1. BASIC SCOPE OF DOCUMENT 3

2. BASIC EQUATIONS FOR HEAT TRANSFER 3

3. TYPE SELECTION MATRIX 3

3.1 SELECTION MATRIX: 5

4. MATERIAL SELECTION 6

5. DESIGN REQUIREMENTS 8

5.1 WHICH FLUID ON WHICH SIDE? 8
5.2 TUBE SIZING AND NUMBER OF TUBES 8
5.3 PITCH AND TUBE LAYOUT 9
5.4 TUBE TO TUBE SHEET JOINTS 9
5.4.1 STRENGTH WELDED WITH LIGHT EXPANSION 9
5.4.2 SEAL WELDED WITH HEAVY EXPANSION – 10
5.4.3 ROLLED-IN 10
5.4.4 MAXIMUM TUBE WALL THICKNESS REDUCTION 11
5.5 CO-CURRENT AND COUNTER-CURRENT FLOW, NUMBER OF PASSES 11
5.6 SPECIAL DESIGN REQUIREMENTS 12
5.6.1 BURSTING DISC 12
5.6.2 GAS ON SHELL SIDE 12
5.6.3 OXYGEN, O2 COOLER 12
5.6.4 PASS PARTITIONAL PLATE > 150BAR 12
5.6.5 OTHERS 14
5.6.6 TUBE IN TUBE HEAT EXCHANGERS 16

6. CALCULATION OF HEAT EXCHANGER 17

6.1 MINIMUM PROCESS PARAMETERS REQUIRED FOR THERMAL DESIGN 17
6.2 CHOOSING OF WORST CASE 17
6.3 OVERDESIGN 18

7. COOLER EVALUATION 18

7.1 BASIC EQUATION FOR REQUIRED POWER 18
7.2 BASIC EQUATION FOR HEAT TRANSFER 18
7.3 BASIC EQUATION FOR OVERALL HEAT TRANSFER COEFFICIENT 18

8. CODES AND REGULATIONS 19

8.1 TEMA 19
8.1.1 TEMA R VS. TEMA C: 19
8.2 GB CODE 20
8.3 API 660 (ISO 16812) [3] 20
8.3.1 TEMA VS. API 660: 21
8.4 ASME SECTION VII DIV1/2 PART UHX [4] 21

8.5 PRESSURE TEST 21
9. REFERENCES / USEFUL LINKS 22
9.1 USEFUL LINKS 22
10. REVISION INDEX 23

1. Basic scope of document

This document defines the basic principles of design for heat exchangers of the shell and tube type. It shall be a basic guide for filling in BCA standard specifications and help to deal with client specifications. It is not mandatory but shall be used as a guideline and deviations from this document may be required on a case-to-case basis.

2. Basic equations for heat transfer

For reference and better understanding of the principles of heat transfer, the below equations are given (definitions are as per TEMA):

$$Q[kW] = \dot{m}[kg/s] \cdot cp[kJ/kgK] \cdot \Delta T[K] \qquad Q = A_0[m^2] \cdot U\left[\frac{kW}{m^2 \cdot K}\right] \cdot \Delta t_m[K]$$

where: Q = heat exchanged; m = mass flow, c_p = specific heat capacity, ΔT= terminal temperature difference; A_0= effective heat transfer surface; U=heat transfer ratio; Δt_m = mean temperature difference.

3. Type selection matrix

Types are defined as per TEMA[1] (see 8). For BC's typical applications, almost exclusively horizontal coolers.

The front-end stationary head types are normally B type bonnets.
If regular cleaning on tube side is required –e.g. when coolant contains sand or other particles -, A type channels with removable covers shall be used.
D type may be used for high pressure (>200 bar).

Type front end	***Application / Gas***	***Limitations/Advantages/Special requirements.***
A Channel and removable cover	When coolant is seawater/ dirty water then, gas on shell side.	Bolted channel cover can be removed for tube side cleaning. Often used when water on tube side. With Gas on tube side limited in pressure.
B Bonnet (Integral cover)	Most commonly in applications	Preferred for clean tube side fluids. Cheaper and easy installation.
C Channel Integral with tube sheet and removable cover	Not commonly used. Can be used for hazardous tube side fluid.	Tube sheet end welded to channel. Tube side fluid Leakage is low. Good where frequent shell side cleaning required.
N Channel integral with tube sheet and removable cover	Good for hazardous fluids on shell side.	When the pressure is high, and D type is not available. NEN type for hydrogen can be a good option here. We have reference where pressure can go up to 235 bar.
D Special High-Pressure closure	High Pressure gas applications	Bonnet directly welded to tube sheet.

Mostly ***single flow E type shells are used***. F, G and H shells with longitudinal baffles are only suitable when cooling water is on shell side. They are not suitable for gas on shell side with high gas forces of reciprocating compressors.

Shell Type	***Application / Gas***	***Limitations/Advantages/Special requirements.***
E One pass Shell	Most commonly in applications	Small shell side velocity so long exchanger.
F Two Pass Shell with Longitudinal Baffle	High shell side flow rate, high shell side velocity so exchanger in small length.	Gives pure counter current flow with two (even) tube side pass. Chances of leakage across longitudinal baffle weld.
G Split flow	Use for low shell side delta P	Longitudinal baffle perforated and more like distribution plate.
X Cross Flow	Cross flow, very low shell side delta P.	Unbaffled shell.
Not used in BC, since used for other applications	K Kettle Type reboiler, H Double split flow, I/J Divided flow	

The execution of the ***rear end head design*** however may vary, most common for BC plants is BEW, BEM and BEU. Floating heads are required upon high temperature difference between tube and shell.

Type rear end	***Application / Gas***	***Limitations***
M – fixed tube sheet like B stationary head	HPI, LNG, LPG, CO with lower temperature differences	Temperature difference between tube and shell is limited (above 80°C). Thermal expansion of shell versus tube bundle may require expansion bellows
L- Rear head like “A” stationary head. N-Rear head like stationary head	Not used in BC application	
P- outside packed floating head	High pressure Air, O2 or N2 when temperature differences are large	Pressure compensation for higher design pressures. Not Allowed in API660.
S- floating head with backing device	Little experience	Often designed same as U tube but with better cleaning facilities. More Expensive than U-Tube, higher Maintenance cost. Not suitable for high pressure.
U - hairpin tubes	Most used high temperature application, alternative to BEM or BEW coolers	Economic, but cleaning restricted, overall arrangement more expensive, then W or M
T Rear head pull through floating head	Rarely used. Shell side cleaning when required frequently.	Larger bundle to shell clearance required.
W – externally sealed floating tube sheet	Low pressure Air, O2 or N2 when temperature differences are large	up to design pressure 20 bar as per TEMA B & 40 bar as per TEMA C. As per TEMA not foreseen for gas, therefore use it only for water and oil. Allowed only for 1 or 2 pass on tube side.

3.1 Selection Matrix:

Abbreviation:

- Cooling Water **A**: drinking quality and water-glycol mixtures (with Inhibitor)
- Cooling Water **B**: dirty (sand) / seawater
- Cooling Water **C**: with Chlorine
- Cooling Water **D**: with Ammonia

Gas	O2 See **Error! Reference source not found.**	N2	Air/CO2	CO	HPI LNG LPG	LDPE LNGM (2	H2	H2O	OIL	PE/PP	IOCL H2/N2
	nontoxic, non-flammable	Poisson	flammable	flammable	flammable	n.a.	n.a.				
	dry or wet	dry	Corrosive wet	dry	dry	dry					
Gas flow:											
- Tube Side	**A/C/D**	**A/C/D**	**A/C/D**	**--**	**A/C/D**	**A/B/C/D**	**A/C/D**	***CW:*** **A/B/C/D**	***CW:*** **A/B/C/D**	**A/B/C/D**	**A/C/D**
- Shell Side	**B**	**B**	**B**	**A/B/C/D**	**A/B/C/D**	**--**	**B**	***Glycol-H20:*** **A**	***Oil:*** **--**	***A***	**B**
Type:								(1			
AEL				**B**	**B**			**B**			
AES			**B**	(B)	(B)		(B)				**(3B**
AEM											**B**
AEU	**>50bar B**	**B**		(B)	(B)		**B**				
AEW	<30bar (B)	(B)				<30bar (B)		(A/B/C/D)	**A/B/C/D**		
BEM				**A/C/D**	**A/C/D**		(A/C/D)	**A/C/D**		**A/B/C/D**	**A/C/D**
BEP (1pass)	**<50bar A/C/D**	<50bar (A/C/D)	**<50bar A/C/D**			**1.stage A/C/D**					
BES			(B)	(A/C/D)	(A/C/D)						
BEU	**>50bar A/C/D**	**A/C/D**	**>50bar A/C/D**	(A/C/D)	(A/C/D)	**A/C/D**	**A/C/D**			(A)	
BEW	<30bar (A/C/D)	<30bar (A/C/D)	<30bar (A/C/D)						(A/C/D)		
BGU						(A/C/D)					

DEU/ DGU						**>200bar A/B/C/D**	**>200bar** A/B/C/D				
NEN							**A/C/D**				

(.) feasible, but not BCA standard

([1] : plate type HE shall be used, if accepted by client

([2] : LNGM Guidance Matrix TDO 2050001

([3] only for low pressure applications. For exact pressure range please contact suppliers.

4. Material selection

Gas and coolant media handled always define the material of construction along with pressure and temperature design criteria. A cooling water analysis shall be provided for that reason.
If no reliable coolant analysis is available, CuNi10Fe could be used as it provides the best overall durability.
When selecting materials, further consider the following:

The heat transfer rate of pure copper (389 W/mK) is the highest, followed by Admiralty or Naval Brass (130 W/mK) for carbon steel it is around 50 W/mK, that of stainless steel the lowest (13…16 W/mK, depending on grade). This will influence the overall heat transfer rate of the cooler and hence the area required and finally the price.

Check the pH-value of the coolant: values of 6 and lower mean acid service, which requires special consideration. pH-values of 7 to 8.5 can be considered as neutral and considered uncritical. For alkaline service with pH values of 9 and above, do not use copper or copper alloys

Baffle material shall be softer than tube material to avoid wear due to vibration.

Beware of electrolytic corrosion, e.g. do not use copper alloy or austenitic stainless-steel tubes with carbon steel tube sheets, aluminium with copper, etc. In this behalf using a BEM type identical material shall be used.

For tube and tube sheet in oxygen service brass, stainless steel or copper/copper alloys are required due to the high velocities. Special attention shall be given to head design to reduce change of flow direction. Also, for oxygen service U-Tube design Monel partition plate to be used with rolled in design, please refer TDO 2023990 E02 page 68.

Wet and corrosive gases e.g. CO2 require stainless steel material.

High content of chlorides (above 50 mg/l) e.g. in sea water, in cooling water may necessitate special stainless steels or the use of naval brass.

Carbon and low alloy materials can be used for cooling water side if water quality parameters are within a certain range. Refer to document **VBCA200050 "Cooling Water Quality Requirements** for the design range for different water quality parameters.

Gasket type and material should be selected based upon client requirement and process gas application. The gasket made of SS 316 plate completely covered with graphite can be used as the

SS metal does not come in contact with sea water. However, SS 316 spiral wound gasket with SS 316 inner ring cannot be used for sea water as the inner ring is exposed to sea water. If spiral wound gasket is required to be used for sea water, then Alloy 625 or super duplex material should be used.

5. Design requirements

5.1 Which fluid on which side?

Process gas on tube side is generally preferred as it provides easier arrangements.

Reasons for putting a fluid on tube side:

- High pressure: saves cost for shell
- High pulsation: reduces the risk of tube and baffle vibration
- High fouling resistance (gas or coolant) or powder loaded gases: Easier cleaning, reduced fouling due to higher velocities

For CO, always put the gas on the shell side since the risk of leakage is lower.

When a bellow is required on the shell and the gas is likely to contain liquids, the gas shall not be put on shell side due to the requirement of providing a drain at the low point which is difficult to be achieved on the bellow.

5.2 Tube sizing and number of tubes

Tube size and wall thickness is defined as per TEMA, however commercial sizes especially in Europe may differ from that. Typical tube sizes used are 12-14-16-19 to 22 mm approx. Tube wall thickness depends on tube stress (pressure, bundle vibration due to flow forces).

Increasing the tube size will increase the shell diameter and largely influence the price of the cooler. The tube diameter shall hence be selected as small as practically possible. This will depend on the flow and is limited by the allowable pressure loss, available space, cleaning requirements and often codes and specifications.

The smaller the tube the larger the velocity. The larger the velocity, the larger are heat transfer coefficients and the larger is the pressure loss.

Hydrocarbon gases in Petrochemical plants with powder/particle contamination may require larger tube sizes.

Long coolers are always more cost effective, as less machining and smaller shells are required.

The ***bundle length*** is limited by the available space and the admissible pressure loss.

The ***number of tubes*** is defined by the surface area required and the maximum possible bundle length.

Whereas smooth tubes are a standard, finned tubes are sometimes used when large surface area is required within limited space.

The tube thickness mentioned in datasheet is average thickness as default value. But at design stage it mandatory to check the requirement from client whether is minimum/average thickness required. As it may leads to confusion and additional costs in later stages.

5.3 Pitch and tube layout

Pitch – i.e. the distance between tube centrelines – is a decisive factor for heat transfer on the shell side. As per TEMA, pitch is minimum 1.25 times the tube outer diameter. For shell side, pitch can be compared to tube size on tube side: The smaller the pitch the larger the velocity. The larger the velocity, the larger are heat transfer coefficients and pressure loss.

Tube layouts are defined as per TEMA: triangular and square. Both can be rotated. Standard tube layout is triangular as it saves space and provides compact coolers with good heat transfer. Square layout shall only be used if mechanical cleaning on shell side is required by the client. In this case, the bundle must of course be removable.
Rotated layouts are preferable for laminar flow, whereas for turbulent flow, and especially where the pressure drop is limited, regular arrangements should be used.

Bundle Layout – The terms **"full bundle"** and **"normal bundle"** in heat exchangers refer to the extent to which the tube bundle fills the shell.

Normal Bundle When the tubes under the nozzle are removed it is called Normal bundle. The tube bundle is slightly smaller in diameter than the shell, leaving a bundle-to-shell clearance (annular space) around the bundle. This allows easier removal and installation of the bundle and provides space for shell-side cleaning and fluid bypasses. So can be used for dirty shell side fluid where cleaning and regular maintenance is required frequently.

Full bundle When tubes under the nozzle are not removed. The tube bundle occupies almost the entire cross-sectional area of the shell, with minimal clearance between the outermost tubes and the shell wall. It maximizes heat transfer area for a given shell size so used when maximum heat duty is required, and shell-side fouling is not a major concern. So can be used for clean shell side fluid where no frequent cleaning or maintenance is required.

5.4 Tube to tube sheet joints

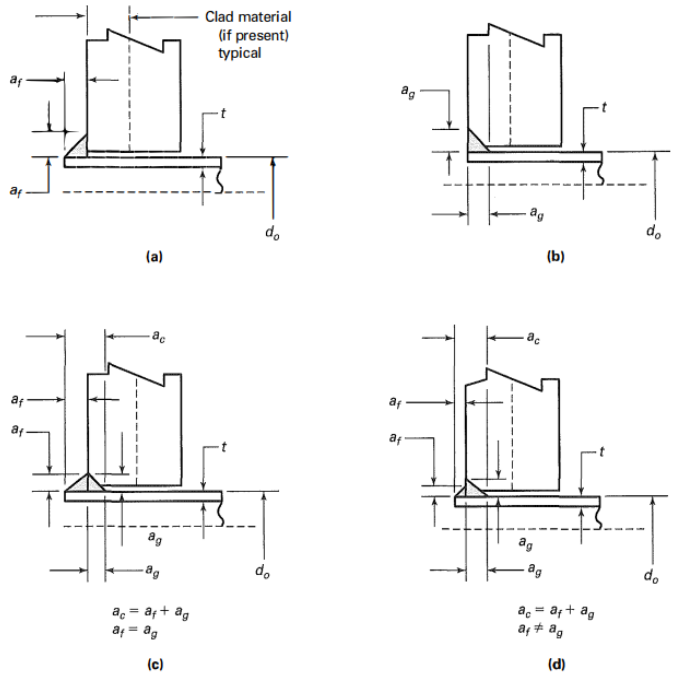
There are principally three ways of connecting the tubes to the tube sheet. The selection also largely depends on the application. For oxygen applications, normally type described in 5.4.2 is used. For LDPE application, most often type described in 5.4.1 is used.

5.4.1 Strength welded with light Expansion.

Strength welded and hydraulically or explosive expansion, (of tube thickness). Standard joint, used for most gas coolers in various application. As per TEMA for strength welded tube to tube sheet joints, ring grooves are not required. Rev01

Some acceptable types of tube to tubesheet strength welds are shown as below:

Figure UW-20.1
Some Acceptable Types of Tube-to-Tubesheet Strength Welds



There is no problem to use either fillet only or groove plus fillet weld provided that the strength requirements of the joint are fulfilled. The groove + fillet might be preferred because it's normally a two-weld run process - root weld run + cap weld run, where the start/stop positions of the two welds are staggered 180 degrees if possible.

After the first run - the tube to tubesheet joint shall normally be examined by the Dye Penetrant Method + a soapy water bubble test for leaks. If you have a lot of tubes - dye pen on the first run might be a problem because it needs cleaning before the cap weld run is made. Codes permit any of the two methods unless you have process reasons to prefer one or the other methods.

5.4.2 Seal welded with heavy Expansion –

Rarely used. Due to high expansion and grooves is this joint complex and most not fabricated in good quality. But this joint can be used for additional tightness required by costumer.

For hydraulic or explosive expanded tube joints, when tubesheet thickness exceeds 76mm, at least two grooves shall be used. Tubesheet with thickness less than or equal to 76mm may be provided with one groove. (See also TEMA R vs. C **Error! Reference source not found.**).

5.4.3 Rolled-in

Rolled in tube to tubesheet joints e.g. with a rotating wedge. Similar requirements as for expanded joints. Used for soft material such as Tube sheet Brass, Nickel alloy in combination with tube in Admiralty brass or CuNi10Fe. Which is the case in most oil cooler.

5.4.4 Maximum Tube Wall Thickness Reduction

Maximum tube wall thickness reduction is based on API660 Section 9.10. Reduction is given in % of tube wall thickness before expansion. Rev01

Material	
Carbon Steel	8%
Stainless Steel	5%
Alloy Steel	6%
Titanium	5%
Copper alloy	8%
Admiralty brass	8%
3003/4004 aluminium	5%

Calculation of tube wall thickness reduction is based on DIN EN 13445-4.

$$T = \left(\frac{d_{if} - d_{ii} - d + d_e}{d_e - d_{ii}}\right) * 100$$

$$T = \left(1 - \frac{d - d_{if}}{d_e - d_{ii}}\right) * 100$$

T: expansion factor / tube wall thickness reduction [%]

d: diameter of the hole inside the tube sheet

dif: internal diameter of the tube after expansion

de: external diameter of the tube before expansion

dii: internal diameter of the tube before expansion

5.5 Co-current and counter-current flow, number of passes

Co-current flow reduces the LMTD (logarithmic mean temperature difference) of the cooler, increasing its size. It is mostly used to reduce the terminal temperature differences and the risk of cold spots and consequent condensation.

Increase of the number of passes is normally done on the tube side, rarer on the shell side (F type). Increasing the number of passes reduces the mean temperature difference Δt_m by a certain correction factor ε. On the other hand, it will increase the fluid velocity and consequently both pressure loss and heat transfer coefficient. In many cases, the allowable pressure loss will limit the use of multi-pass exchangers. Depending on the fluid properties (gas or coolant), change of flow state (laminar or turbulent) etc., either the decrease of mean temperature difference or the increase of the heat transfer coefficient k will prevail. To make multi-pass exchangers economic, the latter must be true – this will allow for a reduction of either the surface area or the coolant consumption.

5.6 Special design requirements

5.6.1 Bursting disc

When the design pressure of the gas side is above the test pressure of the coolant side, a safety valve or bursting disk shall be provided that is to be sized for the tube failure case, i.e. rupture of one tube. Please check the Bursting Disc White book TDO 2051639.

5.6.2 Gas on Shell side

When the gas is on the shell side, consider placing the safety valve upstream of the cooler if there is a risk of the cooler getting stuck e.g. with powder or loosened baffles. Else baffles must have spacers to make axial displacement impossible.

5.6.3 Oxygen, O2 Cooler

For O2 Applications the cooler should be whenever possible 1 Pass, with an axial intel and outlet. If a 2 Pass Cooler must be chosen, the pass partitional plate must made with Monel and inlet and outlet as well axial.

Standard Material for BEW/BEP: Tube sheet in Brass or Nickel alloy and tubes in Admiralty brass or CuNi10Fe.

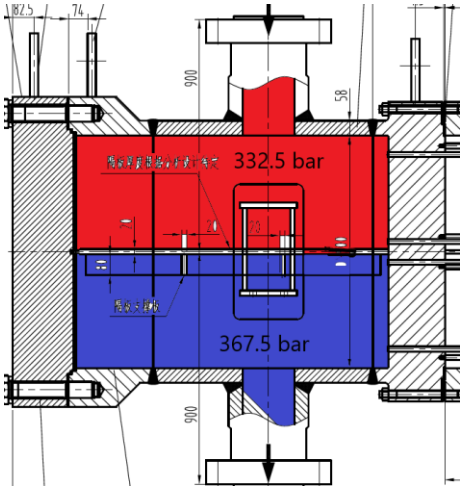
Above 45barg mostly stainless steel will be used.

Tube to tubesheet joint should be the same as other application no additional tightness is required.

5.6.4 Pass Partitional plate > 150bar

The pass partition plate is not only subjected to the forces from pressure loss from hot to cold side but also to the pulsations created by the pushing and pulling of the cylinders with a frequency of about 12 pressure pulsations per second. These pulsations have a significant impact on the plate, especially for coolers operating above 150 bar.

In the past, there have been several claims about broken pass partition plates. Following causes have been identified: Weak design, high pulsation amplitudes from Hyper compressors (LDPE), resonance due to the tube length being a manifold of the wavelength.



Example: Cooler pressure 350barg, Δp

The resonance length *L* can be calculated as follows:

$$L = \frac{c}{2Nf}$$

Where:

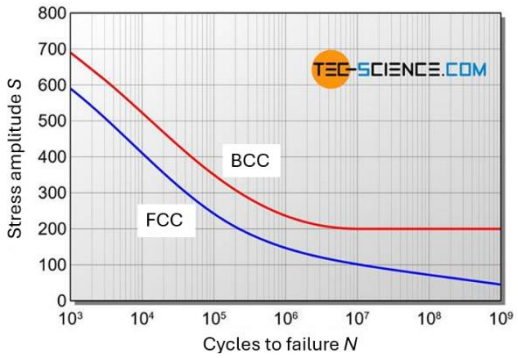
$$L\ [m] = Resonance\ length$$
$$c\ \left[\frac{m}{s}\right] = Speed\ of\ sound$$
$$N\ [-] = Order\ of\ interference\ (integer)$$
$$f\ [s^{-1}] = Frequency$$

If the total tube length of the cooler lies within a range of ±20% of the resonance length or multiples of it, the possibility for enhanced mechanical stresses on the pass partition plate increases, possibly leading material fatigue. Allowable tube lengths can be calculated using this tool:
Amplitude_of_Gas_Calculation.xlsx

In order to understand the fatigue strength of materials, a brief excursion into the topic of material behaviour is necessary. In general, a differentiation must be made between two types of crystal structures:

Structure (English)	**Body-Centered Cubic (BCC)**	**Face-Centered Cubic (FCC)**
Structure (German)	Kubisch raumzentriert (krz)	Kubisch flächenzentriert (kfz)
Phase	Ferrite	Austenite
Fatigue limit	Yes	No
Examples	Carbon steel, titanium alloy	Stainless steel, aluminum, copper

The fatigue limit is the stress level, below which fatigue failure does not occur. It can be visualized using the *S-N* curve (also known as Wöhler curve), which links the stress amplitude *S* to the number of cycles to failure *N* in a logarithmic scale.
The plot on the right shows that the fatigue level only exists for BCC (krz, e.g. carbon steel). Materials with an FCC-structure (kfz, e.g. stainless steel) do not have this distinct limit and will eventually fail from low stress amplitudes.



Thus, the fatigue strength of BCC-structures (e.g. carbon steel) can be confirmed if the material can withstand the defined stress amplitudes for $5*10^6$ cycles.
For FCC-structures (e.g. stainless steel), it is more complicated: In theory, these structures don't have a fatigue level and will eventually break. However, in practice, there is still a fatigue limit at which the probability of failure is very low. It is usually defined at 10^7-10^8 cycles. The reason for this behavior is the overlapping effect of work hardening: New dislocations in the material result in additional resistance to plastic deformation. Still, it is important to keep in mind that this is not the actual fatigue limit.

The failure of the pass partition plate has so far only occurred in LDPE-coolers. There are two reasons for this: the high-pressure pulsations at 5th and 6th stage and the high inertia due to the molar mass (28 g/mol) of LDPE. These coolers are built using carbon steel, hence confirmation for the endurance of $5*10^6$ cycles is required.
Pressures above 150 bar, which require additional considerations regarding pulsations, are also present in LNGM-coolers, manufactured in carbon steel as well. Although no damage to the pass partition plate has been reported so far, a fatigue life cycle analysis for $5*10^6$ cycles is required, too.
For high-pressure hydrogen applications, stainless steel is the cooler material of choice. According to the theory explained above, a stress fatigue test with at least 10^7 cycles would be required. However, the inertia from Hydrogen compared to Methane or LDPE is much lower due to the difference in molecular weight (2 g/mol). Thus, a test for $5*10^6$ cycles is sufficient.

To summarize, the following aspects have to be confirmed for operating pressures higher than 150 bar:

- Tube length according to the above-mentioned criteria regarding resonance
- Fatigue life cycle analysis of $5*10^6$ cycles:
  - 5% p.t.p. at max. operating pressure. (5th stage LDPE)
  - 10% p.t.p at max. operating pressure (6th stage LDPE)
- Additional supports for the pass partition plate (5th, 6th stage LDPE)
- The welding must be checked at sub-supplier (item of ITP).
- Support bracket of cooler, foundation height: As rigid and short (200-400 mm) as possible.
- Additional supports on the head: TEMA D Type (additional weight)
- Avoid SS cladding

5.6.5 Others

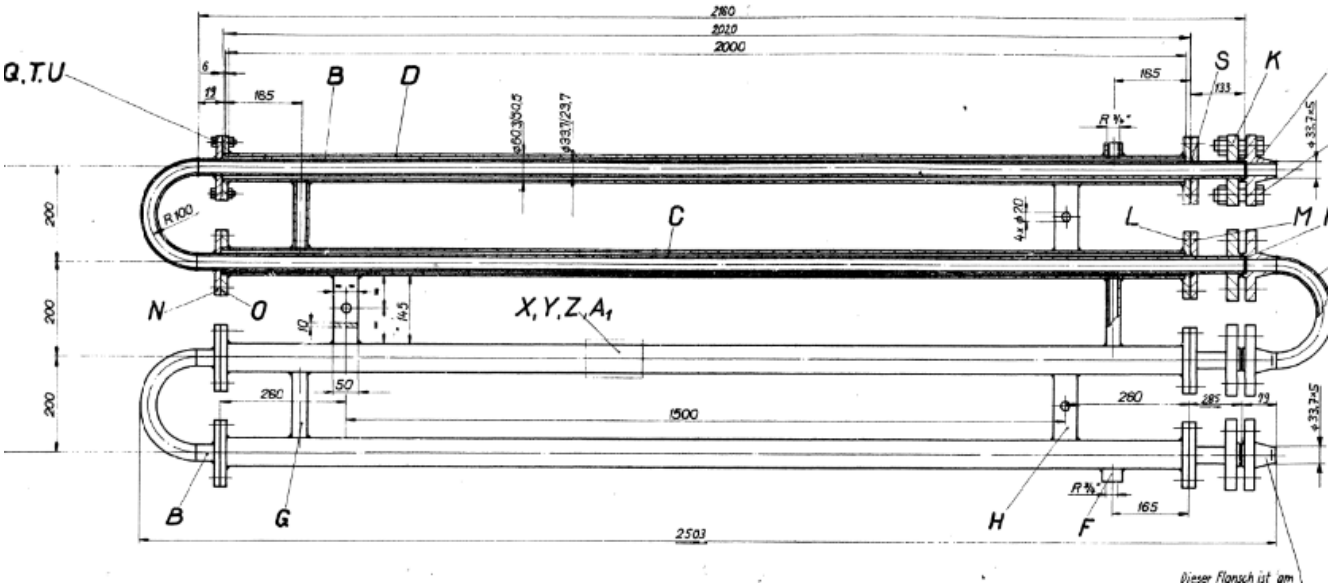
- On gas outlet, coolers shall be equipped with drain valves to allow for checking of tube joint leakage.
- When the gas is wet, saturated or may condensate inside the cooler, the cooler shall be place at an elevation and be self-draining towards the separator.
- Toxic gases like CO, require special leak proof joints or another suitable construction, e.g. double tube sheets.
- Water in tube side velocity shall range from 1.0 to 1.5 and for dirty coolant up to 2.5 m/s to minimize fouling. Rev01
- M.M.T (Mean Metal Temperature) for BEM Type must be calculated[1)] for the Tube sheet calculation.
- When client is IOCL extra care should be taken as the client water quality is stringent, and the specifications are crucial. AES gas on shell side is most requested cooler type but only for low pressure applications for high pressure AEM can be used. BEM gas tube side can be an option where water quality is better. Attention should be given on material selection.
- Tube inserts – These internals are used to make turbulence for tube side fluid for better velocity when fluid has less flowrate and pressure drop. This increases heat transfer coefficient, improving overall exchange efficiency. Also to be used only for clean process fluid. Most widely used types of tube inserts are Twisted tape, Wire matrix inserts, Spiral tube, conical or baffle inserts & Calgavin (SS) (to be confirmed from supplier)

- Impingement Plates –Impingement plates are protective plates installed inside the shell of a heat exchanger, typically near the shell inlet nozzle used when the shell-side fluid enters at high velocity or with solid particles. Their main function is to protect the tube bundle from direct impact by the incoming shell side fluid. They help in distributing the fluid more evenly across the tube bundle, improving heat transfer efficiency and reducing localized wear. Without the plate, the high-velocity fluid can cause erosion, vibration, or even mechanical damage to the tubes directly in its path. This can be plates or rods based on client requirements.
- Tube Plugging - Tube plugging involves sealing off one or more tubes in a heat exchanger to prevent fluid from passing through them. This is typically done when certain tubes are damaged, leaked, or excessively fouled and cannot be repaired immediately. 5-10% tube plugging is allowed as per TEMA. It is only performed sufficient design margin is available. It's not usually preferred for performance downgrade, mostly preferred for tube leaks, break etc. can be done in a step-by-step manner. It is advised to reduce tubes by iteration for performance but need to be careful & suppliers should be approached for similar cases.
- Expansion Joints - Expansion joints are flexible components used in fixed tube sheet heat exchanger systems to absorb thermal expansion, mechanical vibration, pressure fluctuations and other movements. Bellows is most widely used expansion joint for heat exchangers with the similar material of shell.
- If it is overdesigned too much it can be overcool to an undesirable extent, so overdesign should not be higher than 15%. To solve the overdesigned heat exchangers the solutions recommended are
    - change the coolant flow rate, Inlet temperature
    - Plugging/removing tubes.
    - Combination of above.
- Due to the high dynamic forces on the gas side of a reciprocating compressor:
    - Baffle plate thickness min 6 (gas on tube side) respectively 8 mm (gas on shell side).
    - Coolers to withstand residual pulsations of min. 5% p.t.p.
    - Nozzles to be reinforced by collars or integrally, due to the high nozzle loads.
    - Bundle vibration analysis is always required.

[1)] Done by supplier or PEO Department.

5.6.6 Tube in Tube Heat exchangers

Tube in Tube heat exchangers are also sized using ASPEN Exchanger Design & Rating. The *D* type of shell selection is required for tube in tube heat exchangers with multiple shell in series/parallel. The sketch out of Aspen is not yet available for this type but the preliminary sizing can be done for preorder phase projects.



6. Calculation of Heat Exchanger

The Project Engineering Office (PEO) department does thermal calculations in the early state of project with ASPEN EDR, to have the dimension for the General Arrangement. Also, for High pressure coolers and special design. But be aware that special with small cooler supplier are more experienced and the calculation can done by them without the help of PEO.

Even if there is a thermal calculation done by PEO, the supplier shall perform an own thermal calculation and give guarantee for the operation.

6.1 Minimum process parameters required for thermal design

- Determine the worst cases for design (maximum thermal and mechanical loads). See **Error! Reference source not found.**
- Provide corresponding gas flow conditions & gas inlet temperature
- Provide the max operating pressures for these conditions
- Define the expected gas temperature at outlet of cooler
- Define max allowable pressure drop across the cooler
- Define the maximum allowable cooling water inlet and outlet temperatures
- Supply the gas and coolant composition/s
- Define the min or max velocity in cooler tubes and on shell side, as and when applicable
- Define if cooler is required to be parallel flow
- Give client preferences as already known, e.g. cooler type, tube diameter etc.
- Clarify the fouling on shell and tube side. Higher fouling will reduce heat transfer ratio and increase the cooler size. Following table shows typical ranges for fouling resistance (factor).

Gas (clean)	1.76 x 10^{-4} m^2K/W	Gas (dirty)	17.6 x 10^{-4} m^2K/W
Water (treated)	0.8 x 10^{-4} m^2K/W	Water (untreated)	5.5 x 10^{-4} m^2K/W
Oil (low fouling)	1.76 x 10^{-4} m^2K/W	Oil (high fouling)	8.8 x 10^{-4} m^2K/W

  - Refer the document TDO_2011219_E11Typical fluid fouling factors
  - Specify the overdesign if required.

6.2 Choosing of worst Case

Please note that the worst thermal case is not always the case with the maximum heat exchange. The heat exchange area also depends on heat transfer ratio (U) as well as the MTD/LMTD mean temp difference (Δtm), see equation **Error! Reference source not found.**. On the internal summary, Summary coolant data a table Q/LMTD is given, which can help to choose the right case.

6.3 Overdesign

Surface/heat transfer ration margin should always be at least 10% on the worst thermal design case due to sizing tolerances as well as manufacturing tolerances and errors.

7. Cooler evaluation

7.1 Basic equation for required power

$Q = m_1 * cp_1 * \Delta T_1 = m_2 * cp_2 * \Delta T_2$

- Q = heat exchanged [kW]
- m = mass flow [kg/s]
- cp = specific heat capacity [kJ/kgK]
- ΔT= terminal temperature difference [K]
- 1,2 = Fluid hot and cold

7.2 Basic equation for heat transfer

$Q = A0 * U * \Delta tm$

- A0 = effective heat transfer surface [m2]
- U = heat transfer ratio [kW/m2K];
- Δtm = MTD/LMTD mean temp. difference [K]

7.3 Basic equation for overall heat transfer coefficient

$1/U = 1/\alpha\ tube * S\ tube/\lambda * 1/\alpha\ shell * foulingtot.$

- U=heat transfer ratio [W/m2K]
- α tube=heat transfer coefficient on tube side [W/m2K]
- α shell=heat transfer coefficient on shell side [W/m2K]
- S tube= tube thickness [m]
- λ= thermal conductivity of tube material [W/mK]
- fouling tot.= fouling on tube and shell side [m2K/W]

The task is to size and design cooler with optimized way with smaller size and high heat transfer coefficient. The velocity of the fluid is in direct proportion to the heat transfer coefficient. So the higher the velocity lesser area is required for the heat transfer.

8. Codes and Regulations

8.1 TEMA

The Tubular Exchanger Manufacturers Association (TEMA) is an association of manufacturers of shell and tube heat exchangers. TEMA has established a set of standards for Shell and Tube Heat Exchangers. TEMA Mechanical Standards are applicable to unfired shell and tube exchangers. [2].

The Standards recognize three classes of heat exchanger construction:

- Class R for the severe requirements of petroleum processing
- Class C for general commercial application.
- Class B for chemical process service.

There is in fact relatively little difference between the Standards for the three classes; where there are differences, Class R calls for heavier and more conservative construction features, and Class C gives some allowance for lighter design. Class B rarely used in BC.

8.1.1 TEMA R vs. TEMA C:

Para	Particulars	TEMA R	TEMA C **(preferred!)**
1.5	Corrosion allowance Not applicable to tubes See: TEMA 1.5.1.7	CS- 3.2mm	CS- 1.6mm
2.2	Tube size	All TEMA allow the same combination of tube diameter and thickness	
2.5	Tube Pitch	1.25 x Tube Diameter mechanical cleaning -> min. 6.4 mm	1.25 x Tube Diameter Tube diameter less are 15.9 mm and less, expanded only. Pitch may reduce to 1.2 x Tube Diamete
4.4	Min. longitude Baffle thickness (F and G)	6.4 mm for all Material	6.4 for CS
5.11	Min. Inside Depth of floating head (S and T)	Cross over area =1.3 x flow area through the pass the tubes of 1 pass	Cross over area = flow area through the pass the tubes of 1 pass
7.1	Tubesheet Thickness	thickness minus CA, > OD of tubes. Min 19mm incl. CA	thickness minus CA > ¾ OD. of tubes.
7.2	Tube hole Groove	Tube holes for expanded joints for tubes 15.9mm OD and larger shall have groove(s)	For design pressures over 20bar or Temp. over 177°C. Tube holes for expanded joints for tubes 15.9mm OD and larger shall have groove(s)
11.1	Min Bolt size	¾"	½"

8.2 GB Code

The GB code specify the requirement for shell diameter sizes as follow –

For ID < 450			For ID >=450	
OD	CS Thickness	SS Thickness	ID	
φ168	6	5	450	According to design pressure
φ219	6	5	500	
φ273	8	6	600	
φ325	8	8	700	
φ426	9	8		

Additionally, it is important that the "equal area method" is fulfilled as given in GB Code 150.3 section 6.1.1

GB Code 150.3 Section 6.1.1 – For Nozzle or Manhole Diameter.

a) When the internal diameter of cylinder meets Di<1500mm, then maximum diameter of opening shall be dop ≤ Di/2 and dop≤1000mm.
b) The maximum Permissible diameter of the opening in convex head or spherical shell shall meet dop ≤ Di/2.
c) The maximum opening diameter in conical head shall meet dop ≤ Di/3 and Di is the internal diameter of the conical shell at the centre of this opening.

8.3 API 660 (ISO 16812) [3]

API 660 is a shell and Tube heat exchanger standard which are no excluding TEMA but rather gives some additional requirements.

If API 618 is applicable for heat exchanger and client specifies TEMA R, according to chapter 7.10.1.5, then only API 660 is applicable

API618 Chapter 6.1.8 gives also recommendation about the min velocity. BC makes Standard deviation to that. The criteria for temperature rise and water velocity can be very difficult to fulfil with large cooler tube diameters. This can be used to argue for smaller cooler tubes.

8.3.1 TEMA vs. API 660:

Sr no	Subject	API 660 Requirement	TEMA Requirement
1	TEMA types “P” (Outside packed floating head) and “W” (Externally sealed floating tubesheet)	TEMA types “P” (Outside packed floating head) and “W” (Externally sealed floating tubesheet) are not allowed.	Type “W” (Externally sealed floating tubesheet) exchangers shall be used only for water, steam, air, lubricating oil, or similar service. For other limitations refer TEMA.
2	Minimum Tube OD (7.6.1.1)	The minimum outside diameter of the tubes shall be 19.05 mm (3/4”)	As per TEMA minimum tube outside diameter can be 6.4 mm also.
3	Tube Thickness (7.6.1.2)	Minimum tube thicknesses are defined with reference to tube material of construction.	No specific guidelines (numbers) are given in TEMA for average/minimum tube thickness with reference to tube material of construction.

8.4 ASME Section VII DIV1/2 Part UHX [4]

This ASME Section provide construction criteria for the design of Tubesheet (U-tube, Fixed and Floating head) Heat Exchangers. The design of Tubesheet of TEMA 9th Edition is not mandatory. If the client spec defines any previous edition of TEMA, both thickness must be calculated.

The rules contained in Division 2 can be used as an alternative to the minimum requirements specified in Division 1. Both cover the mandatory requirements having an internal or external pressure which exceeds 15 psi (103 kPa). Generally the Division 2 rules are more onerous than in Division 1 with respect to materials, design and non-destructive examinations but higher design stress intensity values are allowed. Therefore Division 2 is used for high pressure applications (above 207 bar) to achieve smaller wall thickness. Division 2 has also provisions for the use of finite element analysis to determine expected stress in pressure equipment.

8.5 Pressure Test

Refer to para 7.3. of White Book Pulsation Dampers (pressure vessels) TDO 2028918 for more information about pressure test.

9. References / Useful Links

[1] STANDARDS OF THE TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION

[1] http://www.tema.org/

[2] http://www.thermopedia.com/de/content/1182/

[3] API STANDARD 660 Shell–and–Tube Heat Exchangers for General Refinery Services

[4] ASME section VIII Div 1 Part UHX

9.1 Useful links

Design Tips: http://www.hcheattransfer.com/tips.html

Example for HEDS: GlobalCube – Guidance documents

Alternative Cooler TDO 2011219 E03

10. Revision Index

Rev 07

Chapter	What
Revised in chapter 8.2	Refined

Rev 08

Chapter	What
3	PP/PE input in the Matrix
4	Gasket
5.6.3	Add of O2 special feature
5.6.4	IOCL

Rev 09

Chapter	What
5, 8	Refined

Rev 10

Chapter	What
3	NEN Type and description of N type updated.
5.4, 5.4.1, 5.6.5	Refined chapters
6.1	Fouling number TDO number
5.6.4	Final agreement missing still under investigation