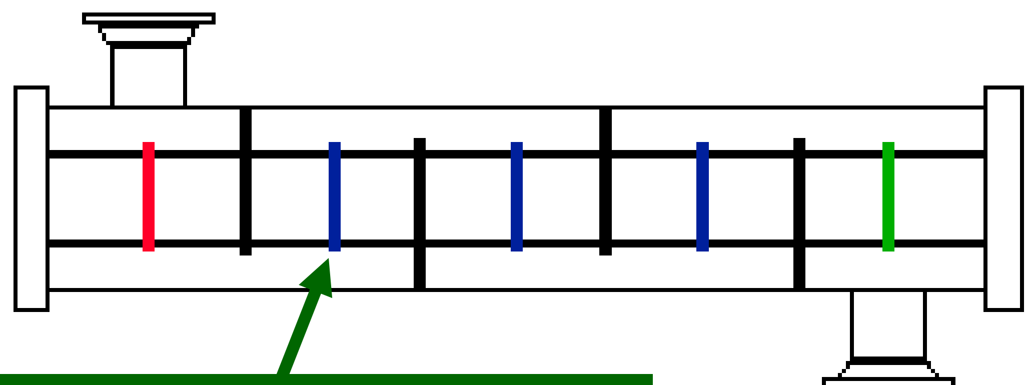


HTFS – Single Segmental

- Single segmental baffles are the most common. The baffle cuts will vary from 15-45% of the shell diameter. A cut of 20-25% will normally provide the highest film coefficient for a given pressure drop.
- For single phase exchangers, up and over flow (horizontal baffles) is best practice if the shell side nozzles are at the top or bottom of the shell. This avoids the region under the nozzle with tubes removed providing a large area in which flow could bypass the bundle. Large bypass areas give reduced bundle mass flux, and hence shell side coefficient for single phase flows.
- For boiling or condensing shell side flows, potential phase separation problems with up and over flows (particularly for condensing) means vertical baffle cuts are preferred.
- For horizontal heat exchangers it is far more important. For the particular case of horizontal condensers, vertical cut baffles (i.e. side-to-side flow) is often chosen since its use permits reasonable liquid-vapor separation (which means that any non-condensables can be easily vented) and provides a small amount of liquid sub-cooling.
- For single segmental baffles, TASC allows a cut of 15% to 45%. Greater than 45% is not practical because it does not provide for enough overlap of the baffles. Less than 15% is not practical, because it results in a high pressure drop through the baffle window with relatively little gain in heat transfer (poor pressure drop to heat transfer conversion). Generally, where baffling the flow is necessary, the best baffle cut is around 25%.
- For F and G shells with up-and-over flow (i.e. horizontal baffle cut) the maximum baffle cut is 25%.

HTFS – No Tubes in Window (NTIW)

– There is also a single segmental baffles where no tubes are placed where the baffle cut creates the baffle window (NTIW). These baffles are normally used to resolve vibration problems. They provide support for the tubes at each baffle segment. Intermediate support plates are often used with no tubes in the window baffles. The support plate is similar to the central piece of a double segmental baffle. It provides additional tube support and very little additional pressure drop. <u>The cut is normally limited to approximately 15%. Otherwise, the sacrifice in the number of tubes/shell would be too costly.</u>



The advantage of a NTIW arrangement is that the tube unsupported span can be minimized to avoid vibration. Additional intermediate support plates can be also added as shown.

HTFS – Double / Triple Segmental and Rod / Strip

- Double segmental baffles are an excellent selection for moderately reducing the shell side pressure drop. Double segmental baffles create more of a parallel type flow than a cross flow that exists with single segmental baffles. There should exist at least a two row overlap between adjacent baffle segments. The baffle cut should be in the range of 15 to 25%.
- Double segmental up-and-over baffles are not allowed in F and G shells.
- Triple segmental baffles are used infrequently primarily because of their high cost. They are used when a significant reduction in the shell side pressure drop is required.
- Rod baffles are licensed through Phillips Petroleum Company. Only a limited number of fabricators are licensed to use them. Each tube is supported at four circumferential points at each baffle. The flow is parallel to the bundle. They provide an excellent remedy when vibration or restrictive pressure drop on the shell side is a problem. They are only used with square patterns.

HTFS – Baffle cut orientation

- For a single phase fluid in a horizontal shell, the preferable baffle orientation horizontal, although vertical and rotated are also acceptable. The choice will not affect the performance, but it will affect the number of tubes in a multipass heat exchanger. The horizontal cut has the advantage of limiting stratification of multi-component mixtures, which might separate at low velocities. The rotated cut is rarely used. Its only advantage is for a removable bundle with multiple tube passes and rotated square layout. In this case the number of tubes can be increased by using a rotated cut, since the pass partition lane can be smaller and still maintain the cleaning paths all the way across the bundle. (From the tubesheet, the layout appears square instead of rotated square.)
- For horizontal single phase fouling applications the vertical cut baffles are preferred, in that it prevents dirt settlement.
- For horizontal shell side condensers, the orientation should always be vertical, so that the condensate can freely flow at the bottom of the heat exchanger. These baffles are frequently notched at the bottom to improve drainage.
- For shell side pool boiling, the cut (if using a segmental baffle) should be vertical.
- For shell side forced circulation vaporization, the cut should be horizontal in order to minimize the separation of liquid and vapor.
- For double and triple segmental baffles, the preferred baffle orientation is vertical. This provides better support for the tube bundle than a horizontal cut which would leave the topmost baffle unsupported by the shell. However this can be overcome by leaving a small strip connecting the topmost segment with the bottommost segment around the baffle window between the O.T.L. and the baffle O.D.

HTFS – Intermediate Tube Supports

- Intermediate supports can normally be used with no-tubes in window (NTIW) designs, since otherwise they would block along the exchanger. The exceptions to this rule are U-tube extra supports and cross flow exchangers (K and X-shells).
- Five types of tube supports can be specified:
  - Midspace intermediate supports
  - Intermediate supports in inlet endspace
  - Intermediate supports in other/return endspace
  - Intermediate supports under central nozzle (e.g. J shell inlet or outlet nozzle)
  - U-bend extra supports
- Split backing ring and pull-through floating head exchangers have a special support / blanking baffle adjacent to the floating head to take the weight of the complete floating head assembly. In this case the tube surface area between floating tubesheet and support becomes ineffective. In order to increase the effectiveness of this surface it is usual to cut slots of various shapes in the baffle.

HTFS – Surface Enhancement

Tubes are also available with externally enhanced surfaces such as low fins. They normally become economical to use when the ratio of the tube side to shell side film coefficient is 3:1. They should not be used with fluids that have high surface tensions. Low fins are most effective in pure cross flow (X shells, NTIW, and segmental baffles). In longitudinal flow, such as triple segmental baffles, rod or strip baffles, the fin valleys are not effectively penetrated by the flow. In such situations pressure drop increases due to the fins acting as a rough surface. While such an increase is not reflected in heat transfer. Use of low fins in such cases is questionable.

Common fin densities by tubing material (density shown in fins/inch):

Carbon Steel	19	Nickel Alloy 600 (Inconel)	28
Stainless Steel	16, 28	Nickel Alloy 800	28
Copper	19, 26	Hastelloy	30
Copper-Nickel 90/10	16, 19, 26	Titanium	30
Copper-Nickel 70/30	19, 26	Admiralty	19, 26
Nickel Carbon Alloy 201	19	Aluminum-Brass Alloy 687	19
Nickel Alloy 400 (Monel)	28		

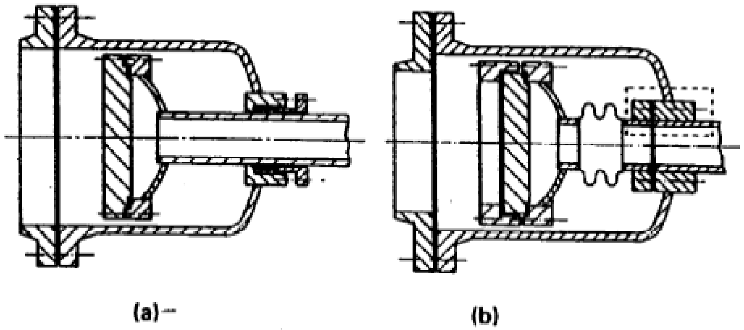
Thermal Design Process

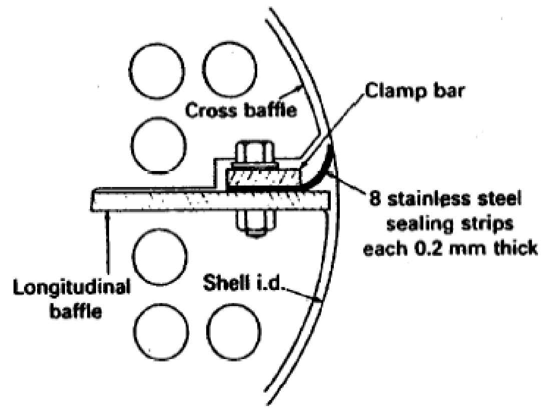
HTFS – Routing of Fluids

- If the shell-and-tube-sides require different materials, the best choice of routing can only be decided after designing both options in many cases. For instance, routing the higher pressure fluid through the shell may produce cheaper unit, particularly if it reduces the exchanger diameter and it is made of carbon steel and fixed tubesheet type.
- Stream with low available pressure drop should be allocated to tube-side or longitudinal shell-side arrangement (i.e. multi-segmental baffles, rods, strips baffles).
- In the other hand, extremely low pressure drop can be obtained in X-shells. The stream with low or sensitive pressure drop requirements may have to be tried in both options.
- High pressure, high temperature and corrosive fluids are best placed on the tube side, which eliminates expensive shells.
- Fouling fluids are preferably placed on the tube side except for U-tubes.
- Hazardous fluids for exchangers with expansion bellows, or with P or W type rear heads should be routed to tube side.
- Viscous fluids are preferably placed on the shell side, where the induced turbulence will result in higher heat transfer.
- Small flow rate and low heat transfer coefficient fluids can be best placed in shell side with considering proper surface enhancement.

HTFS – Temperature Cross

- If the duty involves a temperature cross, consideration should be given to countercurrent flow designs. These involve exchangers having one shell pass and one tube pass (1/1 units) or two shell passes and two tube passes (2/2 units), but floating-head and u-bend units have structural limitations described in below:
- In S and P rear head types, having a single tube-side pass, <u>special construction is required at the floating-head end to accommodate the tube-side nozzle which may be an inlet or an outlet.</u>
- In a fixed tubesheet exchanger the longitudinal baffle may be welded to the shell, but this is not possible with removable bundle exchangers as the baffle must be removed with the bundle. In this case the gap between longitudinal baffles and shell is sealed by packing devices or flexible strips. The flexible strips are placed the shell inlet side of the longitudinal baffle so that the higher pressure assists the strips to minimize close contact with the shell. <u>The worst leakage point for loss of both fluid and heat, however, is still adjacent to the stationary tubesheet where the pressure and temperature gradient is greatest.</u>





HTFS – Tentative Exchanger

- In a design problem, the heat load (Q) and the log mean temperature difference (ΔT_{lm}) must be known. The overall coefficient must be estimated from either experience or published data, such as provided in following table. A tentative surface area (A) is obtained from $A = Q/U\Delta T_{lm}$, after which a tentative design is established based on specified mechanical factors.
- Select a tube size , pitch, and length and determine the number of tubes, shell size, and baffling, if required. (This item will be discussed further in detail)

The overall heat transfer coefficients above are derived from an average of the clean film coefficients α (W/m² K) (marked by an asterisk), and the typical fouling factors r_f(W/m² K)$^{-1}$, given below.

Single-phase fluids		α	$r_f \times 10^4$
Gas at 1 bar abs.[1]*		80–140	2
Gas at 20 bar abs.[1]*		550–820	2
Gas at 50 bar abs.[1]		700–900	2
Gas at 100 bar abs.[1]		800–1200	2
'Light' organic liquid[2]*		1500–1850	4
'Medium' organic liquid[3]		500–1200	5
'Heavy' organic liquid[4]*	Cold	190–230	8
	Hot	130–210	
Ammonia		5000–7500	1
Methanol		5000–7500	1
Aqueous solutions		5000–7500	1
Process water*		4000–8000	5
Treated water*		4000–6000	2

Condensing fluids[5]		Pressure (bar abs.)	α	$r_f \times 10^4$
Organic vapours	'Light', no inerts[2]*	0.1	1200–1800	2
	'Light', no inerts[2]	0.1–1	1800–3500	2
	'Light', no inerts[2]	1–10	3500–7000	2
	'Medium', no inerts[3]	1	1200–2500	2
	'Heavy', no inerts[4]	1	500–1200	2
	'Light' + inerts[2]*	1–10	235–635	2
Ammonia		1	7500–12000	1
Methanol		1	7500–12500	1
Steam*		–	6850–9500	1

Boiling fluids		α HVC[6]	α K and H[7]	$r_f \times 10^4$
Organic liquids	'Light'[2]*	1800–4000	1000–1800	3
	'Medium'[3]	1500–3500	750–1500	3
	'Heavy'[4]	1000–2500	500–1000	5
Aqueous solutions, ammonia, water*		6000–9000	3000–5000	3

HTFS – Tube Side Flow

- If water is coolant, the number of tubes in one pass is chosen to meet the velocity requirements given in Project Technical Specification. (following table is for reference only)
- Typical velocities for fluids of low viscosity lie in the region of 0.5-3 m/s. As the fluid viscosity increases, the tube-side velocity falls and for highly viscous liquids may be as low as 0.1 m/s.
- The velocity range for gases is of the order of 2-15 m/s.

Metal	Velocity range (m/s)
Steel ('fresh' water)	0.8–1.5
Copper	0.9–1.2
Admiralty brass	0.9–1.8
Aluminium brass	1.2–2.4
Aluminium bronze	1.8–2.7
90/10 cupro-nickel	1.8–3.0
70/30 cupro-nickel	1.8–4.5
Monel	1.8–4.5
Stainless steel – type 316	2.4–4.5

Notes:
(1) Salt water unless otherwise stated.
(2) Maximum water temp. = 43 °C salt = 50 °C 'fresh'.

HTFS – Shell Side Flow

- Having temporarily settled the number of tube-side passes, a tentative baffle spacing (for instance half the TEMA recommended value) is assumed, and the shell-side heat transfer coefficient and pressure loss calculated. If the pressure loss exceeds the maximum allowable value, one or combination of following remedial corrective actions shall be undertaken.

Affecting Parameter	Remedial Action
Baffle type	Double or triple segmental
Shell type	J or X type shell
Tube pattern	Rotated square or Square
Tube diameter	Increase to 1” or 1.25”
Baffle cut	Use 30% to 40%
Tube Pitch	Increase to 1.4 to 1.5 x tube OD
Fluid allocation	Switch sides
Arrangement	Increase # of exch. In parallel
Tube type	Plain

HTFS – Controlling Heat Transfer Coefficient

- In order to reduce size and cost of the heat transfer equipment, the thermal design engineer will endeavor to achieve the highest value of U, consistent with the permissible pressure drop and other deign constraints. Of the five conductances from which U is obtained, the conductances of the fouling layers and wall are fixed, leaving the conductances of the two fluid films, α_i and α_o. The thermal design engineer will endeavor to obtain the highest values of α_i and α_o in order to increase U. In many applications α_i and α_o may be of different magnitude and the lower of the two coefficients is the controlling coefficient which must be increased if a significant increase in U is to be obtained.
- It will be evident that every effort must be made to maximize the controlling heat transfer coefficient by utilizing the available pressure loss. In this circumstances one or combination of following remedial corrective actions may be helpful.

Affecting Parameter	**Remedial Action**
Baffle type	Single segmental
Shell type	E or F type shell
Tube pattern	Triangular
Tube diameter	Decrease to 0.625” or 0.5”
Baffle cut	Use 15% to 20%
Tube pitch	Limit to TEMA std spacing
Fluid allocation	Switch sides
Arrangement	Increase # of exch. In series
Tube type	Use enhanced surface

HTFS – Operating and Design Margins

- In design calculations three surface areas arise: **(a)** the 'clean' surface area (A_{cl}), which is calculated from heat transfer correlations, assuming no fouling, **(b)** the 'service' surface area (A_{ser}), which is derived directly from A_{cl} by the inclusion of the required fouling factors, and **(c)** the installed surface area (A_{ins}).
- **Operating Margin:** The ratio A_{ins}/A_{cl} is the operating margin, usually intended to give the exchanger a 'reasonable' period of operation between shutdowns for cleaning. A high operating margin usually arises from the design parameters explained in below:

1. Use of 'large' fouling factors, such that they have a significant effect on design. Following explanatory examples shows the significance of fouling factor effect on operating margin when its order of magnitude is different from the tube inside and outside heat transfer coefficient.

Exchanger Service	Water / Water S/T Exchanger	Steam / Fuel Oil S/T Exchanger
Clean Overall Heat Transfer Coefficient	2840	114
TEMA Specified Fouling Factor	0.000528	0.0097
Service Overall Heat Transfer Coefficient	1136	102.6
Excess Surface Area Required Due to Fouling	2.5 times of clean exchanger	1.11 times of clean exchanger

HTFS – Operating and Design Margins

2. Uncertainties in the process design calculations where the process engineer may specify a larger duty for safety.
3. The user's demand for a fixed tube length, for instance, which can only be met by an unnecessary increase in A_{ins}.
4. Uncertainties in the thermal design calculations, where the thermal design engineer may increase the surface area for safety.

- Although a high operating margin may be regarded as a desirable objective in order to achieve a 'safe' design; however, an excessive margin may cause operating problems in the early life of the exchanger when it is clean. <u>Without temperature control the cold fluid may be overheated, leading to local boiling, corrosion, and fouling.</u> <u>With temperature control, which reduces the cold fluid flow may cause fouling due to its low velocity, in addition to the problems already mentioned.</u>
- <u>**Design Margin:**</u> Having calculated A_{ins} and A_{ser}, what design margin (i.e. A_{ins}/A_{ser}) should be allowed, bearing in mind the problems which may be caused by excessive surface area? There are no fixed rules regarding design margin; however it is recommended to keep the design margin to a minimum and not greater than 1.1, when all the following conditions satisfied:

1. When the reliability of the heat transfer data available to the thermal design engineer is confirmed.
2. When the tube side heat transfer coefficient is controlling the design.
3. When the shell side flow regime is turbulent (heat transfer prediction is much certain for turbulent flow)
4. When the fouling factors is small (i.e. fouling factor does not affect much the required heat transfer area)
5. When the thermal design is performed based on HTRI or HTFS programs (more accurate than other published methods)

Condenser (Theory and Design Features)

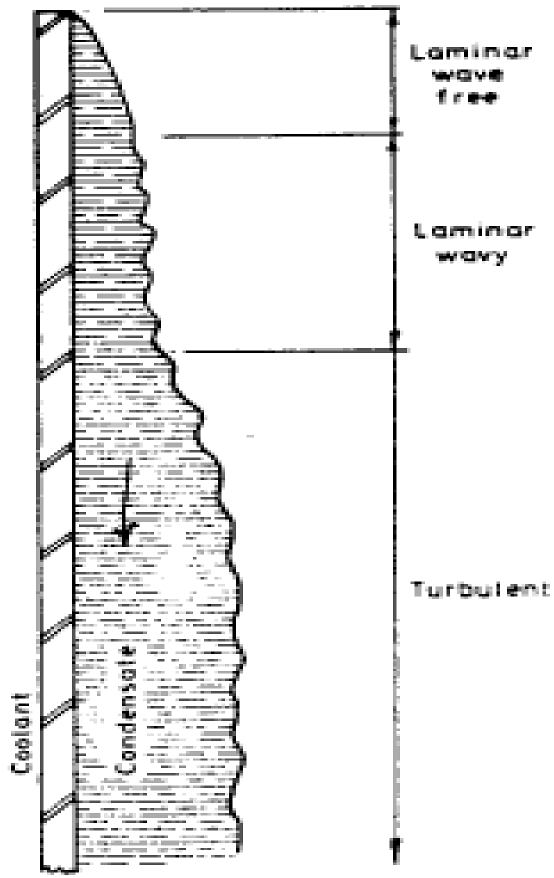
HTFS – Objectives for the Session

Our discussions will include:

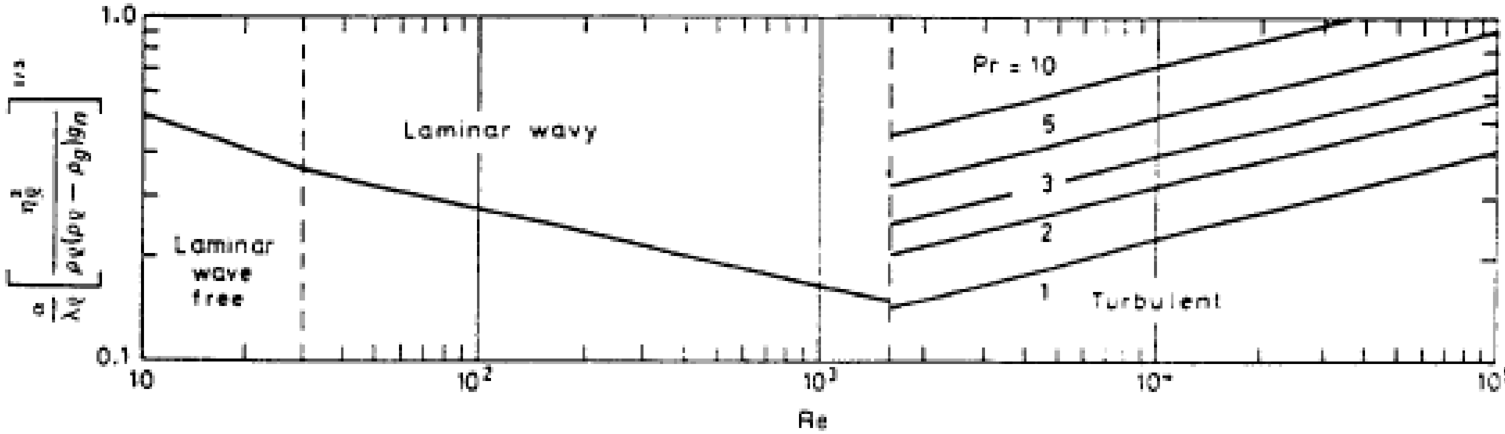
- Condensation -- general principals
- De-superheating & Sub-cooling
- Multi-Component Condensation
- Condenser Operational Problems / Design Recommendations
- Preliminary Condenser Selection

HTFS- Condensation on Vertical Surfaces Including Vertical Tubes

- During condensation, a liquid film develops on the tube wall in the exchanger when the tube wall temperature reaches a point below the dew point of the vapor – Filmwise Condensation. For condensation on vertical surface following distinct flow regime can be observed.
- At the top of the surface, there is a region of very-low condensate Reynolds number, where the flow is both laminar and wave free. In this region, the heat transfer coefficient decreases with thickening of the condensate.
- At some point down the surface, the Reynolds number grows to the point where instabilities form at the vapor-liquid interface, thus giving waves on the film. In this region, the heat transfer coefficient will still decrease with thickening the condensate, although the rate of decrease will be less due to the mixing effect of the waves.
- Still farther down the surface, the Reynolds number grows to the point where turbulence occurs. The turbulence causes an extra thickening of the film in addition to that caused by accumulated condensate. However, the thickening is usually more than compensated for by the better heat transfer in turbulent flow. The heat transfer coefficient therefore tends to increase in this region.

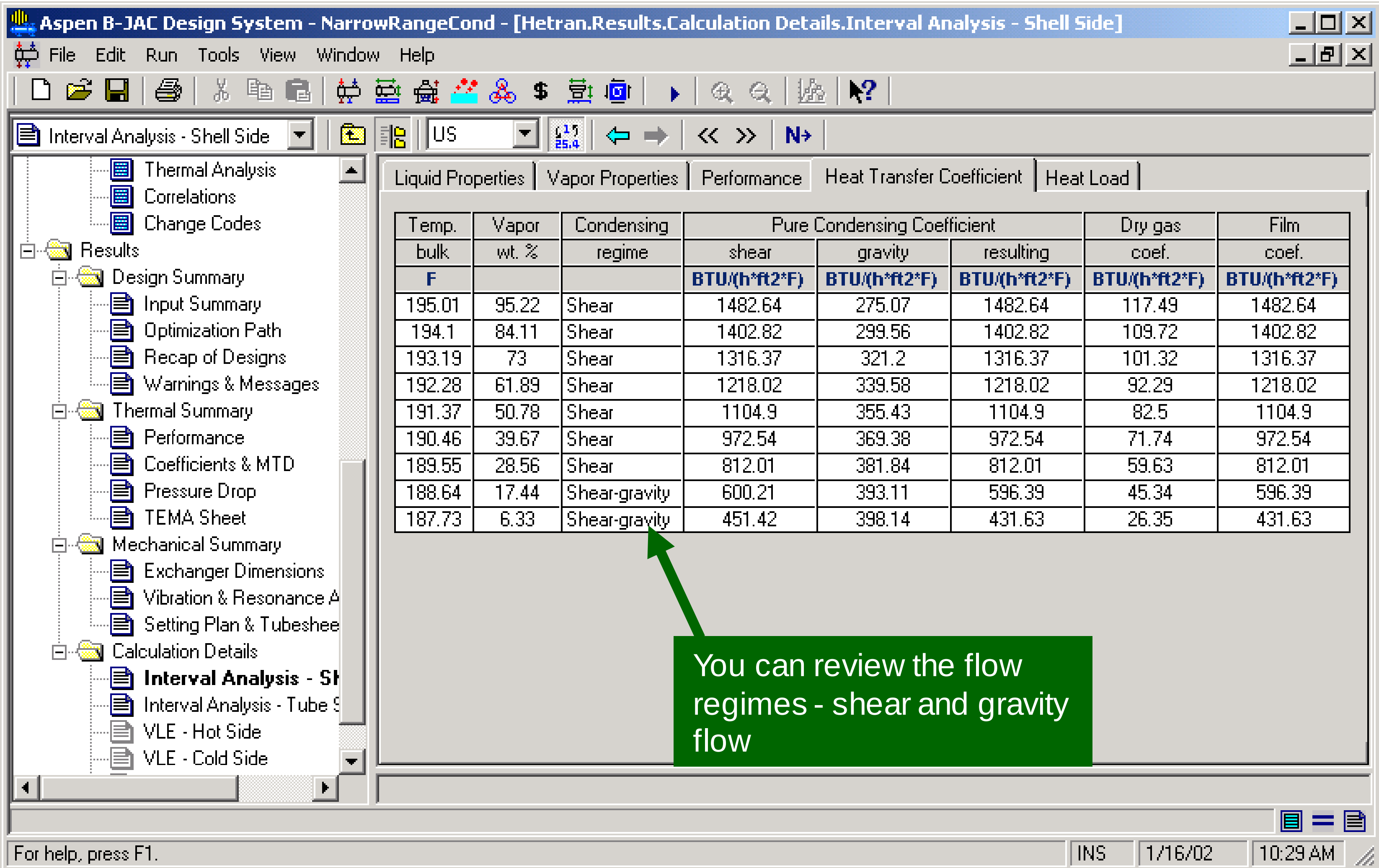


HTFS- Condensation on Vertical Surfaces Including Vertical Tubes



- The condensing heat transfer film coefficient will be controlled by <u>gravity</u> or by the <u>shear</u> velocity of the vapor. A downward vapor velocity will tend to increase the coefficient both by thinning the film and by increasing the likelihood of turbulence.
- Some software packages (e.g. B-Jac) provide information on the flow regime in their calculation details, which would be helpful for thermal designer for deciding on the exchangers heat transfer surface affecting parameters (e.g. number of tubes, tube size, and tube length). (A sample of B-Jac detailed calculations is provided for further clarity)

HTFS- Condensation on Vertical Surfaces Including Vertical Tubes

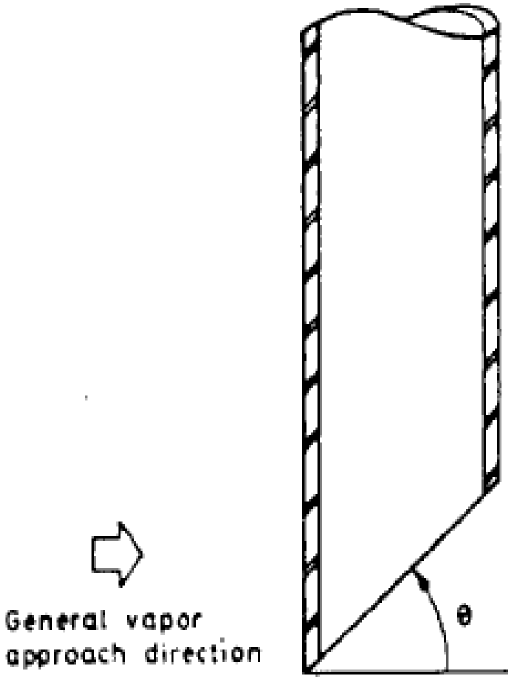


Temp. bulk	Vapor wt. %	Condensing regime	Pure Condensing Coefficient shear	gravity	resulting	Dry gas coef.	Film coef.
F			BTU/(h*ft2*F)	BTU/(h*ft2*F)	BTU/(h*ft2*F)	BTU/(h*ft2*F)	BTU/(h*ft2*F)
195.01	95.22	Shear	1482.64	275.07	1482.64	117.49	1482.64
194.1	84.11	Shear	1402.82	299.56	1402.82	109.72	1402.82
193.19	73	Shear	1316.37	321.2	1316.37	101.32	1316.37
192.28	61.89	Shear	1218.02	339.58	1218.02	92.29	1218.02
191.37	50.78	Shear	1104.9	355.43	1104.9	82.5	1104.9
190.46	39.67	Shear	972.54	369.38	972.54	71.74	972.54
189.55	28.56	Shear	812.01	381.84	812.01	59.63	812.01
188.64	17.44	Shear-gravity	600.21	393.11	596.39	45.34	596.39
187.73	6.33	Shear-gravity	451.42	398.14	431.63	26.35	431.63

You can review the flow regimes - shear and gravity flow

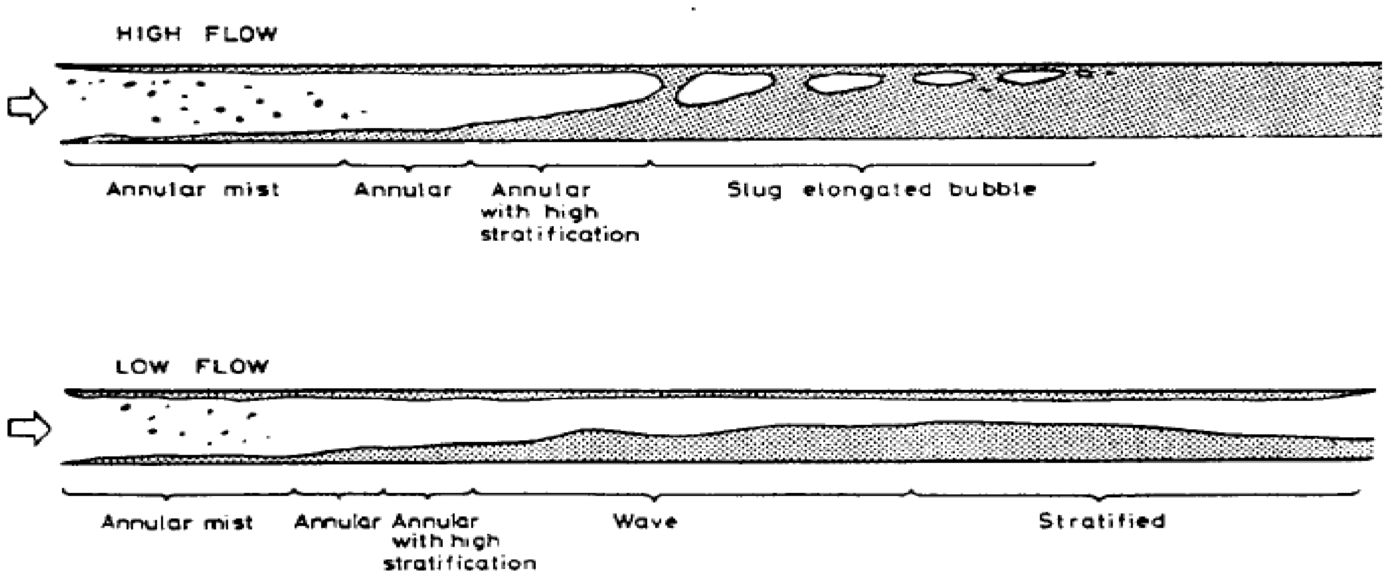
HTFS- Condensation on Vertical Surfaces Including Vertical Tubes

- **Flooding phenomena**: In vertical in-tube condensation with upward vapor flow phenomenon occurs when the condensate is prevented from draining from the bottom of the surface by the upward flow of vapor. This phenomenon is occurred with large waves and disturbances at the bottom of tube with the liquid being held up periodically by the inflow of the vapor. The pressure drop over the condenser tube usually rises sharply at flooding and heat transfer will be affected noticeably.
- It is normal to operate condensers with the vapor velocity well below the flooding velocity. n Design mode TASC uses the flooding velocity to determine the minimum number of tubes. Since flooding velocities tend to be low, especially at low pressures, TASC will often design a shell with a relatively large diameter and short tubes.
- In practice the flooding velocity increases if the tube ends are cut at an oblique angle so that the opening faces the vapor inlet nozzle.
- Another consequence of the low flooding velocity is that the gas-phase resistance to heat transfer may be high if there is a long condensing range.



HTFS- Condensation Inside Horizontal Tubes

- Following figure illustrates the flow patterns that typically occur during tube side condensation. The range of flow patterns observed depends on the total flow in the tube.
- It is obviously evident that the annular flow pattern has greater heat transfer coefficients than the stratified flow pattern.



HTFS- De-superheating and Sub-cooling

- **<u>De-superheating</u>:** If inlet vapor temperature is greater than the dew point, TASC will consider the de-superheating heat transfer in one of two ways.

1. Wet wall de-superheating: Occurs when the bulk temperature of a stream is above the dew point, but the local wall temperature is below the dew point. If the wet wall calculation is selected, the TASC corrects the heat transfer rate in the de-superheating zone to allow for condensation occurring at the wall.
2. Dry wall de-superheating: Occurs when both the bulk temperature of stream and the local wall temperature are above the dew point. In such case the program uses the single phase gas coefficient until the bulk vapor temperature reaches the dew point. <u>Usually dry wall coefficients are lower than wet wall coefficients, and hence more conservative.</u>

- **<u>Sub-cooling</u>:** The cooling of the condensate below the bubble point. Sometimes it is desirable to avoid re-flash of the fluid in the piping. It is, however, better to provide a separate heat exchanger for any required sub-cooling.
- Normally the outlet nozzle of exchanger is sized for free drainage; However, If the amount of sub-cooling is greater than 14°C, it is recommended to design the exiting nozzle/piping to maintain a liquid level in the exchanger.
- You may specify whether or not you want sub-cooling effect be considered in heat transfer and pressure loss calculation. Allowing for it will lead to higher coefficients of heat transfer as well the frictional pressure losses.

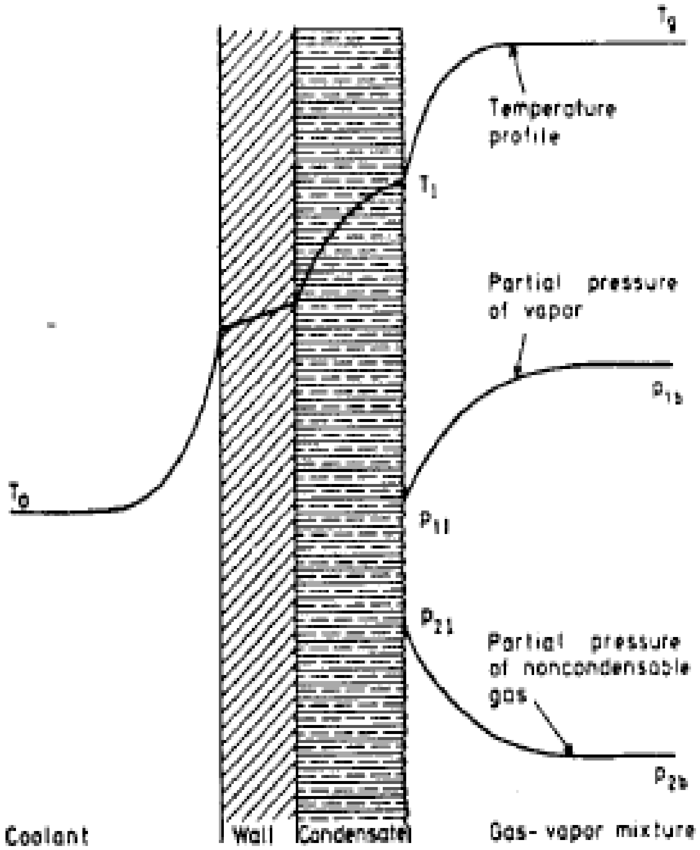
HTFS- Condensation of Vapor Mixtures

- Mixture condensation is differs from pure vapor condensation in two ways: **(a)** The temperature at which condensation occurs changes throughout the condenser. **(b)** Mass transfer effects are introduced in addition to the heat transfer ones.

- There are two possible scenario for vapor mixture condensation:

1. Condensation of single vapor in presence of non-condensable gas

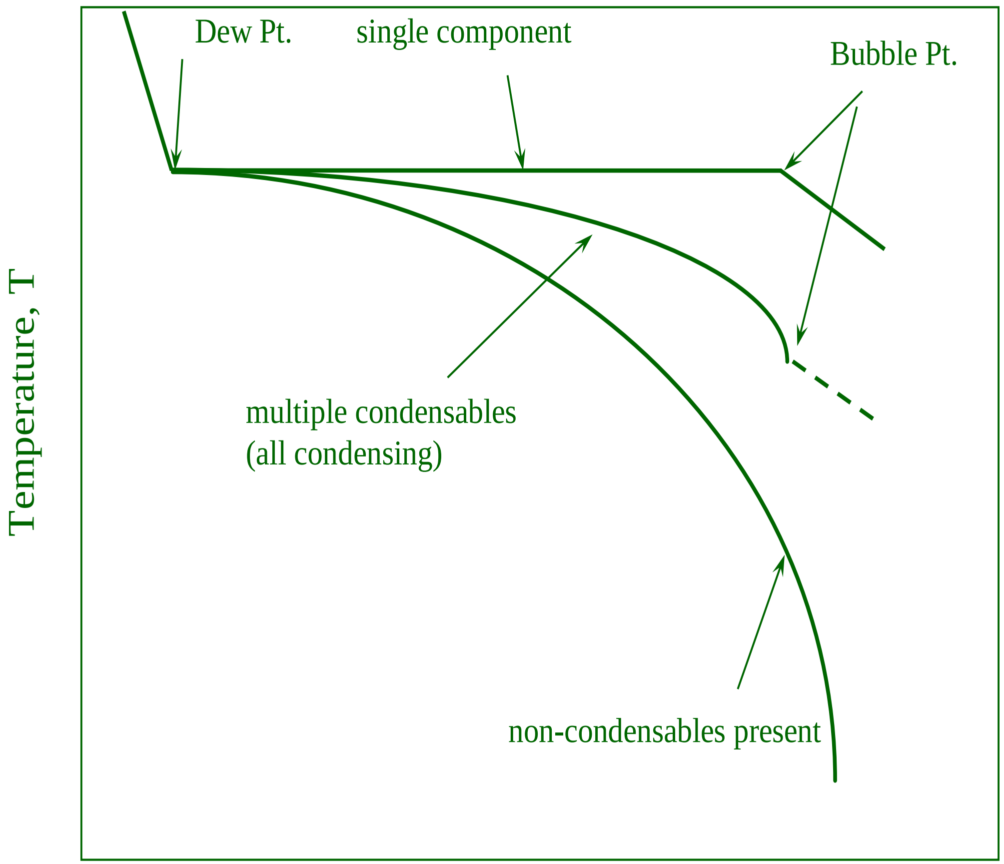
- The partial pressure of vapor in the bulk gas-vapor mixture is P_{1b} and the dew point temperature corresponding to this is T_g. The interface temperature T_I is less than T_g because the vapor concentration and corresponding dew point are lower there. This is because the non-condensable gas is swept toward the interface by the vapor flux, only to be left there as the vapor condenses. An equilibrium is set up with the non-condensable gas held stationary as it tries to diffuse away from the interface against the vapor flow. In turn, the vapor has to diffuse through the non-condensable gas under the influence of its partial pressure gradient. <u>The practical significance of this in condenser design is that the temperature difference between the condensing side and the coolant is reduced with a corresponding reduction in the local heat transfer rate.</u>



HTFS- Condensation of Vapor Mixtures

2. Condensation of multi component condensable vapors

- The condensation occurs over a wide temperature range due to the presence of more than one component in the vapor stream.
- Equilibrium condensation curves exhibit a drop in equilibrium condensing temperature with mixture enthalpy because the dew point of the remaining vapor falls as the less volatile vapors are condensed. Furthermore, the more volatile components tend to accumulate at the interface, thus giving a layer through which the less volatile components must diffuse.
- The typical equilibrium condensation curve is presented for comparison of three possible condensation scenarios.



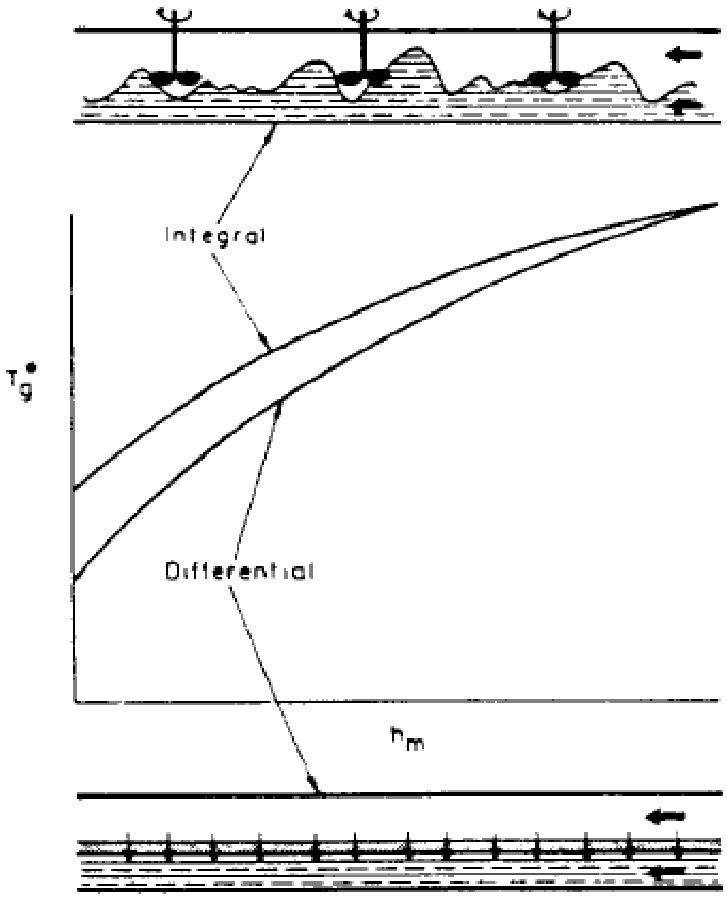
HTFS- Classification of Condensation Curves

- Equilibrium condensation curves may be classified into two main types:

1. **<u>Integral:</u>** In calculating the integral curve, <u>it is assumed that condensate and vapor keep together as they flow the condenser and that they are intimately mixed.</u>

2. **<u>Differential:</u>** In the differential type, it is assumed that the condensate, once formed, is separated from the vapor although it continues to flow parallel to it and, it is assumed, brought to the same temperature.

- There is no clear evidence as to which type of curve, integral or differential, should be used in a given circumstance. <u>When condensing inside tubes, it might be expected that, because the liquid and vapor are kept close together, the integral curve applies. When condensing in a horizontal shell, however, the liquid may separate from the vapor and the differential curve may seem more reasonable.</u>



HTFS- VLE Calculations

- The VLE calculations first determine bubble and dew points of the stream. These define the temperature range over which the stream is two phase. A set of temperatures is then selected at which 'flash' calculations are performed. A flash calculation determines the amount of each phase which is present (and hence the stream quality or vapor mass fraction) and also the composition of each phase. These compositions are in general different from the overall composition of the stream which you have specified. Given the composition of each phase, mixture calculations can be performed to determine the stream properties. The methods of performing VLE calculations presented in below:
- **Ideal Gas Law -** For ideal solutions
- **Equation of State Methods**
  - **Peng-Robinson** and **Soave-Redlich-Kwong** methods - For weakly polar components (such as hydrocarbons, nitrogen, CO2, CO)
  - **Chao-Seader** - For petroleum fractions (at pressures less than 1000 psia and temperatures greater than 0 F)
- **Interactive Parameter Methods** - User must provide interactive parameters from 3rd party source
  - **Van Laar** - Supports combination of polar and non-polar compounds with positive deviations form Raoult's law.
  - **Wilson** - For strongly non-ideal mixtures; alcohols & hydrocarbons
  - **NRTL** - Good for immiscibles
  - **Uniquac** - Good for small & large molecules; including polymers, parameters are less temperature dependent, and immiscibles

HTFS- VLE Calculations

- It is not possible to give a comprehensive guide to what methods are best for what substances, but if you define a Stream Type, before setting up a Data Source, then a default method is set up for you, which should in general be sensible for streams of the type you have chosen.
- Specialist physical properties packages, such as those in Process Simulators, in general contain a wider range of VLE methods, tailored to particular components. If you have access to data from such packages, it is usually better to input it directly, rather than use the simple VLE methods provided in the HTFS Physical Properties Package. A facility is provided for importing data from a file generated by a Process Simulator.

HTFS- Condenser Operational Problems / Design Recommendations

- Wide range of operating condition which is always accompanied with variations in vapor flow, coolant flow, and temperature variations results in pressure changes and changes in effective condensing area and create problems. Overdesign and less fouling will also result in excess areas and add to the control problem.

- **Pressure Control:** For partial condensation, pressure control is obtained simply by a valve on the discharge line. For total condensation, pressure control is more difficult. **a)** For total condensation at atmospheric pressure, the discharge end is simply vented to the atmosphere and, for vacuum operation, control is maintained by bleeding air into the vacuum system. For pressure control, it may be necessary to bleed in a supply of non-condensable gas at a load reduction and to vent it at a load increase. **b)** The condensing area can be changed by flooding with condensate, but this works poorly because of the slow response. **c)** Changing the condensing rate by varying the coolant flow and the temperature drop is possible, but is a poor control method because of slow response.

- **Limited Vapor Load:** In shell side horizontal condensers under partial load, the conditions in 'dead' zones may cause corrosion problems as the composition of vapors and its dew point may be substantially different than in the main vapor stream.

- **Sub-cooling:** Sub-cooling in shell side horizontal condensers is difficult and results in cold tubes in the condensate pool and hot tubes in the vapor space. The resulting tube stresses have been known to wrap shells.

HTFS- Condenser Operational Problems / Design Recommendations

- **Flooding:** Neglecting the potential for flooding in reflux condensers may cause severe operational problems such as unsteady condenser operation, fluctuation in operating pressure and uneven condensate drainage. On the other hand designing the condenser for low flooding velocity may increase the gas-phase resistance to heat transfer if there is long condensing range.

- **Venting:** The accumulation of a small percent of non-condensable gases in condensers can significantly reduce the condensing coefficients. Venting for in-tube condensation is simple, as the flow path is fixed; however, for shell-side condensation, the gases can segregate in pockets and are difficult to remove unless sufficient pressure drop (vapor velocity) is used to force these gases to the vent outlet. The venting problem is most severe in the typical crossflow condenser with either horizontal or vertical tubes.

HTFS- Design Recommendations

- Use "wet wall" calculations when the tube wall temperature is below the saturation temperature of the condensing vapor. Use a "dry wall" calculation when the de-superheating load is greater than 25% of the total heat load.
- Use a pressure dependent VLE curves when condenser is operating under a vacuum.
- Use vertical cut baffles with horizontal shell side condensers to aid in the condensate draining freely from the heat exchanger.
- For horizontal tubeside condensation use small inclination of about 1-5 degrees to assist in the condensate drainage.
- TEMA J or X type shells should be considered as an option for low vacuum designs.
- The vent(s) should normally be placed near the exit, and precautions should be made so that no condensate exits the vent lines.
- When condensing multi-component mixtures having a substantial boiling or dew-point range or when there are soluble gases present, it is necessary to control the condensate and vapor flow so as to enable the low boilers to condense or when stripping to prevent their condensation or absorption. The best control is with tubeside condensation, since with shellside condensation drops away from the cooling surfaces and the vapor.

HTFS- Design Recommendations – Vertical Tubeside

- **Advantages**

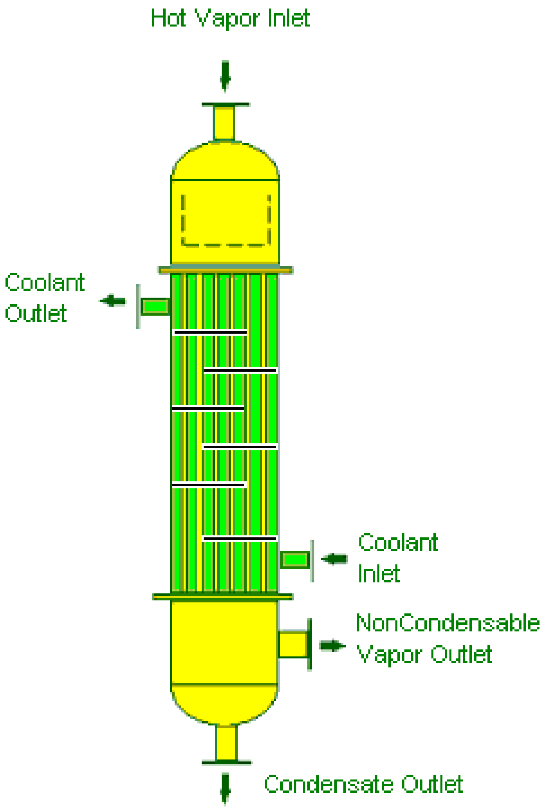
1. Good for high pressure, temperature
2. The condensate washes all the surfaces, which is an advantage under certain corrosive situation.
3. Low pressure drop, gravity aided condensate film downflow
4. Integral condensation
5. Good control of venting, good for sub-cooling
6. Well understood with accurate predictive methods
7. Handles dirty or polymerizing vapors.

- **Disadvantages**

1. Vertical construction & shellside difficult to clean
2. Tubesheet vents required.
3. Poor for pressures below 25 mmHg absolute.

- **Notes:**

1. For axial inlet nozzles , the nozzle entrance velocity head pressure should be compared to the condenser tube pressure drop to ensure that misdistribution is not severe. If needed, a perforated impingement plate with 5-10% hole area placed 0.5-1 nozzle diameters downstream will help.



HTFS- Design Recommendations – Vertical Shellside

- **Advantages**

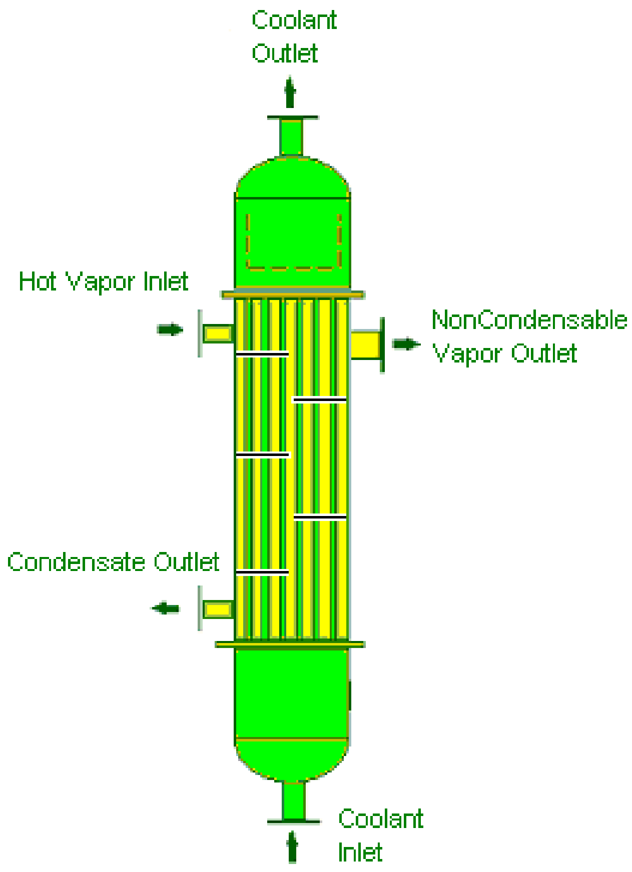
1. Good performance in gravity flow (baffles break-up stream)
2. Good for boiling tubeside coolant (1 pass upflow)
3. Good condensate sub-cooling
4. Multi-passing and variable baffle spacing can be used
5. Can handle freezing condensate

Disadvantages

1. Surface is flooded below condensate outlet
2. Tubeside cleaning is difficult
3. Difficult to vent inerts
4. Tube vibration problems

- **Notes**

1. Some designers advocate that three notches be made in the support plate tube holes to permit the condensate to drain throuh the plate.
2. An upflow vapor is rarely used and, if used, should not have any baffles or support plates.



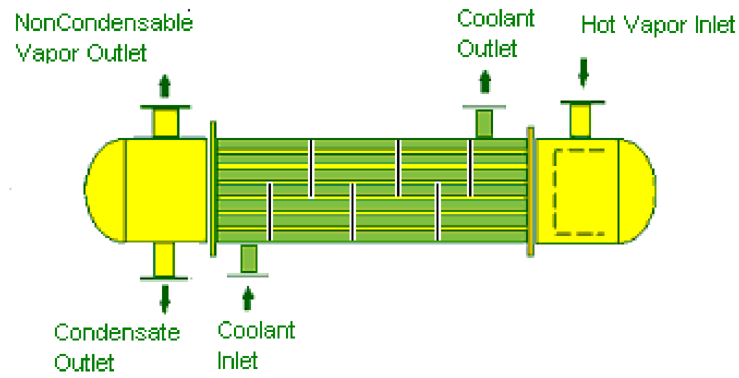
HTFS- Design Recommendations – Horizontal Tubeside

- **Advantages**

1. Good for high temperatures, pressures and corrosive vapors
2. Venting of inerts relatively effective

- **Disadvantages**

1. Possible flow instability
2. Shellside cleaning difficult
3. Should be inclined for drainage
4. Avoid multiple tube passes
5. Poor for pressures below 25 mmHg absolute
6. Possible slugging of condensate



- **Notes**

1. With straight tube multipass arrangement the succe passes should be below one another. These condenser single pass, but two-pass and U tubes are also common. than two passes is unusal.

HTFS- Design Recommendations – Horizontal Shellside

- **Advantages**

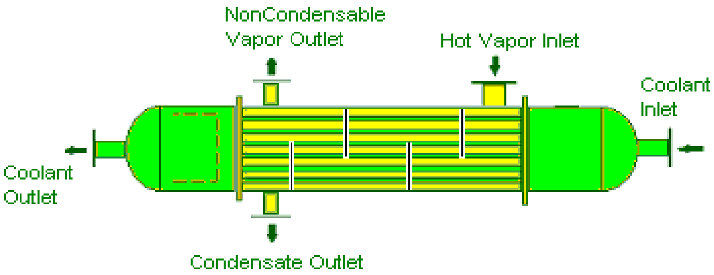
1. Tubeside easy to clean
2. Variety of baffles available to reduce pressure drop
3. High turbulence and high heat transfer possible
4. Low fin tubes may be applicable
5. Multi-passing and variable baffle spacing can be used
6. Can handle freezing condensate

- **Disadvantages**

1. Potential for high pressure drop (delta T penalty), potential for d condensation, potential for inert gas accumulation
2. Sub-cooling difficult to predict

- **Notes**

1. The lower edges of baffles shall be notched to permit drainage.
2. The baffle spacing may be made variable if maintaining a high vapor velocity is desired, such as when noncondensables are present.
3. Baffle cuts smaller than 35% of shell diameter are suggested to avoid baffle tip shortcutting of the flow.
4. The low fin tube may be used when the surface tension of the condensate is lower than 40 dyn/cm.



HTFS- Design Recommendations – Horizontal Shellside J Type

- **Advantages**

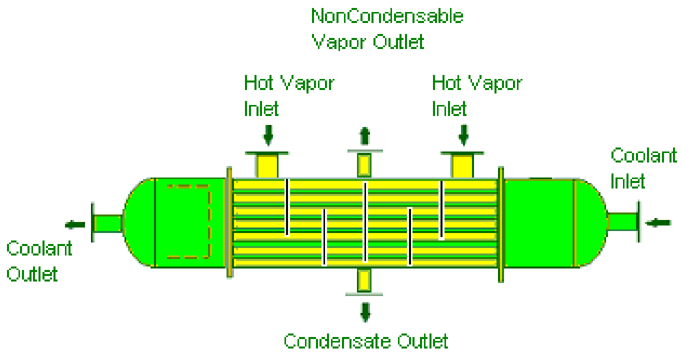
1. Very low pressure drop
2. Low fin tubes applicable
3. Tubeside is easy to clean

- **Disadvantages**

1. Counterflow / temperature cross not possible
2. Sub-cooling difficult to predict

- **Notes**

1. 2 inlets & 1 outlet: large manifold to divide flow but low vapo velocity and smaller inlet nozzles.
2. 1 inlet & 2 outlets: smaller manifold for outlet flows but high entry velocity and larger inlet nozzle.



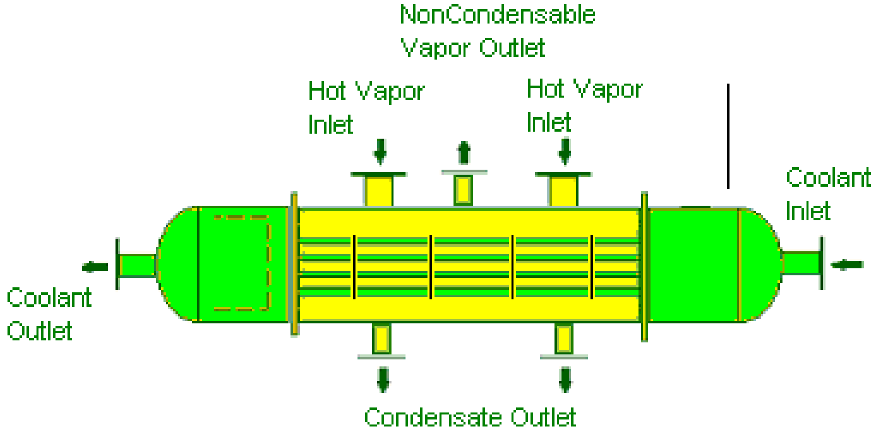
HTFS- Design Recommendations – Horizontal Shellside X Type

- **Advantages**

1. Low pressure drop
2. Low fin tubes applicable
3. Tubeside easy to clean
4. Supports prevent vibration

- **Disadvantages**

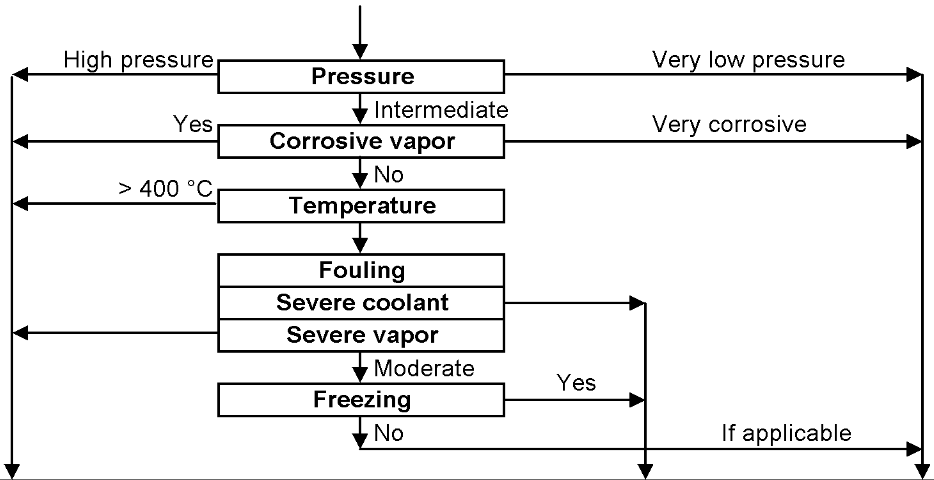
1. Additional piping or vapor distributor usually needed (expensive)
2. Low vapor velocity makes it difficult to vent inerts
3. Not good for wide condensing ranges
4. Not good for temperature crosses or meets
5. Sub-cooling is difficult to predict



HTFS- Preliminary Condenser Selection – Total Condensation

Legends:

1) Acceptablity
   - G: Good
   - P: Poor
   - X: NA
2) Predictability
   - ○ : Average~25%
   - ◍ : Fair < 50%
   - ● : Poor 50+%
   - ⊗ : No method or not recommended

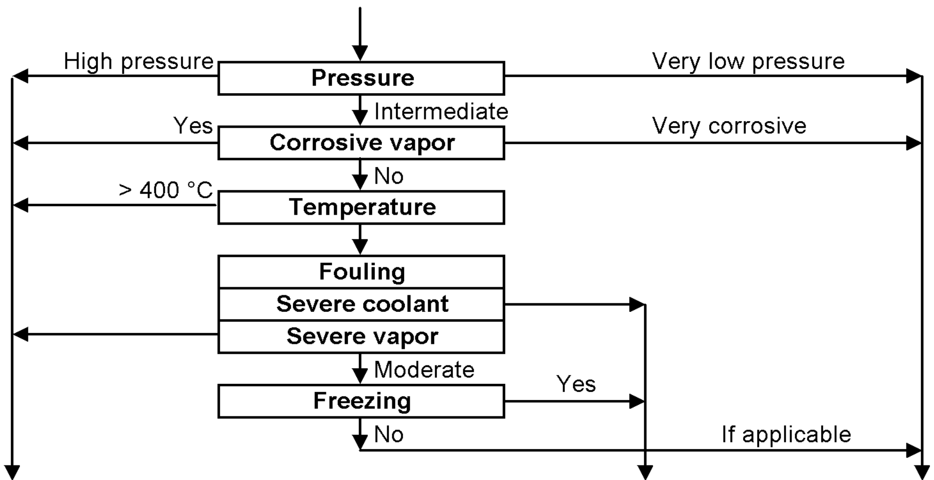


	Tube-side condensation			Shell-side condensation				Direct contact condensation
	Horizontal	Vertical		Horizontal		Vertical		
		Downflow	Upflow	Cross "X"	Baffled	Downflow	Upflow	
Single-component vapor	G ○	G ○	F ◍	G ◍	G ◍	G ◍	G ●	G ◍
Multicomponent vapor	F ◍	G ◍	F ●	G ◍	G ◍	G ●	F ●	P ●
Subcooled condensate	P ⊗	G ○	X ⊗	F ●	P ●	F ◍	X ⊗	X ⊗
Pressure drop								
High	G ◍	G ●	X ●	G ◍	G ◍	G ●	X ●	X ⊗
Low	P	F	G	G	F	G	F	G
Coolant								
Liquid	G	G	G	G	G	G	G	G
Gas	G	G	G	G	G	G	G	X
Boiling	G	G	G	X	X	G	G	X

HTFS- Preliminary Condenser Selection – Partial Condensation

Legends:

1) Acceptablity
   - G: Good
   - P: Poor
   - X: NA
2) Predictability
   - ○ : Average~25%
   - ◍ : Fair < 50%
   - ● : Poor 50+%
   - ⊗ : No method or not recommended



	Tube-side condensation			Shell-side condensation				Direct contact condensation
	Horizontal	Vertical		Horizontal		Vertical		
		Downflow	Upflow	Cross "X"	Baffled	Downflow	Upflow	
Single-component vapor	G ○	G ○	X ⊗	P ◍	P ◍	F ◍	P ◍	X ⊗
Multicomponent vapor	P ◍	G ◍	X ⊗	F ◍	P ●	P ◍	X ⊗	F ●
Noncondensable gas	G ○	G ○	X ⊗	G ◍	G ◍	F ○	X ⊗	F ◍
Pressure drop								
High	G ◍	G ●	P ●	G ◍	G ◍	G ●	X ●	X ⊗
Low	P	G	F	G	P	G	F	G
Coolant								
Liquid	G	G	G	G	G	G	G	G
Gas	G	G	G	G	G	G	G	X
Boiling	G	G	G	X	X	G	G	X

HTFS- Design Example - Definition

Process Design Parameters					
	UNITS	In	Out	In	Out
Fluid Name		Methanol Loop Gas (R2101 Product)		Methanol Loop Gas (Recycle + MUG)	
Total Fluid Quantity	kg/h	733316		733316	
Liquid (Total)	kg/h	n/a	32895	n/a	n/a
Vapor (Total)	kg/h	733316	700421	733316	733316
Water	kg/h	n/a	n/a	n/a	n/a
Temperature	°C	185.4	91.1	61.5	167.7
Molecular Weight (V)		Use properties qouted in Note 3		9.42	9.42
Compressibility (V)				1.03	1.03
Viscocity (V)	cP			0.016	0.019
Specific Heat (V)	kJ/kg.K			3.37	3.43
Thermal Conductivity (V)	W/m.K			0.121	0.15
Density (L)	kg/m3			n/a	n/a
Viscosity (L)	cP			n/a	n/a
Specific Heat (L)	kJ/kg.K			n/a	n/a
Thermal Conductivity (L)	W/m.K			n/a	n/a
Surface Tension	mN/m			n/a	n/a
Inlet pressure / Allowable DP	bara / bar	68.9 / 0.6		72.4 / 0.4	
Fouling Resistance	m2.°C/W	0.000172		0.000172	
Heat Exchanged	MW	74.1		73.4	
Mechanical Design Parameters					
Design Pressure	barg	87		87	
Design Temperature	°C	275		275	
Corrosion Allowance	mm	Nil		3	
Material		SS 304L		P11	

Vibration – How to interpret and solve potential vibration indications

HTFS- Objective for The Session

Our discussion will include:

- Vibration damage patterns
- Failure regions
- Natural frequency and vibration damping mechanisms
- Excitation mechanisms
- Avoiding vibration at the design stage
- Avoiding vibration during manufacture

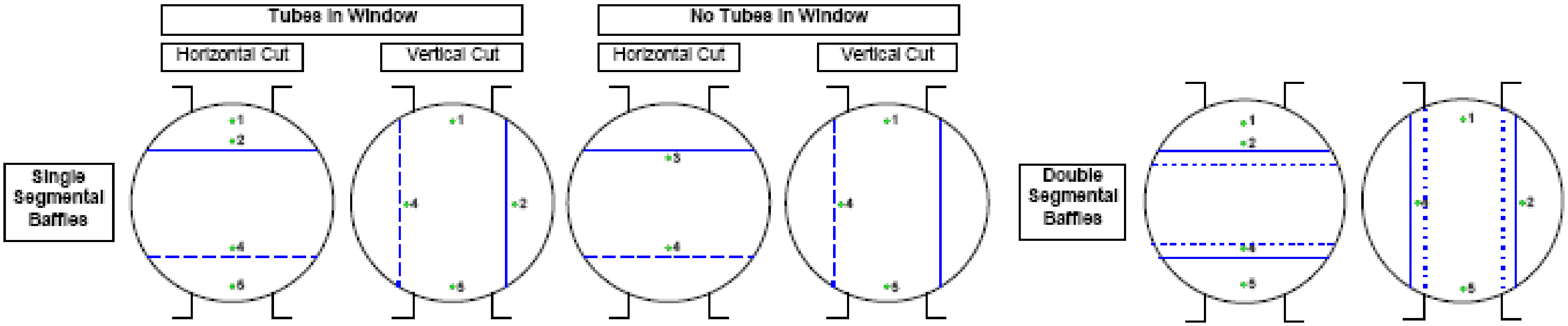
HTFS- Vibration Damage Patterns

- Fatigue due to repeated bending
- Repeated impact between adjacent tubes at mid-span: flat spots occur leading to thinning of the tube walls with eventual splitting
- Cutting at the baffles, particularly if the baffles are thin or harder than the tubes
- Cutting at the tube-hole edges at the inner tubesheet face due to repeated impact between tube and tubesheet
- Loosening of roller expanded or failure of welded tube-tubesheet joints
- Tube material defect propagation and failure

HTFS- Failure Regions

- Tube failures have been reported in nearly all locations within a heat exchanger. Locations of relatively flexible tube spans and / or high flow velocities are regions of primary concern.

1. **U-bends:** Outer rows of U-bends have a lower natural frequency of vibration and, therefore, are more susceptible to flow induced vibration failures than the inner rows
2. **Nozzle entrance and exit area:** Impingement plates, large outer tube limits and small nozzle diameters can contribute to restricted entrance and exit areas. These restricted areas usually create high local velocities which can result in producing damaging flow induced vibration.
3. **Tubesheet region:** Unsupported tube spans adjacent to the tubesheet are frequently longer than those in the baffled region of the heat exchanger, and result in lower natural frequencies. Entrance and exit areas are common to this region. The possible high local velocities, in conjunction with the lower natural frequency, make this a region of primary concern in preventing damaging vibration.
4. **Baffle region:** Tubes located in baffle windows have unsupported spans equal to multiples of the baffle spacing. Long unsupported tube spans result in reduced natural frequency of vibration and have a grater tendency to vibrate.
5. **Obstruction:** Any obstruction to flow such as tie rods, sealing strips and impingement plates may cause high localized velocities which can initiate vibration in the immediate vicinity of the obstruction.
6. Experience has shown that there are certain tube rows within an exchanger that are most likely to suffer vibration damage. These are basically the uppermost and lowermost tube rows with respect to the flow and the tube rows just inside or just above the edge of the baffle cut. TASC considers following tube rows in vibration analysis:



HTFS- Natural Frequency and Vibration Damping Mechanisms

- Calculation of the natural frequency of the heat exchanger tube is an essential step in estimating its potential for flow induced vibration failure. The factors affecting natural frequency of individual unsupported span can be summarized as:

1. Tube elastic and inertial properties and tube geometry (tube OD and thickness)
2. Span shape (straight span or U-bend span)
3. Type of support at each end of the unsupported span
4. Axial loading on the tube unsupported span

- Damping effects are extremely complex and individual tubes can have very different characteristics. In the flow-induced vibration situation, if the energy input cannot be. dissipated through damping, the amplitude of vibration will increase with time leading to a .runaway. condition. If there is a balance, the amplitude remains constant. If the damping exceeds the energy input, the amplitude diminishes with time.

- The main mechanisms of damping are:

1. Fluid damping is caused by viscous and pressure drag as the tube moves relative to the fluid
2. Baffle support damping is caused by friction due to impacting and sliding of the tube at the baffle hole resulting from the inevitable clearance
3. Squeeze film damping is the dissipation of energy due to the periodic displacement of fluid from the tube/baffle gap as the tube vibrates
4. Material damping is the natural energy dissipation which occurs when the tube is flexed

- Three typical damping defined by logarithmic decrements is 0.1 and 0.03, 0.01. The highest value, 0.1, reflects heavy damping; the value of 0.03 serves as a medium value, while 0.01 would represent light damping. Typical log decrements for a single phase liquid stream on the shell side tend to be close to 0.1 (heavy). For gases, typical values are closer to 0.03. Two-phase fluid damping is more difficult to analyze, but may be assumed to lie between these ranges.

HTFS- Excitation Mechanisms

- There are four basic flow induced vibration mechanisms that can occur in a tube bundle. These are the **fluid elastic instability**, **vortex shedding**, **turbulent buffeting**, and **acoustic resonance**. The first three mechanisms are accompanied by a tube vibration amplitude while acoustic resonance causes a loud acoustic noise with virtually no increase in tube amplitude.

- **Fluid elastic instability:** Fluid elastic instability is the most damaging in that it results in extremely large amplitudes of vibration with ultimate damage pattern. This phenomena is evidenced by tubes vibrating in an orbital motion which is produced by flow across the tubes causing a combination of lift and drag displacements of the tubes at their natural frequencies. Typically, once fluid elastic whirling commences, it can lead to a runaway condition if the energy fed to the tubes exceeds that which can be dissipated by damping. The design approach in this case is to avoid fluid elastic vibration by means of following actions:

1. Decrease fluid velocity
2. Increase the tube natural frequency by decreasing the span length
3. Increase the damping by reducing the tube-to-baffle clearance or increasing the baffle thickness
4. Increase the tube pitch
5. Remove the tubes in the window region which have double length spans

HTFS- Excitation Mechanisms (continued)

- **Vortex shedding:** Vortex shedding is the phenomenon caused by the periodic shedding of vortices as the fluid flows over a non-streamlined object such as a heat exchanger tube. Wake oscillates with frequency proportional to flow velocity.

- Vortex shedding is a fluid-dynamic phenomenon and does not depend on the movement of the tubes for its existence. However, if the tubes vibrate at, or near, the vortex shedding frequency, the vortices are shed at the tube vibration frequency. This feature is termed a “lock-in” effect which will be happen if :

$$\mathbf{0.8 < f_v / f_n < 1.2}$$

- **Turbulent buffeting:** Turbulent buffeting relates to the fluctuating forces acting on the tubes under extremely turbulent cross flow conditions for fluids at high Reynolds numbers.