POLITECNICO DI TORINO

Faculty of Engineering

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Master's Degree thesis

Thermal and mechanical design of a steam generator for the petrochemical refinery service

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1. INTRODUCTION

In the refinery process the energy recovery can be a real challenge for the engineering activity and it has a major economic and technological importance. Indeed, achieving substantial falls in the temperature of very high flows may not be immediate, especially when the tubeside fluid is a mixture of hydrocarbons, which can be corrosive and hard to handle. On the other hand, a refinery facility can have an in-house power generation plant or some petrochemical processes which require vapour to be kept operational, as example the steam cracking method. The steam generation can be maintained exploiting the unnecessary available heat by means of a heat exchanger, and the situation mentioned above can occur. Heat exchangers, which at a first glance may seem simply to build, are actually very complex and full of peculiarities. Furthermore, when special materials have to be used, even a small error in design can be reflected in thousands of euros of damage. Design and simulation programs, as the one used in the study case, perform the pre-set computations with the indicated design code. However, to obtain the best configuration for the case under analysis and to avoid errors, a correct and conscious management performed by the engineer is required. Generally, given the magnitude of this sector, this is possible only with a fair amount of experience.

The goal of this thesis is to develop the thermal and mechanical design of a heat exchanger, being conscious of the theories on which the calculus codes are based and the manufacturing and financial problems. The objective of the heat exchanger's thermal design is to find the necessary exchange area that allows the required heat flux between the fluids. Moreover, the best heat exchanger configuration for the case have to be analysed and found. The mechanical design, instead, aims to ensure the physical resistance of each component against the internal stress generated from both thermal and pressure gradients. In addition, a host of traditional considerations must be addressed in designing heat exchangers. As example, the minimization of fixed cost and some manufacturing considerations have to be taken into account during the design activity. I decided to write this thesis in collaboration with SIMIC.Spa, which recently received a steam generator commission from an important EPC contractor ("Engineering, Procurement, Construction"), with intended use in one of the biggest Indian Petroleum refineries. However, due to the confidentiality agreement that has been signed, the name of the interested parts and of the project cannot be exposed. In order to develop the design procedure, I've thoroughly studied the regulations framework, the heat exchange science and the mechanical resistance theories applied to pressure vessels. Therefore, I've carried out an important in-deep analysis of heat exchanger configurations and design options to provide a complete overview of the topic.

A significant problem for those who approach for the first time to the sector of the pressure vessels is the regulatory framework. A non-standard scenario is present in the world, in which there are laws, directive and regulations depending on the delivery place. Very often, also plant requirements or preferences dictated by the client are present. Due to the possible overlay between the laws, preferences and codes used during the design activity, I decided to insert a chapter regarding the regulatory framework and the hierarchical order that must be observed. In the proposed study case, the ASME code and TEMA technical standard have to be observed. Subsequently, the chapter number three introduces the heat exchangers devices, the classification and the terminology used during design and construction. In this section all the important features and construction peculiarity are described. Even if the most used classification method is analysed, it can be noticed how every device can have its peculiar configuration in order to accomplish the different goals and settings required by the final working layout. Indeed, the reader will realize how every device requires to be carefully analysed and designed with particular and dedicated considerations.

The following chapters are the core of the thesis and they approach the thermal and mechanical design of the heat exchanger. For both sections there is a first theoretical part in which the most important concepts are presented. The thermal theoretical part begins with the popular concepts about conduction and convection. After the explanations of the thermal design methods, the chapter continues with the in-deep analysis of the heat transfer in boiling and other phase change configurations. This technical difficulty is also present in the study case and introduces some tricky concepts that generally are not analysed during the degree course. Follows a thermal design chapter in which the design activity of the study case, accomplished with the support of ASPEN shell and tubes, is reported. Starting from the data provided by the client, the inputs are implemented and all the design choices are explained. Here, many important parameters have been analysed and some interesting technical considerations can be found. Subsequently, the computational design procedure has been launched, and the program provides the first set of results. This group of possible configurations will be thoroughly analysed with respect to the parameters of selection that have to be satisfied. The best design will be chosen among them, and it will set the heat exchanger geometry and characteristics. At the end, the vibrational analysis, with both the HTFS and the TEMA methods, have been carried out in order to verify the acceptability of the design. Afterwards, the thermal design refinement, called Rating mode, is made and the dimensions of the interface pieces are set at standard values, e.g. the nozzle diameters. Follows the complete simulation of the last solution, which leads the final thermal output to be ready for the upload in the mechanical part of the program. In this way, the whole thermal design is fixed, and the output will be used for the mechanical design activity of the study case.

Before the second part of the study case just mentioned, a mechanical theory chapter reports some insights and theories about pressure vessel and heat exchanger design. Starting from some preliminary considerations regarding the admissible stress and the most common used theories of failure, the verifications criteria with plasticity collaboration has been analysed. Afterwards, a chapter regarding the general calculation criteria have been reported. Some general considerations about membrane stress in revolution shell, edge effects and stress concentration around holes have also been carried out. The theories of cylinders under internal pressure have more deeply been developed and reported as example. However, many theories and analysis could be incorporated to complement the theoretical approach. The main goal of this section, as well as for the previously reported thermal theoretical part, is to demonstrate how the calculation codes are science-based and derive from globally approved theories. In the same conceptual way, the calculation report of the program, present in the second appendix, includes all the passages and computation done during the mechanical design, which in turn are based on the calculation codes. Also, it can be seen how the designer can always maintain the control on the design activity checking calculations that the program have been done. The last chapter completes the design activity of the study case, discussing the mechanical design of the heat exchanger under consideration. An important mention to the materials, and examples of the ASME designation, has been done in this section. At the end some iterative procedures, similar to the rating mode of the thermal part, were necessary in order to uniform the plate thickness and pipe dimensions. All the vessel dimensions resulting from the last program calculation have been reported in the final chapter, while the discussion is concluded with the technical drawings and the calculation report that have been inserted in the appendices.

2. LEGISLATIVE FRAMEWORK FOR PRESSURE VESSELS

Manufacturers must comply with the jurisdictional or government regulations for different product standards, each targeting specific types of equipment. The heat exchanger devices, independently from their type or purpose, fall into the pressure vessels category. Due to the potential dangers and associated risk connected with the high working pressure and temperature of these devices, the reference directives and the standards design codes are complex and articulate. As a result, an overview of the main classes of regulations is reported below in the order of priority.

2.1. LAWS, DIRECTIVES AND TECHNICAL REGULATIONS

The laws, directives and technical regulations are continental or national documents with the highest level of priority. That's why they are predominant over the standards and technical standard specification. They are issued by legislative authorities, like governments or committees and they set out goals that must be achieved by the country in which they are effective. All the contained requirements are not specific or technical, but they deal with safety or general-purpose issues. So, every device which will be installed in a country must satisfy the local laws in force. The national legislation can impose a specific design code or require the designed device to be verified in accordance with a specific one. As example, the European directive is commented below.

2.1.1. EUROPE - PED 2014/68/EU: Pressure Equipment Directive

The European Pressure Equipment Directive (PED) 2014/68/EU is a pre-requisite for CE-Marking and a guideline for the design and manufacture of pressure equipment. This means that it has substituted all the pre-existent national laws about pressure equipment inside the EU. It applies to pressure vessels, steam boilers, pipelines, heat exchangers, storage tanks, pressure relief devices, valves, regulators and other pressure equipment with a maximum allowable pressure superior to 0.5 bar. All devices are classified in different risk categories (Category I, II, III, IV) in function of their maximum allowable pressure and the danger and amount of fluid they have to process. According to the PED risk category of the product, it can be chosen the procedure to obtain the CE marking. All the devices that will be installed in the European country shall mandatorily have the CE marking and so comply with all the requirement of the PED.

Nevertheless, some pressure equipment that present a relatively low hazard from pressurization could be covered by other directive as the Transportable Pressure Equipment Directive (TPED 2010/35/EU), the Simple Pressure Vessels Directive (SPVD 2009/105/EC) or other directives, e.g. ATEX Directive, Machinery Equipment Directive, Electro Magnetic Compatibility Directive or the Low Voltage Directive.

Summarizing, the Pressure Equipment Directive does not impose a specific regulation to follow during the design activity. However, if the device will be installed inside EU it needs the CE certification, so it shall comply with all the requirement of the PED. If a code different from EN is used, a document demonstrating that all the PED's requirements have been satisfied must be issued. Contrariwise, if the EN regulations are used during the design activity, the PED requirements are automatically assumed to be observed and the CE certification can be directly obtained.

2.2. STANDARDS AND CODES

Standards, also called codes, are documents issued and approved by a recognized and official standardization organization. They provide rules, guidelines or characteristics for products or related processes and production methods for common and repeated use, with which compliance is not mandatory. However, laws and regulations may refer to standards and compulsory have to make compliance with them. [1] These documents may also include or deal with symbols, terminology, packaging and marking or labelling requirements, as they apply to a product, process or production method. For the knowledge of the reader, it is reported that it is generally named "code" the standard in which all the main rules and guidelines of product's sector are contained. Of course, during the design activity, the code is not the only standard considered. A section with the principal organizations and codes for the pressure vessels is commented below.

2.2.1. ASME – Boiler & Pressure Vessel Code (BPVC)

ASME, the American Society of Mechanical Engineers, is an international developer of codes and standards associated with the science, and practice of mechanical engineering. Starting with the first issuance, in 1914, of the Boiler & Pressure Vessel Code, ASME's codes and standards have grown to more than 500 offerings currently in print. These offerings are accepted for use in more than 100 countries around the world and cover a breadth of topics, including pressure technology, nuclear plants, elevators, construction, engineering design, standardization, and performance testing. [2] The code developed from ASME that regulate the pressure vessels, in which vapor generator heat exchangers are contained, is the Boiler & Pressure Vessel Code (BPVC). The following is the structure of the January 2019 Edition of the BPV Code:

- ASME BPVC Section I - Rules for Construction of Power Boilers
- ASME BPVC Section II – Materials
  - Part A - Ferrous Material Specifications
  - Part B - Nonferrous Material Specifications
  - Part C - Specifications for Welding Rods, Electrodes and Filler Metals
  - Part D - Properties (Customary)
  - Part D - Properties (Metric)
- ASME BPVC Section III - Rules for Construction of Nuclear Facility Components
  - Subsection NCA - General Requirements for Division 1 and Division 2
  - Appendices
  - Division 1
    - Subsection NB - Class 1 Components
    - Subsection NC - Class 2 Components
    - Subsection ND - Class 3 Components
    - Subsection NE - Class MC Components
    - Subsection NF – Supports
    - Subsection NG - Core Support Structures
  - Division 2 - Code for Concrete Containments
  - Division 3 - Containment Systems for Transportation and Storage of Spent Nuclear Fuel and High-Level Radioactive Material
  - Division 5 - High Temperature Reactors
- ASME BPVC Section IV - Rules for Construction of Heating Boilers
- ASME BPVC Section V - Nondestructive Examination

- ASME BPVC Section VI - Recommended Rules for the Care and Operation of Heating Boilers
- ASME BPVC Section VII - Recommended Guidelines for the Care of Power Boilers
- ASME BPVC Section VIII - Rules for Construction of Pressure Vessels
    - Division 1
    - Division 2 - Alternative Rules
    - Division 3 - Alternative Rules for Construction of High-Pressure Vessels
- ASME BPVC Section IX - Welding, Brazing, and Fusing Qualifications
- ASME BPVC Section X - Fiber-Reinforced Plastic Pressure Vessels
- ASME BPVC Section XI - Rules for Inservice Inspection of Nuclear Power Plant Components
    - Division 1 - Rules for Inspection and Testing of Components of Light-Water-Cooled Plants
    - Division 2 - Requirements for Reliability and Integrity Management (RIM) Programs for Nuclear Power Plants
- ASME BPVC Section XII - Rules for the Construction and Continued Service of Transport Tanks
- ASME BPVC Code Cases - Boilers and Pressure Vessels

The study case developed in this thesis refers to the ASME BPVC Section VIII for pressure vessels and the accessory section II, V and IX, respectively for the materials, non-destructive examination and welding qualifications.

Particular attention must be adopted for the section II which provides specifications for the materials suitable for the construction of pressure vessels. It consists in four parts:

- **Part A - Ferrous Material Specifications**

  The specifications contained in this Part specify the mechanical properties, heat treatment, heat and product chemical composition and analysis, test specimens, and methodologies of testing for ferrous material. The designation of the specifications starts with 'SA' and a number which is taken from the ASTM 'A' specifications. [2]

- **Part B - Nonferrous Material Specifications**

  The specifications contained in this Part specify the mechanical properties, heat treatment, heat and product chemical composition and analysis, test specimens, and methodologies of testing for nonferrous materials. The designation of the specifications starts with 'SB' and a number which is taken from the ASTM 'B' specifications. [2]

- **Part C - Specifications for Welding Rods, Electrodes, and Filler Metals**

  It provides mechanical properties, heat treatment, heat and product chemical composition and analysis, test specimens, and methodologies of testing for welding rods, filler metals and electrodes used in the construction of pressure vessels. The specifications contained in this Part are designated with 'SFA' and a number which is taken from the American Welding Society (AWS) specifications. [2]

- **Part D - Properties (Customary/Metric)**

  It provides tables for the design stress values, tensile and yield stress values as well as tables for material properties (Modulus of Elasticity, Coefficient of heat transfer, et al.) [2]

2.2.2. EUROPEAN STANDARDS EN13445 – Unfired Pressure Vessels

A European Standard is a standard developed by one of the three recognized European Standardization Organizations (ESOs): CEN, CENELEC or ETSI. Each Standard is identified by a unique reference code containing the letters 'EN'. After the emission by the CEN, a European Standard must be receipt from a National Regulatory Authority, e.g. UNI, and inserted into the National Technical Rules. Then, the name of the National Regulatory Authority is added to the name of the standard. For instance, the European code that contains all the main rules and guidelines for the pressure vessels is the EN13445 – Unfired Pressure Vessels, that becomes UNI-EN 13445 after been inserted into the Italian regulation. As I have introduced, the European code is harmonized with the Pressure Equipment Directive (2014/68/EU or "PED"), so, if the equipment is designed following the EN code, automatically the European PED directive is satisfied. The EN 13445 is divided in:

- EN 13445-1: Unfired pressure vessels - Part 1: General
- EN 13445-2: Unfired pressure vessels - Part 2: Materials
- EN 13445-3: Unfired pressure vessels - Part 3: Design
- EN 13445-4: Unfired pressure vessels - Part 4: Fabrication
- EN 13445-5: Unfired pressure vessels - Part 5: Inspection and testing
- EN 13445-6: Unfired pressure vessels - Part 6: Requirements for the design and fabrication of pressure vessels and pressure parts constructed from spheroidal graphite cast iron
- EN 13445-8: Unfired pressure vessels - Part 8: Additional requirements for pressure vessels of aluminum and aluminum alloys
- EN 13445-10:2015: Unfired pressure vessels - Part 10: Additional requirements for pressure vessels of nickel and nickel alloys. PUBLISHED 2016.6.30
- Parts 7 and 9 do exist but they are merely technical reports.

2.2.3. NATIONAL CODES

Before CEN issued the EN standards, every country in Europe had its own national codes, regularly updated. Nowadays, with the implementation of the EN standards, some country like Italy and Netherlands have stopped updating their national codes, Ispesl VSR and Stoomwezen respectively, that are now dismissed. Despite that, other countries continue evolving their standards. Due to the fact that sometimes these procedures could be expressively required by the client or for legislative issues, the national codes have to be taken into account. Some important national codes still valid for the pressure equipment are:

- AD 2000 MERKBLATTER: Germany
- CODAP: France
- PD 5500: United Kingdom
- Gost: Russia
- JIS: Japan

An example in which a national code must be taken into account is when the piece's final destination is the Russia. The Russian legislation does not impose a standard for the design. So, you can design the equipment following any codes but, at the end, you need to verify that the resulting thicknesses are bigger than the result that you would be obtained if designed using the GOST Russian national codes.

2.3. TECHNICAL STANDARDS SPECIFICATIONS

A technical standard is a document issued by clients, engineering companies or manufacturers containing some technical requirements. They don't need to be satisfied for some juridical reason, but for some technical purpose. For example, if many orders for a big project have to be split between different firms, a technical standard can be released in order to fix the geometry end the dimensions of the items' interface part. In this way the compatibility with the other pieces of the project or interchangeability is guarantee. TEMA (Tubular Exchanger Manufacturers Association) standards is probably the most famous example of these collection of requirements. It's not issued by an official organization, but it was created by the association of the major American companies of heat exchanger in order to compensate where the standards did not contain methods for design some specialized parts. Now a days, they aren't something of juridical recognized, but sometime, for the design activity, they are asked by the clients in association with a code.

3. GENERALITIES ON HEAT EXCHANGERS

3.1. CLASSIFICATION AND MOST COMMERCIAL CONFIGURATIONS

A heat exchanger is a heat-transfer device that is used to transfer internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat-transfer surface, and ideally they do not mix. On the other hand, if the two flows are directly in contact because no element of thermal resistance is present between them, the device is named direct-contact heat exchanger (e.g. steam bubbled into water,Figure 1).

Inside this document only the heat exchangers with a dividing wall between the fluids will be analysed. The classification of a heat exchanger can be done according to different parameters. Below a scheme of the different possibility of heat exchangers' classification is reported. There are an enormous variety of configurations, but most commercial exchangers reduce to one of the three basic types treated in the next page.

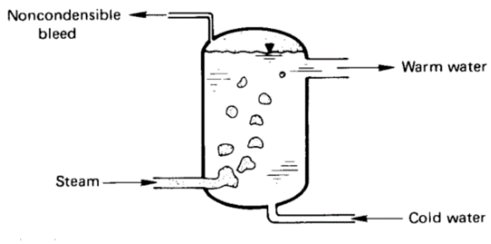
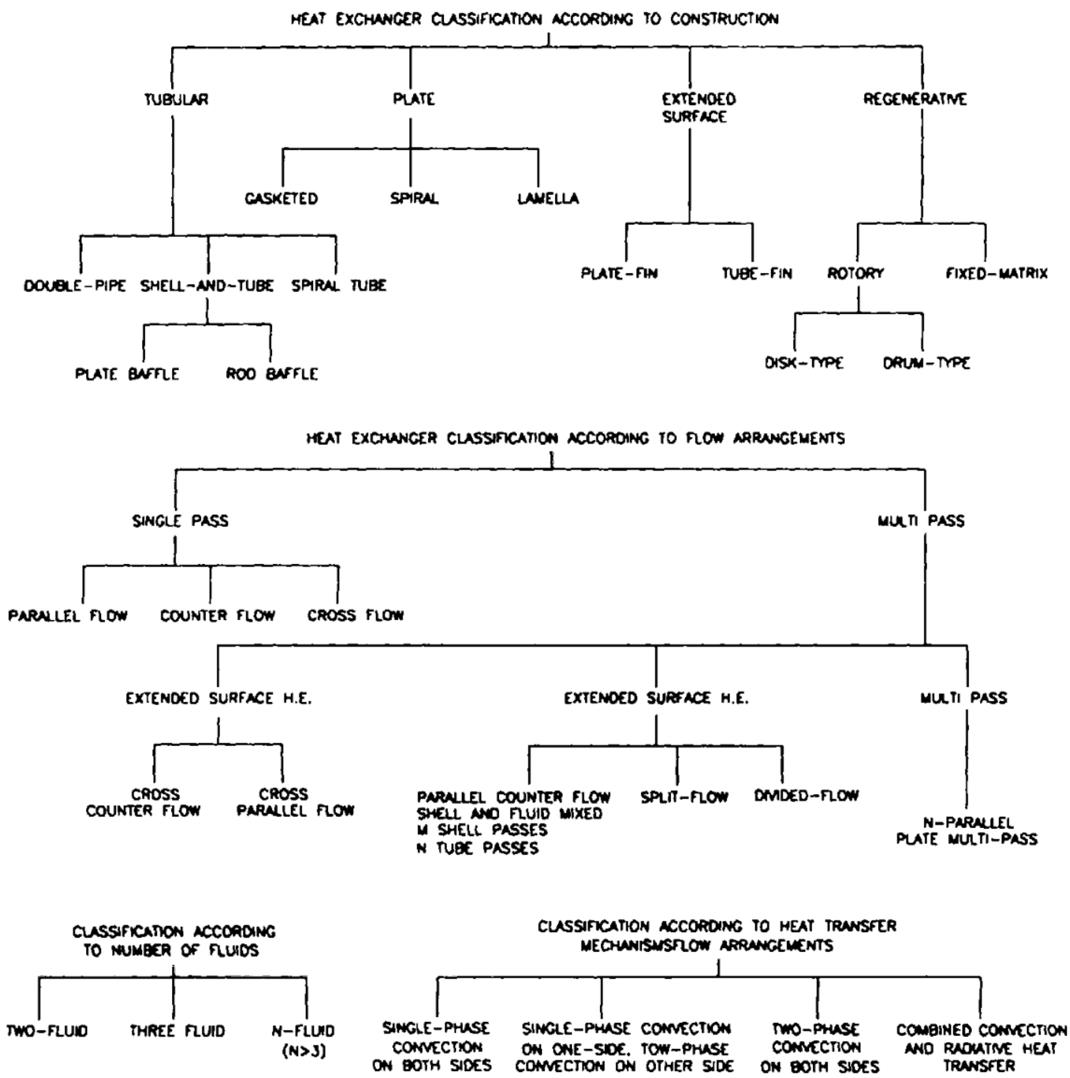
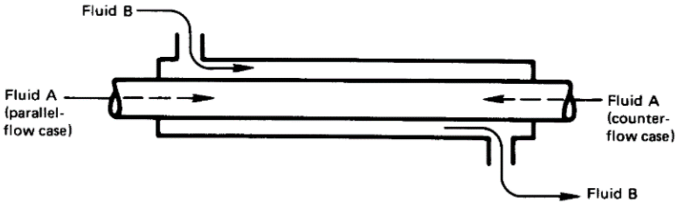


Figure 1: A direct-contact heat exchanger. [16]



The simple parallel or counterflow configuration. These arrangements are versatile and can have different shape.Figure 2 shows how the counterflow arrangement is bent around in a so-called Heliflow compact heat exchanger.



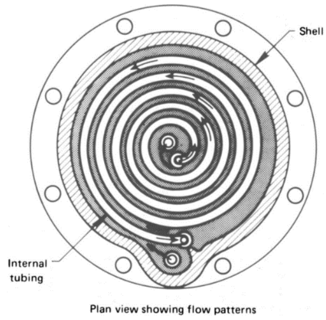


Figure 2: Parallel and counterflow heat exchanger scheme. On the right: Heliflow compact counterflow heat exchanger. [16]

The cross-flow configuration. Figure 3 shows a typical cross-flow configurations in which the two fluid are not mixed together. If baffles are present each flow must stay in the prescribed path through the exchanger and is not allowed to “mix” to the right or left.

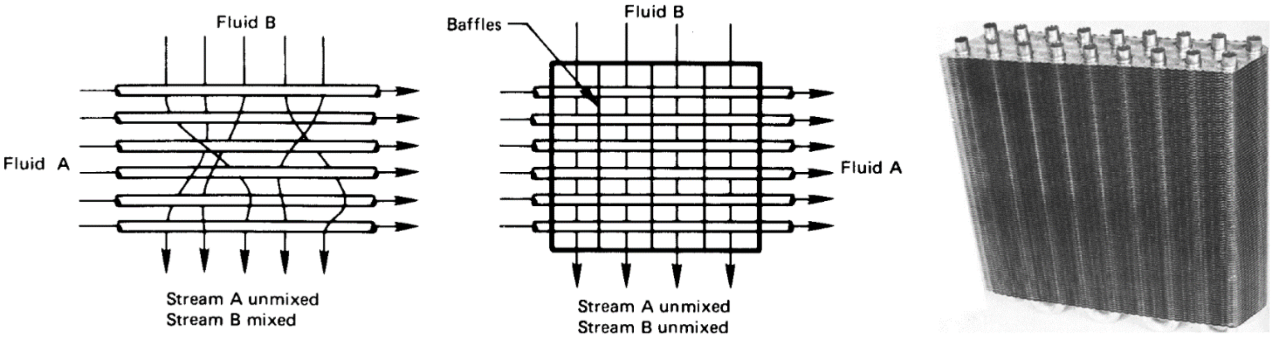


Figure 3: two kind of cross-flow exchangers scheme. on the right a typical plate-fin cross-flow elemet. [16]

The shell-and-tube configuration. Most of the large heat exchangers are of the shell-and-tube form, and them will be study in deep during the next paragraphs. An example is reported in the Figure 4 that shows an exchanger with tube-bundle removed from shell.

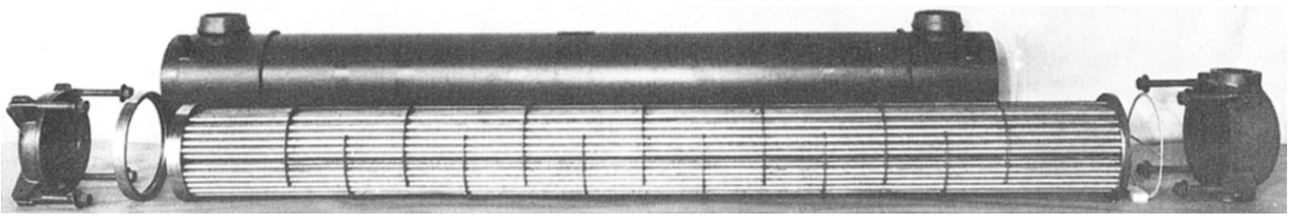
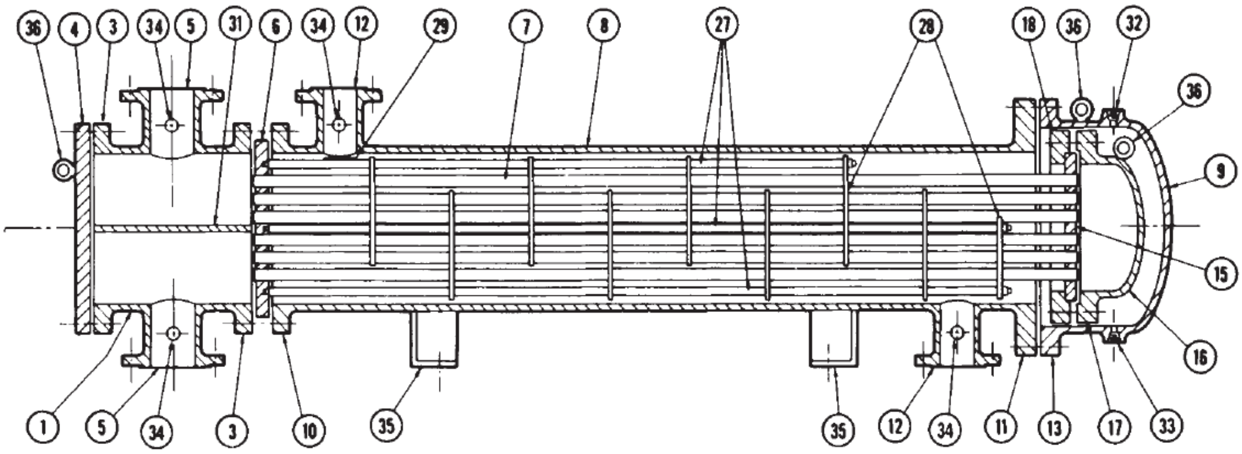


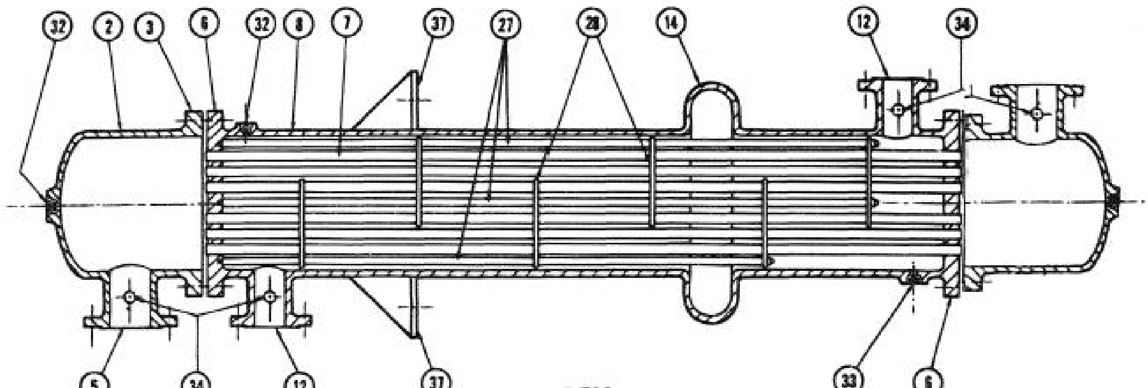
Figure 4: typical commercial one-shell-pass, two-tube-pass heat exchangers. [16]

3.2. SHELL-AND-TUBES CONFIGURATION AND TERMINOLOGY BY TEMA



All the images present in this chapter, except where properly indicated, have been taken from [3].

1. Stationary head – channel
2. Stationary head – bonnet
3. Stationary head flange – channel or bonnet
4. Channel cover
5. Stationary head nozzle
6. Stationary tubesheet
7. Tubes
8. Shell
9. Shell cover
10. Shell flange – stationary head end
11. Shell flange – rear head end
12. Shell nozzle
13. Shell cover flange
14. Expansion joint
15. Floating tubesheet
16. Floating head cover
17. Floating head flange
18. Floating head backing device
19. Split shear ring
20. Slip-on backing flange
21. Floating head cover – external
22. Floating tubesheet skirt
23. Packing box flange
24. Packing
25. Packing gland
26. Lantern ring
27. Tie rods and spacers
28. Transverse baffles or support plates
29. Impingement plate
30. Longitudinal baffle
31. Pass partition
32. Vent connection
33. Drain connection
34. Instrument connection
35. Support saddle
36. Lifting lug
37. Support bracket
38. Weir
39. Liquid level connection
40. Floating Head Support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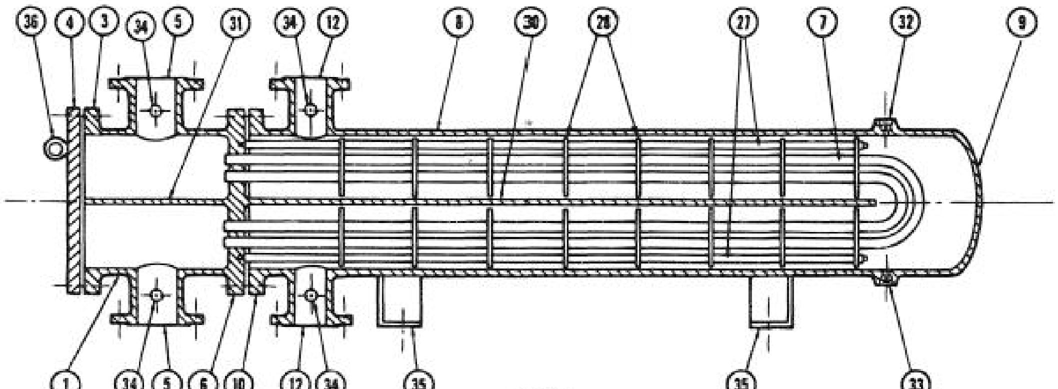
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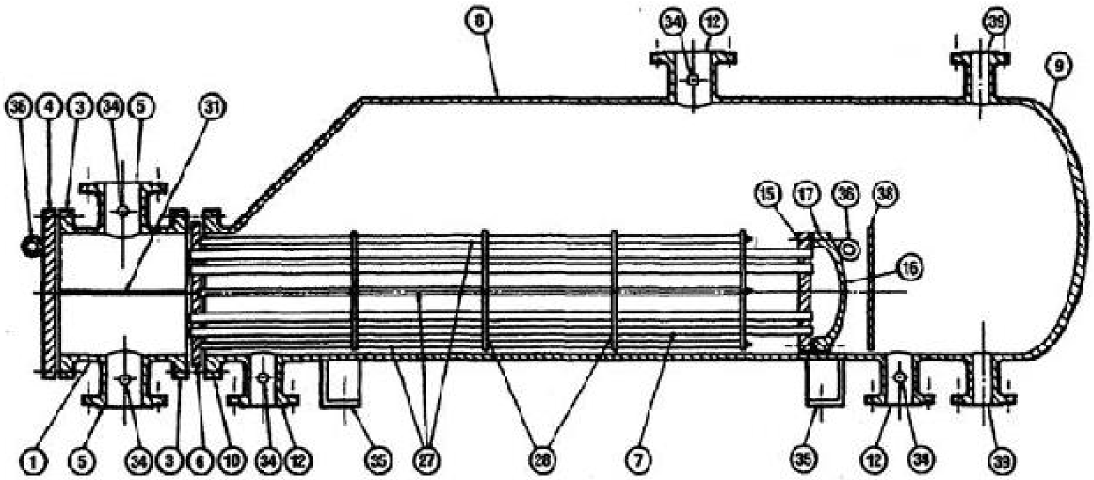
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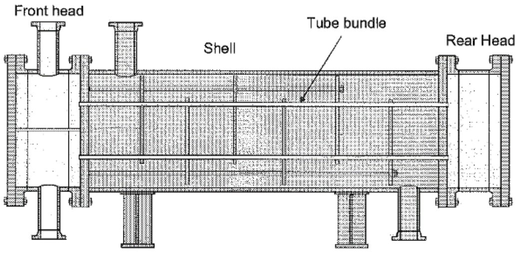
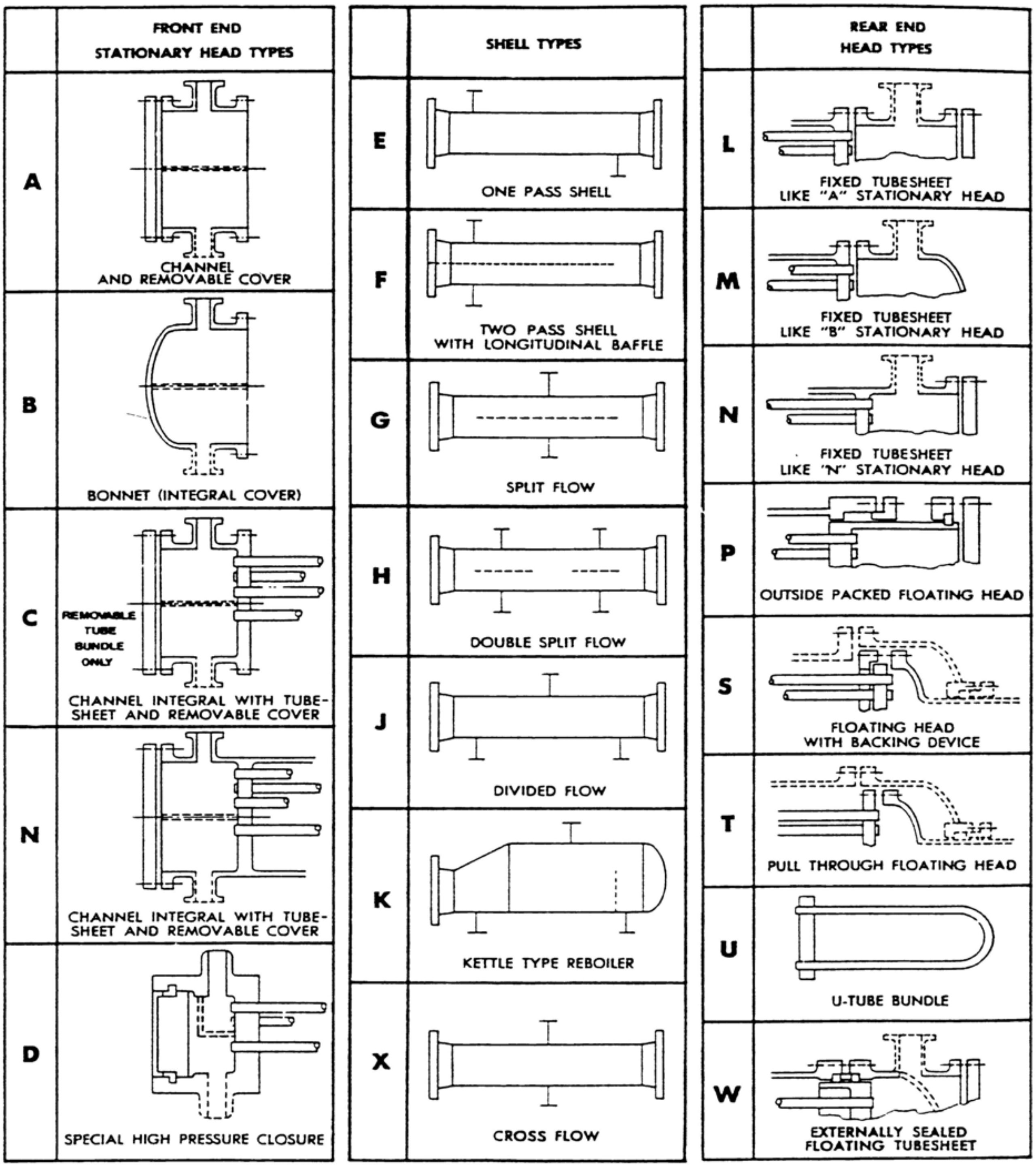
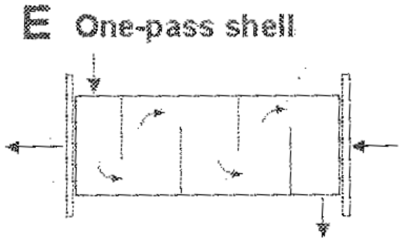
With the purpose of establishing standard nomenclature and terminology, the TEMA has introduced some frequently used standards. As a matter of fact, it is recommended that heat exchanger type has to be designated by a number of letters as described below.

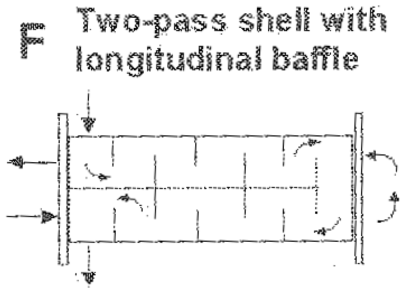




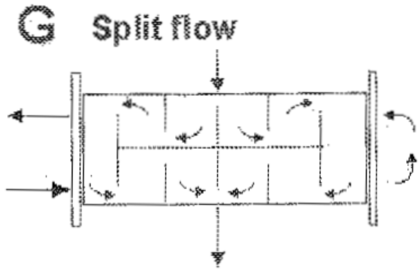
SHELL TYPES



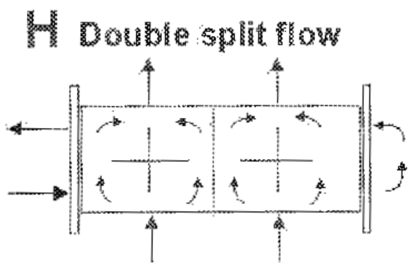
E-shell: this is the most common type, where the shellside fluid enters at one end of the shell and leaves at the other in one pass. Generally it is considered to be the standard. If a single tube pass is used, providing there are more than two or three baffles, near to counter current flow is attainable and temperature crosses (cold fluid exit temperature is higher than hot fluid outlet temperature) can be handled, i.e., low Log Mean Temperature Difference (LMTD).



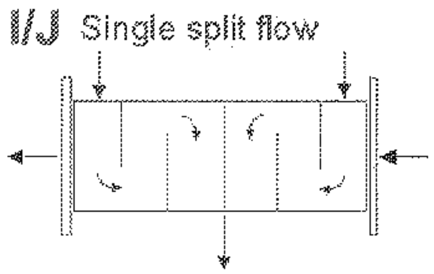
F-shell: this shell has longitudinal baffle extending most of the way along the shell, dividing it into two halves. The shellside fluid enters at one end of the shell, flows in the top half to the other end and then back in the lower half, thus giving two shellside passes. In most F types there are also two passes on the tubeside, thus ensuring countercurrent flow. If a fixed tubesheet construction is used, then the longitudinal baffles can be seal welded to the inside of the shell, thus preventing leakage from the first shell pass to the second. If a removable bundle is fitted, then leakages can be controlled (but not eliminated) by fitting packing material. Note than removing and re-inserting the bundle can damage this seal and result in significant leakage across the baffle from one shell pass to the other. If an F-shell is acceptable to the customer, then a two-tube pass version is an alternative to a single pass E-shell. Also, if two or more tube passes are used (even number), any given number of E-shells in series may be replaced by half the number of F-shells, avoiding very long exchangers.



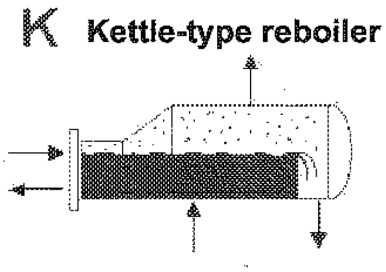
G-shell: this is sometime known as the “split flow” shell. Fluid enters the shell through a nozzle placed at the center of the shell. The shell has a central longitudinal baffle that divides the flow into two, so that each half makes two passes in its half of the shell before combining and exiting through a central nozzle. The main advantage, compared to an E-shell, is that a single G-shell will often handle terminal temperatures which would often require two E-shells in series. Note then the minimum number of baffles that may be specified for a G-shell is five. This is to maintain the crossflow path. Limiting the number of baffles to five ensures that there is a baffle under the nozzle plus two baffles on each leg of the longitudinal baffle (i.e., 2+1+2 for G shell).



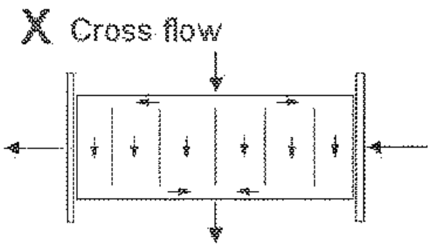
H-shell: sometimes called the “double split flow” type, this shell has two inlet and exit nozzles located ¼ and ¾ the way along the shell. There is a longitudinal baffle in each half of the exchanger, so that on entry each half of the flow is split again, either going to the end of the shell and back, or to the middle and back. Because of this flow split, this type has a low shellside pressure drop, and it is normally used in horizontal thermosyphon reboilers. Note then the minimum number of baffles that may be specified for a H-shell is 10. This is to maintain the crossflow path. Limiting the number of baffles to 10 ensure that there is a baffle under the nozzle plus two baffles on each leg of the longitudinal baffle (i.e., 2(2+1+2) for H shell)



I- or J-shell: also known as “divided flow”, this shell has a single central inlet nozzle and two outlet nozzles, one at each end of the exchanger. After entering the exchanger, the flow is split into half by a centrally located baffle. Since half the fluid flows through each half of the exchanger, the shellside pressure drop for a given service is much lower than an equivalent E-shell. Thus J-shell are commonly used for low pressure condensers and other services with low allowable pressure drop. They are also known as J12 (or 1 nozzle in, 2 nozzles out). Although this type is usually depicted as having one inlet and two outlets, shell with two inlet and one exit (called “I-shell”, or J21) are also used.



K-shell: these are “Kettle” reboilers and in practice are used exclusively for vaporizing service. The inlet of the fluid to be vaporized normally is at the bottom of the shell. The shell diameter is larger than the bundle and boiling liquid flows up through the bundle, with any unevaporated liquid falling back to the bottom of the shell, before recirculating up through the bundle. The liquid level is maintained above the bundle by a weir plate (a level control valve may be used instead of a weir plate). The vapour formed is separated from the liquid in the enlarged shell and leaves through a nozzle at the top. Demister pads are sometimes placed at the vapour outlet to remove any entrained liquid.

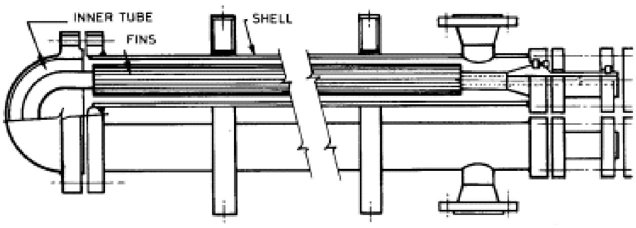


X-shell: this is usually known as a “cross flow” shell, where the shellside fluid makes one pass diametrically across the shell. Some have single central nozzles at the top of the shell and a single exit nozzle at the bottom. Others can have multiple nozzles at the top and bottom. The shellside pressure drop for an X-shell is very low, and therefore is used for services where the shellside volumetric flow rate is high or where the allowable pressure drop is very low, such as vacuum condenser. Also, an X-shell with four or more tube passes in ribbon band layout approximates to pure countercurrent flow. Therefore, depending on the pass layout, it can have a superior effective temperature difference with respect to an E-shell.

There are number of different shell types that are not covered officially by TEMA. These are:

- **Double pipe exchanger**: These consist of a long, small diameter shell, with a single tube placed within it. The tube usually has longitudinal fins extending to the shell to increase the heat transfer area. Double pipe heat exchanger often has a hairpin inner tube, with a separate shell on each leg.

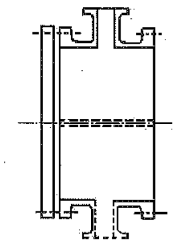
- **Multi-tool hairpin exchanger**: these consist of a bundle of hairpin tubes, with a separate shell on each leg of the hairpin and a special cover over the U-bend of the hairpin. The tubes may be longitudinally finned, but usually they are plain tubes with baffles to give cross flow. The number of tubes in the bundle is usually much less than in a conventional heat exchanger.



FRONT HEAD TYPES

The choice of the front head depends upon the pressure of the tubeside fluid and whether or not the tubeside requires frequent mechanical cleaning. It mainly affects the mechanical design, the cost and the weight; while it has little effects on the thermal design activity. The front end is the tubeside inlet end. If an even number of tubeside passes is specified, it can also be the tubeside exit end. For this reason, the front end always has at least one nozzle and it is also referred to as the stationary head.

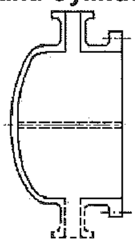
A – Channel and Removable Cover



A-type (channel and removable cover): in this type the channel barrel is flanged at both ends. One flange is bolted to the tubesheet and a usually flat cover plate is bolted to the other, thereby permitting cleaning of the inside tubes without removing the whole channel or associated piping. For inspection or repairs of the tube-to-tubesheet joints, particularly those near the edge, the removal of the channel is usually necessary. Disassembly of the whole channel is only necessary if the bundle has to be removed for the shell tube cleaning. In spite of the relative high cost due to two flanges joints, the A-type is widely used, especially in petroleum refineries where it tends to be regarded as standard.

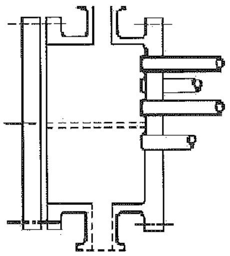
B-type (bonnet – integral cover): the channel barrel is flanged at one end only, the other end being permanently closed by either a welded-in flat plate or semi-elliptical head. This head is cheaper and lighter than the A-type but is not recommended for exchangers which require frequent tubeside cleaning, since the entire head must be removed and the piping connection dismantled. B-types are generally used for exchangers with clean tubeside fluids or for small fixed tubesheet units where removal is relatively easy. Note: the difference between A- and B-type is the removable cover, not the shape.

B - Ellipsoidal Cover and Cylinder

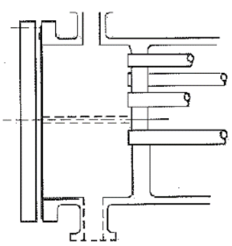


C-type (channel integral with tube sheet – removal cover): this is similar to the A-type, except for the tubesheet end of the channel is not flanged, but is welded directly to the tubesheet, which is then bolted to the shell flange. This enables the whole channel and tube bundle assembly, complete with piping connections to be left in place whilst the shell is drawn away, usually on wheels specially fitted for this purpose. This head type is used when the bolted joints must be minimized for hazardous tubeside fluids or for heavy high-pressure bundles where it is easier to remove the shell and where cleaning of the shellside is more frequent than the tubeside. One major disadvantage is that having removed the cover plate for access to the tubesheet it is extremely difficult to make repairs to the outer tubes. For this reason, it is usually necessary to specify a large bundle-shell clearance than would otherwise be necessary, thus making this a costly option.

C - Channel integral with tube sheet



N - Flat Cover Non removable bundle



N-type (channel integral with shell): this type is similar to the C-type except for the integral tubesheet that is not extended to form a flange, but it is welded to the shell. Like the A-type, it has the advantage that piping connections do not have to be broken to clean the inside of the tubes, but it does have the same disadvantage of the C-type for maintenance of the tube-to-tubesheet joints. Mechanical shellside cleaning is impossible.

D-type: the D-type in TEMA is used to describe a specially designed, non-bolted, closure for high pressure (>150 bar / 2100 psi). It is a generalized term since there are several such designs and some of them are patented. A common alternative for high-pressure exchangers is to use a B-type head, welded to the tubesheet, thus eliminating bolted joints. Providing the exchanger is large enough, access to the tubesheet may be achieved via a nozzle fitted with a manway cover.

Although not given a TEMA designation, **conical heads** are often used for exchangers with one pass on the tubeside. They consist of a single cone, flanged on both ends, the flange at the larger end being bolted to the tubesheet and the other flange being bolted to the piping.

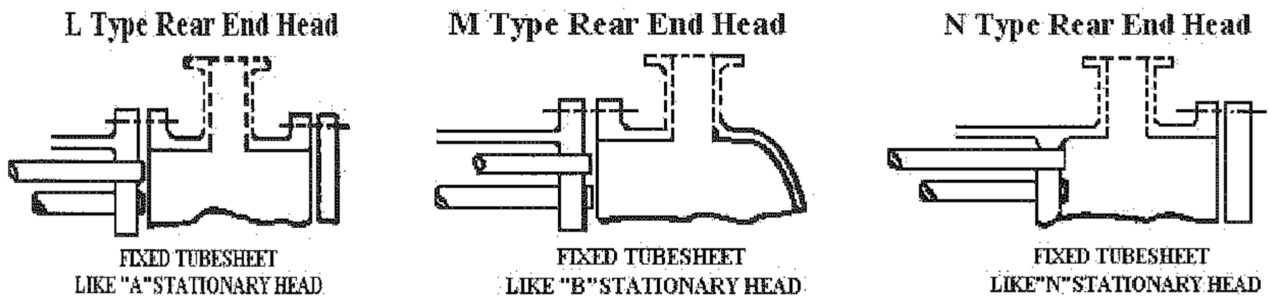
REAR HEAD TYPES

Although there are eight rear head types for a shell and tube heat exchanger designated by TEMA, in practice they correspond to three general types:

- Fixed tube sheet (L, M, N)
- U-tube
- Floating head (P, S, T, W)

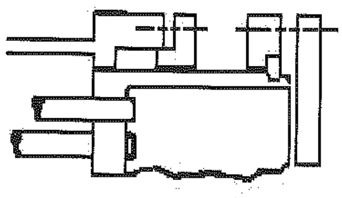
The choice of the rear head is primarily a mechanical design consideration, where it affects whether the bundle and the tubesheet are fixed or can be withdrawn from the shell for the mechanical cleaning. It can impact on the thermal design due to the clearance between the bundle and the shell.
For the exchangers operating at average pressures, the fixed tubesheet is the cheapest of the three and hence the most commonly used. At higher pressures, the U-tubes, which have only one tubesheet, are the cheapest type. Floating head types are more expensive and are used when fixed tubesheets or U-tubes exchangers cannot be accommodated. The various rear end types will be described. Note that the rear head will only be fitted with a nozzle if there is an odd number of tubeside passes.

The rear end is often referred to as the return head, particularly when there are two or more tube passes.



L-, M- and N-type: these are fixed tube sheet exchanger and correspond to an A, B and N-type front end heads. L and N types would normally only be used for single (or odd) tube-pass exchangers, where they permit access to the tubes without dismantling the connections. For exchangers with an even number of tubeside passes generally an M-type is considered.

P Type Rear End Head

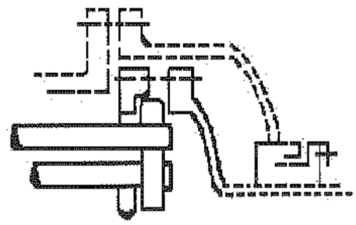


OUTSIDE PACKED FLOATING HEAD

P-type (outside packed floating head): the gap between the shell and the floating tubesheet is sealed by compressing packing material contained between the rear head and an extended shell flange, by means of a ring bolted to the latter. The packed joint is prone to leakage and is not suitable for hazardous or high-pressure service on the shellside.

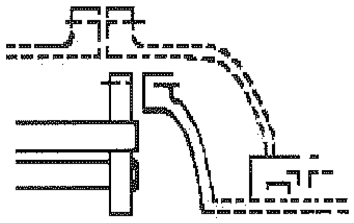
S-type (floating head with backing device): this type is usually referred to as a “split ring floating head” or sometimes abbreviated to SRFH. The backing ring is made in two halves to allow removal, so, the floating tubesheet can be pulled through the shell.

S Type Rear End Head



FLOATING HEAD WITH BACKING DEVICE

T Type Rear End Head

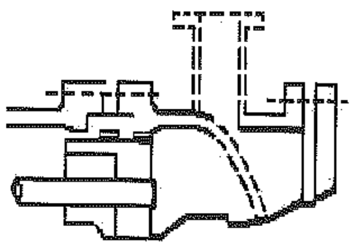


PULL THROUGH FLOATING HEAD

T-type (pull through floating head): this is referred to as “pull through”. Unlike the S-type, the rear end can be pulled through the shell without first having to remove the floating head. To achieve this, the shell diameter has to be greater than that of the corresponding S-type, making the T-type more expensive, except for kettle reboilers. This concept also affects the thermal design. T-type are easier to be dismantled than S-types. Because of the split backing ring, the rear tubesheet can be smaller than the one used with a T type, meaning the shell diameter is smaller. Although it is still larger than with fixed heads.

W-type (externally sealed floating tubesheet): sometimes referred to as an “O-ring” or “lantern ring” type due to the lantern ring seals between the floating tubesheet and the shell and channel respectively. The packed joints are almost certain to show some leakage and therefore are suitable for low pressure, non-hazardous fluids on the shell and tubeside.

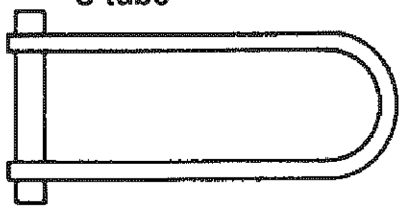
W Type Rear End Head



EXTERNALLY SEALED
FLOATING TUBESHEET

U-type (U-tube bundles): with the U or “hairpin” tubes only one tubesheet is required. Two pass U-tube units are also useful for handling tubeside two phase mixtures which could separate with consequent misdistribution in the return headers of two pass straight tube types. Attention must be payed if tubeside mechanical cleaning is required, because many company specifications do not consider U tubes as mechanically cleanable.

U tube



Summarizing:

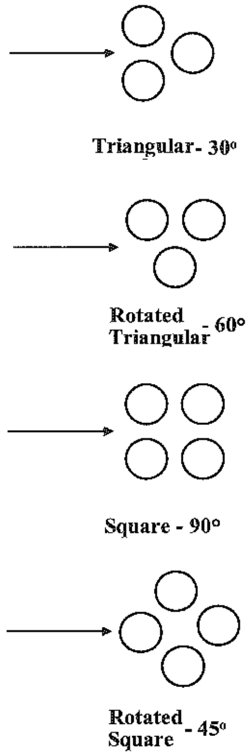
- Types of shell:
  - E- and F-shell are standards
  - G- and H-shell normally are only used for horizontal thermosyphon reboilers
  - J- and X-shell are employed if allowable pressure drop cannot be achieved in an E-shell
  - K-shell is only used as reboilers
- Types of front head
  - A-type is standards for dirty tubeside fluids
  - B-type is standards for clean tubeside fluids
  - C-type have to be considered for hazardous tubeside fluids, heavy tube bundles or frequent shellside cleaning
  - N-type must be considered for fixed tubesheet exchangers with hazardous shellside fluids
  - D-type (or bonnet welded to tubesheet) is used for high pressure
  - Conical heads have to be considered for single tube pass (axial nozzle)
- Types of rear head:
  - L, M and N are chosen if there is no overstressing due to differential expansion and shellside does not need mechanical cleaning
  - Fixed tubesheet with bellows can be selected if the shellside fluid is not hazardous, has low pressure (lower than 80 bar) and it does not lead to need of mechanical cleaning.
  - U-tube is used when countercurrent flow is not required (unless F shell) and the tubeside will not require mechanical cleaning.
  - S-type (Split Backing Ring Floating Head)
  - T, P, W floating head

3.3. CONSTRUCTION PECULIARITY

TUBE DIMENSION AND PATTERN

Inside TEMA standard it is possible to find the conventional lengths for each application case, the tube pitch and the number of recommended passes. Sometime particular attention must be payed selecting the tube pattern. The main concern is the consideration for shellside mechanical cleaning. Bundles with 90- or 45-degree layouts are much easier to clean with respect to a 30- or 60-degree layout, but a larger shell diameter is required to house the same number of tubes. Therefore, for removable bundles (which implies the need to be able to mechanically clean the shellside) it would normally be used the 45/90°. Instead, for non-removable bundles, the 30/60° would normally be used. Moreover, 45-degree layouts are slightly more prone to vibration problems (acoustic resonance) with gasses on the shellside.

As last remark, we can say that generally there is a little difference between 30 and 60 degrees and 45 and 90 degrees. Therefore, 30 and 90 degrees are normally used, while 60 and 45 can be adopted where the other configuration results in vibrational problem.



FINNED PIPES AND TUBE INSERTS

Sometime, in order to increase the shellside heat transfer coefficient U and reduce the required area, fin pipes are used. The pipes can be internally or externally finned, radially or longitudinally, low or high fin height. The finned tubes can be found on the market in a lot of configurations, but, especially if the fins are internal, an increment in pressure drop must be considered. The finned tubes theory will not be treated because the study case will not require this solution.

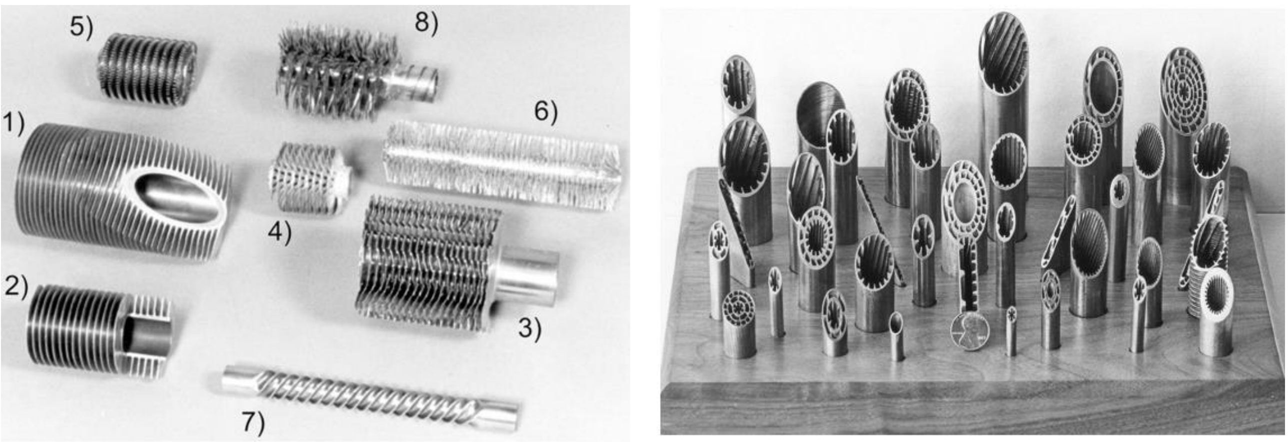
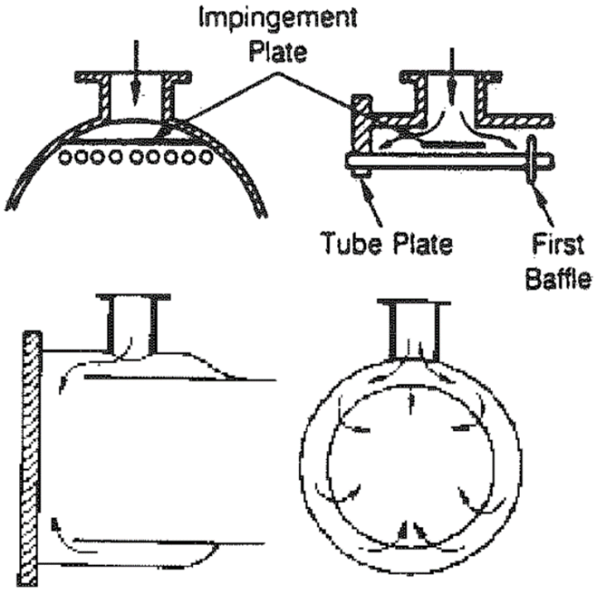


Figure 5: on the left, eight example of externally finned tubing: 1) and 2) typical commercial circular fins of constant thickness; 3) and 4) serrated circular fins and dimpled spirally-wound circular fins, both intended to improve convection; 5) spirally-wound copper coils outside and inside; 6) and 8) bristle fins, spirally wound and machined from base metal; 7) a spirally indented tube to improve convection and increase surface area. On the right an array of a commercial internally finned tubing. [5]

On the other hand, any obstacle placed inside a tube will cause a local increase in the fluid velocity. Some enhancement devices, for example the twisted insert, can increase the relative velocity between the fluid and the surface by imparting a rotational component to the fluid flow. Centrifugal force acts on the density gradients along the tube radius. For a fluid being heated, the centrifugal force tends to move the cooler dense fluid from the centre to the tube wall, enhancing heat transfer. For a fluid being cooled the opposite effect will occurs.

NOOZLES PROTECTION AND IMPINGEMENT PLATE

Generally, the nozzle sizes have to be as smaller as possible to keep costs down. However, it should be remembered that any pressure loss in the nozzle would be used more effectively in the shell or tubes. Checks should be made to ensure pressure drop is not wasted in the nozzles, where it could be used, for instance, to decrease the baffle pitch or to increase the number-of-tube passes in order to enhance tube transfer. Where pressure drop is not a consideration, then the maximum allowable fluid velocity usually limits the minimum nozzle size. This is a metallurgical problem since excessive velocities can lead to erosion, especially if the fluid contains solid in suspension. The velocities tolerated are much higher for gases than liquids and is more helpful to consider the ρv^2 value.



In order to protect the first tube row from high velocity fluid “jetting” onto the tube bundle, an impingement plate is placed just below the inlet nozzle of the shellside. It is usually a square or circular plate, solid or perforated, tack welded to the first tube row in the bundle.

Vapour belt, instead, are used for high volumetric gas flows to uniformly distribute the vapour around the bundle. The aim is to reduce the velocity at the inlet and hence to reduce the pressure drop and the likelihood of vibration and erosion. The example shown in the image consists of an outer annulus on the shell where itself acts as an impingement plate. The vapour flows circumferentially around and longitudinally towards the shell lip before entering the bundle region.

An alternative arrangement consists of an annulus placed around the shell, where the fluid enters into the exchanger through a series of slots placed circumferentially around the shell. In this case the open end of the shell is not needed as shown in the picture at bottom right. The last possible configuration is the impingement roads. They consist in rods bolted on two supports, or baffles, that have the same role of the impingement plate. However, they cause a lower pressure drop in the shellside.

As reference, some rules laid down by TEMA for the addition of impingement plates are reported:

- ρv^2 should not exceed 2232 kg/ms^2 for non-corrosive, non-abrasive, single phase fluids
- ρv^2 should not exceed 744 kg/ms^2 for liquids which are corrosive, abrasive, or at their boiling point
- Always required for saturated vapours and two-phase mixtures
- Shell or bundle entrance or exit area to be such that ρv^2 does not exceed 5953 kg/ms^2 [3]

BAFFLES

Baffles are installed on the shellside to give a higher heat transfer rate by means of the increment of the turbulence of the shellside fluid. They also support the tubes, thus reducing the chance of damage due to vibration. Most of the several different baffles types will be reported below. It must be observed that, beside increasing turbulence, baffles are also able to impose the right angles of flow with respect to the tubes, e.g cross flow.

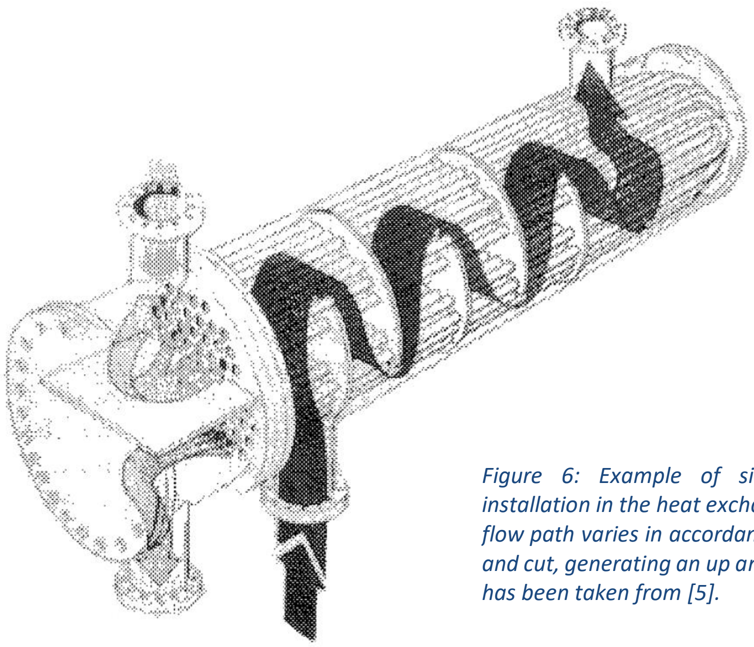
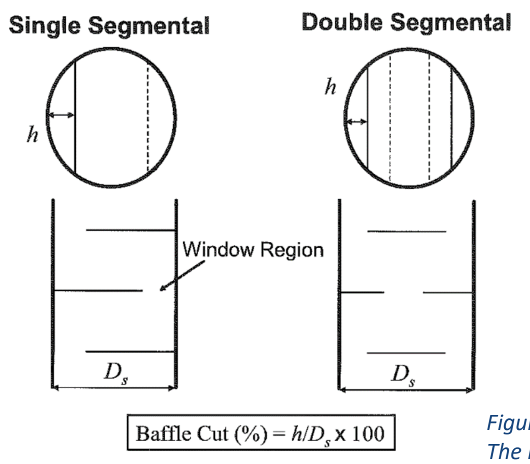


Figure 6: Example of single segmental baffle installation in the heat exchanger shell. The shellside flow path varies in accordance to the baffle position and cut, generating an up and down flow. The image has been taken from [5].

Single segmental: this is the most common baffles type. They may be arranged to provide side-flow (e.g. horizontal condenser) or up and over-flow (e.g. single pass units). The baffle cut normally ranges from 15 to 45%.

Double segmental: these baffles are normally used when there is a requirement for a low shellside pressure drop which cannot be met by a single segmental baffle even with a large baffle cut (e.g. 45%). The lower pressure drop is achieved by splitting the shellside flow into two paths through the exchanger. The baffle cut is normally in the range of 15 to 25%.



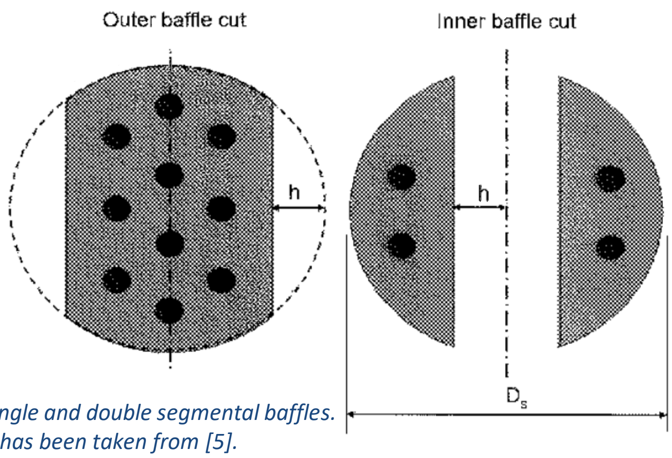
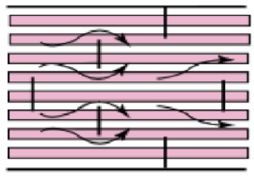


Figure 7: Single and double segmental baffles. The image has been taken from [5].

Orifice baffle: here, there is sufficient clearance between the tube and the baffle hole to allow flow past the baffle without excessive pressure drop. The baffles do not support the tubes, or at least provide limited support to the few tubes they touch. This arrangement should either be used with vertical tubes or some of the baffle raised to press the tubes against the sides of the holes (i.e., baffle offset so touch tubes support).

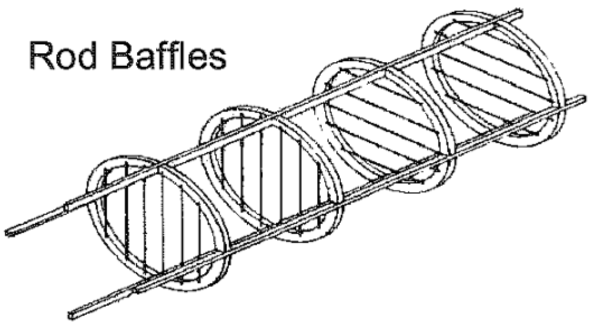
Helical baffle: in this design the baffles are not positioned perpendicular to the shell wall. Indeed, they are quadrants at an angle such that the shellside flow follows a spiral path along the heat exchanger. This design has the advantage of giving a lower shellside pressure drop and reducing the susceptibility to fouling. However, the velocity at the centre of the shell is very low and basically longitudinal along the exchanger. At the outside of the bundle, instead, the flow velocity is relatively high. This leads to a significant difference in heat transfer between the fluid at the centre and the outside of the bundle.



Triple segmental: triple segmental baffles result in a low crossflow, such that the resulting flow is mainly longitudinal. The pressure drop is therefore very low. [4]

Disc and doughnut baffle: similar to the double segmental baffle they are primarily used for process with a low side pressure drop. However, in practice it is not used as much as the double segmental.

Rod baffle: this is a technique for supporting tubes with a matrix of rods instead of the conventional perforated baffle plate. One Rod Baffle consists of a set of rods welded to a ring of diameter just greater than the bundle. A baffle sets consist of four of these baffles, spaced along the exchanger axis, providing positive 4-point support of the tubes. An exchanger will then have a number of these baffle set according to the tube length.



The rods are arranged so that there are two rows of tubes between each pair of rods in the baffle, with the next baffle being offset by one row of tubes. The rods pass between the tubes row with minimal clearance, so that a rod passing a tube provides support.

This baffle was designed principally to eliminate tube damage due to vibration, since it gives primarily axis flow along the bundle, rather than crossflow.

No-tubes-in-window: this refers to single segmental baffles which have no tubes I the so-called “window” left by the baffle cut. As it can be seen in the Figure 8, every tube is therefore supported by every baffle, unlike tubes-in-window setup where tubes in the window region are only supported by every other baffle. This lead to have a lower length of the tubes not supported and to reduce the risk of vibration problems. Intermediate baffles can also be inserted if a further modification of the vibrational modes is required.

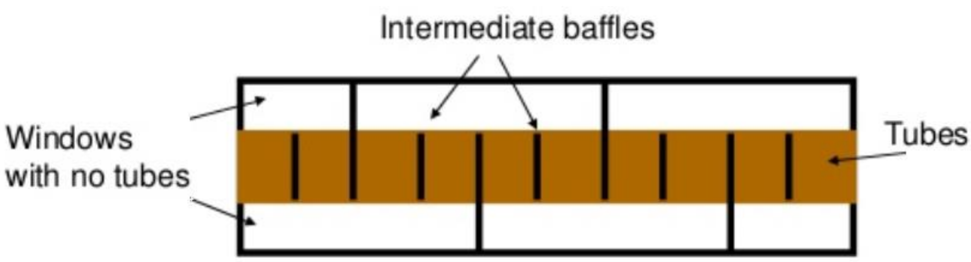


Figure 8: No tubes in the windows - with intermediate support baffles.

BAFFLE CUT ORIENTATION

The term 'Orientation' means the position of the cut edge of the baffle with respect to the shell inlet nozzle centre line. Even if any orientation is theoretically possible, the two are generally used are:

- Baffle cut parallel to shell nozzle centre line (0 degree cut angle, side-to-side flow).
- Baffle cut perpendicular to shell nozzle centre line (90 degree cut angle, up-and-down flow).

For horizontal shells with top or bottom inlet nozzles, 0 degree cut angle is generally referred to as "vertical cut" (or side-to-side flow) and 90 degree cut angle as "horizontal cut" (or up-and-down flow). In case of horizontal condensers, 0 degree cut angle (vertical cut) is often chosen since its use allow reasonable liquid-vapour separation. In most other cases, including all vertically mounted exchangers, the preferred arrangement is 90 degree cut angle since the use of 0-degree results in a larger bypass area and require the installation of sealing strips.

The maximum permissible baffle cut instead is determined by consideration of the maximum allowable unsupported span since, above a given cut, some of the tubes in the centre of the bundle will not be fully supported by any of the baffles. The exact value of the maximum possible cut depends on the geometry of the tube bundle, but is usually taken as 45% for single segmental, and 25% for double segmental. Baffle cuts below the maximum are usually chosen such that the free flow area in the baffle window is roughly equal to the crossflow area at the exchanger centre line since it avoids excessive turn-around pressure losses. Small baffle cuts can lead to poor shellside flow distribution and minimum values of 15% (tubes in window) and 10% (no-tubes-in-window) are recommended. [5]

BAFFLE SPACING

If the calculated shellside pressure drop exceeds the maximum allowable the design engineer will usually increase the baffle spacing and/or baffle cut until the pressure drop is reduced to an acceptable value. As the baffle spacing is increased, however, the resulting larger unsupported tube span renders the tubes increasingly susceptible to damage due to sagging or flow induced vibration. Inside TEMA standards there are reported some maximum unsupported length for various tube size and materials, but they should in no way be regarded as a safe limit for avoiding flow induced vibration.

For baffles with no tubes in the window, there is no theoretical limit on the baffle spacing. This because intermediate supports can be employed to reduce the unsupported span to any required value. If, however, baffle spacing with no tubes in the window is increased to the point where only one baffle is possible, then the design engineer should consider using 'rod baffle' design instead.

Not only a small baffle cut, as already discussed, can lead to have a poor shellside flow distribution, but also a small baffle spacing. In the days before computer aided shellside flow analysis, the minimum spacing traditionally used was one fifth of the shell diameter. Now the design engineer can check whether low baffle spacing are going to lead in turn to an excessively low crossflow fraction and act accordingly. Note that an exchanger with large number of closely spaced baffles could be difficult to fabricate. For this reason, TEMA standard also recommends an absolute minimum space between the baffle. As usual, spacing lower than the specified can be used, especially in the small heat exchanger, if the installation requirements are taken into account.

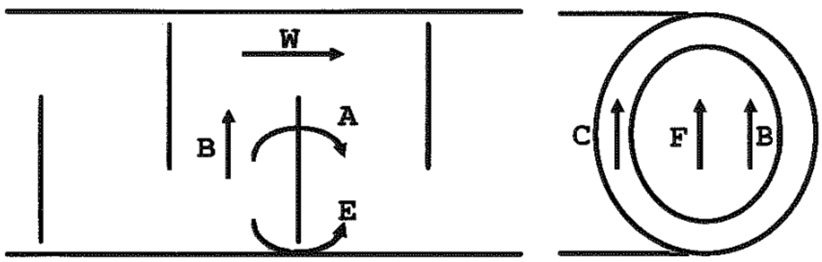
CLEARANCE, LEAKAGES AND BYPASS

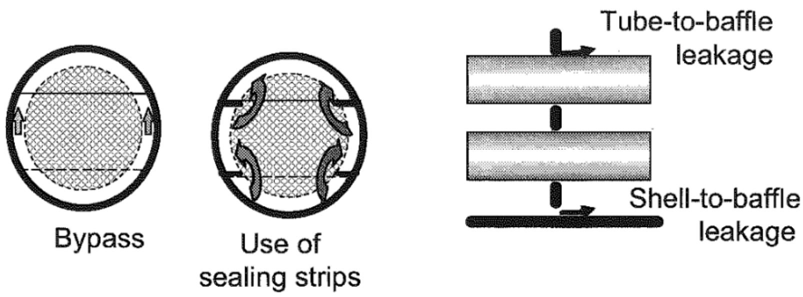
Clearances between various components of a heat exchanger are required for their construction, even if their presence lead to have leakages and bypass.

Leakage and bypass reduce the cross flow and hence lower the heat exchange coefficient. They also cause axial mixing which may reduce the LMTD with close temperature approach. Sealing strips often are used to reduce bypass, while the computer programmes nowadays estimate how much flow will pass along the various leakage ad by-pass path.

In particular, we can have:

A) Baffle hole – tube OD
B) Crossflow
C) Shell ID – bundle OTL
E) Baffle OD – shell ID
F) Pass lanes
W) Window





NOTE ON THE REMOVABLE BUNDLE

An important requirement that can be foreseen is the removable bundle. This configuration choice would have the following advantages:

- the external tubes surface will be easy to clean after a disassembly procedure;
- because the tubes are not bonded on both sides, the necessary thickness of the stationary tubesheet is reduced. This lead to have less internal stress on the internal pipe and a considerable reduction in the cost of the heat exchanger.

Contrariwise, we can have the below reported disadvantages:

- The internal tubes surface cannot be clean in the classical U shape removable bundle;
- A removable bundle requires a flanged joint in order to merge the stationary tubesheet with the stationary head and the shell. This flanged joint is one of the most critical part of the heat exchanger with removable bundle due to the higher stress induced by the thermal and pressure gradient. These gradients are generated from the necessity to have the inlet and outlet nozzles of the piping fluid on the same stationary head, as a consequence of the U shape of the bundle.

4. CONCEPTS OF HEAT TRANSFER AND THERMAL DESIGN

4.1. FOUNDAMENTALS OF CONDUCTION

4.1.1. FOURIER LAW AND GENERAL EQUATION FOR THERMAL CONDUCTION

Considering a three-dimensional body as in Figure 9, the general space- and time-dependent **temperature distribution field T** is represented by a scalar $T = T(x, y, z, t)$ or $T(\vec{r}, t)$ and defines instantaneous isothermal surfaces: T_1, T_2, and so on.

In association with the scalar field T a very important vector, called **temperature gradient ∇T**, is also considered. It has magnitude and direction of the maximum increase of temperature at each point, and it is defined as:

$$\nabla T \equiv \vec{\imath}\frac{\partial T}{\partial x} + \vec{\jmath}\frac{\partial T}{\partial y} + \vec{k}\frac{\partial T}{\partial z}$$

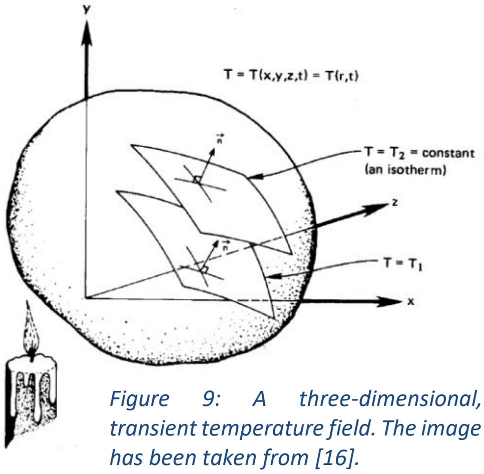


Figure 9: A three-dimensional, transient temperature field. The image has been taken from [16].

In order to analyze the thermal exchange that occur between the body and the environment we need a phenomenological equation that links the **heat flux exchanged by the surface, q** (W/m^2), with the temperature in every point. This empirical equation is called **Fourier's Law**, can be written using the three space components, and take into account the material of the body by means of a constant λ.

$$\boxed{\bar{q} = (q_x, q_y, q_z) = -\lambda \nabla T}$$

The heat flux is so defined as a vector quantity, proportional to the magnitude of the temperature gradient and opposite to it in sign. The previous equation resolves itself in three components:

$$q_x = -\lambda_x \frac{\partial T}{\partial x} \qquad q_y = -\lambda_y \frac{\partial T}{\partial y} \qquad q_z = -\lambda_z \frac{\partial T}{\partial z}$$

The constant, λ or k, is called the **thermal conductivity** $[W/mK]$ and depends on temperature, position and direction. Because most materials are very nearly homogeneous and most of the time isotropic, the assumption of k=constant is generally taken. However, in each case the reliability of this assumption must be proven assessing whether or not λ is approximatively constant in the range of interest.

In the next page the approximate range and the temperature dependence of thermal conductivity will be showed for various substances. In the Figure 10 the thermal conductivity of some representative substance is reported in scale. Matter with lower thermal conductivity are classified as insulant material, while the more conductive substances, that can be seen on the right, presents a value of λ many orders of magnitude larger than insulants. In the Figure 11, instead, the dependence of λ from the temperature can be approximately seen. A preliminary consideration of the material behavior can be done. As a matter of fact, a constant thermal conductivity could be considered for most of the materials in a small high temperature rage, while its value changes a lot for negative temperature range.

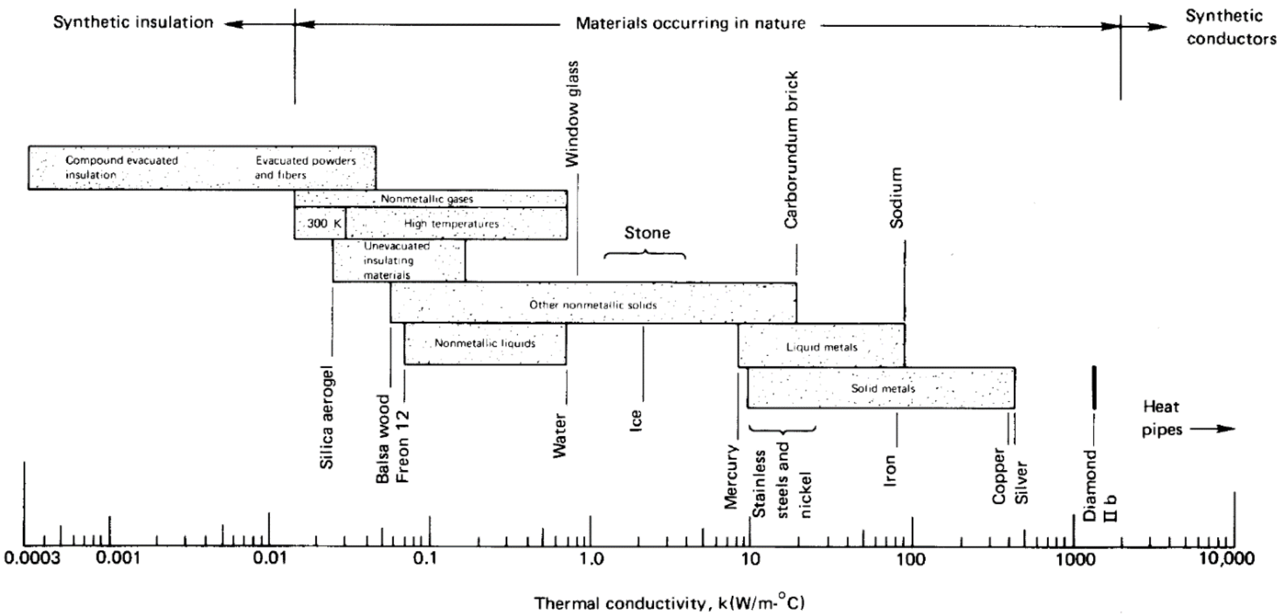


Figure 10: The approximate ranges of thermal conductivity of various substances. [16]

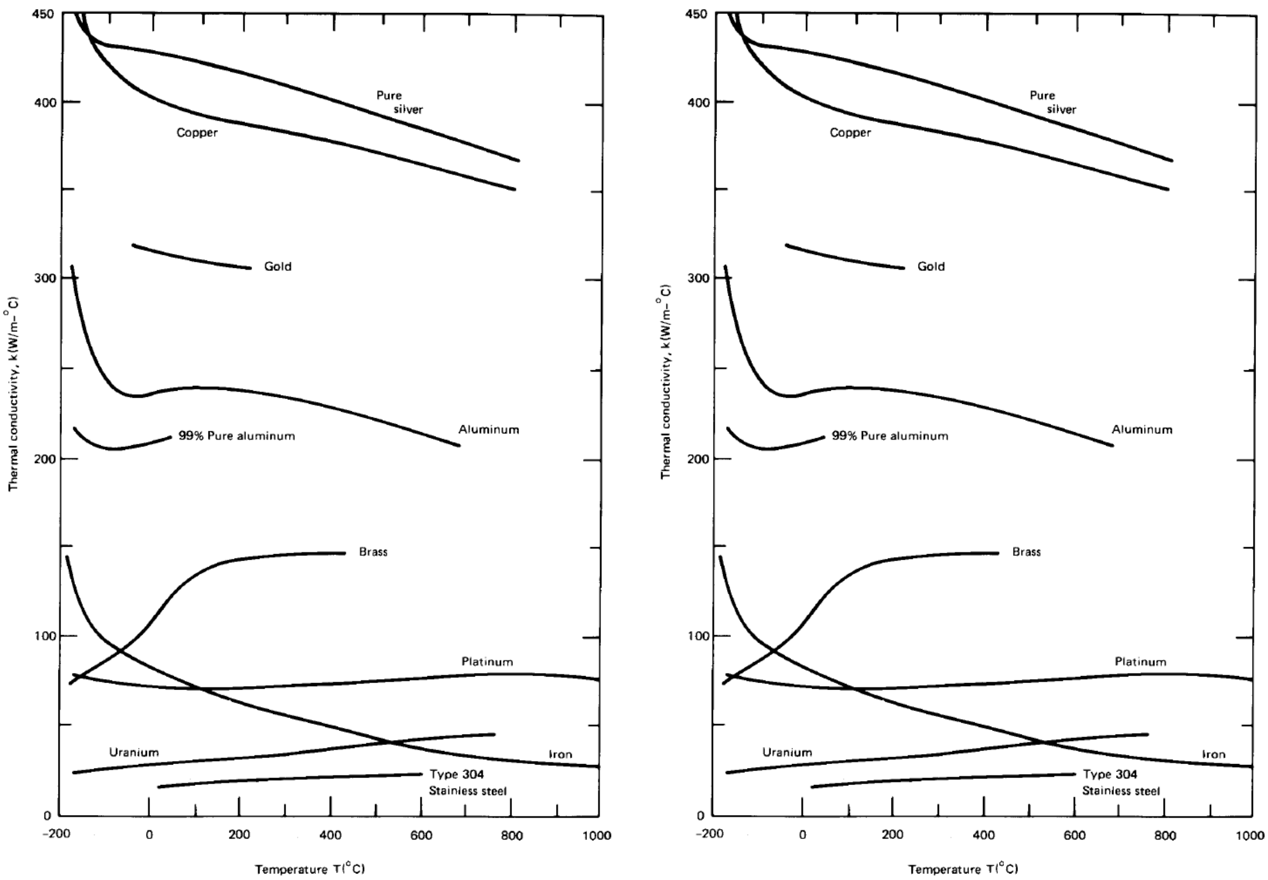


Figure 11: Temperature dependence of thermal conductivity of metallic solids (left) and liquid and gases (saturated or at 1atm, right). [16]

Given that two dependent variables, T and $\bar{q}$, are involved in the Fourier's law, some passages are required to find a solution. To eliminate $\bar{q}$, and first solve for T, the First Law of Thermodynamics is introduced.

Applying it on a three-dimensional control volume, as shown in Figure 12, an element of the surface dS is identified and two vectors are shown on it. One is the unit normal vector, $\vec{n}\ (with\ |\vec{n}| = 1)$, and the other is the heat flux vector, $\bar{q} = -\lambda \nabla T$, at a point on the surface. A **volumetric heat release** $\dot{\boldsymbol{q}}(\vec{\boldsymbol{r}})\ \boldsymbol{W/m^3}$ is distributed through the region. This might be the result of chemical or nuclear reaction, of external radiation into the region, of electrical resistance heating or of still other causes.

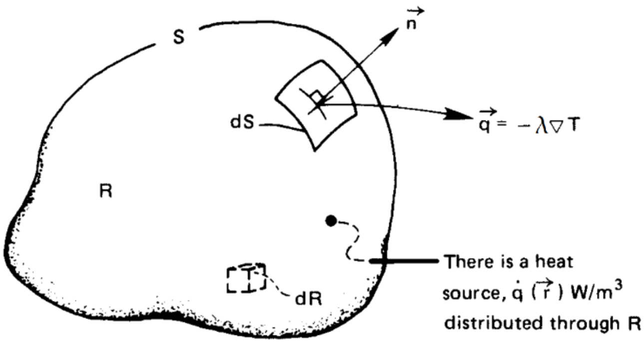


Figure 12: Control volume in a heat4-flow field. [16]

The **total heat flux Q** in Watts can be written as the sum of the heat flux conduced out of dS and the volumetric heat flux generated (or consumed) within the region dR, integrated in the surface and in the volume respectively.

$$Q = -\int_S (-\lambda \nabla T) \cdot (\vec{n} dS) + \int_R \dot{q}\, dR$$

Writing the First Law of Thermodynamic the rate of internal energy of the region R can be modified and expressed as:

$$Q = \frac{dU}{dt} = \int_R \rho c \left(\frac{\partial T(\vec{r}, t)}{\partial t} \right) dR$$

Combining the last two equations we get an expression in which a surface integral must be converted into a volumetric one by means of the Gauss's theorem. Because the region R is arbitrary, also the integrand inside the brackets of the equation obtained must be equal to zero.

$$\int_S \lambda \nabla T \cdot \vec{n} dS = \int_R \left[\rho c \frac{\partial T}{\partial t} - \dot{q} \right] dR \quad \rightarrow \quad \int_R \left(\nabla \cdot \lambda \nabla T - \rho c \frac{\partial T}{\partial t} + \dot{q} \right) dR = 0 \quad \rightarrow \quad \nabla \cdot \lambda \nabla T + \dot{q} = \rho c \frac{\partial T}{\partial t}$$

Re-writing it and introducing the ***thermal diffusivity*** $\boldsymbol{a = \lambda/\rho c\ [m^2/s]}$ we get the more complete version of the three-dimensional **heat diffusion equation**:

$$\boxed{\nabla^2 T + \frac{\dot{q}}{\lambda} = \frac{\rho c}{\lambda} \frac{\partial T}{\partial x} = \frac{1}{a} \frac{\partial T}{\partial t}}$$

Where $\nabla^2 T$ is called Laplacian and, in the various coordinate systems, can be expressed as

$$Cartesian\ coordinate \qquad \nabla^2 T = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}$$

$$Cylindrical\ coordinate \qquad \nabla^2 T = \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 T}{\partial \vartheta^2} + \frac{\partial^2 T}{\partial z^2}$$

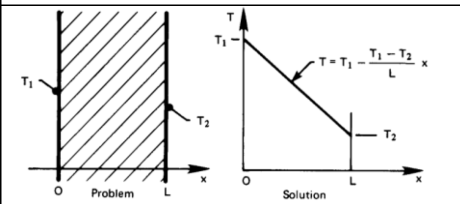
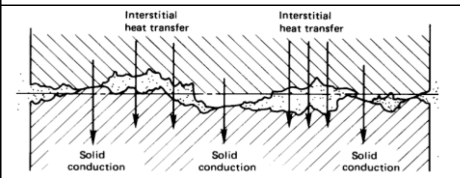
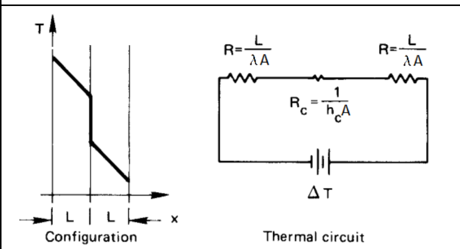
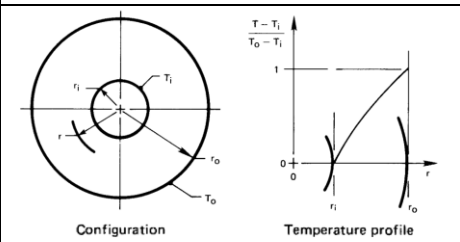
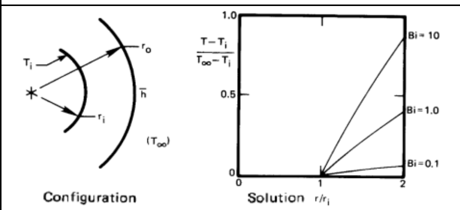
$$Spherical\ coordinate \qquad \nabla^2 T = \frac{1}{r} \frac{\partial}{\partial r} \left(r^2 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin \vartheta} \frac{\partial}{\partial \vartheta} \left(\sin \vartheta \frac{\partial T}{\partial \vartheta} \right) + \frac{1}{r^2 \sin^2 \vartheta} \frac{\partial^2 T}{\partial ø^2}$$

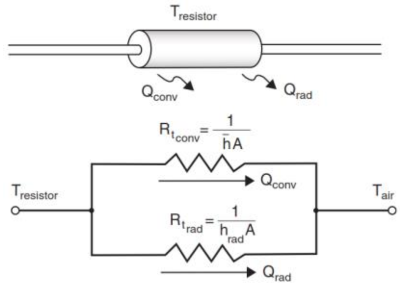
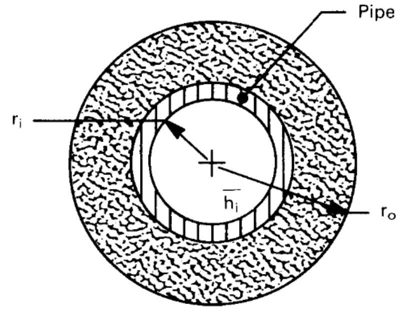
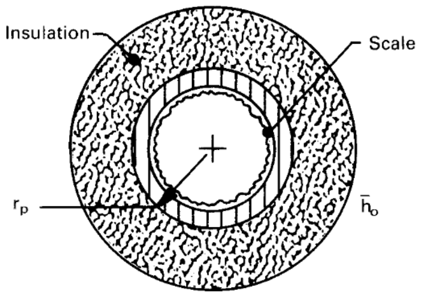
4.1.2. SOLUTION OF THE HEAT EQUATION AND THERMAL RESISTANCE VALUES

In every case of application, at first, the temperature field is computed integrating the heat equation. Then, if also the heat flux has to be calculated, T is differentiated and inserted in the Fourier's Law to get q. Q, expressed in Watt, can be simply obtained multiplying q for the respective area A, e.g. the formula on the right. Comparing the shape of the last formula obtained, with the Ohm's Law, a similitude can be noticed.

$$\begin{matrix} Ohm's \\ law \end{matrix} \quad I = \frac{E}{R} \qquad\qquad \begin{matrix} example\ of\ heat \\ conduction\ solution \end{matrix} \quad Q = \frac{\Delta T}{L/_{\lambda A}} = \frac{\Delta T}{R_t}$$

By means of this procedure, called electrical analogy in which Q=I and ΔT=E, we are able to define the **specific thermal resistance Rt $[K/W]$.** A brief summary of some standard case is reported in the table below.

	IMAGE	THERMAL RESISTANCE [K/W]	NOTES
SINGLE SLAB		$R_t = \frac{L}{\lambda A}$	The resistance is inversely proportional to λ and it depends by the geometry of the problem.
INTERSTICES		$R_t = \frac{1}{h_c A}$	The interfacial conductance $h_c\ [W/m^2\ K]$ describe the behavior of the gas filled interstices present between two solids in contact.
DOUBLE SLABS		$R_t = \frac{L}{\lambda A} + \frac{1}{h_c A} + \frac{L}{\lambda A}$	The total thermal resistance of the case can be computed by the sum of all the resistances present inside the configuration. So, in the final formula, two resistance for the slabs and one for the interstices are present.
CYLINDRICAL TUBE		$R_t = \frac{ln(^{r_o}/_{r_i})}{2\pi l\lambda}$	Comparing to the linear case of a single slab, the resistance is still inversely proportional to λ, but it reflects a different geometry, since A=2πrl.
CONVECTION		$R_t = R_{cond} + R_{conv} = \frac{ln(^{r_o}/_{r_i})}{2\pi l\lambda} + \frac{1}{\bar{h}A}$	The presence of convection on the outside of a cylinder causes a new thermal resistance that must be added at the tube resistance.

THERMAL RADIATION		$R_t = \frac{1}{h_{rad}A}$	Given the radiation Q exchanged by two objects and the radiation heat transfer coefficient, $h_{rad} = 4\sigma T_m^3 F_{1-2}$, the radiation thermal resistance has the same shape of the convective one. The symbols F_{1-2} is the view factor from surface 1 to surface 2.
FOULING RESISTANCE	New pipe Older pipe (after some use)	The empirical fouling resistance Rf must be summed at the others in order to take into account the resistance of the layer of scale that has been formed with the time. $R_t = R_f$	

Fluid and Situation	*Fouling Resistance* R_f (m^2K/W)
Distilled water	0.0001
Seawater	0.0001 – 0.0004
Treated boiler feedwater	0.0001 – 0.0002
Clean river or lake water	0.0002 – 0.0006
About the worst waters used in heat exchangers	< 0.0020
No. 6 fuel oil	0.0001
Transformer or lubricating oil	0.0002
Most industrial liquids	0.0002
Most refinery liquids	0.0002 – 0.0009
Steam, non-oil-bearing	0.0001
Steam, oil-bearing (e.g., turbine exhaust)	0.0003
Most stable gases	0.0002 – 0.0004
Flue gases	0.0010 – 0.0020
Refrigerant vapors (oil-bearing)	0.0040

Introducing this concept, we can write the total heat flux Q by means of the **overall heat transfer coefficient** **U** $[\boldsymbol{W/m^2K}]$ as:

$$Q = UA\Delta T \qquad where \qquad U = \frac{1}{A\sum R_t} = \frac{1}{\left[\left(\frac{1}{h_o} + r_{f,o}\right)\left(\frac{1}{E_f}\right) + r_w + r_{f,i}\left(\frac{A_0}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_0}{A_i}\right)\right]}$$

The overall heat transfer coefficient is very convenient because based on the composite resistance concepts. However, attention must be paid at which area is referred to. The reference area of U, and so the reference areas of all the thermal resistances that compound it, must be the same area used for the computation of Q, so, the formula $Q = UA\Delta T$ must be consistent. The last equality is the application to a tube of a general heat exchanger. E_f is the fin efficiency of the tubes if this construction peculiarity is present.

As already said, in a fairly general use of the word, a heat exchanger is anything that lies between two fluid masses at different temperatures. In this sense a heat exchanger might be designed either to impede or to enhance heat exchange. If the heat exchanger is intended to improve heat exchange, U will generally be much greater than 40 (W/m^2K). If it is intended to impede heat flow, it will be less than 10 (W/m^2K).

4.2. HEAT CONVECTION AND HEAT TRANSFER COEFFICIENT COMPUTATION

The convection is a process in which a moving fluid carry heat away from a warm body with which it is in contact. When cold air moves on a warm body, it constantly sweeps away the one that has been warmed by the body and replaces it with cold air. The aim of the heat convection analysis is to predict the heat transfer coefficients h and ħ, $[W/m^2\,K]$. Their prediction is necessary, in order to find out the heat exchange, when it happens in a system in which the motion of a fluid around the body is present. So, in the cases in which there is the replacement of hot fluid with cold, or vice versa.

The first step before h can be predicted is the mathematical description of the **boundary layer**. Because fluids flowing past solid bodies adhere to them, a region of variable velocity must be defined between the body and the free fluid stream. This region, called boundary layer, has thickness δ and it is arbitrarily defined as the distance from the wall at which the flow velocity approaches to within 1% of u_∞. The dimensional functional equation for the boundary layer thickness, on flat surface, is:

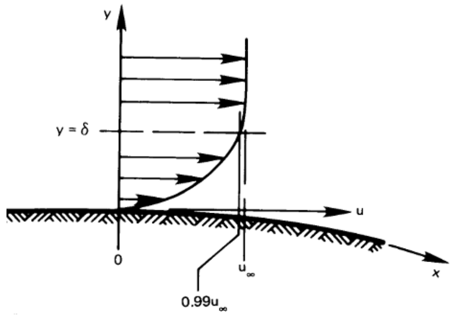


Figure 13: A boundary layer of thickness δ. [16]

$$\delta = fn(u_\infty, \rho, \mu, x)$$

Where x is the length along the surface, ρ is the fluid density in kg/m^3 and μ is the dynamic viscosity in $kg/m \cdot s$.

If the wall is at temperature T_w, different from that on the free stream T_∞, also a **thermal boundary layer** of thickness δ_t can be defined. Equating the heat removed from the wall by the fluid, using the Fourier's law of conduction, at the same heat transfer expressed in terms of convective heat transfer coefficient, the follow equation can be written:

$$-\lambda_f \left.\frac{\partial T}{\partial y}\right|_{y=0} = h(T_w - T_\infty)$$

where λ_f is the conductivity of the fluid. It can be noticed that this condition defines h within the fluid instead of specifying it as known information on the boundary. This equation can be rearranged using L, the characteristic dimension of the body under consideration, in the form:

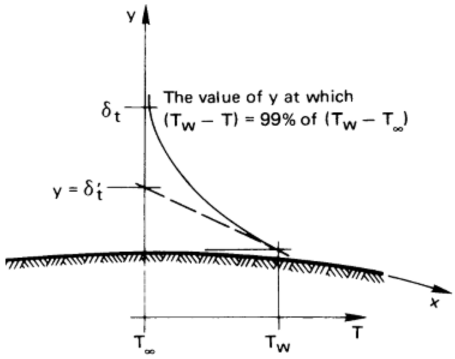


Figure 14: The thermal boundary layer during the flow of cold fluid over a warm plate. [16]

$$\left.\frac{\partial\left(\frac{T_w - T}{T_w - T_\infty}\right)}{\partial(y/L)}\right|_{y/L=0} = \frac{hL}{\lambda_f} = Nu_L$$

In the previous formula Nu_L is the Nusselt number. It that can be rewritten expressing its physical meaning, where it ends up to be proportional to the thickness of the thermal boundary layer.

$$Nu_L = \frac{L}{\delta_t'}$$

Now that the kinematic and thermal boundary layers are defined, it is possible predict the flow field. It can be found by solving the equations that express conservation of mass and momentum written for the defined regions. Because this theoretical part is not the principals subject of this thesis, the derivations of these formulas are not reported. What follow is the two-dimensional continuity equation for incompressible flow and the two forms of momentum equation for the steady, two-dimensional and incompressible flow.

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0$$

$$\frac{\partial u^2}{\partial x}+\frac{\partial uv}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\nu\frac{\partial u^2}{\partial y^2} \qquad or \qquad u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\nu\frac{\partial^2 u}{\partial y^2}$$

A complete derivation of the last equation would reveal that the result is valid for compressible flow as well. Therefore, if there is no pressure gradient in the flow, it can be rewritten as:

$$\frac{\partial u^2}{\partial x}+\frac{\partial uv}{\partial y}=u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=\nu\frac{\partial^2 u}{\partial y^2}$$

This equation can be solved by means of the introduction of a stream function, $\psi(x,y)$, that allows to reduce the number of dependent variables, or by means of the momentum integral method. The first procedure lead to the Exact solution that, for a flat surface where u_∞ remains constant, can be defined as:

$$\frac{\delta}{x}=\frac{4.92}{\sqrt{u_\infty x/\nu}}=\frac{4.92}{\sqrt{Re_x}}$$

Re_x is called Reynolds number and it characterizes the relative influences of the inertial and viscous forces in a fluid problem. This formula means that if the velocity is great or the viscosity is low, δ/x will be relatively small, and the heat transfer will be relatively high.

The second one, instead, is an approximated method that simplify the solution given the boundary layers similarity, and leads to a

$$\frac{\delta}{x}=\frac{4.64}{\sqrt{Re_x}}$$

The exact solution for $u(x,y)$ reveals that *u* can be expressed as a function of a single variable, *η*, and it leads to what it is called the similarity solution:

$$\frac{u}{u_\infty}=f\left(y\sqrt{\frac{u_\infty}{\nu x}}\right)=f(\eta)$$

Now that the flow field in the boundary layer has been determined, the heat conduction equation must be extended to take into account the motion of the fluid. Starting from the conservation of energy, it is possible to write the energy equation for incompressible fluids reported on the left. After, it can be modified for a two-dimensional flow without heat sources in a boundary layer, obtaining the equation on the right.

$$\rho c_p\left(\frac{\partial T}{\partial t}+\vec{u}\cdot\nabla T\right)=k\nabla^2 T+\dot{q} \qquad and \qquad u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\frac{\partial^2 T}{\partial t^2}$$

The temperature field in the boundary layer is the solution of the equation on the right and it can be inserted in the Fourier's law in order to compute *h* as follow

$$h=\frac{q}{T_w-T_\infty}=-\frac{\lambda}{T_w-T_\infty}\frac{\partial T}{\partial y}\bigg|_{y=0}$$

4.3. LOGARITMIC MEAN TEMPERATURE DIFFERENCE AND CORRECTION FACTOR

In general cases U vary with the position in the exchanger and with the local temperature. At first, we can suppose it as a constant value, still reasonable assumption in compact single-phase heat exchangers. In these cases, the overall heat transfer can be written in term of mean temperature difference between the two fluid streams:

$$Q = UA\Delta T_{mean}$$

The problem then reduces to finding the appropriate mean temperature that will make this equation true. Analysing the simple parallel and counterflow configurations, the temperature variation can be plotted for both, as reported in Figure 15. Notice that temperatures change more rapidly in the parallel-flow configuration and so less length is required. However, the counterflow arrangement achieves generally more complete heat exchange from one flow to the other. The determination of ΔT_{mean} for such arrangements proceeds as follows. Defining the differential heat transfer within either arrangement as:

$$dQ = U\Delta T dA = -\left(\dot{m}c_p\right)_h dT_h = \pm\left(\dot{m}c_p\right)_c dT_c$$

Introducing the total heat capacity of the hot and cold streams, $C_h = \left(\dot{m}c_p\right)_h$ and $C_c = \left(\dot{m}c_p\right)_c$ [W/K], we find the below reported equation. If we integrate it for both cases, parallel and counterflow, it gives us the relations between the hot and cold fluid temperature in every point:

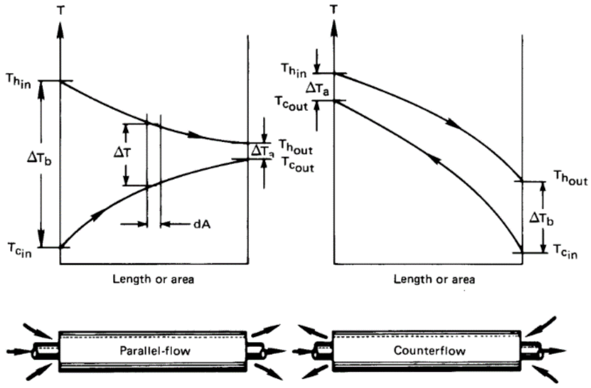


Figure 15: Temperature variation through parallel and counterflow single-pass heat exchanger. [16]

$$\mp C_h dT_h = C_c dT_c \quad \rightarrow \quad \begin{cases} parallel\ flow\ [-C_h] \quad B.C.:\ T_h = T_{h,in} \quad T_c = T_{c,in} \quad \rightarrow \quad T_h = T_{h,in} - \frac{C_c}{C_h}\left(T_c - T_{c,in}\right) = T_{h,in} - \frac{Q}{C_h} \\ counterflow\ [+C_h] \quad B.C.:\ T_h = T_{h,in} \quad T_c = T_{c,out} \quad \rightarrow \quad T_h = T_{h,in} - \frac{C_c}{C_h}\left(T_{c,out} - T_c\right) = T_{h,in} - \frac{Q}{C_h} \end{cases}$$

Where Q is the total heat transfer from the entrance to the point of interest. The above equations can be solved for the local temperature differences:

$$\Delta T_{parallel} = T_h - T_c = T_{h,in} - \left(1 + \frac{C_c}{C_h}\right)T_c + \frac{C_c}{C_h}T_{c,in}$$

$$\Delta T_{counter} = T_h - T_c = T_{h,in} - \left(1 + \frac{C_c}{C_h}\right)T_c - \frac{C_c}{C_h}T_{c,in}$$

Replacing the founded formulas in $dQ = C_c dT_c = U\Delta T dA$ and making further integration, shall be obtained:

$$Q = UA\left(\frac{\Delta T_a - \Delta T_b}{\ln\left(\Delta T_a/\Delta T_b\right)}\right)$$

The ΔT_{mean} for use in the initial equation is thus the logarithmic mean temperature difference (LMTD):

$$\Delta T_{mean} = LMTD = \frac{\Delta T_a - \Delta T_b}{\ln\left(\frac{\Delta T_a}{\Delta T_b}\right)}$$

It must be observed than, if U were variable the integration of the total heat transfer Q would give a different expression as result. This means that, as already said, in most of the cases the variability of U would have to be considered and it changes the value of the ΔT_{mean}. As example of this phenomenon, the vaporization phase showed in Figure 16 is a typical situation in which the variation of U within a heat exchanger might be great, and so it cannot be considered constant. The figure below shows a typical situation, e.g. vaporization phase, in which the variation of U within a heat exchanger might be great. In this case, the heat exchange mechanism on the water side is completely altered when the liquid is finally boiled away. If U where uniform in each portion of the heat exchanger, then we could treat it as two different exchangers in series. However, in the heat exchangers in which U varies continuously with position, as in large industrial shell-and-tube configurations, the analysis presented must be done using an average value of U, defined as:

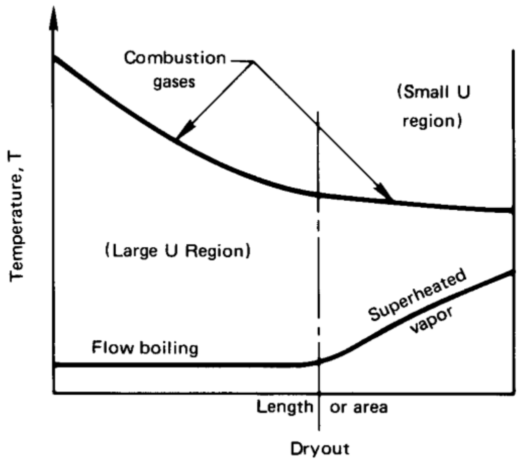


Figure 16: A typical case of a heat exchanger in which U varies dramatically. [16]

$$\overline{U} = \frac{\int_0^A U dA}{A}$$

Another limitation in the use of an LMTD is its restriction to single-pass parallel and counterflow configurations. As a matter of fact, each configuration must be analysed separately and the results are generally more complicated than the expression reported in the previous investigation. All the calculations for every possible configurations have been already done in 1940 by Bowman, Muller and Nagle, and after by TEMA. Now a day, thanks to the organization of these data, the restriction given from different layout can be simply overcome multiplying the LMTD by a coefficient F.

$$Q = UA(LMTD) \cdot F(P, R)$$

Where:

$$P = \frac{T_{t,out} - T_{t,in}}{T_{s,in} - T_{t,in}} = \begin{matrix} relative\ influence\ of\ the\ overall\ temperature\ difference\ (T_{s,in} - T_{t,in}) \\ on\ the\ tube\ flow\ temperature. It\ must\ obviously\ be\ less\ than\ unity. \end{matrix}$$

$$R = \frac{T_{s,in} - T_{s,out}}{T_{t,out} - T_{t,in}} = equals\ the\ heat\ capacity\ ratio\ \ C_t/C_s$$

$$T_t = temperature\ of\ the\ tube\ flow$$

$$T_s = temperature\ of\ the\ shell\ flow$$

Therefore, the factor F is an LMTD correction that varies from unity to zero. It is defined in such a way that the LMTD temperature should always be calculated for the equivalent counterflow single-pass exchanger with the same hot and cold temperatures. In the case in which one flow remains at constant temperature, then either P or R will be equal to zero, and the configuration of the heat exchanger becomes irrelevant. In this case the simple LMTD will be the correct ΔT_{mean} and F must go to unit. As example, two curves taken by TEMA are presented below. Please note than the Figure 17 and Figure 18 only include the curves for R≤1, and them must be modified if the tube-and-shell heat exchanger, given its large number of baffles, behave like a series of cross-flow exchangers. For the curves with R>1 has been noticed than the value of F may be obtained using a simple reciprocal rule: F(P,R)=F(PR,1/R)

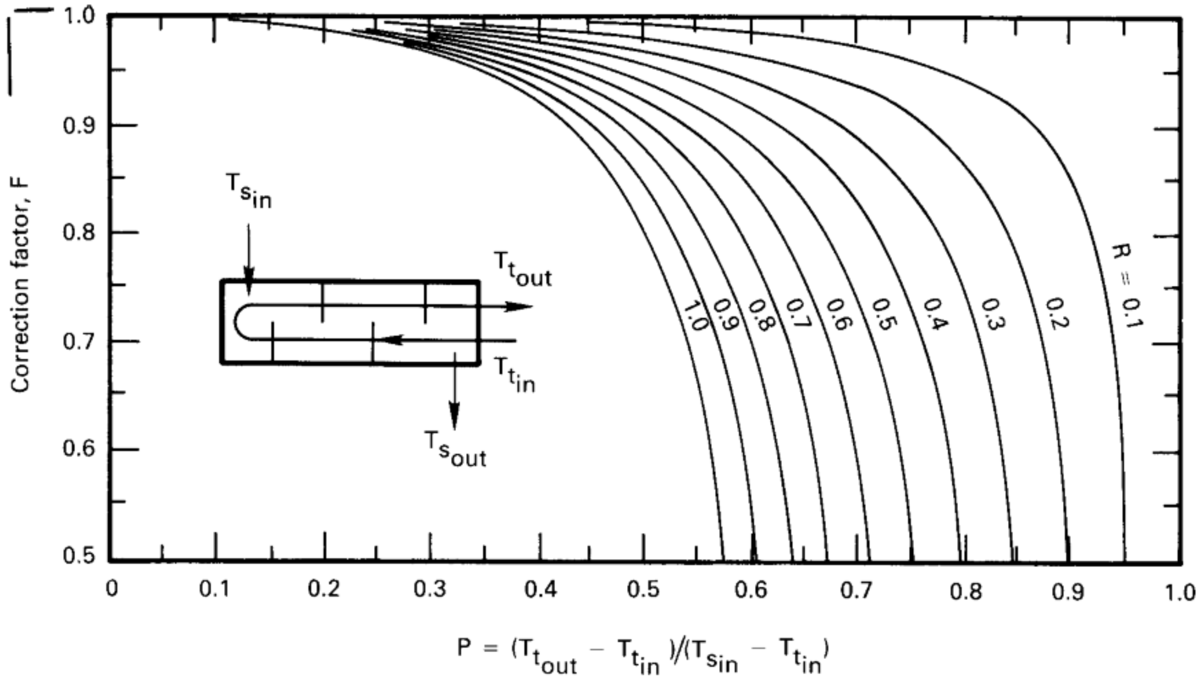


Figure 17: F for a one-shell-pass, four-, six-,…-tube-pass exchanger. [16]

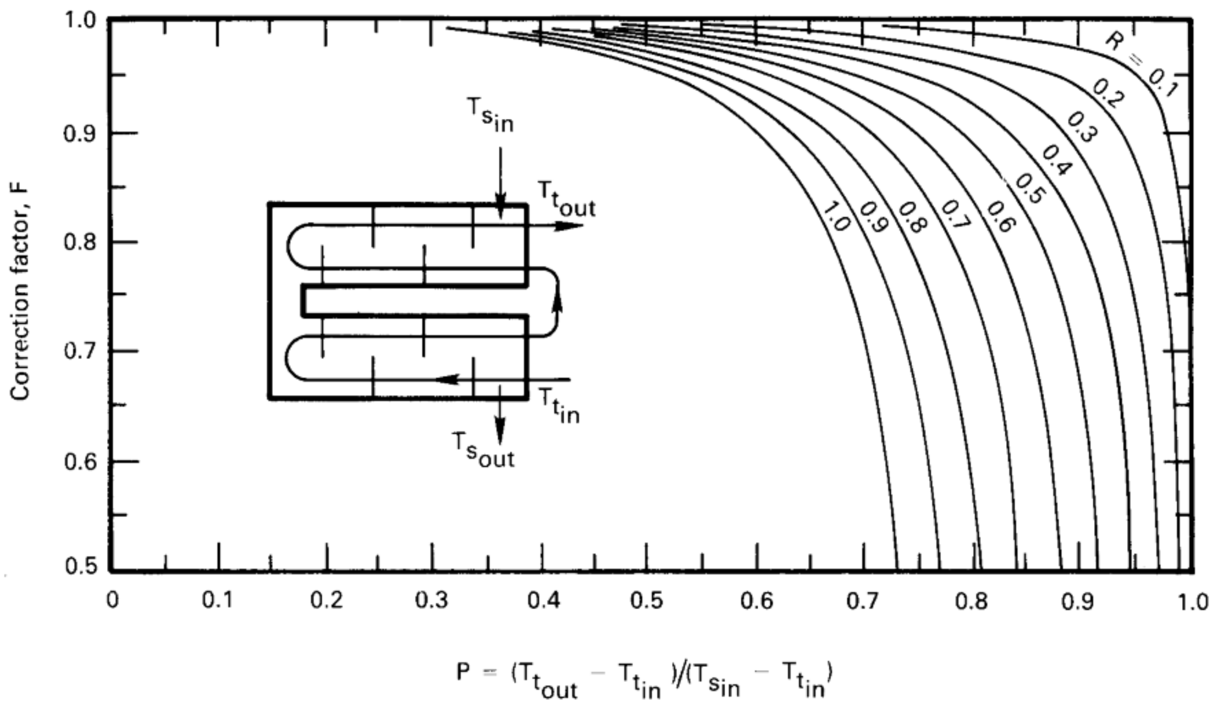


Figure 18: F for a two-shell-pass, four or more tube-pass exchanger. [16]

4.4. EFFECTIVENESS – NTU METHOD

During design activity often the heat exchanger configuration is known, while the outlet temperatures of the pipes are unknown. In this case we cannot proceed with the LMTD method because we don't know the temperature difference, consequently the computation of the LMTD result impossible. The only analytical way to solve the problem is to adopt an iterative method in which the outlet temperatures are estimated such as to make $Q_h = Q_c = C_h \Delta T_h = C_c \Delta T_c$. The correct outlet temperatures will be find when Q_h is equal to the heat exchanged computed by the LMTD, $Q = UAF(LMTD)$.

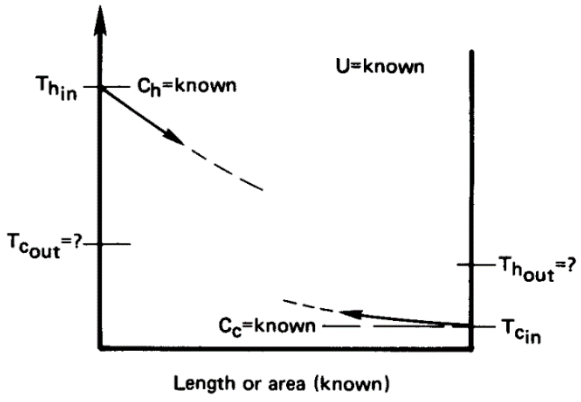


Figure 19: Example of a design problem. [16]

Since in the compact heat exchangers the overall heat transfer coefficient is far more likely to remain uniform, the effectiveness-NTU method can be used to simplify the problem.

Defining the number of transfer unit NTU as dimensionless group that can be viewed as a comparison of the heat capacity of the heat exchanger, expressed in W/K, with the heat capacity of the flow. It can be written physically as:

$$NTU = \frac{UA}{C_{min}}$$

Where C_{min} is the smallest of C_h and C_c.

The effectiveness instead can be founded graphically in function of the C_{min}/C_{max} ratio and NTU

$$\varepsilon = \frac{actual\ heat\ transfer}{\substack{maximum\ heat\ that\ could\ possibly\ be \\ transferred\ from\ one\ stream\ to\ the\ other}} = \frac{C_h\left(T_{h,in} - T_{h,out}\right)}{C_{min}\left(T_{h,in} - T_{c,in}\right)} = \frac{C_c\left(T_{c,out} - T_{c,in}\right)}{C_{min}\left(T_{h,in} - T_{c,in}\right)} = fn\left(\frac{C_{min}}{C_{max}}, NTU\right)$$

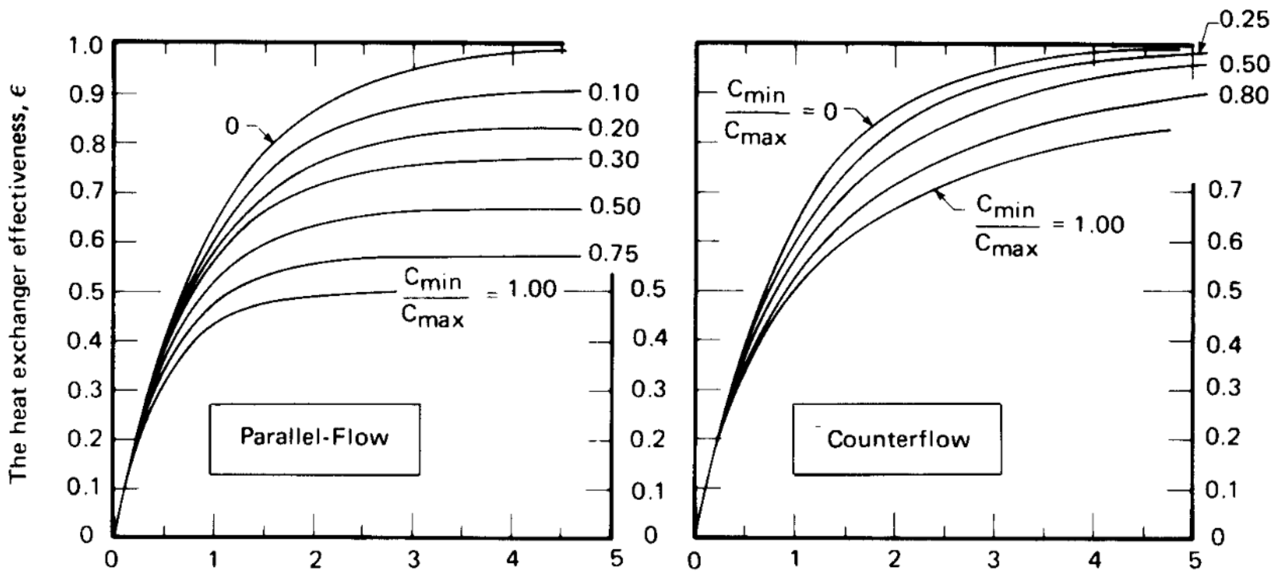


Figure 20: The effectiveness of parallel and counterflow heat exchangers. [16]

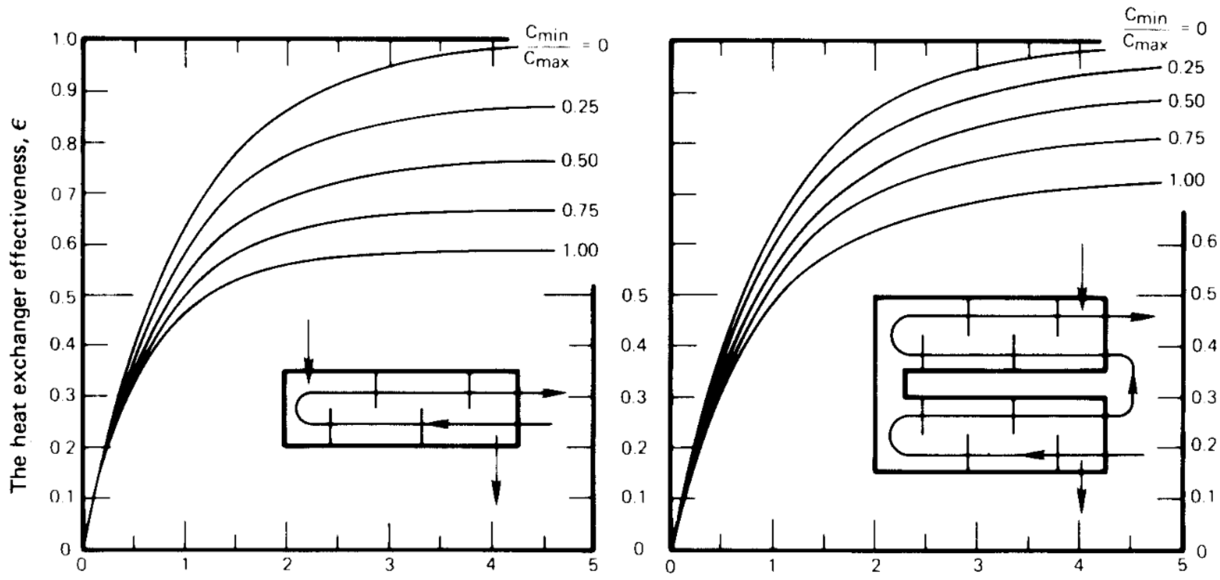


Figure 21: The effectiveness of some other heat exchangers configurations. [16]

It follows that the heat exchanged can be easily computed as:

$$Q = \varepsilon C_{min}\left(T_{h,in} - T_{c,in}\right)$$

Particular attention must be done in the case of uniform temperature. As already said the LMTD does not require correction, so the coefficient F would be unitary.

In ε-NTU approach the volumetric heat capacity rate might approach infinity because the flow rate, or specific heat, is very large or might be infinite because the flow is absorbing or giving latent heat.

As already said, the configuration of the exchanger becomes irrelevant and all the heat exchangers are equivalent. So, the equation for effectiveness in any configuration must reduce to the same common expression:

$$\lim_{C_{max}\to\infty} \varepsilon = 1 - e^{-NTU} \qquad and \qquad \frac{C_{min}}{C_{max}} = 0$$

4.5. HEAT TRANSFER IN PHASE CHANGE CONFIGURATIONS

4.5.1. MODES OF POOL BOILING

The Figure 22 shows, at atmospheric pressure, a complete boiling curve for saturated water. It has been divided into five regimes of behavior which will be now discussed, including the transition phases that divide them.

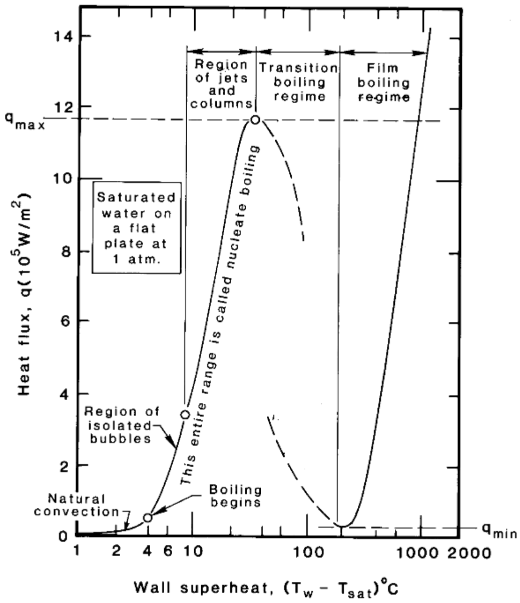


Figure 22: Typical boiling curve and regimes of boiling for an unspecified heat surface. [16]

Natural convection

If water is not in contact with its own vapour, it does not boil at the so-called normal boiling point, T_{sat}. Indeed, the heat continue to be removed by natural convection, and it continues to rise in temperature until bubbles finally begin to form. This occurs when a conventional machined metal surfaces is a few degrees above T_{sat}.

Nucleate boiling

The nucleate boiling regime embraces the two distinct regimes that lie between bubble inception and first transition point.

- *The region of isolated bubbles*: in this range, bubbles rise from isolated nucleation sites. As q and ΔT increase, more and more sites are activated. Figure 23.a is a photograph of this regimes as it appears on horizontal plate.
- *The region of slugs and columns*: when the activate sites become very numerous, the bubble start to merge one to another, and an entirely different vapour escape path comes into play. Vapour formed at the surface joint into jets that feed into large overhead bubbles or "slugs" of vapour. This process is shown as it occurs on a horizontal cylinder in Figure 23.b.

Peak heat flux

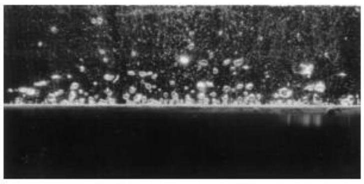
At the upper end of slugs and columns region the temperature difference is low, while the heat flux is very high. Clearly this means that the heat transfer coefficients in this range are enormous and it is very desirable be able to make heat exchange equipment work in this point. However, it is very dangerous to run equipment near q_{max} in system for which q is the independent variable. If q is raised beyond the upper limit of the nucleate boiling regime, such a system will suffer a sudden and damaging increase of temperature. This transition is called by a variety of name: the ***burnout point***; the ***peak heat flux***; the ***boiling crisis***; the ***DNB*** (Departure from Nucleate Boiling) or the ***CHF*** (Critical Heat Flux).

Transitional boiling regime

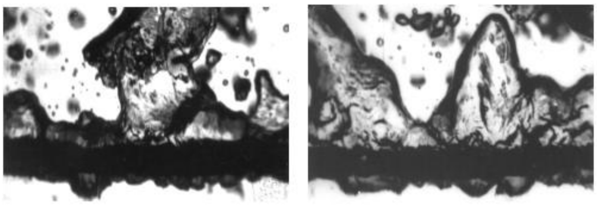
It is a curious fact that the heat flux actually diminishes with ΔT after q_{max} is reached. In this regime the effectiveness of the vapour escape process become worse and worse. Furthermore, the hot surface becomes completely blanketed in vapour and q reaches a minimum heat flux, q_{min}. Figure 23.c shows two typical instances of traditional boiling just beyond the peak heat flux.

Film boiling

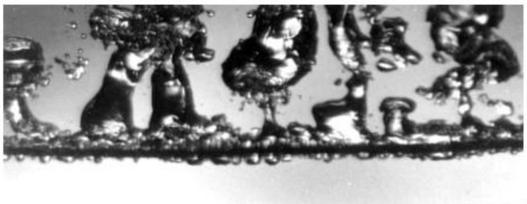
Once a stable vapor blanket is established, q again increases with increasing ΔT. The mechanics of the heat removal process during film boiling, and the regular removal of bubbles, has a great deal in common with film condensation but heat transfer coefficients are much lower because het must be conducted through a vapour film instead of through a liquid film. In Figure 23.d an instance of film boiling can be seen.



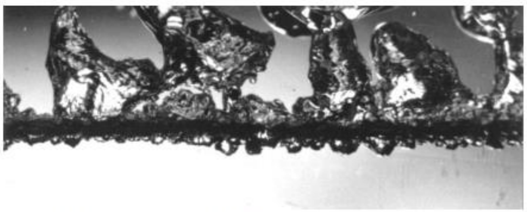
a. Isolated bubble regime—water.



b. Two views of transitional boiling in acetone on a 0.32 cm diam. tube.

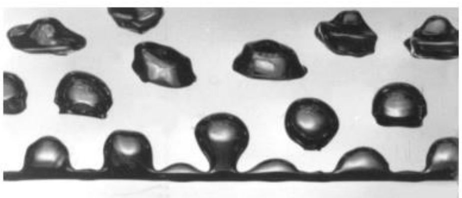


3.45 cm length of 0.0322 cm diam. wire in methanol at 10 earth-normal gravities. q=1.04×10^6 W/m^2



3.75 cm length of 0.164 cm diam. wire in benzene at earth-normal gravity. q=0.35×10^6 W/m^2

c. Two views of the regime of slugs and columns.



d. Film boiling of acetone on a 22 gage wire at earth-normal gravity. The true width of this image is 3.48 cm.

Figure 23: Typical photographs of boiling in the four regimes identified in Figure 22. [16]

4.5.2. NUCLEATE BOILING

INCEPTION OF BOILING

The analysis of the inception of boiling starts from the highly enlarged sketch of a heater surface, shown in Figure 24. Most metal finishing operations engrave tiny grooves on the surface. Moreover, some chattering or bouncing action leave small holes on the plane that has been worked. When a surface is wetted, the surface tension prevents the liquid from entering these holes. So, small gas or vapor pockets, representing the sites at which bubble nucleation occurs, are formed.

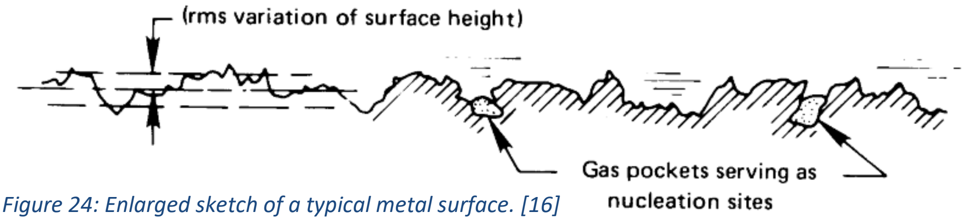


Figure 24: Enlarged sketch of a typical metal surface. [16]

The problem can be highly idealized supposing a spherical bubble of pure saturated steam in equilibrium with an infinite superheated liquid. The *p–v* diagram in Figure 25 shows the state points of the internal vapor and external liquid for a bubble at equilibrium. Notice that the external liquid is superheated to $(T_{sup}-T_{sat})\cdot K$ above its boiling point at the ambient pressure. However, the vapor inside, being held at just the right elevated pressure by surface tension, is just saturated. To determine the size of such a bubble, the conditions of mechanical and thermal equilibrium have been imposed. The bubble will be in mechanical equilibrium when the pressure difference between the inside and the outside of the bubble is balanced by the forces of surface tension, σ. Thermal equilibrium, instead, requires that the temperature must be the same inside and outside of the bubble. Since the vapor inside must be saturated at T_{sat} because it is in contact with its liquid, the force balance takes the form

$$(p_{in} - P_{out})\pi {R_b}^2 = (2\pi R_b)\sigma$$

Which allow to find an expression for the radius of the equilibrium bubble.

$$R_b = \frac{2\sigma}{p_{in} - p_{out}} = \frac{2\sigma}{(p_{sat}\ at\ T_{sup}) - p_{ambient}}$$

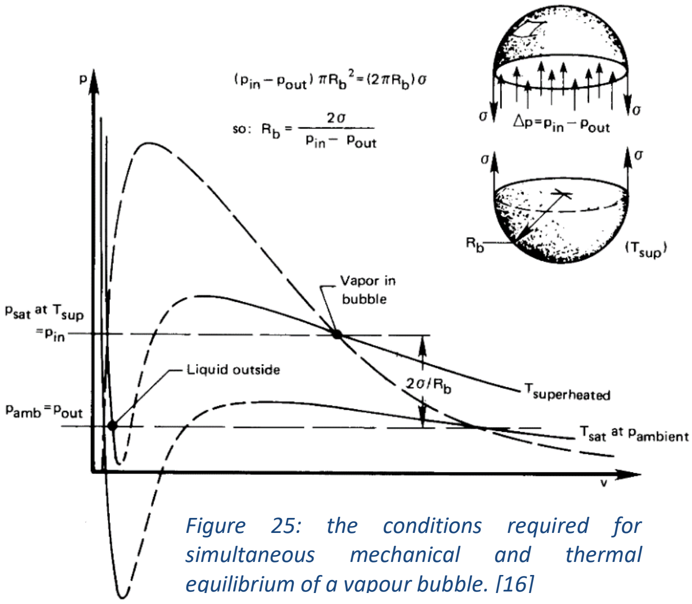


Figure 25: the conditions required for simultaneous mechanical and thermal equilibrium of a vapour bubble. [16]

Note that the equilibrium bubble is unstable. Indeed, if its radius is less than this value, surface tension will overbalance $[p_{sat}(T_{sup}) - p_{ambient}]$. Thus, vapor inside will condense at this higher pressure and the bubble will collapse. If the bubble radius is slightly larger than the one specified by the equation, liquid at the interface will evaporate and the bubble will begin to grow. Thus, as the heater surface temperature is increased, higher and higher values of $[p_{sat}(T_{sup}) - p_{ambient}]$ will result and the equilibrium radius, R_b, will decrease in accordance with the previous equation. It follows that smaller and smaller vapor pockets will be triggered into active bubble growth as the temperature is increased. As an approximation, we can use the equation that describe the radius of the equilibrium bubble to specify the radius of those vapor pockets that become active nucleation sites.

More accurate estimates can be made using Hsu's bubble inception theory, or the still more recent technical literature. In order to present all the data necessary for the discussion, the value of the surface tension σ for several substances are reporter in the table below.

Substance	*Temperature Range* (°C)	σ (mN/m)	$\sigma = a - bT$ (°C)	
			a (mN/m)	b (mN/m·°C)
Acetone	25 to 50		26.26	0.112
Ammonia	−70	42.39		
	−60	40.25		
	−50	37.91		
	−40	35.38		
Aniline	15 to 90		44.83	0.1085
Benzene	10	30.21		
	30	27.56		
	50	24.96		
	70	22.40		
Butyl alcohol	10 to 100		27.18	0.08983
Carbon tetrachloride	15 to 105		29.49	0.1224
Cyclohexanol	20 to 100		35.33	0.0966
Ethyl alcohol	10 to 100		24.05	0.0832
Ethylene glycol	20 to 140		50.21	0.089
Hydrogen	−258	2.80		
	−255	2.29		
	−253	1.95		
Isopropyl alcohol	10 to 100		22.90	0.0789
Mercury	5 to 200		490.6	0.2049
Methane	90	18.877		
	100	16.328		
	115	12.371		
Methyl alcohol	10 to 60		24.00	0.0773
Naphthalene	100 to 200		42.84	0.1107
Nicotine	−40 to 90		41.07	0.1112
Nitrogen	−195 to −183		26.42	0.2265
Octane	10 to 120		23.52	0.09509
Oxygen	−202 to −184		−33.72	−0.2561
Pentane	10 to 30		18.25	0.11021
Toluene	10 to 100		30.90	0.1189
Water	10 to 100		75.83	0.1477

Substance	*Temperature Range* (°C)	$\sigma = \sigma_o [1 - T\,(\mathrm{K})/T_c]^n$		
		σ_o (mN/m)	T_c (K)	n
Carbon dioxide	−56 to 31	75.00	304.26	1.25
CFC-12 (R12)	−148 to 112	56.52	385.01	1.27
HCFC-22 (R22)	−158 to 96	61.23	369.32	1.23
HFC-134a (R134a)	−30 to 101	59.69	374.18	1.266
Propane	−173 to 96	53.13	369.85	1.242

Figure 26: Surface tension of various substances from the collection of Jasper [22] and other sources [21] [24] [25].

REGION OF ISOLATED BUBBLES AND NUCLEATE POOL BOILING HEAT FLUX CORRELATION

In the last century the mechanism of heat transfer enhancement in the isolated bubble regime was hotly argued. The conclusion that have been made was that the bubbles "act as small pumps" that keep replacing liquid heated at the wall with cool liquid. The real problem is to specify the correct mechanism thanks to which this happens. Figure 27 shows the way bubbles probably act to remove hot liquid from the wall and introduce cold liquid to be heated. The two images on the left show a bubble growing and departing in saturated liquid. To grows, the bubble needs to absorb heat from the superheated liquid on its periphery. When the bubble leaves, the cold liquid is entrained onto the plate, which then warms up until nucleation occurs and the cycle repeats. On the right, instead, a bubble growing in subcooled liquid is shown. When the bubble protrudes into cold liquid, a short-circuit for cooling the wall is provided. Indeed, the steam can condense on the top while evaporation continues on the bottom. Then the cold liquid is brought to the wall when the bubble caves in.

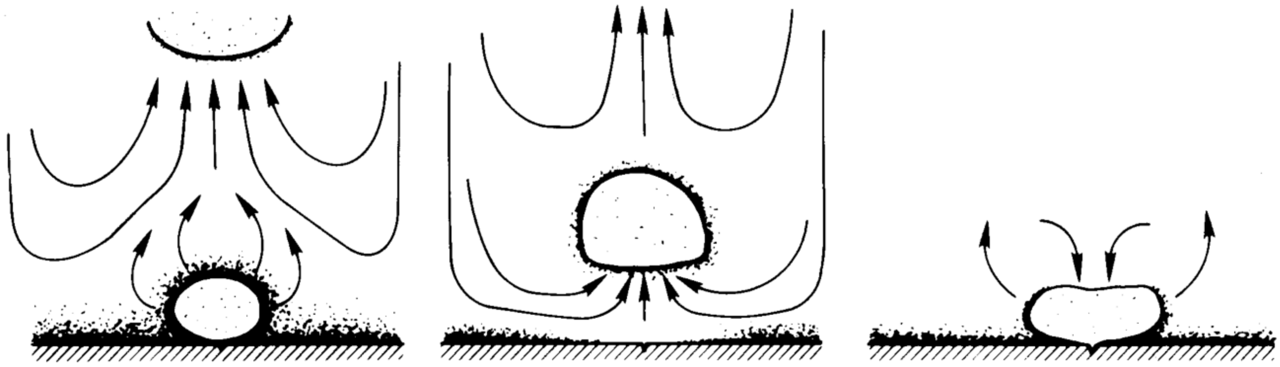


Figure 27: Heat removal by bubble action during boiling. Dark regions denote locally superheated liquid. [16]

It is evident that the number of active nucleation sites generating bubbles will strongly influence *q*. So, a direct dependence can be written in the following ways:

$$q \propto \Delta T^a n^b$$

Where $\Delta T = T_w - T_{sat}$ and *n* is the site density or number of active sites per square meter. The exponents turn out to be, approximately, *a=1,2* and *b=1/3*. The problem with this formulation is that *n* is a nuisance variable because it varies from system to system and cannot easily be evaluated. Indeed, normally *n* increase with ΔT, but it can increase in different way. If all sites were identical in size, they would be activated simultaneously, and *q* would be a discontinuous function of ΔT. When the sites have a typical distribution of sizes, *n* and *q* can increase very strongly with ΔT. Luckily, for a large class of factory-finished materials, *n* varies between ΔT^5 and ΔT^6, so *q* varies roughly as ΔT^3. This makes possible to correlate *q* approximately for a large variety of materials with a formulation like the following

$$\frac{c_p(T_w - T_{sat})}{h_{fg} Pr^s} = C_{sf}\left[\frac{q}{\mu h_{fg}} - \sqrt{\frac{\sigma}{g(\rho_f - \rho_g)}}\right]^{0.33}$$

where all properties are for liquid at T_{sat}. The h_{fg} is the latent heat of vaporization and the constant C_{sf} is an empirical correction for typical surface conditions. Figure 28 includes a set of values of C_{sf} for common surfaces as well as the Prandtl number exponent, *s*.

As initially denoted, there are two nucleate boiling regimes, the region of isolated bubble and the region of slugs and columns. While the first correlation applied only to the first of them, this last equation represents *$q(\Delta T)$* in both. This happens because it is empirical and it is not depending from the rational analysis of either nucleate boiling process. However, it is not very accurate for either of them. Thus, the ability to predict the nucleate pool boiling heat flux is poor due to the dependence of the nuisance variable *n*, and there is still not achieved a reliable heat transfer design relationship for nucleate boiling. Generally, the major problem in the nucleate boiling regime is to avoid exceeding q_{max}, rather than to calculate q given ΔT, which is not often required.

Surface–Fluid Combination	C_{sf}	s
Water–nickel	0.006	1.0
Water–platinum	0.013	1.0
Water–copper	0.013	1.0
Water–brass	0.006	1.0
CCl_4–copper	0.013	1.7
Benzene–chromium	0.010	1.7
n-Pentane–chromium	0.015	1.7
Ethyl alcohol–chromium	0.0027	1.7
Isopropyl alcohol–copper	0.0025	1.7
35% K_2CO_3–copper	0.0054	1.7
50% K_2CO_3–copper	0.0027	1.7
n-Butyl alcohol–copper	0.0030	1.7

Figure 28: Selected values of the surface correction factor and Prandtl number exponent. [16]

4.5.3. PEAK POOL BOILING HEAT FLUX and transitional boiling regime

LOW AND HIGH HEAT FLUX TRANSITIONAL BOILING REGIMES AND TAYLOR INSTABILITY

The process that connects the peak and the minimum heat flux is compounded by the high and low heat flux transitional boiling regimes, that are different in character. The high heat flux transitional boiling is represented by a large amount of vapor produced. It wants to float upward, but it has no clearly defined escape route. The jets that carry vapor away from the heater in the region of slugs and columns cannot serve that function in this regime because they are unstable. Therefore, vapor float up in big slugs, how has been shown in the Figure 23.c. When the bubble detaches from the surface the liquid falls in, touches the surface briefly, and a new slug begins to form. The low heat flux region, instead, it is almost indistinguishable from the film boiling shown in Figure 23.d. However, both processes display a common conceptual key: the heater is almost completely blanketed with vapor, and an unstable configuration of the liquid on top of vapour can be seen.

An example of how heavy fluid falls unstably into a light one can be seen in the Figure 29, in which the water condensing from a cold water pipe collapses into air. The heavy phase falls down at one node of a wave, while at the other the light fluid rises. This collapse process is called Taylor instability, while the length of the wave that grows fastest is called Taylor wavelength, λ_d. It is the wavelength that predominates during the collapse of an infinite plane horizontal interface and it can be predicted by means of the dimensional analysis, which leads to the following result.

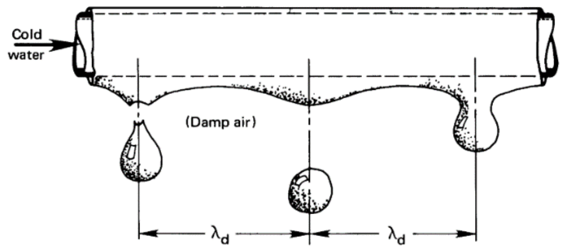
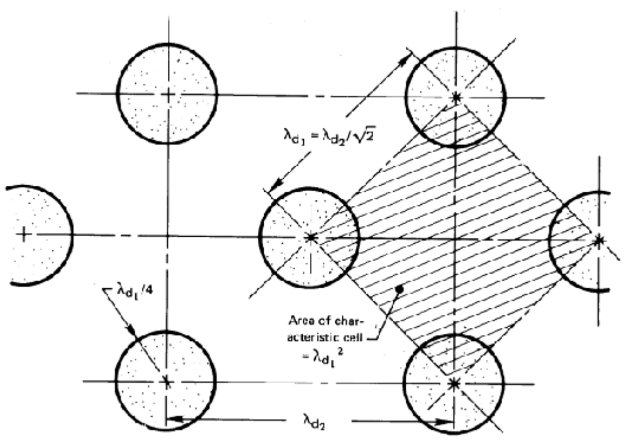


Figure 29: Taylor instability in the interface of the water condensing on the underside of a cold waterpipe. [16]

$$\lambda_d \sqrt{\frac{g(\rho_f - \rho_g)}{\sigma}} = \begin{cases} 2\pi\sqrt{3}\, for\ one - dimensional\ waves \\ 2\pi\sqrt{6}\, for\ two - dimensional\ waves \end{cases}$$

Throughout the transitional boiling regime, vapor ascends into liquid on the nodes of Taylor waves, and at q_{max} this rising vapor forms into jets. These jets arrange themselves on a staggered square grid, as shown in Figure 30. The basic spacing of the grid is λ_{d2}, as for the two-dimensional Taylor wavelength. The distance of the most basic module of jets, instead, is λ_{d1}, and it is correlated to λ_{d2} by means of $\lambda_{d,2} = \sqrt{2}\,\lambda_{d,1}$.



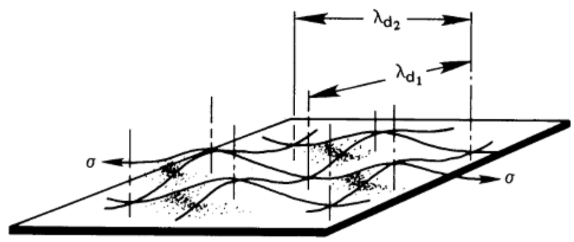


Figure 30: the array of vapour jets as seen on an infinite horizontal heater surface. On top, plan view of bubbles rising from surface; below, waveform underneath the bubbles. [16]

HELMHOLTZ INSTABILITY OF VAPOUR JETS

The Helmholtz instability, which can be seen in Figure 31, explain how the jets become unstable at the peak and why they cave in when the vapour velocity in them reaches a critical value. It is a fluid dynamic instability caused by different layers of the fluid in relative motion one with respect to one other. A high-pressure zone is present where the fluid velocity is low, while a lower pressure zone is the result of higher velocity layer. The state of collapse occurs when the interface that divides two regions undergoes small disturbance. Indeed, particles of fluid that before was at rest end up being in a one where there is a velocity field. This phenomenon generally leads to instability, which, however, does not always occur in vapour jets. Indeed, the surface tension in the jet walls tends to balance the flow-induced pressure forces that bring about collapse. As a matter of fact, since the vapor velocity in the jet must reach a limiting value, u_g, before state of collapse occurs, the jet is not always unstable. The following expression has been taken from [6] and exhibit the relation between the vapor flow, u_g, and the wavelength of a disturbance in the jet wall, λ_H.

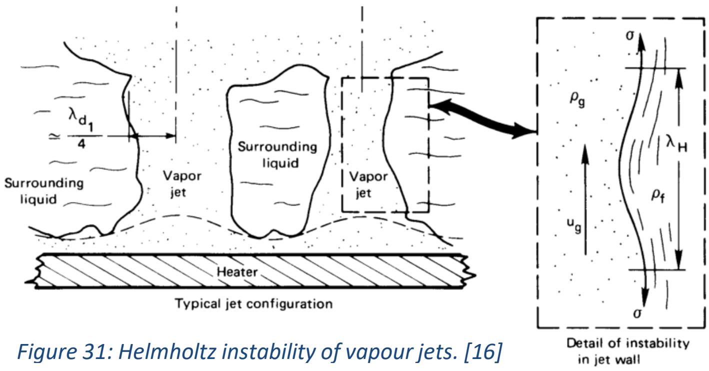


Figure 31: Helmholtz instability of vapour jets. [16]

$$u_g = \sqrt{\frac{2\pi\sigma}{\rho_g \lambda_H}}$$

However, a real liquid–vapor interface will usually be irregular, and therefore it can be viewed as containing all possible sinusoidal wavelengths superposed on one another. Then, in order to develop the most representative wavelength, the one more liable to collapse must be guessed. So, the Helmholtz instability is a theory that only approximate the problem of vapour jets collapse.

PREDICTION OF q_{max}

In order to write a general expression for q_{max}, it has to be observed that the heat flux must be balanced by the latent heat carried away in the jets when the liquid is saturated. Thus, the following formula can be immediately written:

$$q_{max} = \rho_g h_{fg} u_g \left(\frac{A_j}{A_h}\right)$$

where A_h is the heater area that supplies each jet and A_j is the cross-sectional area of a jet. The problem of q_{max} prediction in any pool boiling configuration always comes down to the determination of two parameters. One is the length of the perturbation in the jet wall, λ_H, which will trigger Helmholtz instability and fix u_g. So, the value of u_g that must be inserted in the above equation can be found from the Helmholtz one. The other is the ratio A_j/A_h. The specific expression for every heater configuration can be found imposing a correct value of the jet radius or by means of the dimensional analysis. This last procedure leads to have two pi-groups in function of the Bond number, *Bo*, used to compare buoyant force with capillary force. Attention must be paid in the case in which $Bo < 1/44$, where the process become completely dominated by surface tension and the Helmholtz wave mechanism no longer operate. In this particular case the mechanism of heat removal passes directly from natural convection to film boiling.

4.5.4. FILM BOILING AND RADIATION CONTRIBUTION

The similarity between the film boiling and the film condensation is so great that, in 1950, Bromley was able to adapt the equation for condensation on cylinders to the film boiling prediction. Indeed, modifying the boundary conditions in the film condensation analysis and changing k and v from liquid to vapor properties, he writes the following equation for the $\overline{Nu}_D$. D is the characteristic dimension of the body under consideration, so, in this formulation it is the diameter of the cylinder.

$$\overline{Nu}_D = 0.62\left[\frac{(\rho_f-\rho_g)gh'_{fg}D^3}{v_g k_g(T_w-T_{sat})}\right]^{1/4}$$

The vapor and liquid properties should be evaluated at $T_{sat}+\Delta T/2$ and at T_{sat} respectively. The constant value of 0.62 was fixed by means of the comparison between the equation with experimental data values. However, when the Prandtl number is large, e.g. *Pr≥0.6*, the latent heat needs to be corrected using the formulation below:

$$h'_{fg} = h_{fg}\left[1+\left(0.968-0.163/Pr_g\right)Ja_g\right] \quad where \;\; Ja_g = c_{p_g}(T_w-T_{sat})/h_{fg}$$

Is there also reported the formulation for the film boiling on the spheres

$$\overline{Nu}_D = 0.67\left[\frac{(\rho_f-\rho_g)gh'_{fg}D^3}{v_g k_g(T_w-T_{sat})}\right]^{1/4}$$

In these expressions only the heat transfer by convection through the vapor film is present. Radiation, instead, can be enormous when film boiling occurs much beyond q_{min} in water. In this case is not sufficient to add a radiation heat transfer coefficient $\bar{h}_{rad}$ to $\bar{h}_{boiling}$because this phenomenon will increase the vapor blanket thickness, reducing the convective contribution. One approximate relation for the cylinder application is reported below. However, an accurate correction for every specific case would be needed.

$$\bar{h}_{total} = \bar{h}_{boiling} + \frac{3}{4}\bar{h}_{rad}, \quad where \quad \bar{h}_{rad} = \frac{q_{rad}}{T_w-T_{sat}} = \frac{\varepsilon\sigma(T_w^4-T_{sat}^4)}{T_w-T_{sat}} < \bar{h}_{boiling}$$

Where ε is a surface radiation property of the heater called emittance. It is worth noting that radiation is seldom important when the heater temperature is less than 300°C.

The analogy between film condensation and film boiling cannot be assumed as valid during film boiling on a vertical surface, because on the liquid-vapour interface the Helmholtz instability occurs for a certain distance from the leading edge. However, it has been proved that using $\lambda_{d,1}/\sqrt{3}$ in place of D in the first equation, it can be obtained a very satisfactory prediction of $\bar{h}$ for rather tall vertical plates. The analogy adopted also deteriorates when it is applied to small curved bodies. This happens because the vapor film in boiling is thicker than the liquid film during condensation. Consequently, in curved bodies a correction is needed even if it could be ignored in film condensation. This curvature correction, to be made when R < 1.86, is:

$$\overline{Nu'_D} = \left[\left(0.715+\frac{0.263}{R'}\right)(R')^{1/4}\right]\overline{Nu}_D$$

4.5.5. MINIMUM HEAT FLUX

For the prediction of the minimum heat flux q_{min} it is assumed that, as $(T_w - T_{sat})$ is reduced in the film boiling regime, the rate of vapor generation becomes too small to support the Taylor wave action which characterizes film boiling. The equation for horizontal heaters was firstly predicted from Zuber and after corrected, on the basis of experimental data, by Berenson in the following formula

$$q_{min} = 0.09\rho_g h_{fg} \sqrt[4]{\frac{\sigma g(\rho_f - \rho_g)}{(\rho_f - \rho_g)^2}}$$

The parallel prediction of the equation for horizontal wires was made by Lienhard and Wong, and lead to the below correction

$$q_{min,wires} = 0.515 \left[\frac{18}{R'^2(2R'^2 + 1)}\right]^{1/4} q_{min}$$

The problem with all these expressions is that, when the film boiling heat fluxes are higher than the minimum, some contact frequently occurs between the liquid and the heater wall. In this case, the boiling curve deflects above the film boiling curve finding a higher minimum than the ones reported above. The values of the constants shown above should therefore be viewed as practical lower limits of q_{min}.

5. THERMAL DESIGN OF THE HEAT EXCHANGER: CASE STUDY

5.1. INTRODUCTION TO THE THERMAL DESIGN

In order to obtain the required heat flux between the fluids, a certain amount of exchange area is needed. The goal of the heat exchanger's thermal design is to find that value and the best heat exchanger configuration for the case. Apart from predicting heat transfer, a host of traditional considerations must be addressed in design heat exchangers, for example the minimization of pumping power and the minimization of fixed cost. After a brief introduction on thermal design, the document will be focus only on the used program. In this section are provided all the explanation about the choice that has been taken during the thermal design activity.

5.1.1. LOGICAL APPROACH TO THE MANUAL THERMAL DESIGN

To better understand the curse of the design process, faced with such an array of trade-offs of advantages and penalties, a list of design considerations for a large shell-and-tube exchanger design is here reported:

- Decide which fluid should flow on the shellside and which one should flow in the tubes. Normally, this decision will be made to minimize the cost. Later some useful advices are listed:
    - Put dirty stream on the tubeside because is easier to clean inside the tube;
    - Put high pressure stream on the tubeside to avoid thick and expensive shell;
    - When special materials are required for one stream, put that one on the tubeside to avoid expensive shell;
    - Cross flow gives higher coefficients than in plain tubes, hence put the fluid with lowest coefficient on the shellside;
    - The more viscous oil would flow in the shell in order to minimize the pumping cost;
    - Corrosion behaviour, fouling, and problems of cleaning fouled tubes also weigh heavily in this decision.
- Early in the process, the designer should assess the cost of the calculation comparison with:
    - The converging accuracy of computation;
    - The investment in the exchanger;
    - The cost of miscalculation.
- Make a rough estimate of the heat exchanger size using, for example, U values taken from tables and/or anything else that might be known from experience. This serves to circumscribe the subsequent trial-and-error calculations. It will help to size flow rates and anticipate temperature variations, avoiding again subsequent errors.
- Evaluate the heat transfer and pressure drop.
- Evaluate the needed exchange area imposing the required value in the heat transfer formula.
- Evaluate the cost of various exchanger configurations that appear reasonable for the application.

These are the steps generally followed by the process engineers to optimize the preliminary design of a heat exchanger.

5.1.2. APPROACH TO THE COMPUTER AIDED THERMAL DESIGN

The manual thermal design was used in the small exchangers' design, typically compact cross flow used in transportation equipment. Larger shell and tube exchangers pose two kinds of difficulty in relation to U, which nowadays make the use of computer program necessary. The first one is the variation of U through the exchanger, which we have already discussed. The second difficulty is that convective heat transfer coefficients are very hard to predict for the complicated flows that move through a baffled shell. As introduced before, the determination of $\bar{h}$ in a baffled shell remains a problem that cannot be solved analytically. Instead, it is normally computed with the help of empirical correlations, or with the aid of large computer programs that include relevant experimental correlations. The problem of predicting $\bar{h}$ when the flow is boiling, or condensing, is even more complicated.

As already introduced, nowadays computer programs are used to overcome these problems. All of them are based on one of two main algorithms for thermal design:

- HTFS, Heat Transfer and Fluid flow Service method: (e.g. Aspen Shell and Tube Exchanger)
- HTRI, Heat Transfer Research, Inc: (e.g. Xist)

During the study case, the heat exchanger thermal design will be done using the Aspen Shell and Tube Exchanger. This program uses an HTFS algorithm and allow us to make the thermal design optimization applying a series of refining steps.

5.2. CUSTOMER INPUT DATA

The thermal design of the heat exchanger begins from the evaluation of the customer input data. As previously said the whole real heat exchanger's document cannot be inserted or attached, due to the confidentiality agreement that have been signed. However, the summary table of the necessary input data is reported below.

Applicable To:	○ Approval	● Construction	○ As Built	● ASME	● TEMA
Service	**MP Steam Generator**				
Manufacturer:	**SIMIC S.p.A.**				
PERFORMANCE OF ONE UNIT					
Fluid Allocation		SHELLSIDE		TUBESIDE	
Fluid Circulated		**Boiler Feed Water**		**HCBN**	
Total Fluid Entering	kg/h	**8.846**		**431.779**	
Liquid	kg/h			**206.344**	**275.328**
Vapour	kg/h - MW			**225.435** **129,2**	**156.451** **128,5**
Non Condensables	kg/h - MW				
Steam	kg/h		**8.404**		
Water	kg/h	**8.846**	**442**		
Fluid Vaporized / Condensed	kg/h				
Temperature	°C	**121,0**	**251,32**	**271,0**	**265,0**
Operating Pressure	Mpa (g)	**4,0**		**1,3**	
Allowed Pressure Drop	MPa	**0,03**		**0,07**	
Fouling Resistance	$m^2 \cdot °C \cdot h/kcal$	**0,0004**		**0,0004**	
CONSTRUCTION OF ONE SHELL					
		SHELLSIDE		TUBESIDE	
Design Pressure	Mpa (g)	**7,8**		**6,0**	
Design Temperature Max / Min	°C	**280,0 / 5,0**		**300,0 / 5,0**	
Corrosion Allowance	mm	**3,0**		**3,0**	
Shell Material:	**Carbon Steel**	Shell Cover Material:	**Carbon Steel**	● Integral	○ Removable
Channel Material:	**Carbon Steel**	Channel Cover Material:	**Carbon Steel**	○ Integral	● Removable
Tubesheet Stationary Material:	**Carbon Steel**	Tubesheet -Floating Material:			
Floating Head Cover Material:		Impingement Plate Material:			

TUBESIDE

HOT SIDE PROPERTIES		**1**	**3**	**4**	**6**	**7**	**9**	**10**
		Inlet						Outlet
Temperature	°C	271	270	269	268	267	266	265
Vapor Weight Fraction		0,52	0,49	0,47	0,43	0,41	0,38	0,36
Enthalpy	kcal/kg	10,79	8,39	7,19	4,79	3,60	1,20	0,00
Latent Heat	kcal/kg	39,40	37,85	38,01	38,48	38,71	39,19	39,31
Vapor Density	kg/m³	51,761	50,857	50,407	49,513	49,069	48,187	47,749
Vapor Viscosity	cP	0,010	0,010	0,010	0,010	0,010	0,010	0,010
Vapor Thermal Conductivity	kcal/(h·m·°C)	0,030	0,030	0,030	0,030	0,030	0,029	0,029
Vapor Specific Heat	kcal/(kg·°C)	0,661	0,659	0,658	0,656	0,654	0,652	0,651
Total Liquid Density	kg/m³	495,6	497,4	498,4	500,2	501,1	502,9	503,7
Total Liquid Viscosity	cP	0,127	0,128	0,128	0,128	0,129	0,129	0,130
Total Liquid Thermal Conductivity	kcal/(h·m·°C)	0,080	0,080	0,080	0,080	0,080	0,080	0,080
Total Liquid Specific Heat	kcal/(kg·°C)	0,812	0,870	0,852	0,835	0,829	0,820	0,817
Total Liquid Surface Tension	dyne/cm	2,91	2,98	3,02	3,09	3,13	3,20	3,24
Total Liquid Critical Pressure	kg/cm²(a)	26,0	26,0	26,0	26,0	26,0	26,1	26,1
Total Liquid Critical Temperature	°C	326	325	325	325	325	325	325

5.2.1. OTHER REQUIREMENTS AND PRELIMINARY INPUT COMMENTS

In addition to the datasheet, three further requirements useful for the thermal design and for the choice of the configuration has been expressed from the client. In particular, these are the specifications:

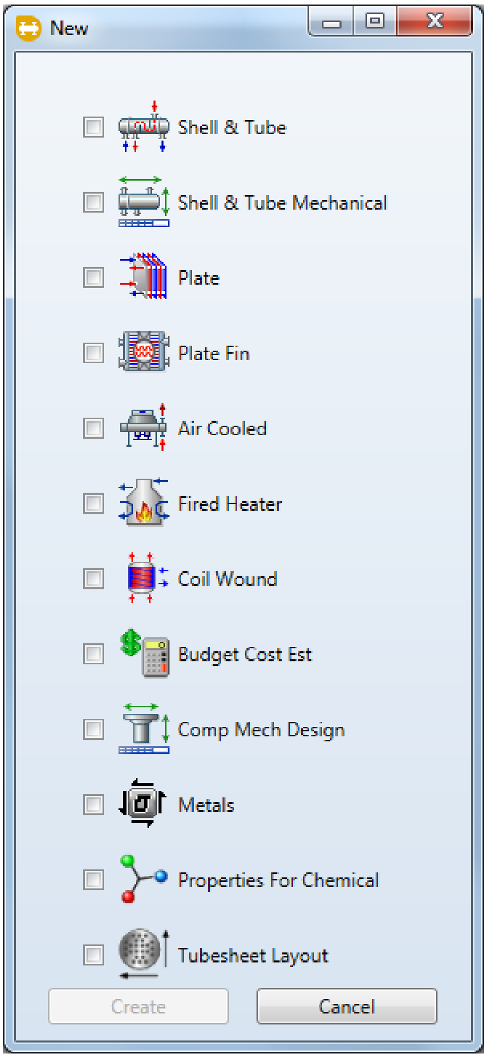
- The heat exchanger must be a horizontal shell and tube type;
- Due to the impurity of both liquid the heat exchanger shall present removable bundle in order to guarantee the possibility of the shellside mechanical cleaning;
- The tubes of diameter 25,4mm and 2,4mm minimum thickness would be appreciate if used in the removable bundle;
- It is not required the vacuum working condition, so no vacuum design parameters were indicated.

Given all the input data necessary to proceed with the thermal design, some preliminary comments on it can be done. The aim of this heat exchanger, given the liquid and steam outlet mass flow rate, is clearly the production of saturated vapour. The water that is converted in steam represents the cold fluid, and it will be of course placed in the shellside in order to guarantee the correct behaviour of the steam generator. A Kettle shell type, when well designed, would allow the correct expansion of the steam before the injection in the pipeline. The HCBN in liquid and vapour physical state represent the hot fluid and has been placed in the tubeside. While the corrosion characteristics of HCBN depends on the fluid, in the datasheet there is not specified which hydrocarbon is. Crude oil for example can be a problem due to the high Sulphur content. Reacting with hydrogen molecules it forms hydrogen sulphide [H_2S]. This compound release again hydrogen during the time that, being in forms of ions, will cause problem of hydrogen embrittlement. Other HCBN instead can be very low corrosive or even increase the corrosion resistance of the steel. Whereas the indications provided in the datasheet underline carbon steel as material for all the heat exchanger parts, it is appropriate to think that if the HCBN would be crude oil it would probably be de-sulphured in a previous instance. Common practice in such cases is to add three millimeters of oversize. This value come from the 0,1 millimeters of average corrosion per year estimated for a not high-aggressive hydrocarbon. Since during design activity the estimated life of a heat exchanger is 30 years, the three millimeters are the result of the obvious calculation. However, the properties of the HCBN fluid are provided in the table, while in the datasheet the fouling resistance of 0,0004 m^2°Ch/kcal is reported. Because both fluids, in the shellside and tubeside, have a high fouling coefficients, the heat exchanger correctly requires a removable bundle in order to allow the cleaning procedure in the shellside. While the U shape of tubes can be selected in order to have a cheap detachable bundle, it doesn't allow the mechanical cleaning inside the tubes. For this reason, it is important to make sure that chemical cleaning is possible from the tubeside. The other possible solution is the straight removable bundle that would allow the mechanical cleaning in the tubeside, but would be more expensive giving the prevented thermal expansion. Indeed, if the bundle would not be free to expand, internal stress due to thermal gradient will be introduced and taken into account during the mechanical design. This would lead to a very thick tubesheets that always have a high influence on the cost of the device. With regards to the horizontal orientation of the heat exchanger it of course depends from the position of the device inside at the plant. It has been decided from the process engineers that have been probably considered all the variables finding the most comfortable solution for the case. For what concern the thermal and mechanical design, this configuration lead to have a lower hydrostatic head, being the heat exchanger a pool boiling type. Indeed, given the level of the water constantly present in the shell a fairly lower hydrostatic pressure is generated in the horizontal configuration. The main design possibilities will be thoroughly discussed when the correspondent input of the program used in the design activity will be presented.

5.3. INTRODUCTION TO ASPEN EXCHANGER DESIGN AND RATING V.11

In the following paragraphs the program Aspen Exchanger Design and Rating is presented. This section would not be a complete guide for the use of the program, but only a brief overview in order to understand how it works and how the design choices, presented in the following sections, have been implemented in the program.

Launching the program and selecting a new project, the choice of the software section to be used is required. The possibilities are:



Shell & Tube: section of the program that will be used in the first part of the study case. It allows the thermal design of the shell and tube heat exchanger.

Shell & Tube Mechanical: allow to draft the mechanical design useful to estimates the overall cost. It is the section that will be used during the second part of the study case.

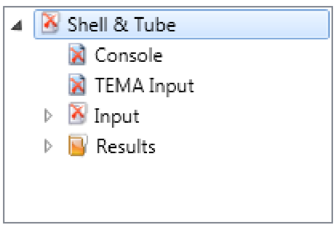
Plate, Plate Fin, Air cooled, Fired Heater and Coil Wound: leads to the thermal design of plate, fin plate and air-cooled heat exchanger respectively. For thermal design of boiler with flame the Fired Heater section must be chosen, while coil wound is a new functionality that has been added in the latest versions that allow to design coil wound heat exchanger. All these modules will not be necessary during the study case.

Budget Cost Estimation: it is a very practical section that allow to estimate the cost of the heat exchanger.

Component Mechanical Design: to select in case of preliminary mechanical design of a single component. It turn to be useful in the case of replacement part design.

Metals and **Property of Chemical**: are modules that contain all the property of the metals and chemical substance in tables of data. These modules can be automatically recalled from the program during the design computations.

Selecting the Shell & Tube module and clicking on "*create*" a new project is open, and a navigation pane with different form is made available on the left. The hierarchically higher forms are:



Console: automatically filled resume of the data implemented in the project

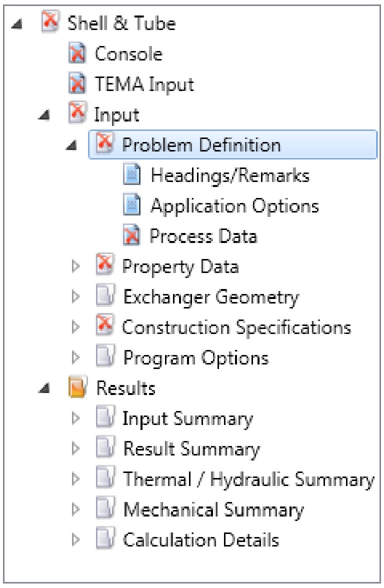
TEMA input: because the TEMA recommended a standard format of datasheet, this card offers the possibility to input all the data in the TEMA datasheet frame shape.

Input and **Result**: these sections have a lot of subcases. They are used for the implementation of the input data and design choices. Every used sheet and function will be presented in the next section and place side by side with the inserted data and choice explanation.

5.4. DESIGN MODE: IMPLEMENTATION OF THE INPUT DATA AND EXPLANATIONS

5.4.1. PROBLEM DEFINITION

As first step, going in *Home/Set_units*, the correct unit of measure must be set as *SI, International System*. Since the program has an American origin, the default choice is *US, United States customary units*.



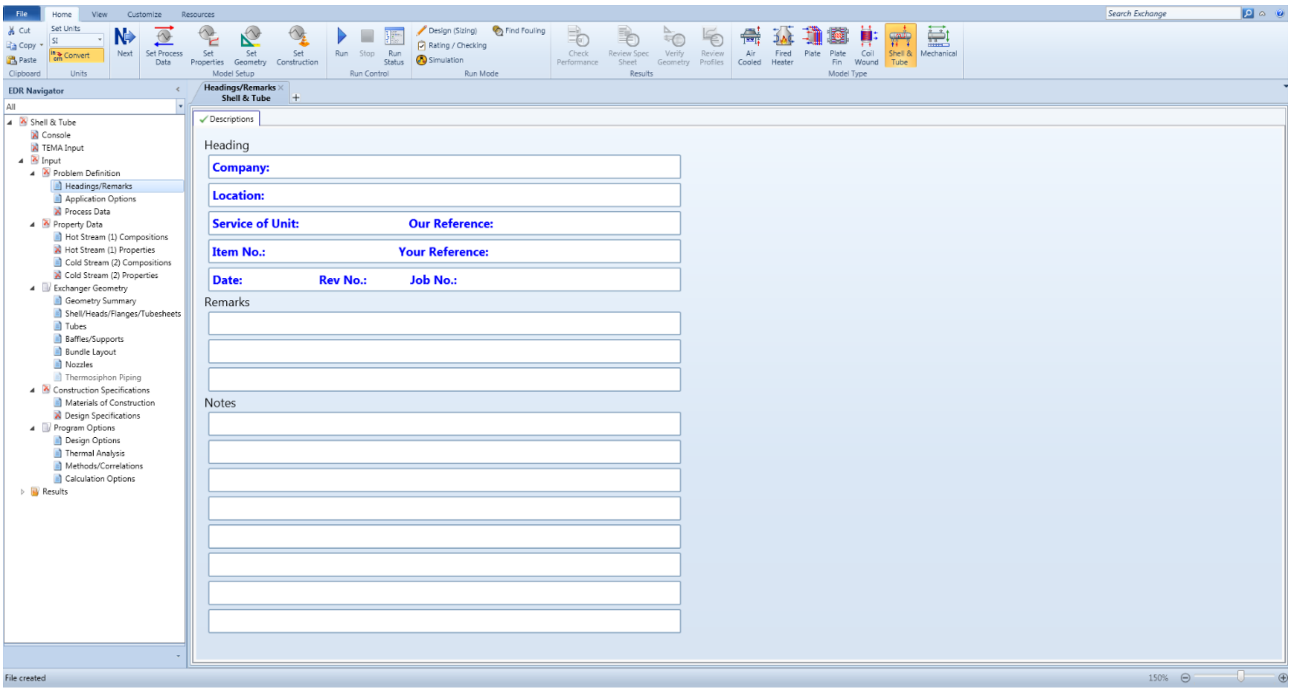
Before starting to explain the program features, is useful to remark some graphical and logical characteristics. In particular, red crosses are shown on the icons of the schedule in which mandatory data are required. They change in function of the selected calculation mode and allow to see, graphically, where are placed the missing data to start the design process.

Another visual feature of the program is the meaning associated with the text graphics. When a choice is made, or data is inserted, the text becomes bold, and the program cannot automatically change that feature when the calculation is performed. The program's default choices, instead, are written in italic font and can be always modified, also by the program, during the design activity.

The first form of the input section is the **problem definition**, which has the three following subtabs:

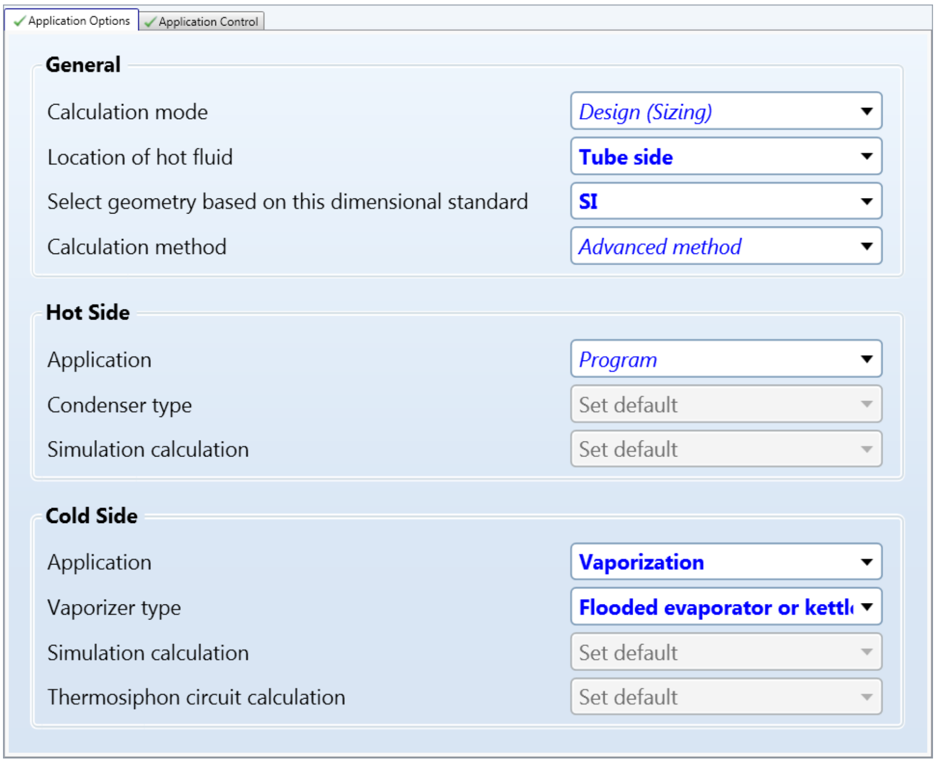
HEADING/REMARKS

Heading/Remarks is the place where all the company and project data are inserted in order to store correctly the devices during the years. It also allows to add some remarks, that will be shown in the final datasheet, and notes, which instead are designer annotation not shown anywhere.



APPLICATION OPTIONS

Application Options is the first important schedule in which choices must be done.



Calculation mode allows to choose the design step used by the program. In function of the design accuracy selected, always more input parameters are required. In particular, the four calculation modes are:

- **Design/Sizing**: it is the first mode required at the begin. It requires very low amount of input and it makes a draft design of the project, with the logic of exploiting the following step as refinement. In this first stage the heat exchanger type is defined.
- **Rating/Checking**: this step is a refinement of the Design mode. The selected output of the previous mode is taken, and a verification design is made considering more input data.
- **Simulation**: it is an application in which all the input data are required, and a complete simulation of the designed heat exchanger is launched. It is useful to find the causes of a problem in a heat exchanger already existent or to evaluate other possible use, e.g. working with other fluids or mass flow rates.
- **Find Fouling**: it is a different calculation mode used when there is a heat exchanger completely modelled that must be allocated to another use, e.g. employing other liquids. During the simulation all the parameter must be inserted as input, unless the fouling factor. The aim of this simulation is to find the maximum allowable fouling factor that, with that device, still allow the indicated heat exchange. After, it must be verified that the fouling factors of the new fluids are lower than the maximum allowable fouling factor obtained from the simulation.

Design mode is therefore the first option to be selected in order to proceed with the first step of the thermal design.

Talking about the **Location of hot fluid** the study case is a steam generator, so, I known that the water will be the cold fluid, and it will be placed in the shellside because it must have the space to expand during the phase change. Therefore, in this case the hot fluid is the HCBN, and it will be placed in the tubes side. Note that once the tubes side is selected, the text becomes bold with all the previously done consideration.

After the choice of the unit of measure, the **calculation method** has to be selected between advanced or standard. This function was inserted in order to solve the very long computation time required in the past decades. Nowadays, with more powerful computers, the *advanced method* choice can be always done, due to the fact that this first step will not require more than some minutes of calculation.

Now the **Hot and Cold Side** parameters must be inserted. The hot side, represented by the tubes side, is characterized by the HCBN in liquid and vapour physical state for both, inlet and outlet, despite the condensation process. In this case, is useful to allow the program to set the best option by itself during calculation. So, the default option has not to be changed and *Program,* in italic font, is shown. The cold side instead is characterized by the evaporation of the water in the shell; therefore, *vaporization* is selected in the application menu. Another choice that can be done is the vaporizer type. Given the required fluid and steam mass flow rate, the heat exchanger must be a kettle reboiler, so, *kettle type* can be selected.

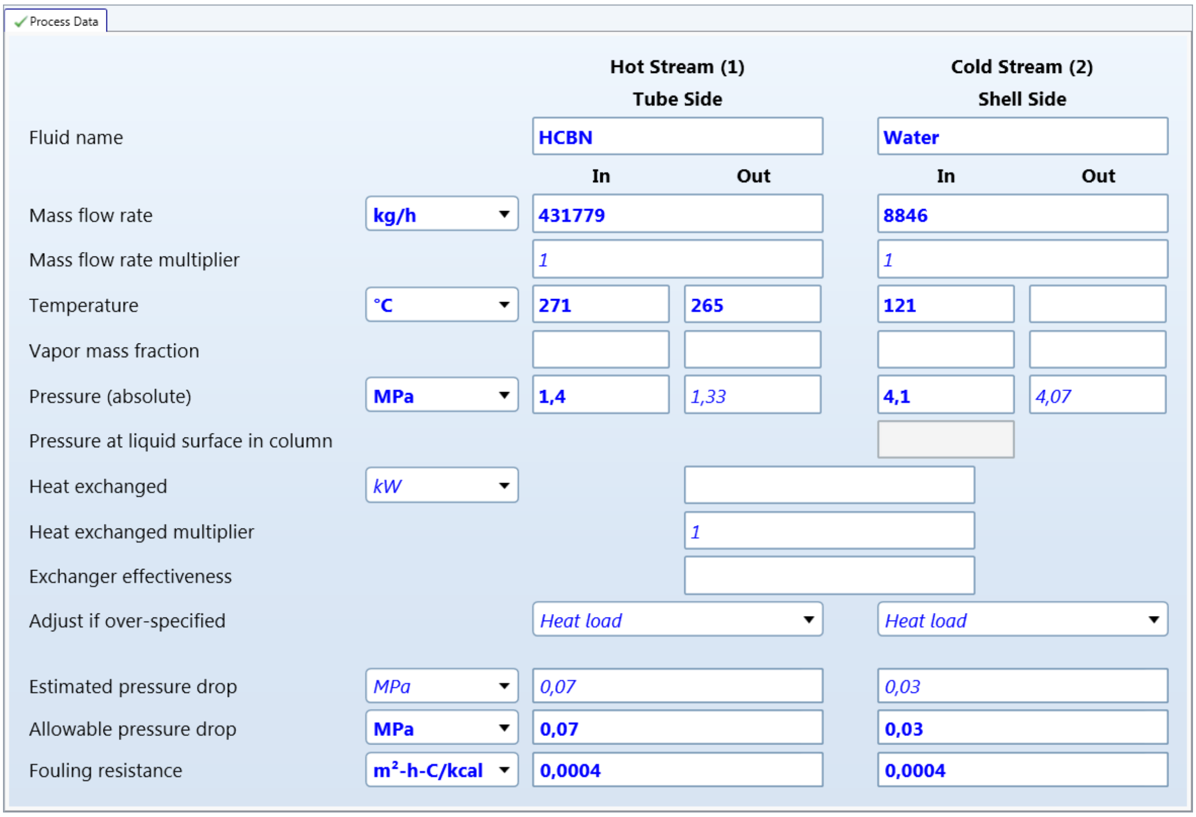
Note than in the section application options there is also a secondary sheet named **Application Control**. In this section are collected the setting that govern the mathematical passages of the design, and there are no parameters to be entered here.

PROCESS DATA

This is the section in which the parameters from datasheet must be implemented. Indeed, fluids name, flow rates, temperature and pressure must be inserted. About temperature, we decide to not force the outlet temperature of the water, in order to allow at the program to change it. This decision has been taken because the value contained in the datasheet is slightly different from the value contained inside at the database program. This difference in the data can lead to miscalculations if not properly managed. Talking about pressure instead I can have two indication on the datasheet: (a) or (g).

(a) Stands for *Absolute pressure*, and it indicates that the value takes into account the atmospheric pressure (so, it will be 101325 Pa higher than relative pressure, e.g. inner pressure measured in vacuum environment);
(g) Stands for *Gauge pressure*, and it indicates that the value is the measure of the difference between the inner pressure and the external-atmospheric one. (e.g. inner pressure measured in air environment).

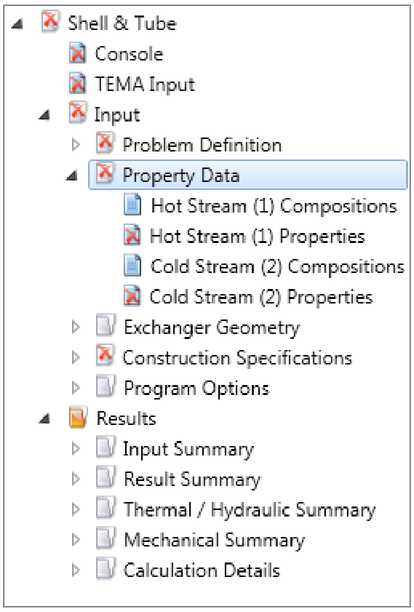
The program always indexes with the label *"(absolute)"* where absolute pressure is expected, while, where there is not specification, the gauge pressure must be inserted. The **output pressures** are automatically computed by the program, which base the calculation on the pressure drop, estimated in function of the inlet pressure. The **allowable pressure drop**, instead, must be manually inserted in the correct field after having choice the coherent unit of measure.



In this case, the tubeside and shellside fouling coefficients was been provided inside the datasheet. However, when it is not, the values can be taken by TEMA's tables in function of the types of fluids.

Note the presence of the label "*adjust if over-specified*". It asks to specify which parameter the program will force if the heat exchanged result to be higher than the one indicated. In this case the client does not have provided that data, so it will be one of the outputs of the design procedure.

5.4.2. PROPERTY DATA

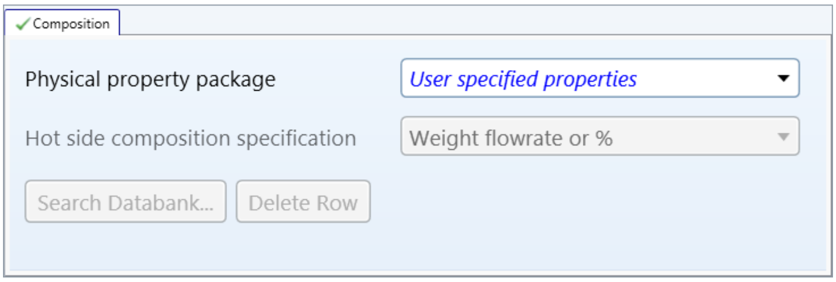


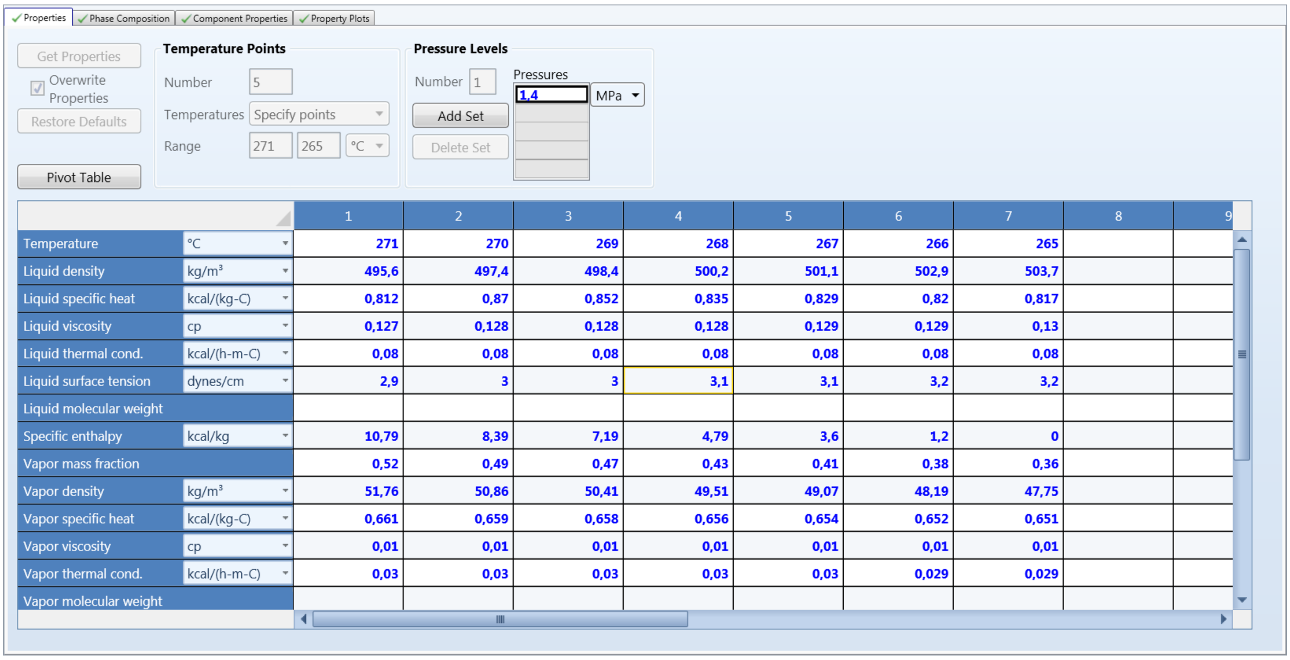
The following group of forms is designed to accept the input data of the fluids flowing in the shell and in the tubes. Attention must be paid because the program refers to them with the "hot and cold" adjectives, instead of the flowing position. In this study case both fluid property entry possibilities will be presented. In order to make the next screenshots more visible, the list of selectable forms is shown on the right as it can be seen in the real program.

HOT STREAM COMPOSITION AND PROPERTIES

In ASPEN there are two methods to indicate the composition of a fluid. The first is seen in this paragraph, because it allows the insertion of the fluid property manually. For the water, instead, the second method, explained in the following paragraph, will be used.

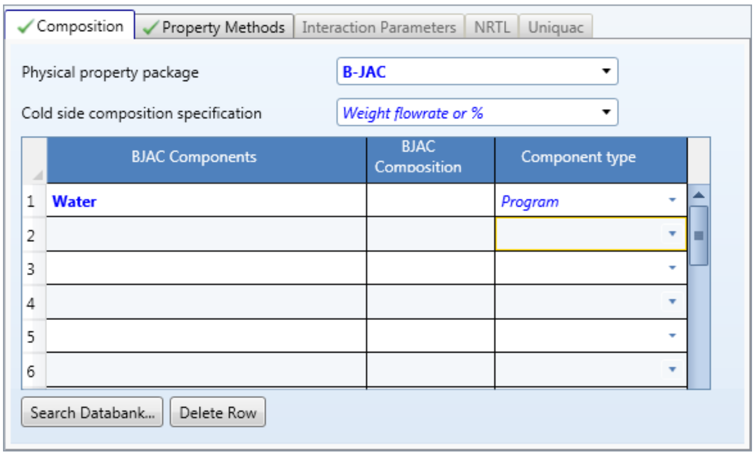
The hot stream is represented by a mixture of hydrocarbons (HCBN) of which we don't know the component substances and the quantity of them. However, the customer has provided us a table with all the properties values at the inlet pressure in function of the temperature. These values can be inserted manually in the program selecting "*user specified properties*" in the **Hot Stream (1) compositions** form. Then, the table visible in **Hot Stream (1) Properties** can be filled with the provided values. Note than the value has been inserted for seven temperature steps, each of which refers to the inlet pressure level.



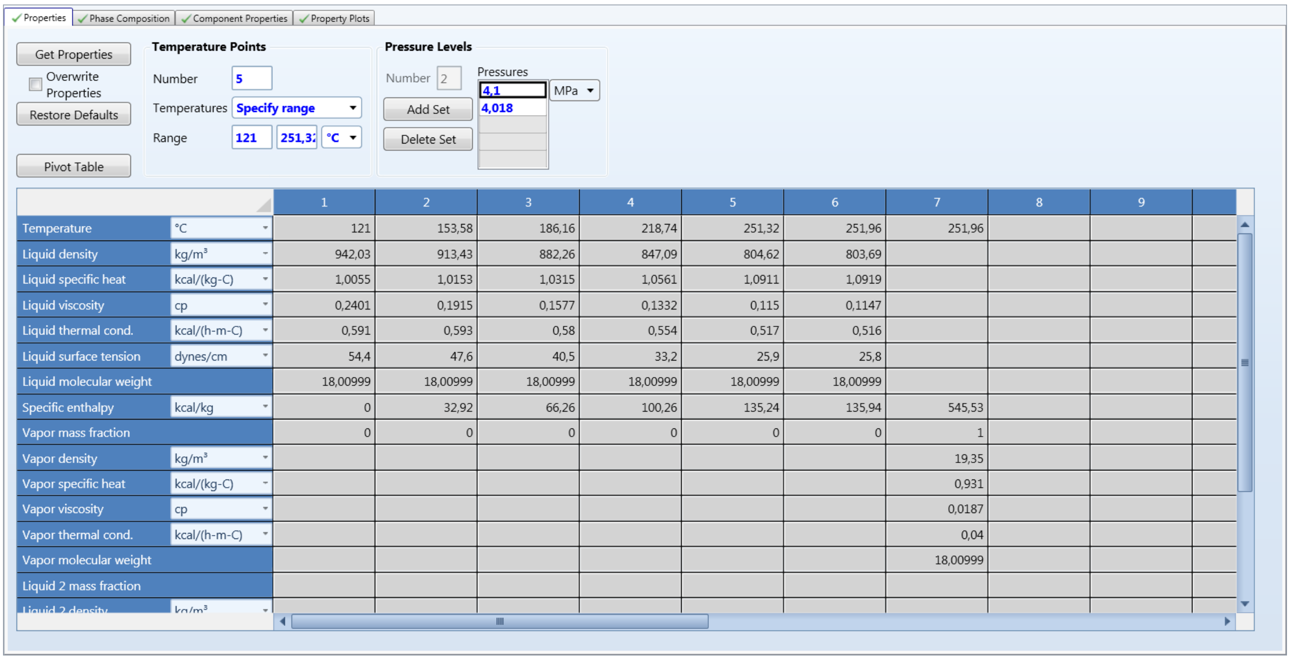


		1	2	3	4	5	6	7	8	9
Temperature	°C	271	270	269	268	267	266	265		
Liquid density	kg/m³	495,6	497,4	498,4	500,2	501,1	502,9	503,7		
Liquid specific heat	kcal/(kg-C)	0,812	0,87	0,852	0,835	0,829	0,82	0,817		
Liquid viscosity	cp	0,127	0,128	0,128	0,128	0,129	0,129	0,13		
Liquid thermal cond.	kcal/(h-m-C)	0,08	0,08	0,08	0,08	0,08	0,08	0,08		
Liquid surface tension	dynes/cm	2,9	3	3	3,1	3,1	3,2	3,2		
Liquid molecular weight										
Specific enthalpy	kcal/kg	10,79	8,39	7,19	4,79	3,6	1,2	0		
Vapor mass fraction		0,52	0,49	0,47	0,43	0,41	0,38	0,36		
Vapor density	kg/m³	51,76	50,86	50,41	49,51	49,07	48,19	47,75		
Vapor specific heat	kcal/(kg-C)	0,661	0,659	0,658	0,656	0,654	0,652	0,651		
Vapor viscosity	cp	0,01	0,01	0,01	0,01	0,01	0,01	0,01		
Vapor thermal cond.	kcal/(h-m-C)	0,03	0,03	0,03	0,03	0,03	0,029	0,029		
Vapor molecular weight										

COLD STREAM COMPOSITION AND PROPERTIES

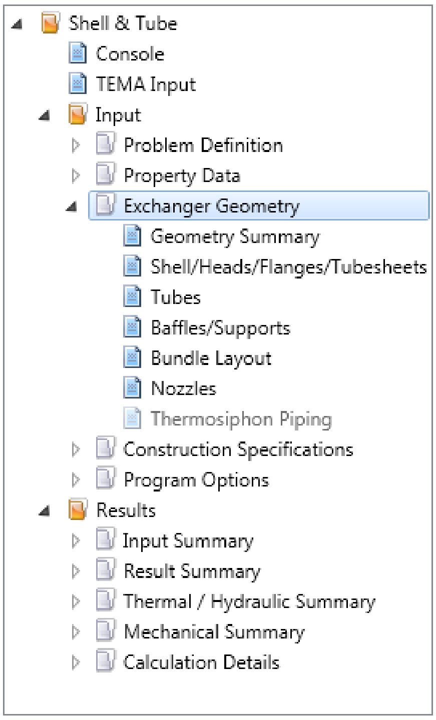


The second method is used when the fluid, or the percentual values of mixed fluids, are known. Every fluid present in the mixture can be chosen from one of the three database present on ASPEN: *ASPEN Properties*, *COMThermo* and *B-JAC*. Once that the substances have been loaded from the database, the molecular percentage composition, or the mass weight percentage composition, has to be provided. In this case, being the cold fluid that vaporizes compounded by water only, one unique component has been selected, using the B-JAC database. Clicking on "*get properties*" the program interrogates the previously presented modulus and it loads automatically the property of the fluid. The updated data are reported in the next form, **Cold stream (2) Properties**, in function of the temperature and for the indicated set of pressure (e.g. inlet and outlet pressure). Also, the phase composition and the graphs of all the inserted property data can be visualized in the other forms.



		1	2	3	4	5	6	7	8	9
Temperature	°C	121	153,58	186,16	218,74	251,32	251,96	251,96		
Liquid density	kg/m³	942,03	913,43	882,26	847,09	804,62	803,69			
Liquid specific heat	kcal/(kg-C)	1,0055	1,0153	1,0315	1,0561	1,0911	1,0919			
Liquid viscosity	cp	0,2401	0,1915	0,1577	0,1332	0,115	0,1147			
Liquid thermal cond.	kcal/(h-m-C)	0,591	0,593	0,58	0,554	0,517	0,516			
Liquid surface tension	dynes/cm	54,4	47,6	40,5	33,2	25,9	25,8			
Liquid molecular weight		18,00999	18,00999	18,00999	18,00999	18,00999	18,00999			
Specific enthalpy	kcal/kg	0	32,92	66,26	100,26	135,24	135,94	545,53		
Vapor mass fraction		0	0	0	0	0	0	1		
Vapor density	kg/m³							19,35		
Vapor specific heat	kcal/(kg-C)							0,931		
Vapor viscosity	cp							0,0187		
Vapor thermal cond.	kcal/(h-m-C)							0,04		
Vapor molecular weight								18,00999		
Liquid 2 mass fraction										
Liquid 2 density	kg/m³									

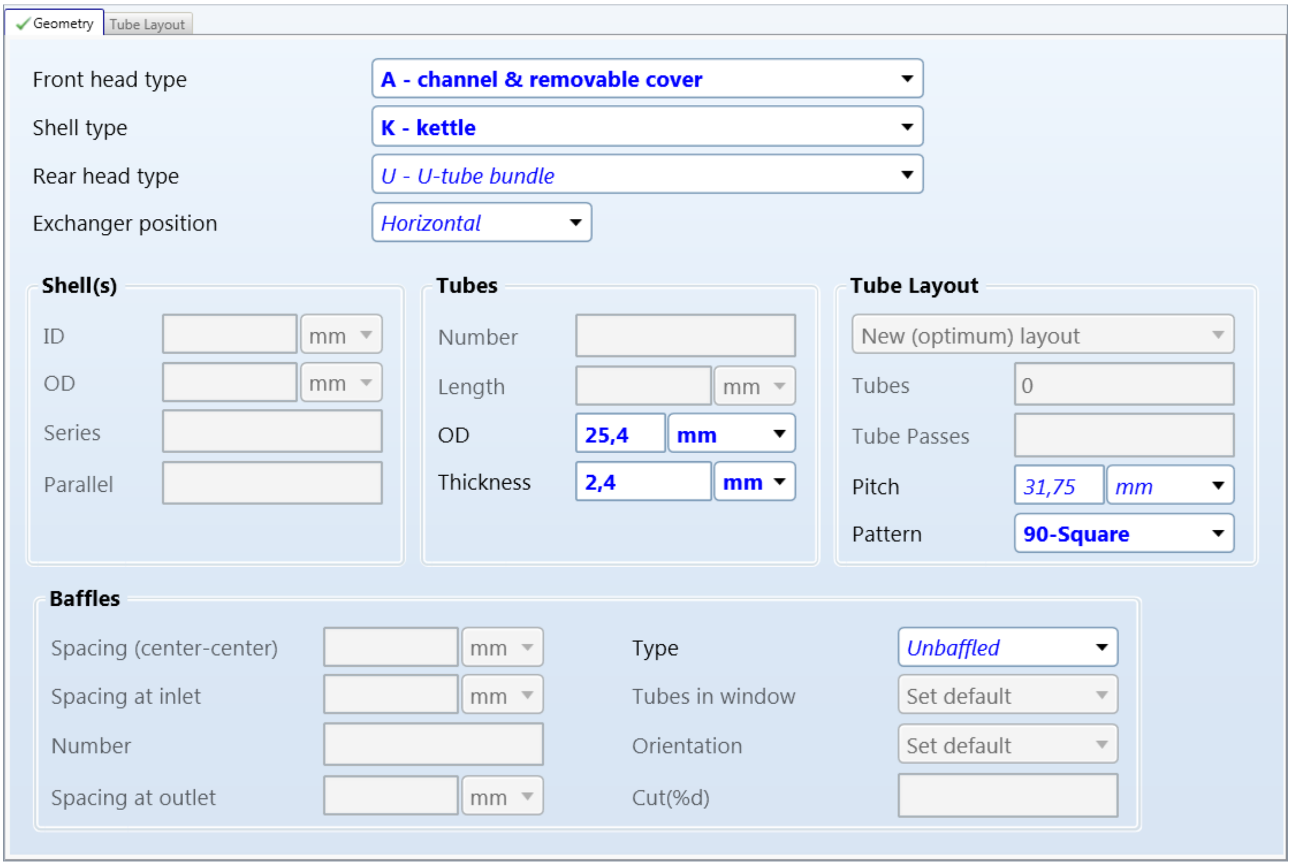
5.4.3. EXCHANGER GEOMETRY



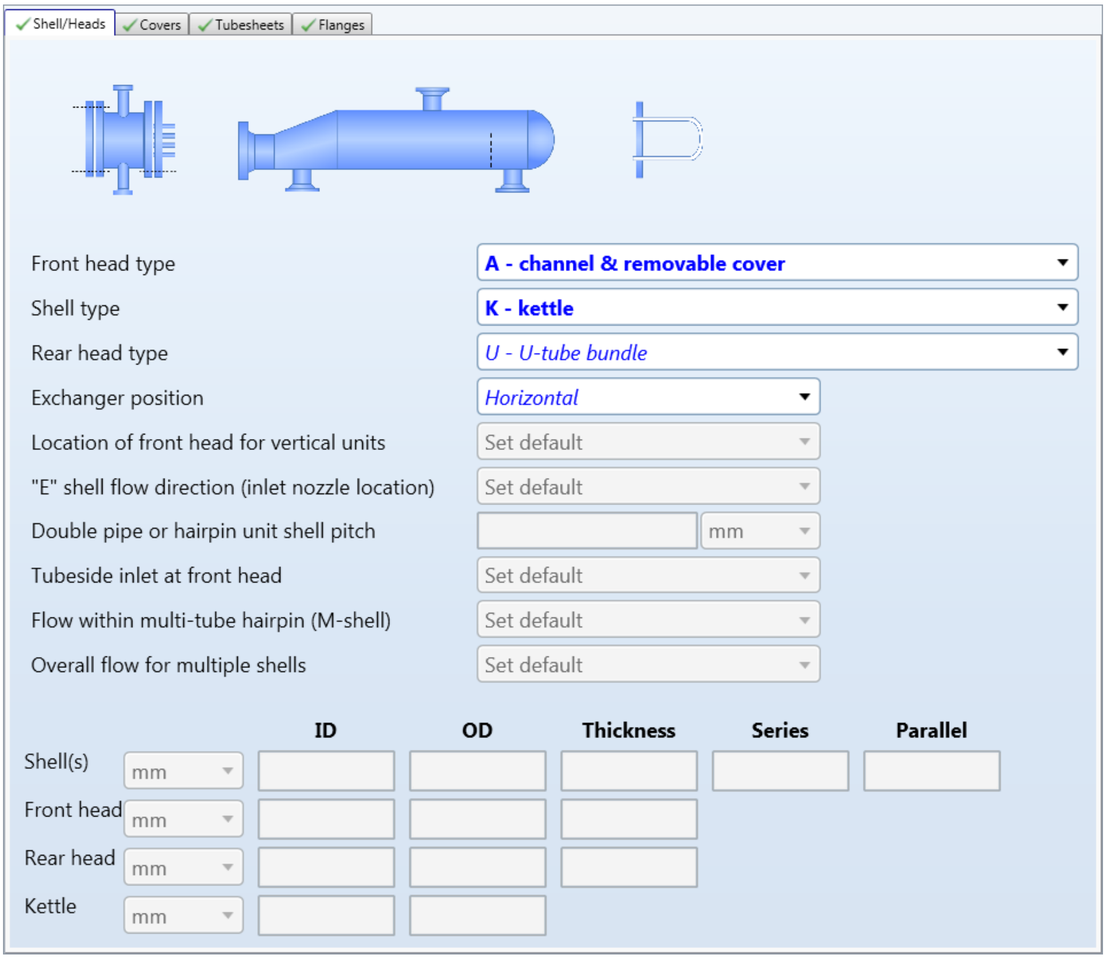
The following group of forms and sub-forms is designed to accept the input data of the shell and tubes heat exchanger geometry and configuration. In particular will be possible to implement all the characteristic regarding the shell and bundle type, the flanges, the tubesheet, the baffles and the nozzles. In order to make the next screenshots more visible, the list of selectable forms is shown on the right as it can be seen in the real program.

GEOMETRY SUMMARY

This is a summary sheet in order to make the data acquisition easier. However, some data can be directly edit from here. The following forms are more specific and allow the insertion of very detailed data. It is appropriate to remember that not all the possible insertions will be thoroughly investigated, while someone will be considered during the subsequent design step.



SHELL/HEADS/FLANGES/TUBESHEET



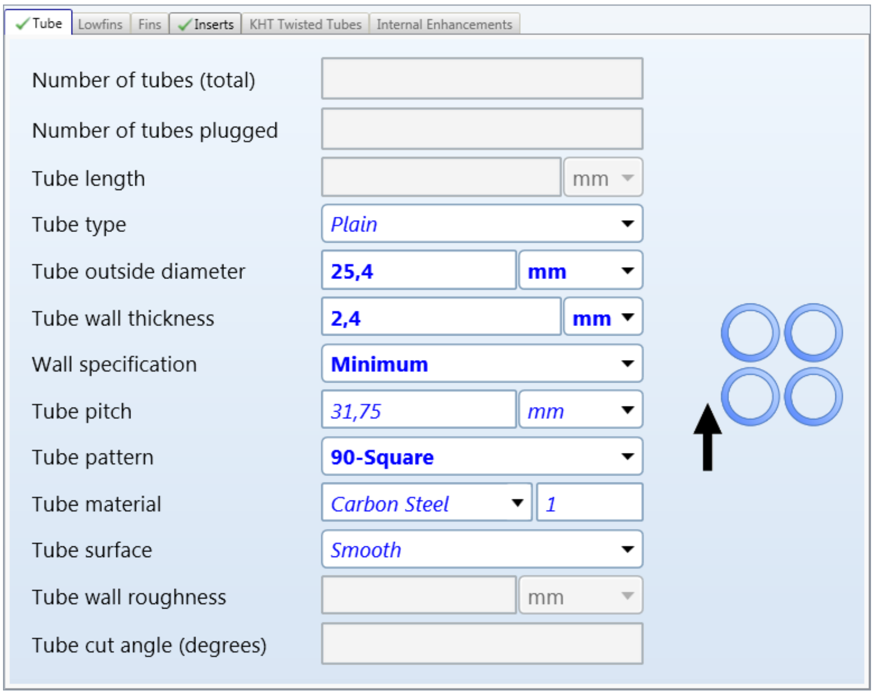
In this form, the first statements on the heat exchanger geometry have to be done. In particular, the first section requires to insert the front and rear head type and the shell type. On the shell, type the choice is mandatory. Since the heat exchanger type is a vapour generator, it must be a "K – Kettle" type in order to allow the correct expansion of the steam. Because both fluids, in the shellside and tubeside, have a higher fouling coefficient, the heat exchanger requires a removable bundle in order to allow the cleaning procedure in the shellside. Notice that the U shape of the tubes is the classic one adopted in removable bundle configurations, mainly for economical reason. Indeed, it is the default choice of the program if K-kettle type is selected. The most important drawback of this configuration is that it doesn't allow the mechanical cleaning inside of the tubes. For this reason, it is important to make sure that chemical cleaning is possible from the tubeside. Straight removable bundle can be done, but they don't allow the thermal expansion of the tube, resulting in very thick and expensive tubesheets. The U removable bundle carries with him the classical ellipsoidal weld shell cover, that cannot be dismounted. The chemical cleaning can be done flushing the whole device without opening the head. However, have the full access at the tubesheet represent a high advantage. Indeed, it is one of the points in which the dirt concentrates more. Moreover, the tubesheet-tube joint is also a critical point for the welds, so, frequent inspections are generally planned. Due to these considerations, and the possibility to access at the tubesheet without disconnect the inlet and outlet pipes, the A configuration has been chosen over B. The C type, instead, was excluded because it allows a less practical dismounting with the impossibility to exclude and remove the head channel before the bundle. For this reason, the *A – channel & removable cover* has been selected in the menu. As last operation, in conformity with the specified requirements, *horizontal heat exchanger* position have to be inserted. All the other grey spaces do not require input because are parameters further implemented in the other steps or resulting as output from design step. Note than the program would allow me to insert manually the inner and outer shell diameter. This enable the simulation of leakages and by-pass computing the gap between the baffles and the shell. With the insertion of this parameters, the leakages and by-pass simulation lead to introduce coefficients that, taking into account during calculation, will reduce the heat exchange.

The parameter described above are the only parameters mandatorily required in this form during *"Design (sizing)"* calculation mode. In the other section the tubesheet type preferences could be selected but the classic type is generally used with exception for very specific cases. However, the tubesheet joints configuration is a parameter on which may be important to linger. Indeed, the tubes can be connected at the tubesheet in different way. In particular, they can be:

- *Expanded only*: During the assembly phase the tube is inserted in the tubesheet hole, on which some circular locations may have been already machined. After, an internal working produces one or more annular relieves that lead the tube to expand in the pre-worked locations. The joint is created by means of the generated internal stresses and the corresponding deformations. This method results great for the mechanical coupling resistance and sealing capacity, but only until a certain value of pressure.
- *Seal weld*: with this choice a simple weld is indicated. Conceptually it does not provide high joint mechanical resistance, but a complete sealing capability. It is only used for application at atmospheric pressure.
- *Strength welding*: indicates a weld performed with characteristics such that, other than a complete sealing, also the mechanical resistance of the joint is ensured.
- *Expanded and seal welding*: it characterizes a joint in which both, a weld and the expanded reliefs are present. In this case the weld takes care of the sealing capability of the coupling, while the forced engagement ensures the mechanical resistance.
- *Strength welding and expansion reliefs*: it is the opposite case of "Expanded and seal welding" in which the weld ensure the mechanical resistance and sealing, while the expansion reliefs act as support. This is used for high pressure equipment.

Because of the high shell and tubeside pressures in the study case, a *"Strength welding with expansion reliefs"* is required. This will ensure the mechanical resistance of the joint and the complete separation of the water from HCBN. Moreover, the expanded reliefs are also useful because allow the tubes to stay in correct position during the welding procedure. Also, different types of weld are possible. Where possible, the simplest method is to leave three millimetres of the tubes protruding the tubesheet and made fillet welds. However, this method can be adopted only in case of mid-lower pressure devices. In case of high-pressure devices, the machining of welding bevels and a full penetration weld is required. Another application in which this procedure is not allowed is, for example, in the case of device working it the plastic polymerization (e.g. polyethylene). Indeed, the three millimetres protruding the tubesheet can represent an obstacle for the stream that solidify around the obstacle, leading to the passages' obstruction. However, the study case is not one of this cases, so the three millimetres solution can be adopted, and this information must be implemented in the program. In this section, if required, also the expansion joint parameters can be inserted. The expansion joint is a shaped part to be inserted in the middle of the shell, and that allows the thermal expansion of the shell. Generally, it is used in the double welded tubesheet in order to insert a degree of freedom and reduce the thickness of the tubesheet. It is important try to reduce the use of expansion joints because they introduce a weak point in the heat exchanger, e.g. fatigue crack. Moreover, they can only be used for low pressure equipment, e.g. lower than 20 bar. For the U-shape reboiler the expansion joint is not necessary because the bundle already has the required degree of freedom that allow thermal expansion.

TUBES

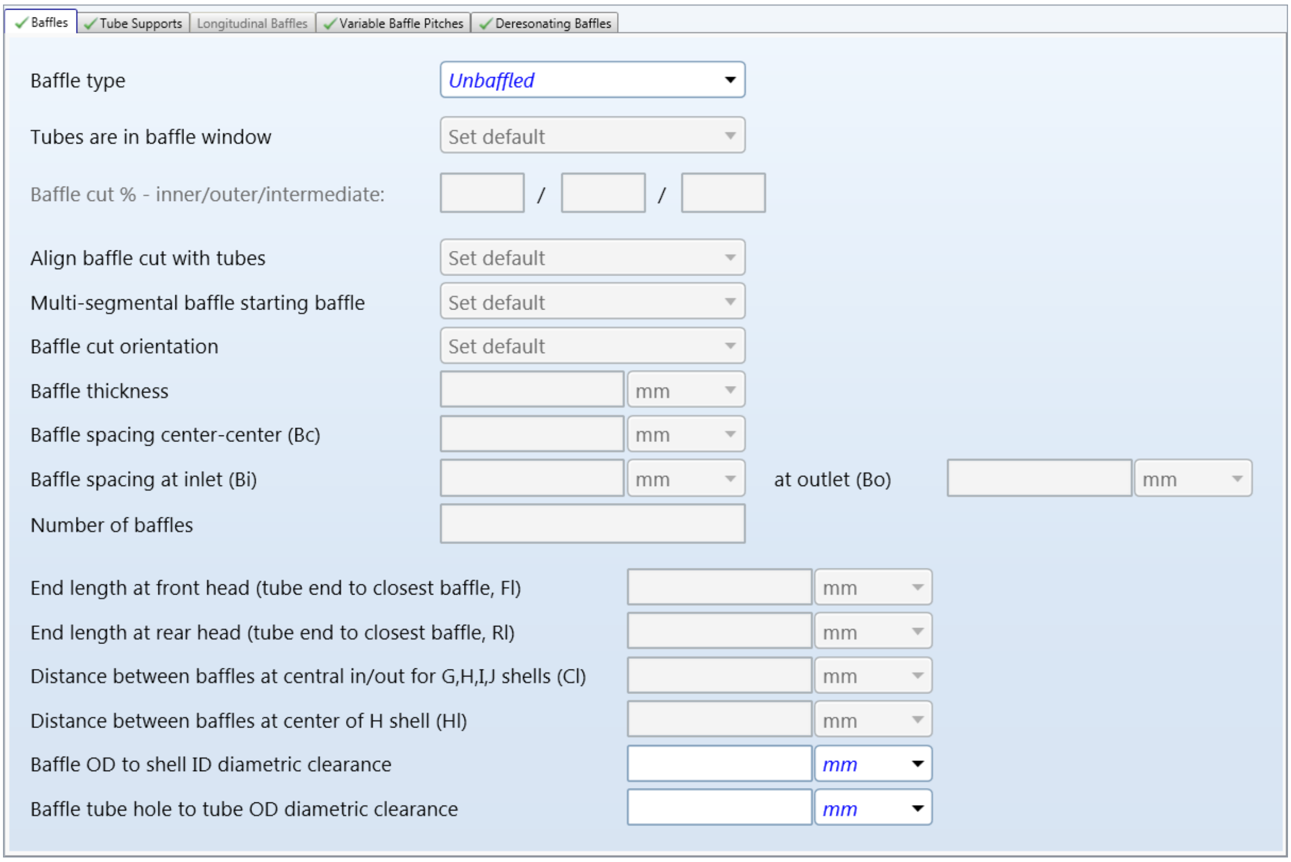


In the Tubes form the tubes characteristics and dimensions can be implemented. Since we are in *Design (sizing)* calculation mode *Plain tubes* have to be selected. Indeed, *finned tubes* will be inserted, only in the case of necessity, during a further step. Later on, the customer preferences of 25,4 mm diameter and 2,4 mm thickness must be inserted instead of the most used 19,05x2,11mm tubes, that has been proposed by default. The next choice is depending from the specification of the tubes that are going to be bought. *Average* or *Minimum wall* can be selected. With the *Average wall* option, the supplier will provide a tube with a tolerance plus-minus 12,5% on the thickness. This means that it could vary between the 2,1÷2,7mm range. *Minimum wall* option, instead, means that the tube only has the upper tolerance. So, in the previous example its thickness could vary between 2,4÷2,7mm. Given the customer request, *minimum wall* option must be chosen.

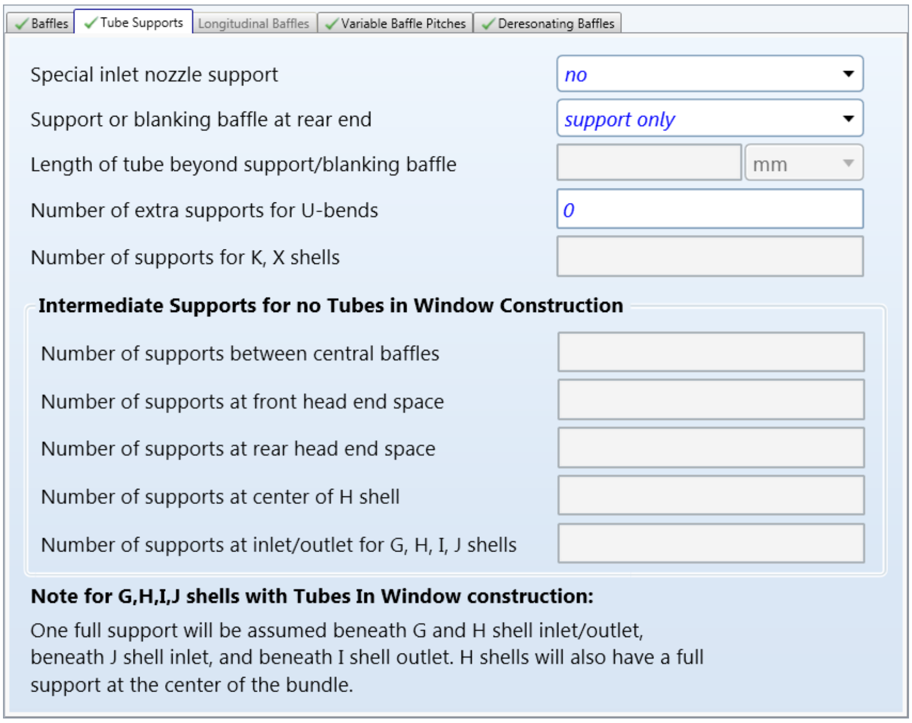
The differences among tube patterns have already been discussed during the "*Generality on heat exchanger*" section. Despite the higher volume that this layout requires, a *90° square* pattern must be implemented in order to allow the external mechanical cleaning. For the thermal design the *carbon steel* indication it is enough for the material. Further specification will be inevitably done during mechanical design.

The *smooth tube surface* parameter refers to the finishing degree level. The label "*smooth*", that will be our indication, refers to a great surface finishing; "*commercial*" instead to a rough one. A specific roughness degree level can also be specified in cases of very smoothly surfaces. This last option lead to a more accurate computation of a tubeside pressure drop if a specific level of roughness has been inserted. A curiosity to be notice is that in English with the word tube is indicated the heat exchanger pipes, that present also an accurate external surface in order to reduce the pressure loss. This requirement it is not present on the general use pipes, that have rough (e.g. commercial) external surfaces.

BAFFLES/SUPPORTS

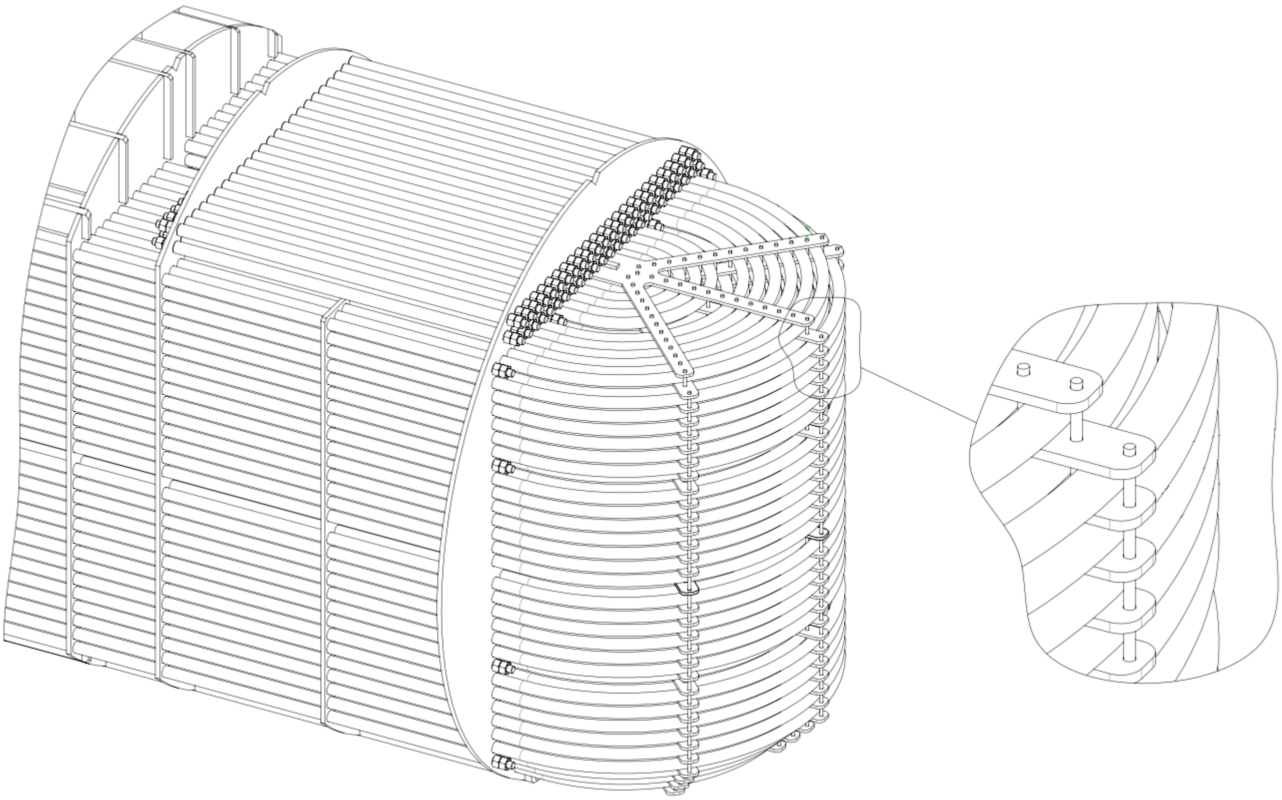


It must be noticed that the program makes distinctions between baffles and tube supports. While with the term *baffles* are indicated the part responsible for the guidance of the fluid, the *supports* only have the to carry the tubes in their correct position. Because this device is a Kettle pool boiling, so, it doesn't have a real shell stream. Only the tube supports will be present, and "*unbaffled*" must be chosen.



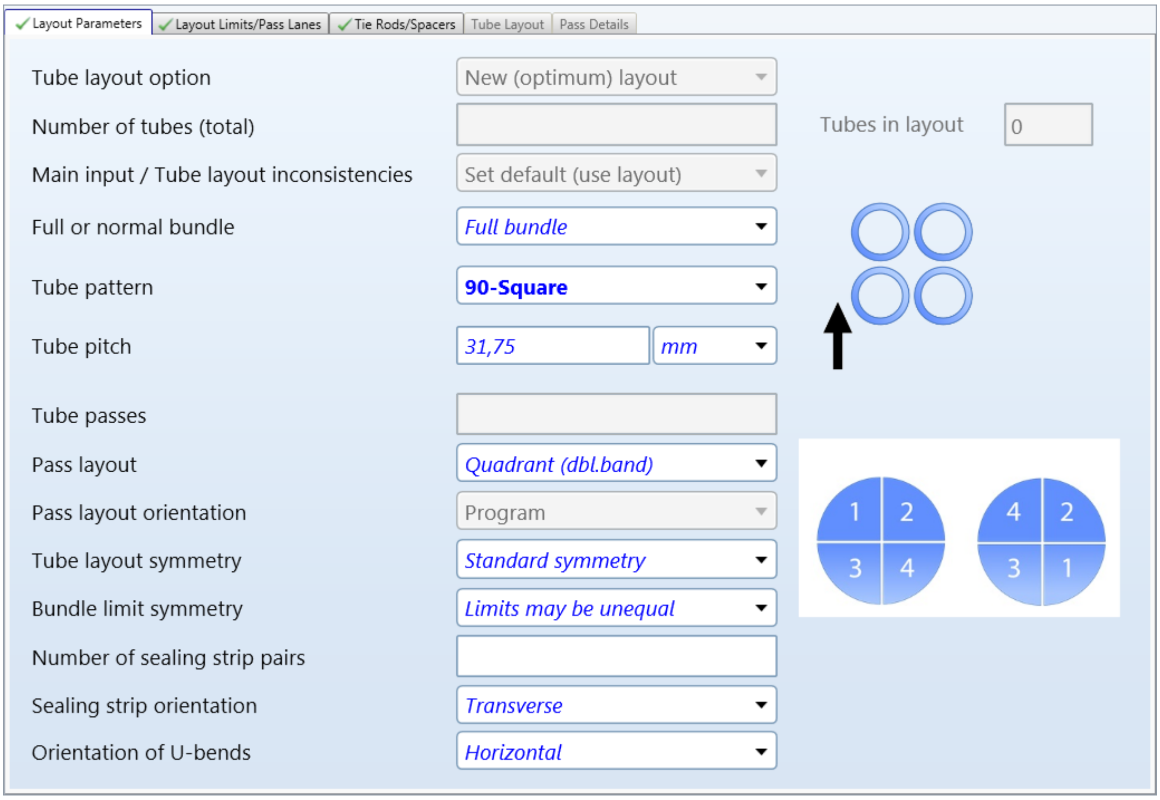
Changing the form from *“Baffles”* to *“Tube Supports”*, no special inlet nozzle must be selected. It can be observed that the tubed curvature present at the end of the U-type bundle is a very critical point for vibration. So, a baffle must be inserted there selecting “*support only*” for the second space.

The number of supports and the length between them instead have to be left uncompleted. Them will be freely choice by the program or be subjected to further refinement in case of vibrational problem. Also, several extra supports for the U-bend can be inserted in order to avoid vibrational problem at the end. These devices are angular support placed in the turning of the bundle, as can be seen in the image below. Their aim is to divide the critical part in some smaller segments, changing in this way the vibrational mode of the bundle.



The other cards contained in this level are used only in the case in which variable pitches are required. This arrangement is used only if no one of the other remedy solve the vibrational problem in the bundle, so, they will not be described during the design step.

BUNDLE LAYOUT

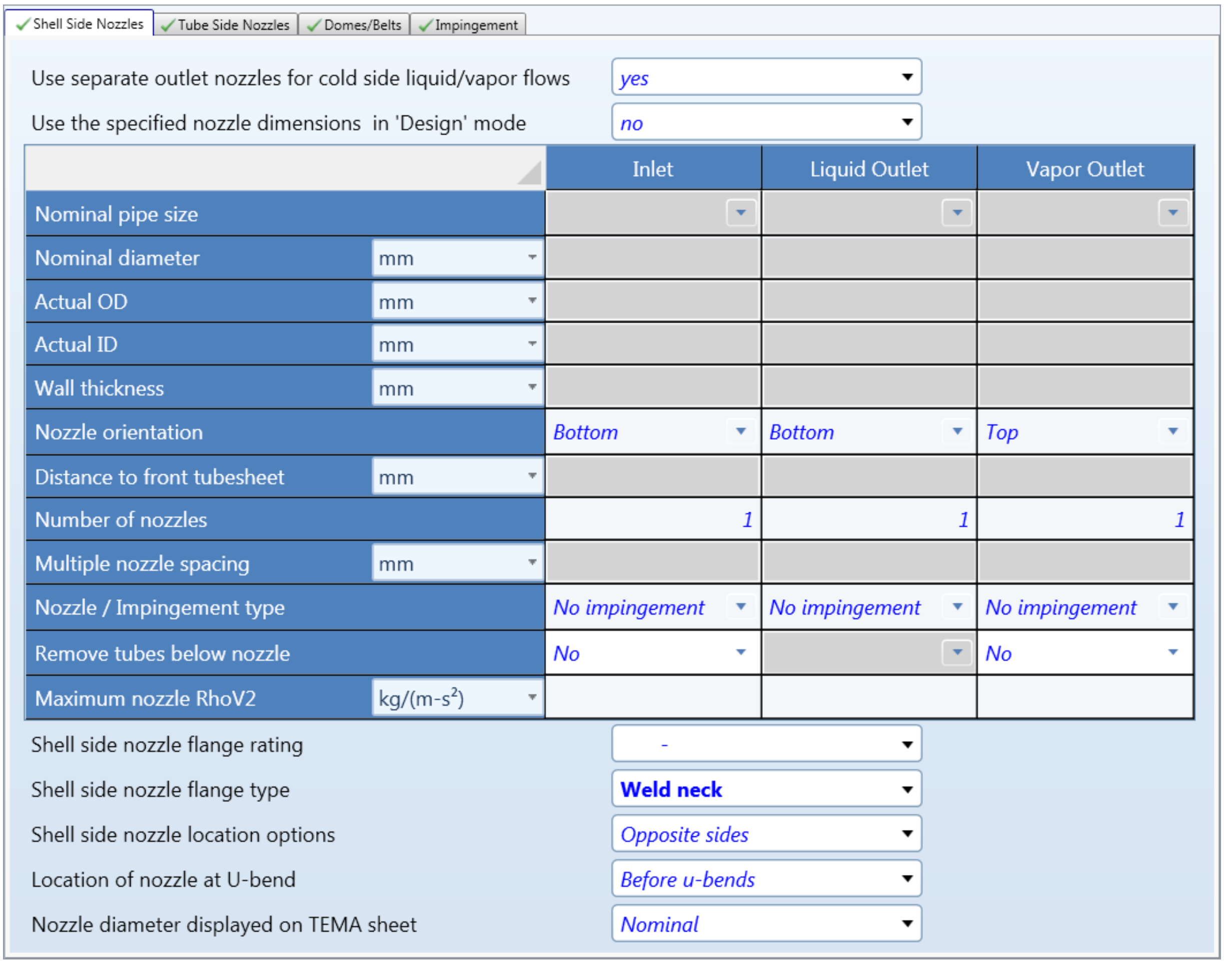


In this form, always present in the "Exchanger Geometry" section, it is possible to implement the bundle configurations. A first parameter that can be selected is *Full* or *Normal bundle*. The difference between these two types is that, in the first one, the whole bundle is filled with tubes, while in the second one the tubes are not present in the area towards the nozzles. Generally, this is an important choice because the tubes placed suddenly before the shell nozzles may generate a pressure drop. On the other hand, as already discussed, the flow coming from the nozzle can lead to a high erosion of the piping. Being this a kettle reboiler, where the shellside fluid is almost stationary, a *full bundle* configuration can be adopted.

Now, the pass layout can be analysed. Being a U bundle, at least two passes of tubes will be present. More passes can be done, and they would allow a better exploitation of the temperature difference. However, the shellside pressure drop will be increased, and the same thing will happen at the cost. Furthermore, the software, Aspen Shall and Tubes, does not allow to insert more passages if the allowable pressure drop has been exceeded. These considerations lead to an analysis of convenience whether it is better to increase the number of passes or change the heat exchanger configuration. Also, a different type of pass layout can be chosen between *Quadrant*, *Ribbon* or *Mixed*. Quadrant is the most common configuration, while Ribbon and Mixed are merely used in order to solve vibration problems. Indeed, the bundle layout does not influence in a significant way the heat exchanged, despite the tube passes. As last observation, the default choice, Horizontal orientation of the U-bend, can be left.

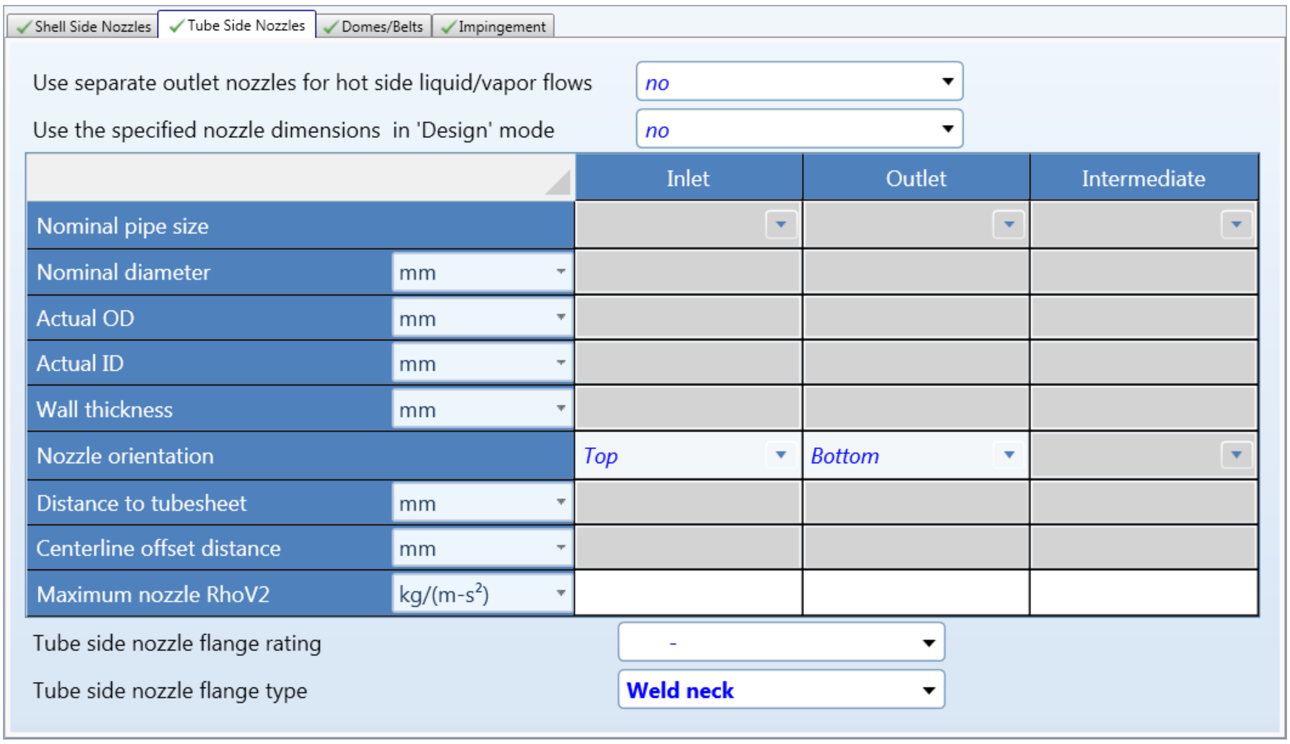
NOZZLES

The Nozzle module is the last form in the *"Exchanger geometry"* set. It contains four sub-forms, two of which will be analysed now. The first is the "Shell Side Nozzles". Being the heat exchanger a kettle reboiler, it vaporizes the water with the pool boiling configuration. The pool boiling is a layout in which the bundle is always completely covered with the water that, contained in the shellside, represents the cold fluid. The vaporizing fluid enters from an inlet nozzle placed at the bottom, and it will exit from the shell in both the liquid and the vapour phase. The liquid outlet, also placed at the bottom, has the function of guarantee the correct recirculation of the water and to adjust its level. The control of the water level is an important aspect of the configuration because it must always be above the tubes level, but it cannot increase too much. As a matter of fact, an excess of water in liquid phase would occupy the space required by the vapour to expand, leading to a dangerous increment in the shellside pressure. A simple but functional method to regulate the outlet flow rate, and so the level of the water, is to insert a weir. When the elevation of the fluid overcomes the obstruction, it flows out through the liquid outlet nozzle placed beyond. Nowadays this method is less used, because the regulation is assigned at the pumps that, guided by level sensors, regulate the inlet flow rate. In every case, some blowdown nozzles are always present. The choice that has been done is to use separate outlet nozzles for the liquid and vapour flows. The vapour phase outlet has been placed upward, and it can be compounded by several nozzles. In this case the mass flow rate in the shellside is not excessively large, so, probably only one nozzle will be required. However, if more than one outlet will be necessary, the program will generate a warning. In the image below, the choices already discussed are shown. The dimensions of the nozzles have been left free, and will be optimized by Aspen. However, most of the time the heat exchanger to be designed will be inserted in a plant already working. In that case the dimensions of the nozzle are dictated by those already mounted.



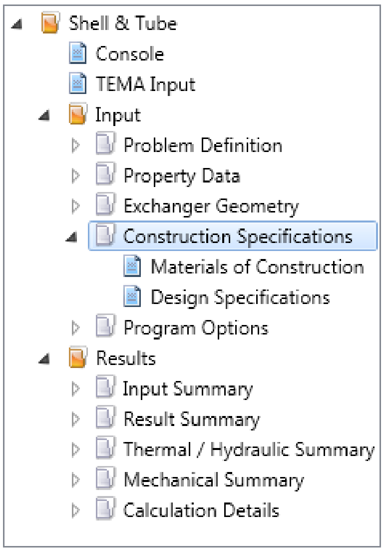
The impingement plates, already discussed during the theory part, are inserted to avoid the damage of the tubes caused by the inlet flow of the shellside. However, its presence will increase the pressure loss. For this reason, their insertion is postponed at the following steps, in the case in which the ρv^2 index will be too high or vibration problems will be found. Regarding the flange type, the possible choices are *"Weld neck"*, *"Slip on"* or *"Lap joint"*. In petrochemical context the choice almost always falls on *"Weld neck"*. As last thing, the preferences on the location of the nozzles can be pointed out. In particular, for the shellside, the inlet and the outlet nozzles are delineated to be opposite in side, rather than on the same side. Furthermore, the indication "before the U-bend" will ensure that the flow will not be disturbed by the presence of the bundle.

Moving on the *"Tube Side Nozzles"* form, more or less the same decision has to be done. In this case, the hot fluid is flowing. It is represented by HCBN in liquid and vapour physical state for both, inlet and outlet. However, separate nozzles cannot be used because it flows in the bundle, entering from the top, and going out from the bottom. In the same way of shellside argumentation, the dimensions of the nozzle have been left free and the weld neck flanges have been denoted. In the image below can be seen how these choices were implemented. Please note that also other parameters can be inserted, and some other forms would be available in the case in which impingement devices were selected.



5.4.4. CONSTRUCTION SPECIFICATIONS

The construction specification set is compounded by the "Materials of Construction" and "Design Specifications" forms.



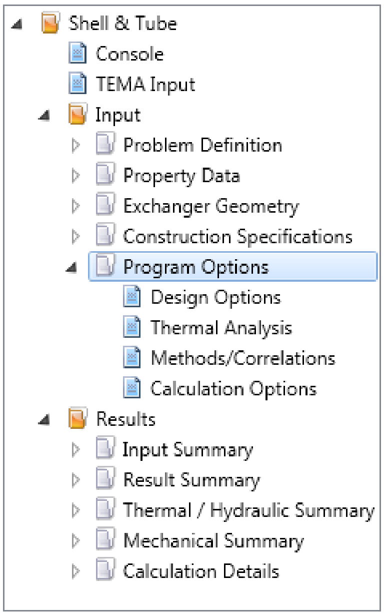
MATERIAL CONSTRUCTION

In "Material of Construction" the materials for all the heat exchanger parts can be outlined. In the program setting the default value can be inserted, and it will automatically be assigned at all the components. However, several times different parts require different materials, and each value must be inserted manually. In the case study, all the parts will be made of carbon steel, so, if the default value has been set in a proper way, modifications are not required. Other forms allow to insert the accessory parts, e.g. gasket, which will not influence the thermal design, but will be inserted inside the final database. If a particular material is required for the tubes, and it is not present in the database, the property needed for the thermal calculation can be specified manually in this section.

DESIGN SPECIFICATION

The values which can be inserted here have a minor importance for the thermal calculation, due to the fact that their influence is not very high. "*Normal*" service class have been inserted because the application of the heat exchanger cannot be considerable as "*Lethal*" or for "*Cryogenic temperatures*". "TEMA class R" means that the intended use is for petrochemical refinery service, as stated from the client. Some important parameters, instead, are the design Pressure and Temperature. These values are different from the operative temperature, and they are used to make a preliminary sizing of the device. The minimum Design Metal Temperature, as well as the test parameters and the 3mm of corrosion allowance, will be faced soon. However, as already said, they are not parameters that strongly influence the thermal design. As last declaration, the full radiography controls on the tubeside and shellside can be assigned.

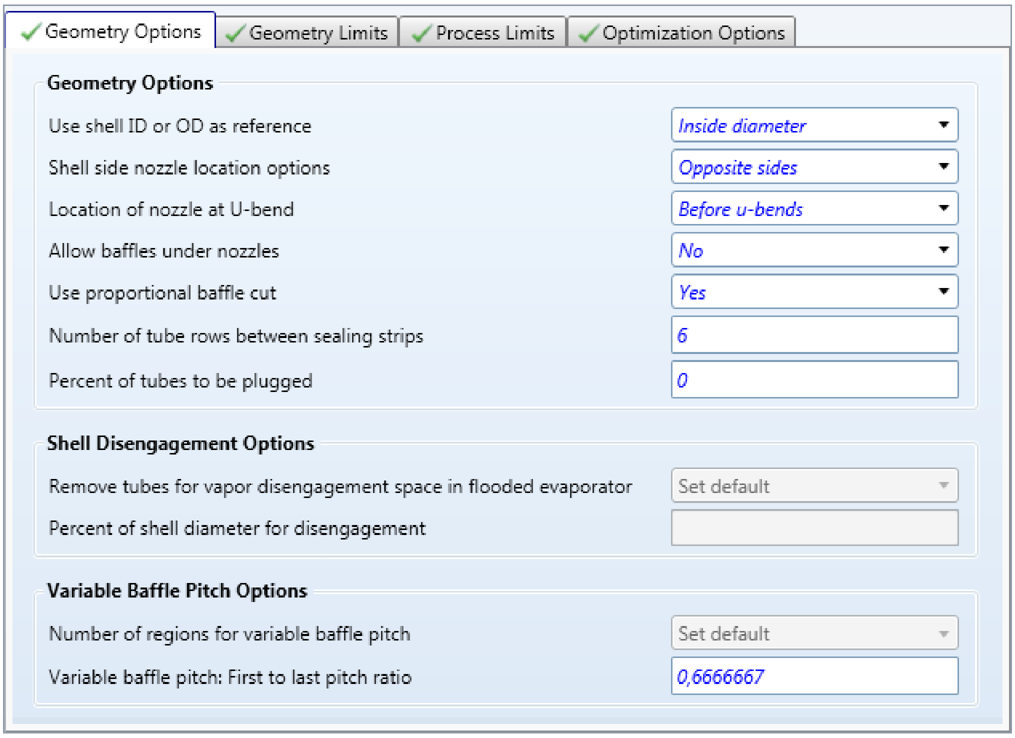
5.4.5. PROGRAM OPTIONS

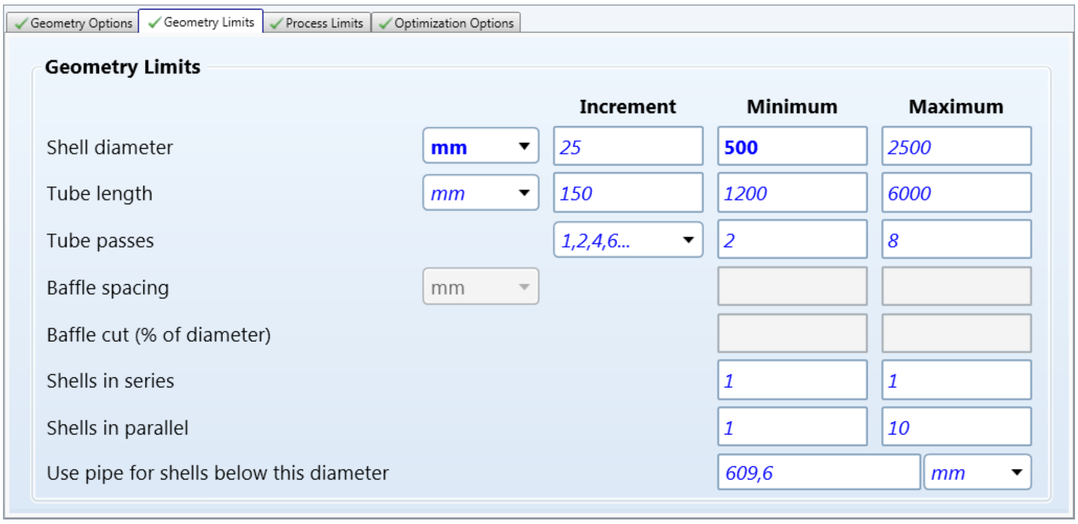


Until now the process and the heat exchanger parameters have been set. The program options section allows to insert the limits within whom the program will operate. Many forms and sub-forms are available in this part.

DESIGN OPTIONS

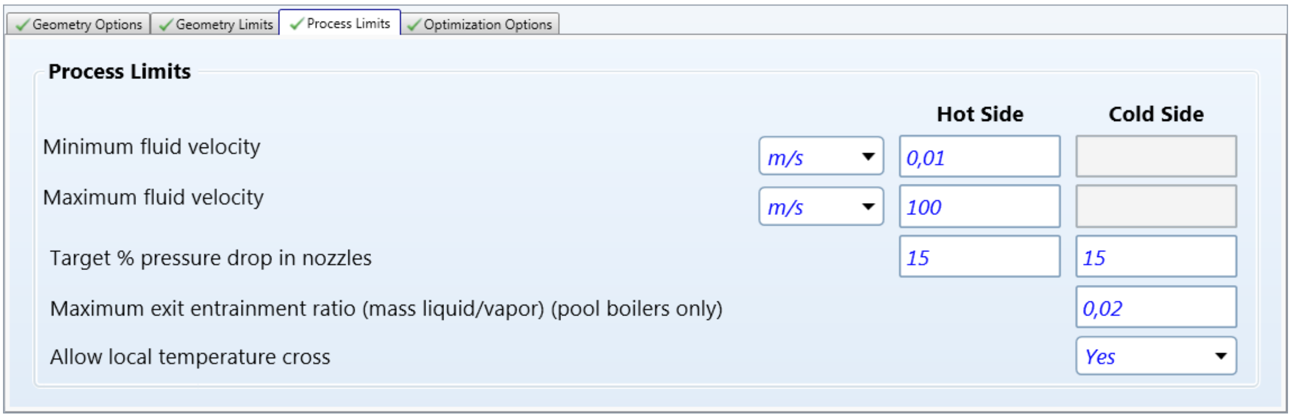
Starting from the first sub-form, *"Geometry Option"*, it can be specified on which diameter the program must base its calculation for the preliminary mechanical design. Indeed, if the heat exchanger is a small size device, the shell would be obtained from a pipe, and the external diameter must be designed as reference. The maximum diameter of commercial pipe easy to find on the market is 609.6mm, 24 inch, so, if the shell exceeds this dimension, a rolled plate would be necessary, and the inlet diameter would be the concern. The last possible option is that the device has a shell so thick that must be made from a machine worked forged piece. At this point, some nozzle configuration options are proposed again, and some new one can be implemented. As example, the choice of don't "allow the presence of baffle under nozzles" can be done. In this way, the inlet or outlet flow will not be disturbed. An interesting discussion can be done on the parameter "percentage of tube to be plugged". This value indicates the number of tubes which can be plugged, ensuring the required heat exchange, when failure occurs on them. As a matter of fact, it could happen that some tubes, during the years, can break, corrode, fail for welding crack or for erosion, and they need to be excluded in order to guarantee the correct operation of the device.



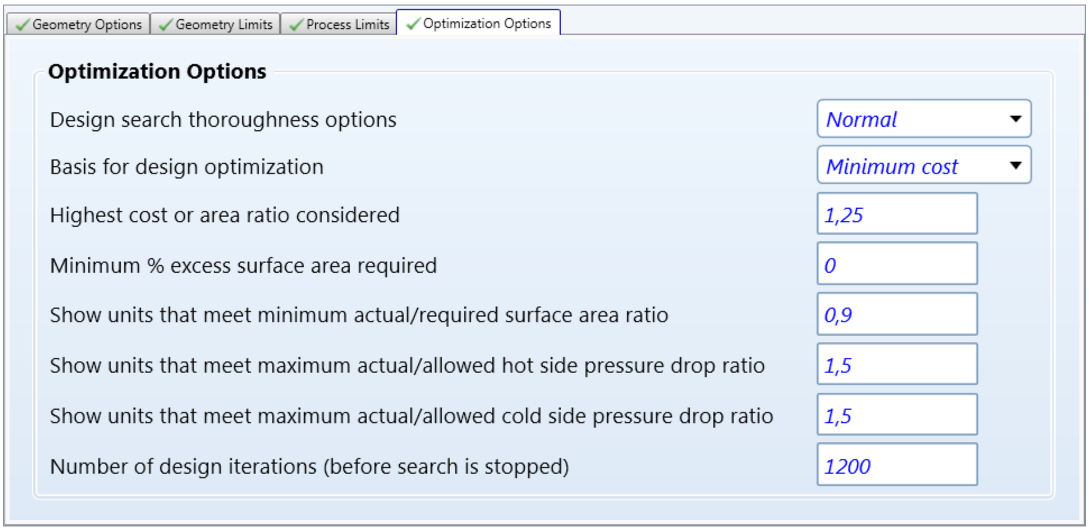


The second sub-form is *"Geometry Limits"*. A very important parameter, here, is the possible range for the diameter computation and its increment. The program, during the design activity, will iteratively calculate the possible solutions in this interval of values. The same conceptual thing must be inserted for the tube length, where in the case of U bundle, it refers at the length of the bended tube. Indeed, in this case, it is recommended not to exceed the six meters, due to the fact that the maximum length of the commercial tubes is about fifteen meters, and they must be bend in the U shape. Proceeding, the tube passes, and the number of possible shell in series or parallel can be set or limited. Being a U bundle, the minimum number of passes must be two, while the maximum is left at the default value, eight. Once that all these parameters have been set, and the calculation have been launched, the program will iterate the computation in order to search for all the possible solutions in the inserted ranges. In case in which no solution will be possible, or convenient, it will consider the possibility to make more devices working in series or in parallel. Two devices in series divide among themselves the heat drop, so it is useful when high difference in temperature fluid must be managed. Two devices in parallel, instead, allow to split the mass flow rate. It is worth to observe that a kettle device cannot be made by more shells in series, because in the further stages the water will enter already in the vapour phase. Indeed, in the case in which superheated steam would be produced in more stages, the second one can simply be a BEU heat exchanger type. So, the range of the possible shell in series has to be limited to one, while the number of parallel devices can be left as proposed by default.

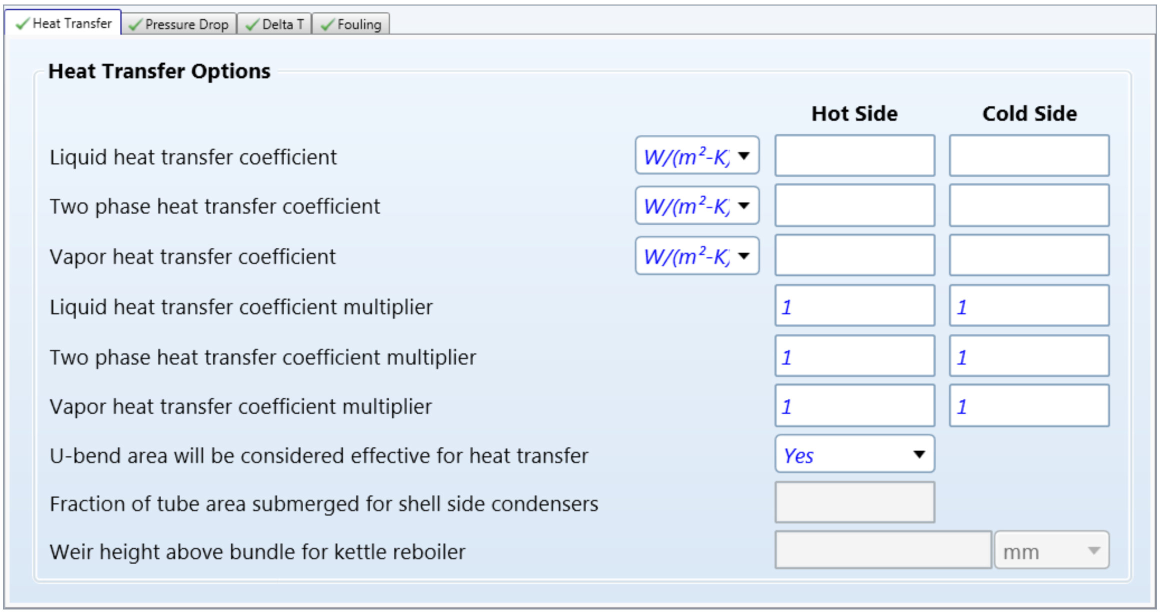
Proceeding, the "*Process Limit*" sub-form can be selected. The minimum and maximum fluid velocity becomes an important parameter only in case of very dirty fluids. Indeed, the velocity of the fluids impacts on the heat transfer coefficient, on the tube erosion and on the possibility of fouling. For this reason, a minimum velocity of 1 *m/s* must be always inserted, with exception for very dirty fluids, which require a higher value. Please note that, giving the kettle configuration, the shellside velocities are not required. Next parameter allows to set the percentage of the pressure drop that will be present in the nozzle. Usually it is around 10÷15% of the total pressure drop, so a concordant value has been laid down. Now, the entrainment ratio value must be analysed. This option only applies to pool boiling configuration and it conceptually expresses the percentage value of the outlet mass which can be in liquid phase. This value is very important because it leads to compute the difference between the shell diameter and the bundle one. Indeed, higher is the difference between them, lower will be the liquid droplet in the outlet steam. Or, in other words, lower is the entrainment ratio, higher will be the diameter of the shell, in order to allow the steam to expand more and to release the water droplets. Its value can be set at 0,02 if dry steam is required, as in this case. The discussion of this section closes with the temperature cross that can be accepted in a steam generator like this, so, its value will be left on "yes".



The last sub-form is the *"Optimization Options"* in which some parameters can be inserted to discard the possible configurations. The first decision to be implemented is if the design is devoted to minimizing the heat exchanger volume or the cost. This will influence the choice of the program between different output possibilities, e.g. it decides whether to use one big heat exchanger or two smaller device in parallel. A bunch of other important parameters can be set in order to limit the possible outputs. In particular, it can be chosen to visualize only the solution that offers, at least, a well-defined higher percentage of exchange area with respect to the one required. On the other hand, the user can require also to show the solutions that would have, within certain limits, an exchange area lower that the required one, or a higher pressure drop. This allows to take them in consideration as "*Near*" option, considering that some modifications can be done in the further steps.

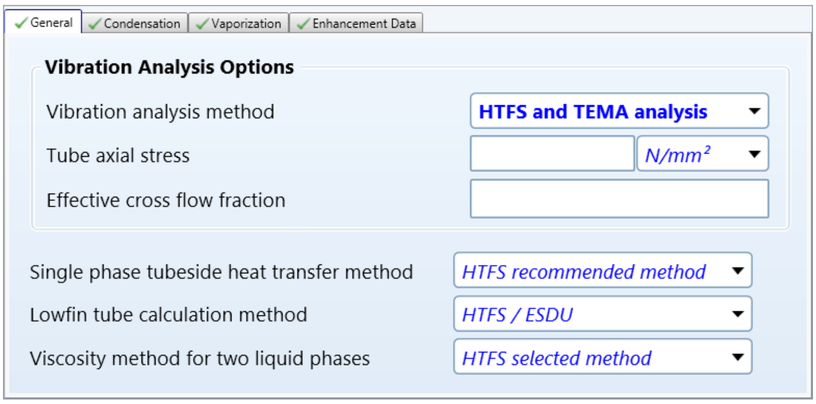


THERMAL ANALYSIS



In the *"Thermal Analysis"* set of sub-forms, the thermal property of the fluid can be inserted. Inside *"Heat Transfer"* the fluids heat transfer coefficients can be added. In that case the program will skip the steps to find them out from the fluids property already indicated. The next important parameter gives the possibility to choose if the U-bend portion of the bundle must be considered as heat exchanging or not. In the kettle configuration the whole tubes area is covered by water, so the whole area must be considered. However, in some other configurations the area is only partially exchanging, so it should be neglected. A last useful parameter is the elevation of the weir above the bundle in order to keep the tubes always covered. However, this value can only be implemented in the rating mode. All the other forms in this section allow to choose some criteria for the mathematical models which will be used in order to develop the simulations. These procedures allow to develop and take into account the variables, e.g pressure drop, thermal gradient and fouling, during the computational step.

METHODS/CORRELATIONS AND CALCULATION OPTIONS



It is now time to set the vibration analysis method. Three options based on two algorithms are possible. The alternatives are to perform an analysis built on the HTFS method, on TEMA or using both of them. The Heat Transfer and Fluid flow Service, HTFS, is a recent and complex method with respect to TEMA. It is more reliable but requires more computation time. Generally, since TEMA standards is required, both analyses are used. The use of the TEMA analysis only is discouraged due to the approximations that would lead to not consider possible cases of vibration. The remaining forms are dedicated to the mathematical method to implement the calculations, and the explanation will require an in-deep analysis of them, which would not be the argument of this thesis.

5.5. DESIGN MODE: RESULTS SECTION

Now that all the input parameters have been inserted, the button "*RUN*" can be pressed. In this phase the program found all the possible solutions in the ranges that have been designed. During the calculation it is possible to click on "*RUN Status"* and a screen with all the configuration analysed in real time is shown. In the last column of this "in-motion" screen, the sentence of the relative configuration is written. Indeed, in this place can be present one of the following statements:

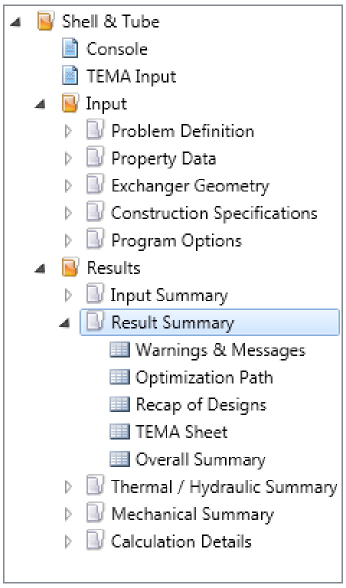
- "OK": it refers to a possible design configuration in which the required heat exchange flux and exchange area are satisfied. Furthermore, the device is present inside the restrictions field, e.g. pressure loss, and all the parameters enter in the specified range.
- "NEAR": represents a design arrangement in which the target parameters are not achieved, e.g. exchange area. The program provides the possibility to proceed with one of these configurations as long as some input parameters are changed. Most of the times it can happen that the best economical configuration is represented as a near one. However, the modification of the input value that leads to an acceptable device greatly increase its price, e.g. finned tubes, making it a no more convenient design.
- "(OK)": is a possible solution that present some vibration problems. Also in this case, one of these solutions can be uploaded for the next steps, provided that further analysis and modifications will be done to solve the issue. Generally, the modifications proposed here don't increase a lot the price.

At the end, the result part of the program is compiled. As for the input, it is compounded by several sections, each of which can be divided in forms and sub-forms. In the first section, *"Input Summary",* a resume of the input data is reported. The other sections proceed gradually to exhibit the results. Indeed, the second "Result Summary" shows the output of the iteration, while the others allow to enter in the selected output in a more detailed way. All the important views and explanations of the obtained result are now reported.

5.5.1. RESULT SUMMARY

WARNING & MESSAGES

The first form of the "*Result Summary*" section refers to the warnings and messages that the program generates during the computation. In the study case, only one "Result warning" has been generated. It can be seen at the end of the page and it points out that the Design 4, that will be shown in the next section, can be a good alternative to the one proposed by the program. Indeed, it declares that it satisfies the area ratio and pressure loss constraints, and it would be cheaper, being its cost 95% of the chosen output. However, it has been marked as (OK) because it presents some vibration problems. As already explained, it could be selected instead of the given output and some modifications can be done in order to solve that problem. As it can be seen, no input or operational warnings have been issued. This means that the program was able to find solutions in the indicated range, without serious conflicts in input data. The notes, instead, refers to minor changes in input data that have been done automatically by the program.



		Description
Results Warning	1711	Design 4 is 95 percent of the cost of the best Design, but fails to meet TEMA unsupported length or nozzle rho-V-squared limits. Designs which meet area ratio and pressure loss constraints, but fail the TEMA criteria appear as (OK) rather than OK in the Design Optimisation Results. You should review whether such Designs are acceptable, or can be simply modified to be so.

__OPTIMIZATION PATH__

In this form, a table shows the best configurations chosen by the program. The candidate number one is highlighted and placed at the bottom. These solutions are preferable to the others because they satisfy in a better way the parameters of selection previously described, e.g. heat exchanger cost or volume. Here is reported the output table of the study case, divided in two different image, in order to make it more visible.

Optimization Path

Current selected case 6 Select

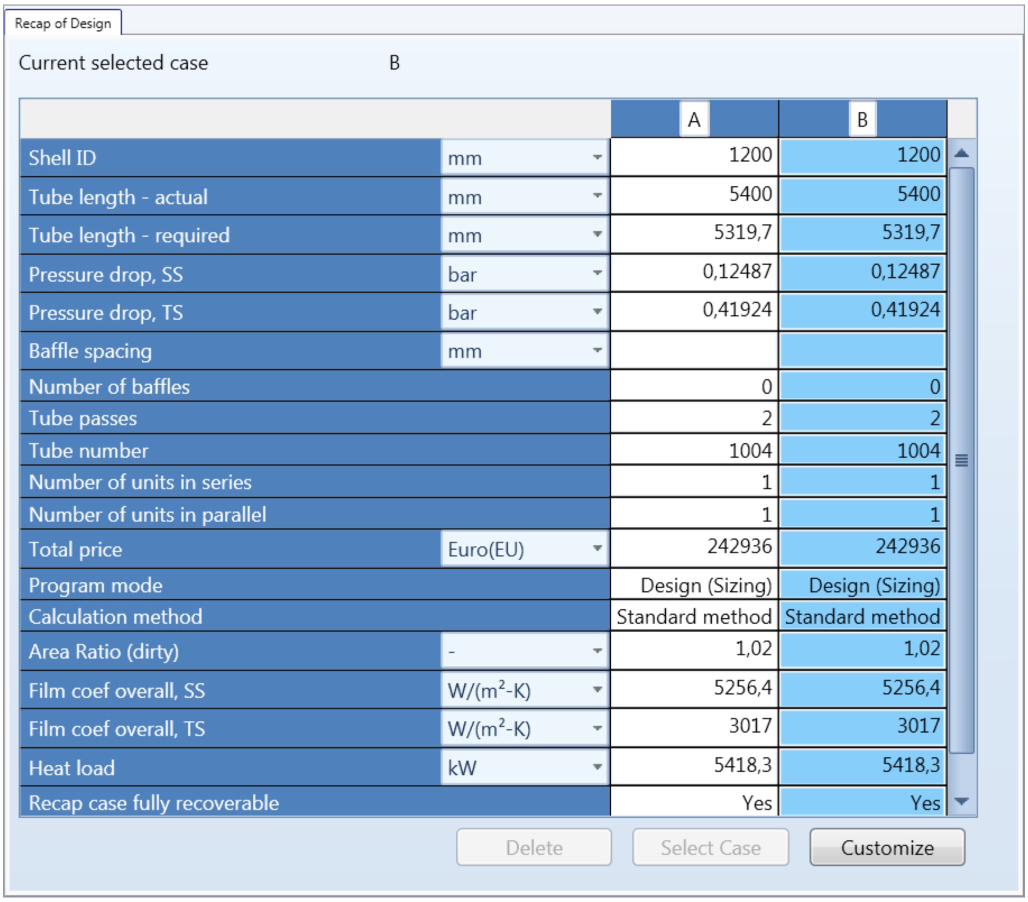
		Shell	Tube Length			Pressure Drop			
	Item	Size	Actual	Reqd.	Area ratio	Shell	Dp Ratio	Tube	Dp Ratio
		mm	mm	mm		bar		bar	
1	1	1075	6000	6474,2	0,93 *	0,11492	0,38	0,67128	0,96
2	2	1100	6000	6246,7	0,96 *	0,11414	0,38	0,62371	0,89
3	3	1125	6000	6040,7	0,99 *	0,11459	0,38	0,58135	0,83
4	4	1150	5850	5740,3	1,02	0,1187	0,4	0,51569	0,74
5	5	1175	5550	5542,9	1	0,11908	0,4	0,4633	0,66
6	6	1200	5400	5319,7	1,02	0,12487	0,42	0,41924	0,6
7	7	1225	5250	5163,2	1,02	0,12377	0,41	0,38856	0,56
8	8	800	6000	6295,8	0,95 *	0,07753	0,26	0,60386	0,86
9	9	825	6000	5950,2	1,01	0,08244	0,27	0,53946	0,77
10	10	850	5700	5661,3	1,01	0,08205	0,27	0,4696	0,67
11	11	875	5400	5389,3	1	0,08264	0,28	0,40859	0,58
12									
13	6	1200	5400	5319,7	1,02	0,12487	0,42	0,41924	0,6

		Baffle	Tube		Units		Total	Operational Issues		
	Item	No.	Tube Pass	No.	P	S	Price	Vibration	Rho-V-Sq	Design Status
							Euro(EU)			
1	1	0	2	792	1	1	205900	No	No	Near
2	2	0	2	828	1	1	215026	Possible	No	Near
3	3	0	2	864	1	1	222070	No	No	Near
4	4	0	2	918	1	1	230416	Possible	No	(OK)
5	5	0	2	956	1	1	235554	Possible	No	(OK)
6	6	0	2	1004	1	1	242936	No	No	OK
7	7	0	2	1040	1	1	250754	Possible	No	(OK)
8	8	0	2	420	2	1	251404	No	No	Near
9	9	0	2	450	2	1	263593	No	No	OK
10	10	0	2	478	2	1	270513	No	No	OK
11	11	0	2	508	2	1	279375	No	No	OK
12										
13	6	0	2	1004	1	1	242936	No	No	OK

The solution automatically chosen is the Design 6. It respects all the requirements and is the cheaper one with respect to the other "*OK*" solutions. Notice that the estimation of the price is made by the program using some very accurate tables of costs, customizable in the "*Customize*" section of the program, that take into account all the parameters, e.g. material, labor cost, etc. As already discussed, one of the other possibilities can be chosen. For example, the solutions number 9, 10 and 11 respect all the requirements, and could be selected if a shell with a lower diameter is preferred. It is nevertheless true that, they would provide a smaller margin on the exchange area, as can be seen from the lower value of the area ratio. Several "*Near*" and solutions with vibration problems can be seen. Some of them have a minor cost than the Design 6 but, during the design activity, the input data was already optimized for the cheapest solution. A modification of the input parameters, in order to make one of them compliant with the requirements, would lead to an increment of the price for that solution. This makes the Design 6 the best overall design. Since the maximum value inserted in input data for the area ratio was 1,02, no one of the solutions presents a higher value than the indicated one. In the case, a more expensive configuration could have been preferred if more margin would be needed. Sometime only solutions with vibration problems, or pressure drop higher than the limit, are found by the program. In such cases, the modifications of the input data are mandatory to find out a possible solution.

RECAP OF DESIGN

In this form a general view of the adopted solution is shown.



Recap of Design

Current selected case B

		A	B
Shell ID	mm	1200	1200
Tube length - actual	mm	5400	5400
Tube length - required	mm	5319,7	5319,7
Pressure drop, SS	bar	0,12487	0,12487
Pressure drop, TS	bar	0,41924	0,41924
Baffle spacing	mm		
Number of baffles		0	0
Tube passes		2	2
Tube number		1004	1004
Number of units in series		1	1
Number of units in parallel		1	1
Total price	Euro(EU)	242936	242936
Program mode		Design (Sizing)	Design (Sizing)
Calculation method		Standard method	Standard method
Area Ratio (dirty)	-	1,02	1,02
Film coef overall, SS	W/(m²-K)	5256,4	5256,4
Film coef overall, TS	W/(m²-K)	3017	3017
Heat load	kW	5418,3	5418,3
Recap case fully recoverable		Yes	Yes

Delete Select Case Customize

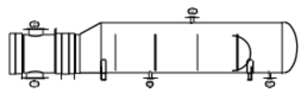
THEMA SHEET

In this part of the program, the output datasheet in TEMA frame shape is provided, and it will be necessary as input for the mechanical design.

TEMA Sheet

Heat Exchanger Specification Sheet

1	Company:						
2	Location:						
3	Service of Unit:	Our Reference:					
4	Item No.:	Your Reference:					
5	Date:	Rev No.:	Job No.:				
6	Size : 1200 / 1591 - 5400 mm	Type: AKU Horizontal		Connected in: 1 parallel 1 series			
7	Surf/unit(eff.) 449,4 m^2	Shells/unit 1		Surf/shell(eff.) 449,4 m^2			
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		Shell Side		Tube Side		
10	Fluid name		Water		HCBN		
11	Fluid quantity, Total	kg/h	8846		431779		
12	Vapor (In/Out)	kg/h	0	8438	224525	155440	
13	Liquid	kg/h	8846	408	207254	276339	
14	Noncondensable	kg/h	0	0	0	0	
15							
16	Temperature (In/Out)	°C	121	251,78	271	265	
17	Bubble / Dew point	°C	251,91 /	/	/	/	
18	Density Vapor/Liquid	kg/m^3	/ 942,03	19,29 / 803,97	51,76 / 495,6	46,32 / 503,7	
19	Viscosity	cp	/ 0,2401	0,0187 / 0,1147	0,01 / 0,127	0,01 / 0,13	
20	Molecular wt, Vap			18,01	167,27	152,6	
21	Molecular wt, NC						
22	Specific heat	kJ/(kg-K)	/ 4,21	3,887 / 4,571	2,767 / 3,4	2,726 / 3,421	
23	Thermal conductivity	W/(m-K)	/ 0,6869	0,0462 / 0,5995	0,0348 / 0,0929	0,0337 / 0,0929	
24	Latent heat	kJ/kg		1715,8	230,3	92,8	
25	Pressure (abs)	bar	41	40,87513	14	13,58076	
26	Velocity (Mean/Max)	m/s	0,31 / 0,39		7,12 / 7,9		
27	Pressure drop, allow./calc.	bar	0,3	0,12487	0,7	0,41924	
28	Fouling resistance (min)	m^2-K/W	0,00034		0,00034	0,00042 Ao based	
29	Heat exchanged 5418,3 kW			MTD (corrected) 16,49 °C			
30	Transfer rate, Service 731,3		Dirty 742,3	Clean 1724,8 W/(m^2-K)			
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**		
32			Shell Side	Tube Side			
33	Design/Vacuum/test pressure	bar	78 / /	60 / /			
34	Design temperature / MDMT	°C	280 /	300 /			
35	Number passes per shell		1	2			
36	Corrosion allowance	mm	3	3			
37	Connections	In mm	1 76,2 / -	1 457,2 / -			
38	Size/Rating	Out	1 38,1 / -	1 508 / -			
39	Nominal	Out - Vapor	1 88,9 / -	/ -			
40	Tube #: 502 U's	OD: 25,4	Tks. Minimur 2,4 mm	Length: 5400 mm	Pitch: 31,75 mm	Tube pattern: 90	
41	Tube type: Plain	Insert: None		Fin#: #/m	Material: Carbon Steel		
42	Shell Carbon Steel	ID 1200	OD 1278 mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel	-	Tubesheet-floating	-		
45	Floating head cover	-		Impingement protection	None		
46	Baffle-cross Carbon Steel	Type Unbaffled	Cut(%d)	Spacing: c/c	mm		
47	Baffle-long -	Seal Type		Inlet	mm		
48	Supports-tube	U-bend	0	Type			
49	Bypass seal		Tube-tubesheet joint	Expanded & strength welded(App.A 'e')			
50	Expansion joint	-	Type	None			
51	RhoV2-Inlet nozzle 353	Bundle entrance 29		Bundle exit 73	kg/(m-s^2)		
52	Gaskets - Shell side	Flat Metal Jacket Fibe	Tube side	Flat Metal Jacket Fibe			
53	Floating head	-					
54	Code requirements	ASME Code Sec VIII Div 1		TEMA class	R - refinery service		
55	Weight/Shell	39624,5	Filled with water 54835,7	Bundle	9615,5	kg	
56	Remarks						
57							
58							



In the first panel can be seen the shellside inlet diameter and the kettle inner diameter, 1200mm and 1591mm respectively, and the length of the tubes, 5400mm. The device is only one and it is an AKU horizontal type, as specified in the input. The first new output shown in the datasheet is the effective heat exchange area, which amounts to *449,4* m^2. Please note that the effective area is different from the required one; it will be analysed later, in the "*Thermal/Hydraulic Summary"* section.

Proceeding, it can be observed that the panel called "*performance of one unit*" represents, in this case, the operation condition of the whole heat exchanger, being it composed by one device only. Comparing the mass flow rate of the liquid with the design that have been inserted, it can be noticed that the inlet mass flow rate of the water is the same. The outlet, instead, is slightly different, *8438 kg/h* of steam with respect to *8404 kg/h* provided as input. However, this difference represents only the 0,004% of the input value, so it is negligible, and can be assumed as a calculation refinement due to the integration procedure. The same corrections have been done on the mass flow rate divisions between the liquid and vapour phases of the HCBN. Also in this case the percentual differences between the output and the input, being it approximately 0,005%, can be accepted as modification. As required, the vaporization is practically complete, while the HCBN condenses in part during the path. The same minor alteration has been made on the temperature outlet of the steam. Below these values, the fluids properties have been reported from the database. Furthermore, a preliminary check on the pressure drop can be done in this section. The computed pressure drop in the shellside is lower than the maximum allowed, and the same comparison can be made for the tubeside. Some important values to be observed are the outputs of the velocity. In the shellside, the velocity of the water is very low, but it can be acceptable due to the pool boiling configuration. In the tubeside, instead, it represents a very important value. In order to enhance the heat transfer, a lower speed would be better. However, low velocities lead to increase the fouling, while a too high speed can generate erosion or vibration problems. As already said, a value higher than *1 m/s* should be adopted if the fluid would be water. However, the HCBN is a dirty fluid, and at least a velocity of *2÷3 m/s* should be present. In this case, the configuration leads to have *7 m/s*, so it would be fine. Then, the heat exchanged, which amounts to *5418,3kW*, and the MTD of *16,49°C* are reported. The transfer rates are the performance values called thermal conductance, *h*, inside the theoretical part. Three different values are here reported. The first is the "*Service*", *731,3* $[W/m^2K]$, computed with a fouling factor that has been corrected, *0,00042* $[m^2K/W]$, based on the external diameter of the tube. This is the most conservative, and it is the one used for the calculations. The second is the "*Dirty*", *731,3* $[W/m^2K]$, and represents the thermal conductance which takes into account a fouling factor of *0,00034* $[m^2K/W]$. The last, instead, is the *"Clean"*, *1724,8* $[W/m^2K]$, computed without inclusion of the fouling effect. From here, it can be noticed how the fouling can decrease the capacity to transfer heat in the device. After the corrosion allowance value, that has been reported from the input, there are the dimensions of the connections. As it had been indicated, for the water there are one input and two outputs, one for the steam and one to drain the non-vaporized water. For the HCBN, instead, only two nozzles are present, but the dimensions are much greater due to the higher mass flow rate. The following data reported is the number of the forks and their length of *5400mm*. They are in total 502 and they lead to have 1004 holes in the tubesheet. Note that in order to produce one fork, a tube of more than *10800mm* is required. In the datasheet are also reported the approximate dimensions of the tubes and shell outer diameters. However, the correct values will be determined with the mechanical design program in the next sections of the thesis. As previously said, the program recognizes the heat exchanger as a unbaffled type, being the devices present in the kettle supports only. Indeed, in this kind of design, the shell diameter is farly greater than the bundle one. This leads to have some unbaffled space on its sides. Correctly, a support on the U-bend has been placed, while the suplementar angular supports are not needed in this case. As last parameters, the ρv^2 and an estimation of the weight are reported.

OVERALL SUMMARY

The “Overall Summary” is a more detailed summary of the thermal data, fluid properties, pressure drops and velocities.

Overall Summary

#			Shell Side In	Shell Side Out	Tube Side In	Tube Side Out				
1	Size	1200 X 5400	mm	Type	AKU	Hor	Connected in	1 parallel	1 series	
2	Surf/Unit (gross/eff/finned)		464,6 /	449,4 /		m^2	Shells/unit	1		
3	Surf/Shell (gross/eff/finned)		464,6 /	449,4 /		m^2				
4	Design (Sizing)			PERFORMANCE OF ONE UNIT						
5			Shell Side		Tube Side		**Heat Transfer Parameters**			
6	**Process Data**		In	Out	In	Out	Total heat load	kW	5418,3	
7	Total flow	kg/s	2,4572		119,9386		Eff. MTD/ 1 pass MTD	°C	16,49 /	16,52
8	Vapor	kg/s	0	2,344	62,3681	43,1779	Actual/Reqd area ratio - fouled/clean		1,02 /	2,36
9	Liquid	kg/s	2,4572	0,1133	57,5705	76,7607				
10	Noncondensable	kg/s	0		0		**Coef./Resist.**	$W/(m^2\text{-}K)$	$m^2\text{-}K/W$	%
11	Cond./Evap.	kg/s	2,344		19,1902		Overall fouled	742,3	0,00135	
12	Temperature	°C	121	251,78	271	265	Overall clean	1724,8	0,00058	
13	Bubble Point	°C	251,91				Tube side film	3017	0,00033	24,6
14	Dew Point	°C					Tube side fouling	2360	0,00042	31,45
15	Vapor mass fraction		0	0,95	0,52	0,36	Tube wall	17212,8	6E-05	4,31
16	Pressure (abs)	bar	41	40,87513	14	13,58076	Outside fouling	2909,9	0,00034	25,51
17	DeltaP allow/cal	bar	0,3	0,12487	0,7	0,41924	Outside film	5256,4	0,00019	14,12
18	Velocity	m/s	0,23	0,39	7,9	6,48				
19	**Liquid Properties**						**Shell Side Pressure Drop**		bar	%
20	Density	kg/m^3	942,03	803,97	495,6	503,7	Inlet nozzle		0,00555	14,59
21	Viscosity	mPa-s	0,2401	0,1147	0,127	0,13	InletspaceXflow		0	0
22	Specific heat	kJ/(kg-K)	4,21	4,571	3,4	3,421	Baffle Xflow		0,00186	4,89
23	Therm. cond.	W/(m-K)	0,6869	0,5995	0,0929	0,0929	Baffle window		0	0
24	Surface tension	N/m	0,0544	0,0258	0,0029	0,0032	OutletspaceXflow		0	0
25	Molecular weight		18,01	18,01	198,92	197,31	Outlet nozzle		0,03064	80,52
26	**Vapor Properties**						Intermediate nozzles			
27	Density	kg/m^3		19,29	51,76	46,32	**Tube Side Pressure Drop**		bar	%
28	Viscosity	mPa-s		0,0187	0,01	0,01	Inlet nozzle		0,03559	8,24
29	Specific heat	kJ/(kg-K)		3,887	2,767	2,726	Entering tubes		0,01913	4,43
30	Therm. cond.	W/(m-K)		0,0462	0,0348	0,0337	Inside tubes		0,3593	83,22
31	Molecular weight			18,01	167,27	152,6	Exiting tubes		0,0065	1,51
32	**Two-Phase Properties**						Outlet nozzle		0,01121	2,6
33	Latent heat	kJ/kg		1715,8	230,3	92,8	Intermediate nozzles			
34	**Heat Transfer Parameters**						**Velocity / Rho*V2**		m/s	$kg/(m\text{-}s^2)$
35	Reynolds No. vapor			164984,6	767892,7	531618	Shell nozzle inlet		0,61	353
36	Reynolds No. liquid		13469,36	1299,24	55812,91	72699,9	Shell bundle Xflow	0,23	0,39	
37	Prandtl No. vapor			1,57	0,79	0,81	Shell baffle window			
38	Prandtl No. liquid		1,47	0,87	4,65	4,79	Shell nozzle outlet		21,28	8697
39	**Heat Load**		kW		kW		Shell nozzle interm			
40	Vapor only		0		0				m/s	$kg/(m\text{-}s^2)$
41	2-Phase vapor		-0,1		-896,3		Tube nozzle inlet		9,15	7608
42	Latent heat		4021,8		-3048,5		Tubes	7,9	6,48	
43	2-Phase liquid		-1,4		-1473,5		Tube nozzle outlet		6,12	4067
44	Liquid only		1398		0		Tube nozzle interm			

#										
45	**Tubes**				**Baffles**		**Nozzles: (No./OD)**			
46	Type			Plain	Type	Unbaffled			Shell Side	Tube Side
47	ID/OD	mm	20,6 /	25,4	Number	0	Inlet	mm	1 / 88,9	1 / 457,2
48	Length act/eff	mm	5400 /	5210,5	Cut(%d)		Outlet		1 / 48,26	1 / 508
49	Tube passes		2		Cut orientation		Intermediate		1 / 101,6	/
50	Tube No.		1004		Spacing: c/c	mm	Impingement protection		None	
51	Tube pattern		90		Spacing at inlet	mm				
52	Tube pitch	mm	31,75		Spacing at outlet	mm				
53	Insert			None						
54	Vibration problem (HTFS / TEMA)		No /	No			RhoV2 violation			No

In addition to the data previously analysed, also shown here for convenience, the area ratio is reported. This parameter is the ratio between the effective and the required area in order to have the necessary heat transfer. Analysing the areas, the gross surface, *464.6 m^2*, is the total tubes area that has been calculated. The effective one, *449,4 m^2*, is the corrected one at the net of the portion that does not effectively transfer heat, e.g. the confined area in the tubesheet. Note that, in the case in which the U-bend would be indicated as not transferring, its whole area too would be subtracted. The finned area is not present because normal plain tubes are required in this design, while the required area will be analysed inside the section *"Thermal/Hydraulic Summary"*. However, the area ratios between the effective and the required surface for fouled and clean state are 1,02 and 2,36 respectively. This means that this design has an oversized area of 2% in fouled condition, and it would not be considered in the case in which a higher percentage had been requested in the input parameter. In the clean state, instead, the effective area is more than two times the required one. Again the high influence of the fouling can be seen, indeed, without fouling the heat exchanged would be fairly higher and the required area lower. Always in this panel there are all the thermal conductance and film coefficients for the fouled and cleaned cases. A preliminary analysis of the pressure drop division can be done. It will be reported in a more detailed version in a future form, but it can be seen that, in the shellside, the *80,52%* of the drop is done by the outlet nozzle. The critical part of the tubeside, instead, is the losses inside at the tubes, which account for the *83,22%*. As already analysed, the total pressure drop for both sides can be accepted because within the limits. If not so, these critical parts would be the ones on which going to act. Proceeding a detailed analysis of the velocity along the path is reported, and some geometrical data are repeated.

5.5.2. THERMAL / HYDRAULIC SUMMARY

- Shell & Tube
  - Console
  - TEMA Input
  - Input
    - Problem Definition
    - Property Data
    - Exchanger Geometry
    - Construction Specifications
    - Program Options
  - Results
    - Input Summary
    - Result Summary
    - Thermal / Hydraulic Summary
      - Performance
      - Heat Transfer
      - Pressure Drop
      - Flow Analysis
      - Vibration & Resonance Analysis
      - Methods & Convergence
    - Mechanical Summary
    - Calculation Details

Inside this set of forms, very detailed outputs are reported. However, a lot of results are repeated

PERFORMANCE

"*Overall performance*" sub-form is the performance part already presented in the "*Overall Summary*". In the *"Resistance Distribution"* module, instead, the required area can be seen. It amounts to *190.5 m^2* in the cleaning condition, and *442,7 m^2* in the fouled one. Inside the *"Shell by Shell Condition"*, the parameters that have been split between series or parallel devices can be analysed in deep. Since the solution adopted for the design consists in only one device, this card reports a merely repetition of the data.

Overall Performance | Resistance Distribution | Shell by Shell Conditions | Hot Stream Composition | Cold Stream Composition

Design (Sizing)		Shell Side		Tube Side	
Total mass flow rate	kg/h	8846		431779	
Vapor mass flow rate (In/Out)	kg/s	0	2,344	62,3681	43,1779
Liquid mass flow rate	kg/h	8846	408	207254	276339
Vapor mass fraction		0	0,95	0,52	0,36
Temperatures	°C	121	251,78	271	265
Bubble / Dew point	°C	251,91 /	/	/	/
Operating Pressures	bar	41	40,87513	14	13,58076
Film coefficient	W/(m^2-K)	5256,4		3017	
Fouling resistance	m^2-K/W	0,00034		0,00042	
Velocity (highest)	m/s	0,39		7,9	
Pressure drop (allow./calc.)	bar	0,3 /	0,12487	0,7 /	0,41924

Total heat exchanged	kW	5418,3
Overall clean coeff. (plain/finned)	W/(m^2-K)	1724,8 /
Overall dirty coeff. (plain/finned)	W/(m^2-K)	742,3 /
Effective area (plain/finned)	m^2	449,4 /
Effective MTD	°C	16,49
Actual/Required area ratio (dirty/clean)		1,02 / 2,36
Vibration problem (HTFS)		No
RhoV2 problem		No

Unit	AKU	2 pass	1 ser	1 par	
Shell size	1200 - 5400	mm		Hor	
Tubes	Plain				
Insert	None				
No.	1004	OD 25,4	Tks 2,4	mm	
Pattern	90	Pitch	31,75 mm		
Baffles	Unbaffled		Cut(%d)		
Total cost		242936	Euro(EU)		

Total Messages: 5
Errors: 0
Input: 0
Results: 1
Operations: 0
Notes & Advisory: 4
Warning and Messages details

Heat Transfer Resistance
Shell side / Fouling / Wall / Fouling / Tube side
Shell Side | Tube Side

Overall Performance | Resistance Distribution | Shell by Shell Conditions | Hot Stream Composition | Cold Stream Composition

Overall Coefficient / Resistance Summary			Clean	Dirty	Max Dirty
Area required (tube OD base)		m^2	190,5	442,7	449,4
Area ratio: actual/required			2,36	1,02	1
Overall coefficient		W/(m^2-K)	1724,8	742,3	731,3
Overall resistance		m^2-K/W	0,00058	0,00135	0,00137
Shell side fouling		m^2-K/W	0	0,00034	0,00035
Tube side fouling			0	0,00042	0,00043
Resistance Distribution	W/(m^2-K)	m^2-K/W	%	%	%
Shell side film	5256,4	0,00019	32,81	14,12	13,91
Shell side fouling	2909,9	0,00034		25,51	25,8
Tube wall	17212,8	6E-05	10,02	4,31	4,25
Tube side fouling *	2360	0,00042		31,45	31,81
Tube side film *	3017	0,00033	57,17	24,6	24,24

* Based on outside surface - Area ratio: Ao/Ai = 1,23

Heat Transfer Resistance
Shell side / Fouling / Wall / Fouling / Tube side
Shell Side | Tube Side

HEAT TRANSFER

In addition to the thermal coefficients already seen, in this form there are reported the Prandtl and Reynolds numbers, as well as the thermal flux. The shell and the tubes mean metal temperatures are very important parameters for the mechanical design. Given that the metal has a high coefficient of thermal conductivity, it will tend to stabilize more or less at the same temperature, regardless of fluid's one that varies point by point. In other words it is the mean of the temperature gradient present in the shell or tubes metal, that is lower than the fluid's one due to the thermal conductivity. Please note that the tubeside mean temperature is increased by the high temperature of the head zone, that is taken into account. A high difference in these two values leads to a high internal stress in the heat exchanger with a double welded tubesheet. Indeed, the prevented dilatation induced by that configuration will lead to a high thickness of tubesheet. These cases are even worsened when the tubes and the shell must be made in different materials, due to the different thermal conductivity coefficients. In the study case the thermal dilatation is not prevented, so this problem will not be present. *"Duty Distribution"* analyses how and where the thermal exchange happens. It can be seen that the *74,23%* of the total heat exchange is latent heat, while only the *25,8%* is used to bring the water at the boiling temperature. These are reasonable values since the aim of the device is to provide saturated vapour. Indeed, the steam will not be oversaturated, or even a percentage in "vapour only" would be shown. In the tubeside the fluid is biphasic either at the inlet and at the outlet, but during the process condenses given the heat released. So, in this case, the HCBN gas release the 16,54% of the total heat released, while the liquid the 27,2%. The remaining 56,26% is provided by the condensing process that transform the vapour into liquid.

Heat Transfer Coefficients | MTD & Flux | Duty Distribution

Film coefficients	W/(m²-K)	Shell Side		Tube Side	
		Bare area (OD) /	Finned area	Bare area (OD) /	ID area
Overall film coefficients		5256,4 /		3017 /	3720
Vapor sensible		/		/	
Two phase		5740,1 /		3017 /	3720
Liquid sensible		4107,6 /		/	
Heat Transfer Parameters		**In**	**Out**	**In**	**Out**
Prandtl numbers	Vapor		1,57	0,79	0,81
	Liquid	1,47	0,87	4,65	4,79
Reynolds numbers	Vapor Nominal		164984,6	767892,7	531618
	Liquid Nominal	13469,36	1299,24	55812,91	72699,9

Heat Transfer Coefficients | MTD & Flux | Duty Distribution

Temperature Difference	°C	Heat Flux (based on tube O.D)	kW/m²
Overall effective MTD	16,49	Overall flux	12,2
One pass counterflow MTD	16,52	Critical heat flux (at highest ratio)	3308,3
LMTD based on end points	61,96	Highest local flux	15,4
Effective MTD correction factor	0,27	Highest local/critical flux	0
Wall Temperatures	°C		
Shell mean metal temperature			251,53
Tube mean metal temperature			258,89
Tube wall temperatures (highest/lowest)			254,27 / 252,11

Heat Transfer Coefficients | MTD & Flux | Duty Distribution

Heat Load Summary	Shell Side		Tube Side	
	kW	% total	kW	% total
Vapor only	0	0	0	0
2-Phase vapor	-0,1	0	-896,3	16,54
Latent heat	4021,8	74,23	-3048,5	56,26
2-Phase liquid	-1,4	-0,03	-1473,5	27,2
Liquid only	1398	25,8	0	0
Total	5418,3	100	-5418,3	100
Effectiveness	0,9348			

PRESSURE DROP

This is the form in which the whole pressure drop analysis is presented in detail. The difference between the gravitational, frictional and momentum change is also reported. The gravitational one is a contribution of the pressure losses at the inlet of the water flow. As a matter of fact, the water already present in the shell generate a hydrostatic head that acts as an obstruction. Another great contribution is the frictional one, generated by the roughness of the materials in contact. Finally, the momentum change contribution is completely negligible since it has two orders of magnitude less. The value and the percentage distribution of the pressure drop already analysed is here reported in a more exhaustive way.

Pressure Drop | Thermosiphon Piping | Thermosiphon Piping Elements

Pressure Drop bar	Shell Side				Tube Side			
Maximum allowed	0,3				0,7			
Total calculated	0,12487				0,41924			
Gravitational	0,08668				0			
Frictional	0,03806				0,43173			
Momentum change	0,00013				-0,01249			
Pressure drop distribution	m/s Near Inlet	m/s Near Outlet	bar	%dp	m/s Near Inlet	m/s Near Outlet	bar	%dp
Inlet nozzle	0,61		0,00555	14,59	9,15		0,03559	8,24
Entering bundle	0,23				7,9		0,01913	4,43
Inside tubes					7,9	6,48	0,3593	83,22
Inlet space bundle Xflow			0	0				
Mid-Space bundle Xflow	0,23	0,39	0,00186	4,89				
Mid-Space windows			0	0				
Outlet space bundle Xflow			0	0				
Exiting bundle	0,39				6,48		0,0065	1,51
Outlet nozzle	21,28		0,03064	80,52	6,12		0,01121	2,6
Liquid outlet nozzle	0,12		4E-05	0,11				
Vapor outlet nozzle	21,28		0,03064	80,52				
Intermediate nozzles								

FLOW ANALYSIS

In the pool boiling heat exchanger configuration, the flow analysis of the shellside is not consistent, but it is still shown because the output table of the program is standard. In other cases where the baffles are present, it would constitute an important analysis and every type of leakage would be showed here.

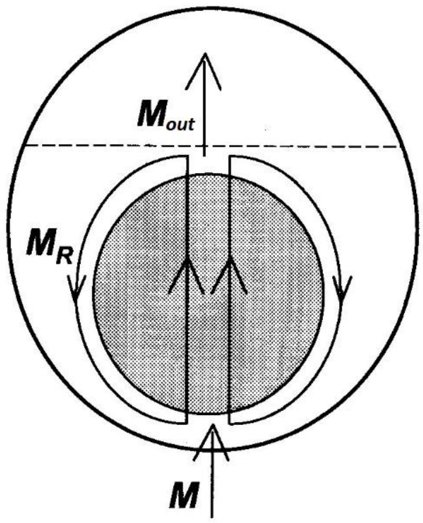
Flow Analysis | Thermosiphons / Kettles / Knockback Condenser

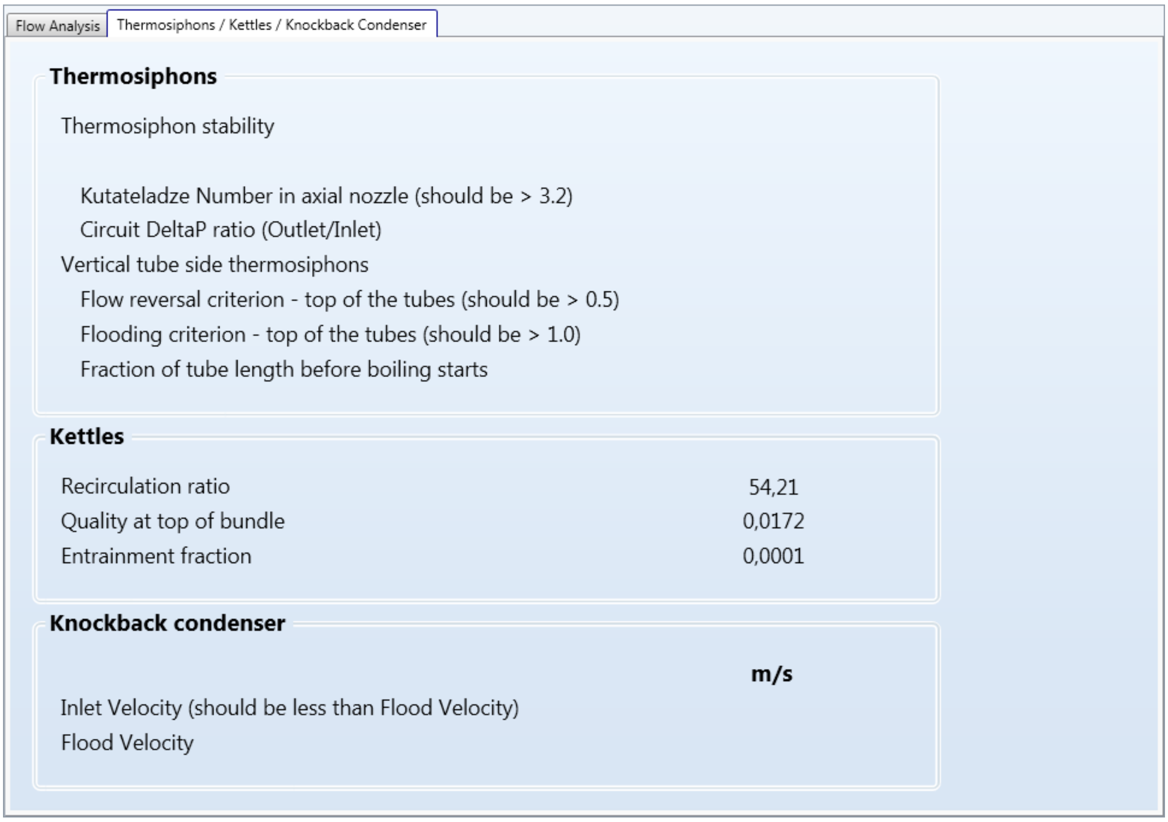
Shell Side Flow Fractions	Inlet	Middle	Outlet	Diameter Clearance mm
Crossflow (B stream)	1	1	1	
Window (B+C+F stream)	0	0	0	
Baffle hole - tube OD (A stream)	0	0	0	0,4
Baffle OD - shell ID (E stream)	0	0	0	6,35
Shell ID - bundle OTL (C stream)	0	0	0	12,7
Pass lanes (F stream)	0	0	0	

Rho*V2 Analysis	Flow Area mm²	Velocity m/s	Density kg/m³	Rho*V2 kg/(m-s²)	TEMA limit kg/(m-s²)
Shell inlet nozzle	4261	0,61	942,03	353	2232
Shell entrance	2908	0,9	942,03	758	5953
Bundle entrance	729767	0,23	806,98	29	5953
Bundle exit	729767	0,39	473,65	73	5953
Shell exit	70159	1,74	19,21	58	5953
Shell outlet nozzle	5734	21,28	19,21	8697	
	mm²	m/s	kg/m³	kg/(m-s²)	kg/(m-s²)
Tube inlet nozzle	144310	9,15	90,79	7608	8928
Tube inlet	167312	7,9	90,79	5660	
Tube outlet	167312	6,48	110,59	4647	
Tube outlet nozzle	180465	6,12	108,62	4067	

The second table instead reports the ρv^2 analysis that allows to understand if the nozzles are correctly designed and if some problems of erosion will occur. The values of ρv^2 obtained in the study case are far minor than the limit reported in the theoretical part. This means that no modifications on the nozzle diameters or impingement plates are required.

In the second sub-form the thermosiphons and knockback condenser sections are empty since the device is a Kettle. Indeed, in the correspondent part some important parameters are reported. The recirculation ratio is the ratio between the mass flow rate recirculated around to the bundle, M_R in the image, and the inlet mass flow rate, M. The quality of the vapour at the top of the bundle, instead, have to be as lower as possible, because the tubes must always be covered by water. A value of 0,0172 can be accepted. The same is needed for the entrainment fraction, already discussed in the program input part. In the case in which the quality of vapour would be to high, an intervention on a process parameter is required, e.g. enhancing the water mass flow rate, or decreasing the hot fluid one. Differently, if the problem is represented by the entrainment fraction, an increment of the inner shell diameter must be done.





VIBRATION & RESONANCE ANALYSIS

As already introduced, in the input section two vibration analysis methods are possible, HTFS and TEMA, and both of them have to be performed in this case. During every analysis, two type of vibrations are evaluated. The first is the fluid elastic instability, while the second one is an evaluation of the resonance. Focusing on the HTFS method, which, as already said, is a complex mathematical based analysis, it performs the calculations for all the points along the tubes. In the output of the study the more critical tubes are reported, e.g. tube 3 and 4, which positions can be seen in the "*Mechanical summary*" sub-form. For every tube the ratio between the actual shellside flowrate and the critical flow rate is reported for different values of damping. All these values must be far from 1, due to the fact that having a critical flow rate causes fluid elastic instability. In this case also in the tube 3 and 4, highlighted as the most dangerous, the values are acceptable. Notice that all the values are the same, being the observed critical tubes mirrored. In case in which these values are coming closer to 1, the configuration would be labelled with (OK), and possible vibration can occur. Proceeding, the natural frequency of the tubes and the method used to compute them are reported.

Fluid Elastic Instability (HTFS) | Resonance Analysis (HTFS) | Simple Fluid Elastic Instability (TEMA) | Simple Amplitude and Acoustic Analysis (TEMA)

Shell number: Shell 1

Fluid Elastic Instability Analysis

Vibration tube number		3	4
Vibration tube location		Inlet row, centre	Baffle overlap
Vibration		No	No
W/Wc for heavy damping (LDec=0.1)		0,26	0,26
W/Wc for medium damping (LDec=0.03)		0,48	0,48
W/Wc for light damping (LDec=0.01)		0,83	0,83
W/Wc for estimated damping		0,22	0,22
Estimated log Decrement		0,14	0,14
Tube natural frequency	cycle/s	13,96	13,96
Natural frequency method		Dominant Span	Dominant Span
Dominant span		U-bend	U-bend
Tube effective mass	kg/m	1,88	1,88

Note: W/Wc = ratio of actual shellside flowrate to critical flowrate for onset of fluid-elastic instability

Tube material density	kg/m^3	7841,74
Tube axial stress	N/mm^2	0
Tube material Young's Modulus	N/mm^2	186153
U-bend longest unsupported length	mm	1835,37

The second sub-form, always concerning the HTFS method, is dedicated to the resonance analysis. At first the dangerous location is delineated, e.g. the midspace, and the distance between the supports, called span length, is reported. However, in the adopted design, resonance does not occur in neither tubes. Also in this case a ratio allows to estimate the distance between the real case and the one in which resonance would occur. Indeed, the proportion between the tube frequency and the tube natural frequency is reported, and it must be as far as possible from one. In a more detailed way, the program reports both values: the natural frequency and the acoustic frequency that will cause resonance in motion and sonorous respectively. In case in which resonance occurs, the problem can be solved in two ways. The first is to modify the mass flow rate, solution particularly suitable when the device is already working, while the second one is to act on some geometrical characteristics, e.g. the span length.

Fluid Elastic Instability (HTFS) | Resonance Analysis (HTFS) | Simple Fluid Elastic Instability (TEMA) | Simple Amplitude and Acoustic Analysis (TEMA)

Shell number: Shell 1

Resonance Analysis

Vibration tube number		3	4
Vibration tube location		Inlet row, centre	Baffle overlap
Location along tube		Midspace	Midspace
Vibration problem		No	No
Span length	mm	1302,62	1302,62
Frequency ratio: Fv/Fn		0,18	0,26
Frequency ratio: Fv/Fa		0	0,05
Frequency ratio: Ft/Fn		0,12	0,17
Frequency ratio: Ft/Fa		0	0,03
Vortex shedding amplitude	mm		
Turbulent buffeting amplitude	mm		
TEMA amplitude limit	mm		
Natural freq., Fn	cycle/s	13,96	13,96
Acoustic freq., Fa	cycle/s	804,33	73,76
Flow velocity	m/s	0,16	0,23
X-flow fraction		1	1
RhoV2	kg/(m-s^2)	21	25
Strouhal No.		0,4	0,4

Moving on the TEMA analysis, conceptually the same evaluations have to be done. The strongest difference between this method and the HTFS one is that, being the TEMA an old procedure developed for the manual computation, a lot of coefficients must be roughly taken from experimental tables and the values are evaluated only in some critical points. This generates a situation in which a configuration can present fluid elastic instability, or resonance problems, that cannot be identified if the point in which it happens is not considered as critical by TEMA. The explored critical points are the inlet, the baffle window and overlap, the outlet and the U-bend. Notice that in the study case the baffles only have the support task, so the windows are not present, and the overlap represents the whole area.

Fluid Elastic Instability (HTFS) | Resonance Analysis (HTFS) | Simple Fluid Elastic Instability (TEMA) | Simple Amplitude and Acoustic Analysis (TEMA)

Tube material density	kg/m^3	7841,74
Tube axial stress	N/mm^2	0
Tube material Young's Modulus	N/mm^2	186153

Fluid Elastic Instability Analysis

		Inlet	C-C Window	C-C Overlap	Outlet	U-bend
Vibration indication		No		No	No	No
Unsupported span	mm	1302,62		1302,62	1302,62	1815,37
Tube natural frequency, fn	cycle/s	29,01		35,94	36,37	16,28
Crossflow velocity	m/s	0,01		0,04	0,1	0,04
Critical velocity	m/s	1,28		2,25	2,99	0,78
Crossflow to critical velocity ratio		0,01		0,02	0,03	0,05
Estimated log decrement		0,08		0,03	0,02	0,01

Fluid Elastic Instability (HTFS) | Resonance Analysis (HTFS) | Simple Fluid Elastic Instability (TEMA) | Simple Amplitude and Acoustic Analysis (TEMA)

Amplitude Vibration Analysis

		Inlet	C-C Window	C-C Overlap	Outlet	U-bend
Vortex shedding indication		No		No	No	No
Turbulent buffeting indication		No		No	No	No
Tube natural frequency, fn	cycle/s	29,01		35,94	36,37	16,28
Vortex shedding frequency, fvs	cycle/s	0,2		0,67	1,67	0,67
Vortex shedding amplitude	mm	0		0	0	0
Vortex shedding amplitude limit	mm	0		0	0	0
Turbulent buffetting amplitude	mm	0		0	0	0
Turbulent buffetting amplitude limit	mm	0,51		0,51	0,51	0,51

Acoustic Vibration Analysis

		Inlet	C-C Window	C-C Overlap	Outlet	U-bend
Acoustic resonance indication		No		No	No	No
Crossflow velocity	m/s	0,01		0,04	0,1	0,04
Strouhal number		0,43		0,43	0,43	0,43
Acoustic frequency, fa	cycle/s					
Vortex shedding frequency, fvs	cycle/s	0,2		0,67	1,67	0,67
Turbulent buffeting frequency, ftb	cycle/s	0,12		0,4	1	0,4
Condition A fa/fvs		0		0	0	0
Condition A fa/ftb		0		0	0	0
Condition B velocity	m/s					
Condition C velocity	m/s					
Condition C						

METHODS & CONVERGENCE

In this form and sub-forms all the used mathematical method indicated during the input, e.g. inside *"Program options"*, or automatically adopted during the computation phase, are reported. For the shake of completeness the image of this part is reported, but, as already said, it will not be thoroughly analysed because it is not one of the central arguments of this thesis.

Methods Summary | Convergence Plot

	Hot Side	Cold Side
Heat transfer coefficient multiplier	No	No
Heat transfer coefficient specified	No	No
Pressure drop multiplier	No	No
Pressure drop calculation option	friction only	friction + gravity
Calculation method	Standard method	
Desuperheating heat transfer method	Wet wall	
Multicomponent condensing heat transfer method	HTFS - Silver-Bell	
Vapor shear enhanced condensation	Yes	
Liquid subcooling heat transfer (vertical shell)	Not Used	
Subcooled boiling accounted for in	Heat transfer & pressure drop	
Post dryout heat transfer accounted for in	Yes	
Correction to user-supplied boiling curve	Boiling curve not used	
Falling film evaporation method	HTFS recommended method	
Single phase tube side heat transfer method	HTFS recommended method	
Lowfin Calculation method	HTFS / ESDU	
Tube Pass Multiplier	1	

5.5.3. MECHANICAL SUMMARY AND CALCULATION DETAILS

In the mechanical summary all the dimensions of the heat exchanger parts are reported in a detailed way by means of tables and sketches. The whole basic geometry, tube, support and bundle characteristics are presented again in a compact form. Furthermore, a preliminary estimation of the cost, weight and tubesheet layout is reported. An important explanation which needs to be done is that this mechanical summary reports the dimensions of the equipment resulting from the thermal design, but not their thicknesses. Indeed, the mechanical design of the heat exchanger has not been performed yet, and will be the topic of the next chapters.

In *"Calculations details"*, similarly to *"Mechanical summary"*, the tables of the thermal analysis and the 2D and 3D plots of the manually inserted and automatically computed fluid properties are reported in a great detail. Finally, it is worth observing that most of these data are inserted in the tables after the rating mode has been launched.

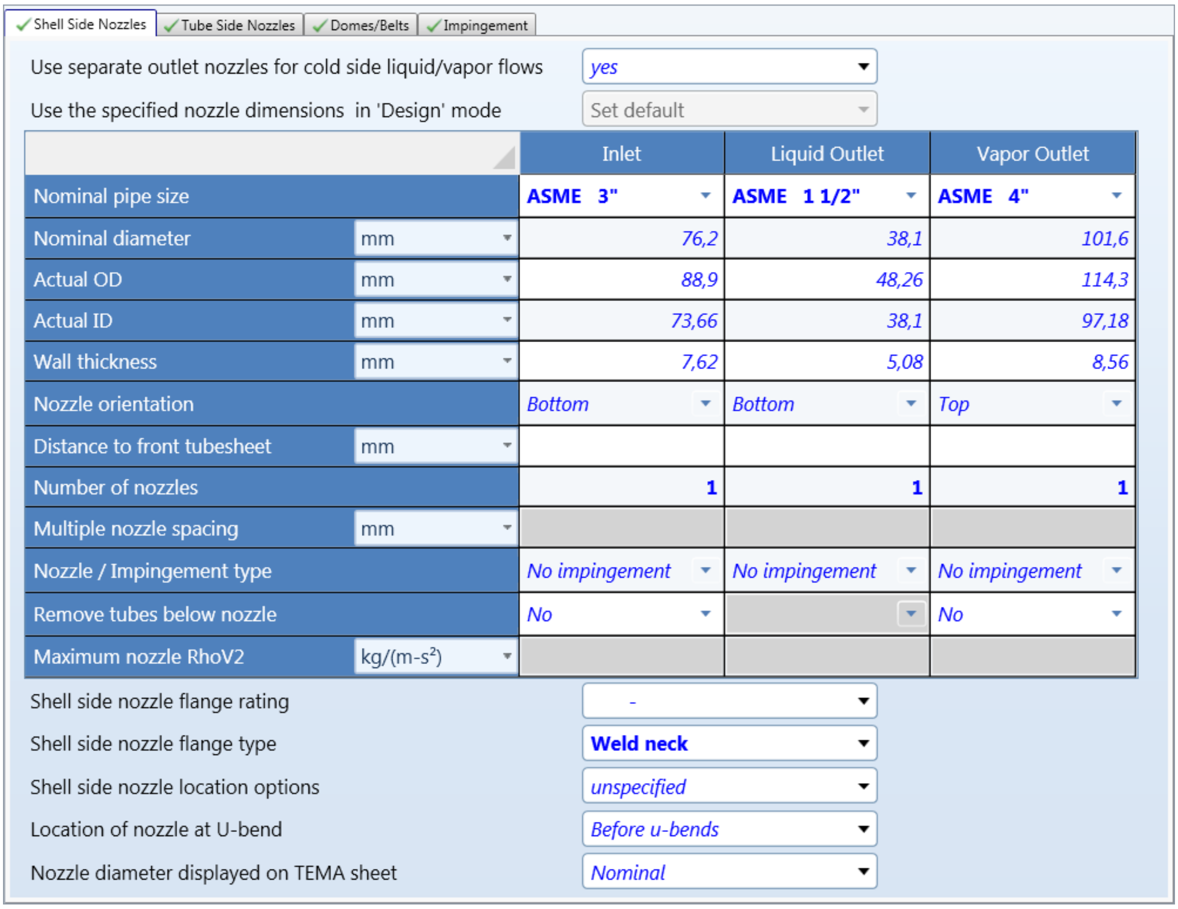
Costs/Weights

Weights	kg	Cost data	Euro(EU)
Shell	21785,4	Labor cost	164188
Front head	8223,6	Tube material cost	17843
Rear head	0	Material cost (except tubes)	62157
Shell cover			
Bundle	9767,2		
Total weight - empty	39776,2	Total cost (1 shell)	244188
Total weight - filled with water	54968,1	Total cost (all shells)	244188

5.6. RATING MODE

Now that the first mode of calculation has been performed, generating a range of proposals from the input data, the selected geometry, Design 6, can be uploaded in the rating mode of the program. After the selection of “*Rating mode*” in the upper bar and after having followed the instructions to upload the chosen design, a preliminary simulation can be launched in order to verify if some basic problems occur. Indeed, it can happen that, being the simulation carried out in this part of the program more accurate, some difficulties that had not been identified during the design mode can be figured out. As result, the Design 6 without any further modification does not have produced warnings in this further simulation. Only a note of calculation refinement, saying that the outlet temperature of the steam has been modified from 251,5°C to 251,7°C, have been provided by the program, while the other results have been accepted from the design mode.

Once that the design has been verified in the Rating mode, some further modifications can be done in order to adapt the dimension to some standard values. Indeed, after having inserted the input parameters and launched the Design mode, the output parameters associated to the selected configuration, e.g. Design 6, could not be modified before. Now that this operation can be done, the size of the nozzles must be verified. Going inside the form *“Exchanger geometry - Nozzles*” their diameters must be equal to the standard diameters of the pipes on the market, otherwise they would be difficult to find. All the sizes selected by the program already correspond to a standard in the ASME code, but its name has not already been selected. This can appear as a useless procedure since the dimensions was already correct, however, it indicates to the program to base the mechanical design on the standard tubes, and not only on a generic tube with those dimensions. Furthermore, the nozzle of the shellside vapour outlet measures 3,5 inches, that is not a very common dimension. For this reason it has been increased to 4 inches, erasing the bold text with the unwanted measure and selecting the desired one.

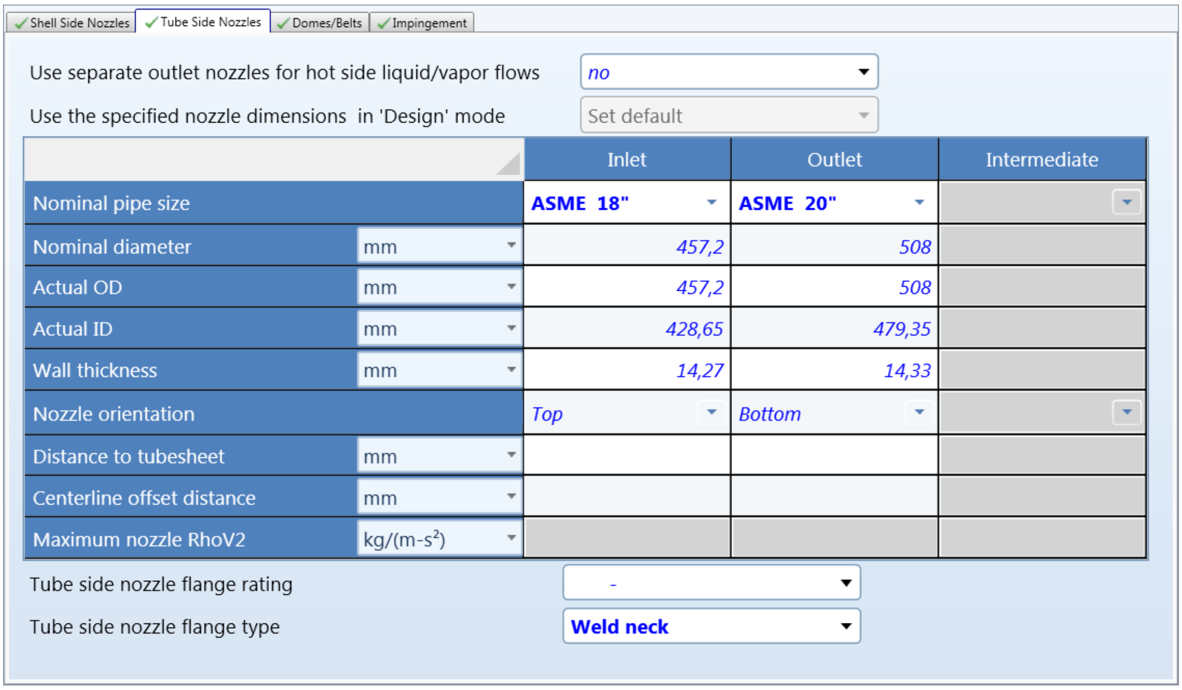

Shell Side Nozzles | Tube Side Nozzles | Domes/Belts | Impingement

Use separate outlet nozzles for cold side liquid/vapor flows	yes
Use the specified nozzle dimensions in 'Design' mode	Set default

		Inlet	Liquid Outlet	Vapor Outlet
Nominal pipe size		**ASME 3"**	**ASME 1 1/2"**	**ASME 4"**
Nominal diameter	mm	76,2	38,1	101,6
Actual OD	mm	88,9	48,26	114,3
Actual ID	mm	73,66	38,1	97,18
Wall thickness	mm	7,62	5,08	8,56
Nozzle orientation		Bottom	Bottom	Top
Distance to front tubesheet	mm			
Number of nozzles		**1**	**1**	**1**
Multiple nozzle spacing	mm			
Nozzle / Impingement type		No impingement	No impingement	No impingement
Remove tubes below nozzle		No		No
Maximum nozzle RhoV2	kg/(m-s²)			

Shell side nozzle flange rating	-
Shell side nozzle flange type	**Weld neck**
Shell side nozzle location options	unspecified
Location of nozzle at U-bend	Before u-bends
Nozzle diameter displayed on TEMA sheet	Nominal

The same passages have to be done for the tubeside nozzles, which already have standard and common dimensions, but have to be designed with the standard models. After the nozzle modification, the geometry configuration must be processed again. Notice that it can only lead to improvements as a decrement of the pressure drop and of the ρv^2. Indeed, no further warnings or notes have been issued, and the thermal design can be considered completed and ready to export the data in the mechanical part.



All the other parameters have not been modified, accepting the final thermal design as proposed from the last rating mode.

6. CONCEPTS OF PRESSURE VESSELS MECHANICAL DESIGN

6.1. PRELIMINARY CONSIDERATIONS

6.1.1. ALLOWABLE STRESS AND SAFETY FACTORS

The following stresses are typically considered to determine the basic allowable stress of steel:

- σ_R= minimum value of the unitary maximum load during the tensile test at room temperature. It is also called rupture stress.
- $\sigma_{(0.2)/t}$= minimum value of the unitary load during the tensile test at temperature t that causes a permanent deformation equal to 0,2% of the initial length between references after removal of load. It practically replaces the yield strength σ_s when the steel does not exhibit the classic yielding phenomenon. If, in the other hand, it would be easy to determine, the value of $\sigma_{(0,2)}$ coincides with the lower value of the yield strength. The symbol σ_s and the term "yield strength" will be used to refer to them, even though they are formally incorrect.
- $\sigma_{R/100000/t}$=average value of the unitary rupture stress for creep after 100,000 h at temperature t.

In the case of austenitic steel there is general agreement that, instead of a permanent deformation of 0,2%, allows to take as a reference value the deformation of 1%. Note that the temperature t is the average wall temperature.

Limiting to focus on the values of $\sigma_{(0,2)}$ and σ_R, in Figure 32 can be noticed that $\sigma_{(0,2)}$ decreases with the temperature rise, while σ_R increases initially within a moderate range of temperatures and then decreases. Moreover, the reduction of $\sigma_{(0,2)}$ has an important impact on the sizing of the vessel and, under certain conditions, on the selection of steel to be used. In fact, the decrease in $\sigma_{(0,2)}$ can be more or less substantial for steel with different compositions.

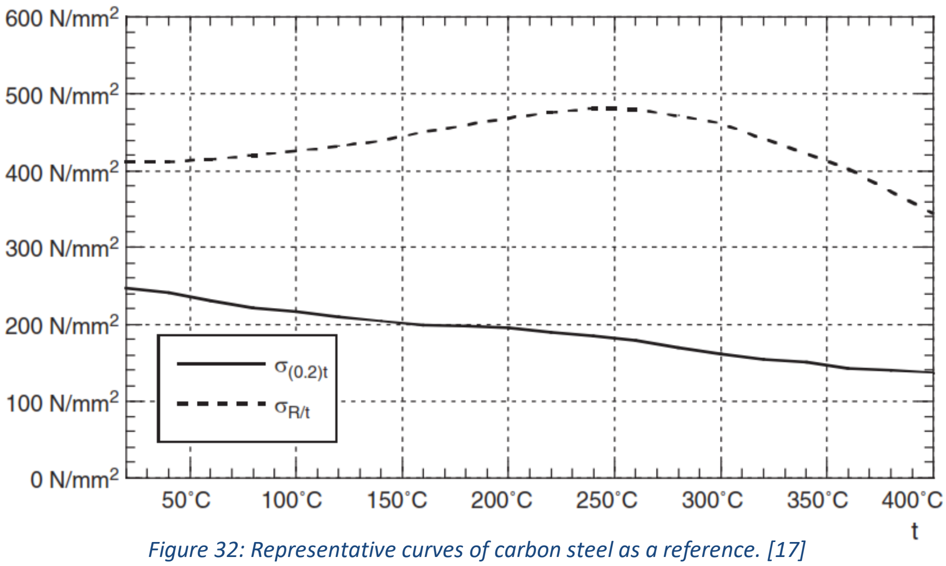


Figure 32: Representative curves of carbon steel as a reference. [17]

The rupture stress during the tensile test refers to room temperature. This may sound surprising since the resistance verification must be executed at design temperature t, to which the other two values above in fact refer. But, has pointed out before, the value of the rupture stress at moderate temperatures is greater than the one at room temperature. Within this temperature range, the adoption of this last value addresses basic safety criteria. The use of this value is in fact justified within the range of moderate temperatures: the goal here is to guarantee, through an adequate safety factor, that stresses acting upon the vessel do not cause a dangerous situation leading to rupture. Moreover, when steel with high levels of yield strength are adopted and the design temperature is either room temperature or anyway moderate, the value of $\sigma_{(0.2)/t}$ is very close to σ_R. The determination of the allowable stress solely based on $\sigma_{(0.2)/t}$ may lead to a value that does not sufficiently protect against rupture. Finally, as far as the rupture stress per creep at 100,000h is concerned, its importance is evident if the design temperature is high. For such stress, given the dispersion of values present even for similar types of steel, one refers to the mean value of the range, generally assuming that the size of the range itself does not go beyond ±20%.

In order to obtain the allowable stress, *f,* the three characteristic stresses are associated to safety factors. The values of these ones lack of a general consensus, and they have undergone numerous modifications over time. The basic allowable stress of the material, *f*, is given by the smallest of the following values:

$$f = min\left(\frac{\sigma_R}{2{,}4};\ \frac{\sigma_{(0.2)/t}}{1{,}5};\ \frac{\sigma_{R/100000/t}}{1{,}5}\right)$$

From a conceptual point of view, the basic value is the one derived from $\sigma_{(0.2)/t}$. The other two values mentioned above occur in special instances, even though $\sigma_{R/100000/t}$ is found quite frequently. It is critical for *f* when its value is lower than the yield strength, since the safety factor has the same value of the one relative to $\sigma_{(0.2)/t}$. Finally, note that the stress *f* has been defined as basic allowable stress of the material. This does not necessarily mean that it corresponds to the allowable stress during resistance verification of a specific piece in a specific position. In some cases, it is acceptable that the ideal stress may reach the yield strength or even twice it, from the point of view of calculation in the elastic field, as will be analyzed during the plasticity collaboration concept. The stress *f*, which in other cases actually corresponds to the maximum stress allowable for the piece, and at any rate to the maximum ideal stress of the membrane, represents a reference point, since it is present in all equations to compute the thickness of the various components.

6.1.2. THEORIES OF FAILURE

Given that it is not necessary nor relevant to examine all failure theories, only those directly related to resistance verification of pressure vessels will be briefly introduced in order to understand what follow next. Pressure vessels are characterized by the existence of stresses along three axes. First of all, due to pressure, there is a principal stress orthogonal to the wall of the vessel, while two additional principal stresses act on the plane orthogonal to the previous one. In the case of cylindrical elements the first of such stresses is radial, the other two are directed, respectively, along the circumference and along the axis of the cylinder. This calls for a failure theory that allows one to correlate such state of stress with the resistance values of the material, derived from the tensile test that in turn is based on a single stress directed along the axis of the specimen. The most generally accepted failure theories for ductile materials, such as steel used to build pressure vessels, are the well-known theory of maximum shear stress or Guest–Tresca, and the one known as distortion energy theory or Huber–Hencky.

According to Guest–Tresca the level of danger is captured by the maximum shear stress, in other words all states having the same maximum shear stress are equivalent with respect to danger. The state of stress relative to the specimen being subjected to single tensile stress is represented in Mohr's plane by the only circle shown in Figure 33. The maximum shear stress acting at 45° with respect to the only principal stress σ_{III} is equal to $\sigma_{III}/2$. Therefore, if a dangerous condition is associated to the yielding of the material and the corresponding stress is σ_s, the shear stress can be computed as

$$\tau_s = \frac{\sigma_s}{2}$$

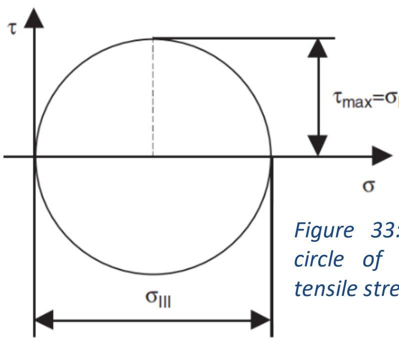


Figure 33: Mohr's circle of a single tensile stress. [17]

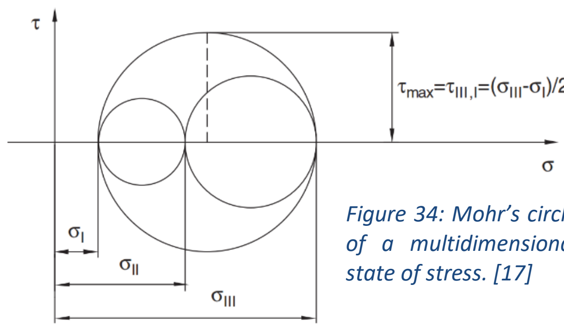


Figure 34: Mohr's circle of a multidimensional state of stress. [17]

If the state of stress is along three axis, σ_I, σ_{II} and σ_{III} are the three principal stresses, and conventionally they increase in value from $\sigma_I$to σ_{III}. This condition is represented in Figure 34. The three maximum shear stresses on the three planes where they operate are hence given, respectively, by

$$\tau_{III,I} = (\sigma_{III} - \sigma_I)/2$$

$$\tau_{III,II} = (\sigma_{III} - \sigma_{II})/2$$

$$\tau_{II,I} = (\sigma_{II} - \sigma_I)/2$$

With the above agreement the maximum value of the shear stress is given by $\tau_{III,I}$. Therefore, the condition of danger is represented by the following

$$\tau_{III,I} = \frac{(\sigma_{III} - \sigma_I)}{2} = \tau_s = \frac{\sigma_s}{2}$$

From a formal point of view a conventional stress, also called ideal stress or stress intensity, is defined as

$$\sigma_{id} = \sigma_{III} - \sigma_I$$

Such stress, in the case of stress condition along more than one axis, takes over the same function that the only applied axial stress has inside the specimen, i.e., it can pinpoint the examined fiber's working condition with respect to danger. In fact, when the ideal stress reaches the value of yield strength, according to the failure theory adopted here, a situation of danger is faced.

According to Huber and Hencky, the level of danger is captured by the distortion energy, i.e., all conditions of stress that produce the same distortion energy are equivalent to the condition of danger. Defining again the three principal stresses as σ_I, σ_{II} and σ_{III}, such energy is given by the following equation:

$$L = \frac{1}{6G}(\sigma_I^2 + \sigma_{II}^2 + \sigma_{III}^2 - \sigma_I\sigma_{II} - \sigma_{II}\sigma_{III} - \sigma_{III}\sigma_I)$$

In the specific case of the specimen since $\sigma_I = \sigma_{II} = 0$, the condition of danger is characterized by $\sigma_{III} = \sigma_s$, and the distortion energy is

$$L_s = \frac{1}{6G}\sigma_{III}^2 = \frac{1}{6G}\sigma_s^2$$

For a state stress along more than one axis, the condition of danger is obtained equating the last two equation. This operation leads to find the following ideal stress:

$$\sigma_s = \sigma_{id} = \sqrt{\sigma_I^2 + \sigma_{II}^2 + \sigma_{III}^2 - \sigma_I\sigma_{II} - \sigma_{II}\sigma_{III} - \sigma_{III}\sigma_I}$$

This ideal stress allows to identify the working condition of the fiber under scrutiny. As for the previously discussed theory, when σ_{id} reaches the value of the yield strength, a dangerous condition takes place. Similarly, the sizing of the piece is obtained by making sure that σ_{id} does not go beyond the allowable stress. Von Mises' theory formally corresponds to the Huber–Hencky's one, even though they start from completely different assumptions under the conceptual point of view. Today this theory is the most generally accepted for resistance verification of pieces for which ductile materials are used, more than the Guest–Tresca theory. However, it can be noted that codes in the most important industrialized countries are based on the theory of Guest–Tresca, given that it is more conservative than that of Huber–Hencky and easier to apply as well.

6.1.3. PLASTICITY COLLABORATION

In the sizing of pressure vessels the possibility of plastic collaboration of steel is widely exploited. This is a topic that, if threated in depth, would result in a vast and not justified analysis, so only the main aspects will be shortly presented.

The principle at the core of plastic collaboration consists of the possibility that less stressed fibers may contribute to the resistance of the piece by helping the most stressed ones. The adoption of the criteria of plastic collaboration goes against the traditional concept of verification in the elastic field, which says that the condition of danger is reached when the most stressed fiber begins to show signs of yielding in the material. More precisely, if the material is ductile, it can withstand major deformations before rupturing. Therefore, the fact that the material yields in one area of the piece does not represent a condition of danger, if the nearby fibers are still far from yielding. When yielding is reached in a portion of the piece in which a stress gradient is present, the yielded fiber, or fibers, are unable to absorb another increment in stress. Neglecting the hardening that happens after the yielding point, as can be seen from the Figure 35, they undergo an increase in deformation to maintain the material continuity, but the stress remains constant. At the same time, the nearby fibers that are still far from yielding are able to absorb increasing stresses. At the end, while peaks of deformation are present given the constraint to the condition of congruence, peaks of stress have disappeared since in every point the stress is equal to the yield strength. The condition of danger is reached when all fibers have exhausted the possibility to absorb an increase in stress; in other words, the condition of danger is represented by the plastic flow of the entire piece. Following this simplified model, taking into account plastic collaboration corresponds in practice to ignoring the peaks, doing the calculations on the base of the stress average values. Actually, the plastic collaboration theory is more complex, and every stress condition should be examined separately through a process known as stress analysis. This helps to identify the exact nature of the peak, in order to determine which criteria to apply to carry out verification. Also, a coefficient of plastic collaboration, ψ, that represent the condition of danger is found. In addition, it is important to consider those peaks that are the consequence of the respect for congruence of deformations among pieces of different geometrical shape connected with each other. If they were ideally isolated, they would be characterized by different values of deformations in correspondence with the junction. Flat, hemispherical or torospherical heads connected to the ends of the cylinders represent a typical case. In these cases the stresses are self-limiting, since they occur only out of necessity to respect congruence, and not to balance external forces. Indeed, if the yielding point is reached and the subsequent deformations are relatively large, the congruence is maintained without leading the material to rupture. In fact, in these situations it would be more logical to carry out the analysis in terms of deformations. In practice, this is done, and the peak stresses are obtained according to the laws of elasticity, with the product of deformation by the modulus of elasticity. The peak deformation can take a value greater than the yield strength σ_S, however, this happens only from a formal point of view. In reality, it is obvious that the stress is equal to σ_S. In these cases, it is acceptable that the ideal stress computed according to the laws of elasticity may be even double the yield strength. Moreover, it is important to keep in mind that the verification criteria and the deriving equations for sizing discussed in the following chapters refer to work conditions that do not imply significant fatigue phenomena, due to the presence of a limited number of cycles.

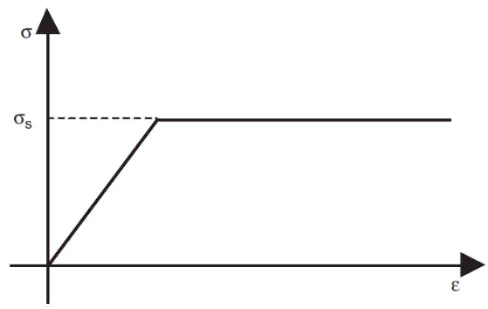


Figure 35: Simplified stress-deformation curve. [17]

6.1.4. VERIFICATION CRITERIA

According to modern verification criteria, stresses can be divided into three categories: primary, secondary, and peak stresses. Then, primary stresses can be further divided into general membrane stresses, local membrane stresses, and primary bending stresses. In summary, there are five types of stresses that now will be analyzed: general membrane, local membrane, primary bending, secondary, and peak.

GENERAL MEMBRANE STRESSES (σ_m)

They correspond to stresses derived from calculation when one considers the element under test as a membrane. More generally, they correspond to the average value of the stresses through the thickness of the vessel. In contrast to local membrane stresses, that we will discuss shortly, the fundamental characteristic of these stresses is that a potential yielding of the material does not cause a redistribution of the stresses, since the same stress is present in all the surrounding fibers. A typical example of general membrane stresses is represented by the average values of the stresses acting in a cylinder area that is not influenced by holes or by the junction with the heads.

LOCAL MEMBRANE STRESSES (σ_{ml})

In this case the average values of the stresses are studied through the thickness in the analyzed section. In contrast with the previous ones, they involve a limited area of the component, and this means that the surrounding fibers are subject to membrane stress of lower value. A potential yielding of the material happens together with a redistribution of the stresses to the surrounding fibers that are still able to contribute to the local resistance of material, since they are not yielded. Typical examples of stresses of this kind are the membrane stresses produced in the cylinder and in the dished heads in correspondence with their junction. Even the membrane stresses that occur in the cylinder and in the nozzle welded together represent an example. As already said, these are the average values of the stresses. In both cases, junction cylinder-heads and in correspondence of the nozzles, stress peaks are generated but they do not fall into this kind of stress category.

PRIMARY BENDING STRESSES (σ_f)

These stresses belong to the category of primary stresses, such as the ones mentioned above, but they are characterized by the fact that their value is proportional to the distance of the fiber from the neutral axis of the section. As the previous ones, they derive from the balance conditions between internal stresses and external forces acting upon the vessel, e.g. pressure or mechanical loads. A typical example is represented by the stresses at the center of a flat head. The stresses produced by bending moments exerted on a vessel with a quadrangular section fall into this category, as well.

SECONDARY STRESSES (σ_{sec})

Their fundamental characteristic is not to be involved in balancing the forces applied to the vessel, but their only purpose is to guarantee the congruence of the deformations. Once the required deformations are produced, even though this happens through the yielding of the material, they do neither cause further deformations nor do they force the intervention of the surrounding fibers, as is the case instead for the local membrane stresses. The stresses in correspondence of the junction between cylinder and heads belong to this category, but not the membrane ones because in that case they are local membrane stresses. The stresses still not related to membranes in the cylinder or in the sphere and in the welded nipple in correspondence of a hole belong to this category, as well. In this last case local peaks due to the presence of sharp edges are excluded, as they belong to the next category.

The stresses due to thermal flux are secondary, as well. In fact, they are also self-limiting, since they are produced solely to re-establish the congruence of the deformations that differ in the various fibers because of the temperature gradient.

<u>PEAK STRESSES</u> (σ_{pic})

Generally speaking, a peak stress is any type of stress that will show a maximum in the section under test. From this point of view peak stresses are the local membrane stresses, as they are locally greater than the general membrane stresses, and the secondary stresses. Going back to the division of stresses in three categories (primary, secondary, and peak), peak stresses do have a different characterization. Only the stresses present in a limited area of section under test, that do not produce significant deformations, and that represent a potential liability only as long as the possibility of rupture is due to fragility or fatigue, belong to this category. They are peaks induced by special geometrical characteristics of the coupled elements that can often be eliminated or at least reduced through an appropriate management of the couplings. Moreover, the stresses due to a rapid variation in the temperature of the fluid inside the vessel belong to this category.

It is important to notice that the stress to be compared with the allowable stress is in any case the ideal stress or the comparison stress according to the Guest–Tresca failure theory. The allowable stresses for the various categories are shown in Figure 36, where the basic allowable stress of the material has been identified with *f*, as already done during its definition. As one can see, the allowable stress is obviously equal to the basic allowable stress of the material for the general membrane stress. Is possible to assume an allowable stress equal to 1.5*f*, therefore generally equal to the yield strength, for local membrane stress and for primary bending stresses. If there are secondary stresses besides primary ones, the resulting maximum ideal stress must not be greater than 3*f*, a value which is generally equal to twice the yield strength. The peak stresses can be neglected if a significant fatigue phenomenon has been ruled out. If this has not been possible, a specific analysis of stresses under fatigue must be performed.

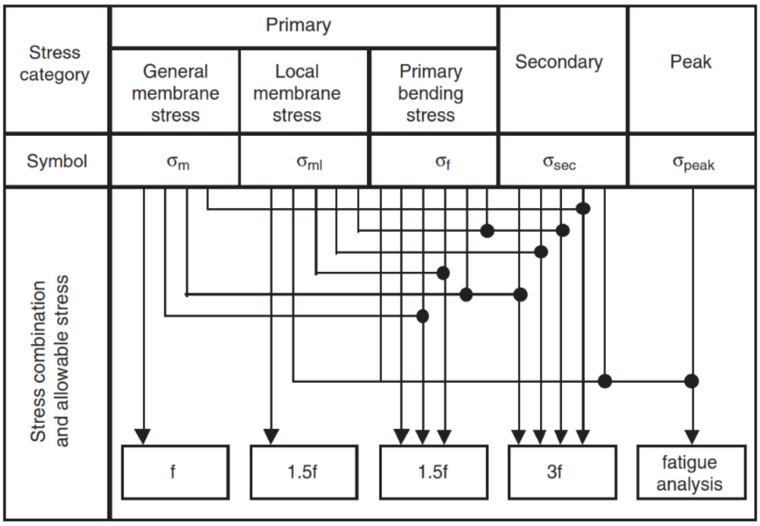


Figure 36: Allowable stress for each stress category. [17]

6.2. GENERAL CALCULATION CRITERIA

6.2.1. MEMBRANE STRESS IN REVOLUTION SHELL

In pressure vessels design, the knowledge of membrane stresses in revolution shells is particularly interesting for the analysis of stress conditions. Indeed, the stresses in cylinders, bended pipes, spheres, spherical heads and cones, provided they are thin, correspond to the membrane stresses. Moreover, they represent the average values of the stresses in vessels of great thickness. Figure 37 shows an element of a generic revolution shell, where R_1 is the radius of curvature according to the meridian and R_2 is the radius of curvature on the plane orthogonal to the meridian. Calling σ_t and σ_m the stresses along the parallel and the meridian, respectively, and *s* the thickness, the forces F_t are on sides A–B and C–D, and can be written as:

$$F_t = \sigma_t s dl_1$$

Their resultant orthogonal to the element is

$$F_1 = 2\sigma_t s dl_1 \frac{d\vartheta_2}{2}$$

On the sides *A – D* and *B – C,* instead, the forces F_m are

$$F_m = \sigma_m s dl_2$$

The resultant orthogonal to the element is

$$F_2 = 2\sigma_m s dl_2 \frac{d\vartheta_1}{2}$$

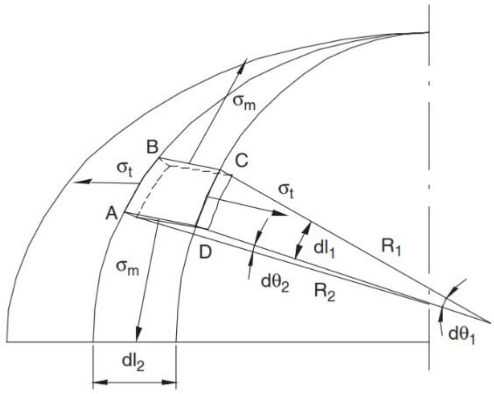


Figure 37: Element of a generic revolution shell. [17]

Given a pressure *p* which produces a force in the same direction of F_1 and F_2, it can be developed as

$$F_3 = p dl_1 dl_2$$

In order to have balance in the direction orthogonal to the element, it must be satisfied

$$F_3 = F_1 + F_2$$

Entering the previously founded equations in the balance one, and rewriting it by means of the geometrical relations below reported, it can be obtained

$$\begin{cases} \sigma_t s dl_1 d\vartheta_2 + \sigma_m s dl_2 d\vartheta_1 = p dl_1 dl_2 \\ dl_1 = R_1 d\vartheta_1 \\ dl_2 = R_2 d\vartheta_2 \end{cases} \rightarrow \sigma_t \frac{s}{R_2} dl_1 dl_2 + \sigma_m \frac{s}{R_1} dl_1 dl_2 = p dl_1 dl_2 \rightarrow \frac{\sigma_m}{R_1} + \frac{\sigma_t}{R_2} = \frac{p}{s}$$

The last equation is completely general, regardless if the axis crosses the membrane or not. However, two generally valid equations that allow to calculate the meridian and circumferential stresses directly can be derived from it if the axis crosses the membrane. Taking the circular area above A – B, as can be seen from Figure 38, the thrust due to the pressure on the portion of the membrane is balanced by the vertical components of the stresses σ_m.

$$\pi R^2 p = 2\pi R s \sigma_m \sin\alpha$$

Inserting in this balance the geometrical relation $R = R_2 \cdot \sin\alpha$, the equation for σ_m can be found. Afterwards, this last formula can be placed in the general one that has been deduced previously in order to find the expression for σ_t. Both results are here reported below.

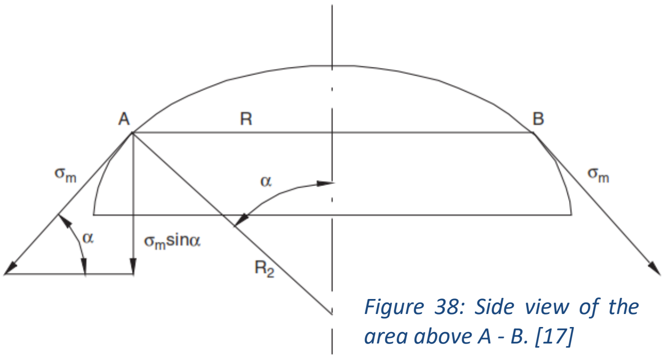


Figure 38: Side view of the area above A - B. [17]

$$\sigma_m = \frac{pR}{2s\ \sin\alpha} = \frac{pR_2}{2s} \qquad \sigma_t = \frac{pR_2}{s}\left(1 - \frac{R_2}{2R_1}\right)$$

These equations, as previously said, allow to compute the meridian and circumferential stresses of a revolution membrane with respect to the axis that crosses itself. For instance, in the case of a cylinder we have $R_2 = D/2$ and $R_1 = \infty$, where *D* is the diameter of the cylinder. The meridian stress in the cylinder is more appropriately indicated as longitudinal or axial stress σ_a, and therefore

$$\sigma_a = \sigma_m = \frac{pD}{4s} \quad and \quad \sigma_t = \frac{pD}{2s}$$

Even if the axis of the cylinder does not actually cross it, the cylinder is enclosed by the heads that release the thrust *F* on the cylinder itself. Therefore, the conditions leading to the general formulas of the two components are still valid.

In the case of sphere $R_1 = R_2 = D/2$, and the resulting equation is

$$\sigma_m = \sigma_t = \frac{pD}{4s}$$

For the cone, a sketch can be seen in Figure 39. Geometrically can be identified $R_1 = \infty$ and $R_2 = r/\cos\alpha$, where *r* is the radius of the generic parallel and *α* the half angle at the tip of the cone. Therefore

$$\sigma_m = \frac{pr}{2s\cos\alpha} \quad and \quad \sigma_t = \frac{pr}{s\cos\alpha}$$

Indicating with *D* the diameter at the base of the cone, the maximum values of the meridian and the circumferential stresses are

$$\sigma_m = \frac{pD}{4s\cos\alpha} \quad and \quad \sigma_t = \frac{pD}{2s\cos\alpha}$$

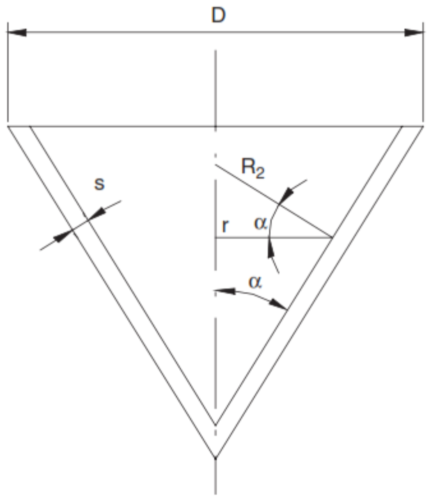


Figure 39: Application of the general equations of the stresses at the cone. [17]

Comparing these equations with those of the cylinder it can be noticed that only the $\cos\alpha$ is missing. Therefore, if the parameter α would be equal to zero, the equations of the cone would correspond to the cylinder ones.

6.2.2. EDGE EFFECTS IN CYLINDERS AND SEMISPHERES

The analysis of edge effects in cylinders and semispheres is fundamental for the study of pressure vessels. Indeed, the state of stress, caused by forces and moments applied in a radial way to the edge of a cylinder or a semisphere, determine the characteristics of the constraint at the junction between cylinder and heads. Moreover, the laws that regulate the distribution of stresses and deformation allow to determine the extent of the collaborating area around a hole and along a nozzle welded to the cylinder. Figure 40 shows a cylinder, with ideally infinite length in the *x* direction, and *r* as average radius. As can be seen, radial forces and moments are evenly distributed on its edge *x=0*.

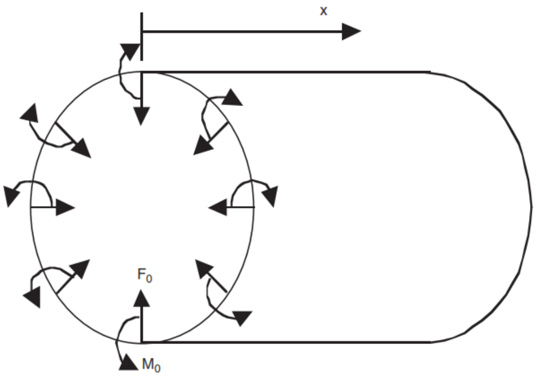


Figure 40: Infinite length cylinder of radius r with forces and moments applied to itself. [17]

Isolating a beam of unitary width from the cylinder, the force F_0 and the bending moment M_0 on the edge of the beam, and per unitary length of the circumference, can be indicated as shown in Figure 41. F_0 and M_0 bend the beam generating a deflection *y* that is a function of *x*. Due to this deflection the radius undergoes a variation in length Δr, equal to *y*, accompanied by a circumferential deformation ε_t and a stress σ_t

$$\varepsilon_t = \frac{\Delta r}{r} = \frac{y}{r}$$

$$\sigma_t = E\varepsilon_t = \frac{Ey}{r}$$

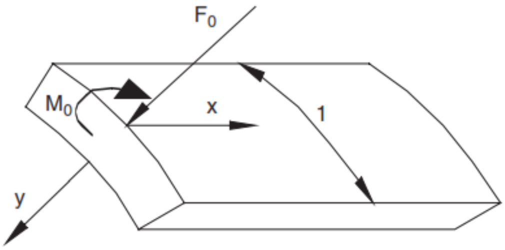


Figure 41: Beam of unitary width. [17]

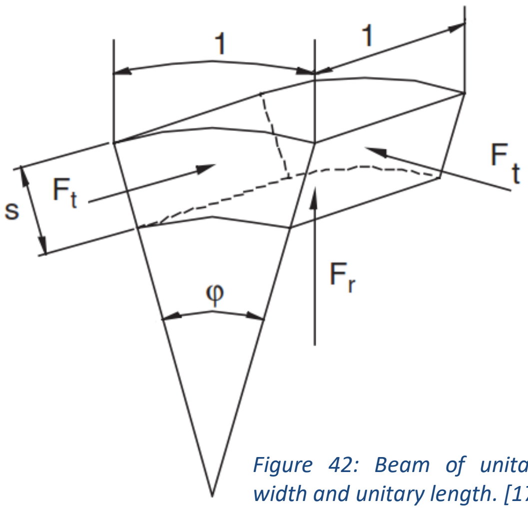


Figure 42: Beam of unitary width and unitary length. [17]

Considering the beam element of unitary length, has shown in Figure 42, the forces F_t on the radial sides, due to the σ_t, are represented by

$$F_t = \sigma_t s = \frac{Es}{r} y$$

A radial force F_r results from the presence of those forces, which form the angle φ between them

$$F_r = 2F_t \frac{\varphi}{2} = F_t \varphi = \frac{Es}{r^2} y$$

As it can be seen, the radial force F_r is proportional to *y*, and the cylinder behaves similarly to a beam.

Considering now an element of length *dx,* shown in Figure 43, the shear forces T and $T + dT$, as well as the moments M_x and $M_x + dM_x$ are on the sides perpendicular to the x-axis. In addition, the radial force dF_r is exerted on the element

$$dF_r = F_r dx = \frac{Es}{r^2} y dx$$

Finally, the moments M_z act upon the radial sides and are derived from congruence requirements. It is now possible to proceed with the balance along the y-axis and the one referred to rotation. Note that the last passage in the moment equation reports the formula obtained by neglecting the terms of higher order.

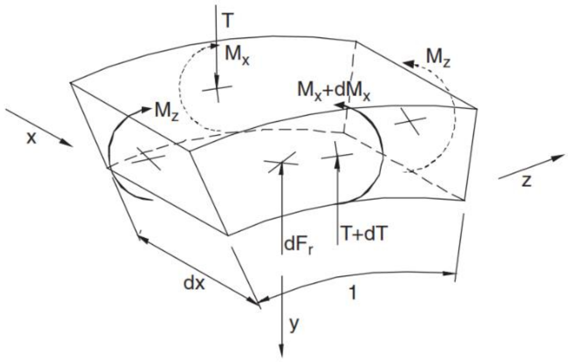


Figure 43: Element of unitary width and dx length. [17]

$$T - \frac{Es}{r^2} y dx - (T + dT) = 0 \; \rightarrow \; \frac{dT}{dx} = -\frac{Es}{r^2} y$$

$$T dx - \frac{Es}{r^2} y \frac{dx^2}{2} + dM_x = 0 \; \rightarrow \; T = -\frac{dM_x}{dx}$$

Deriving the result of the second equation and inserting it in the first one, the following formula can be obtained

$$-\frac{dT}{dx} = \frac{d^2 M_x}{dx^2} = \frac{Es}{r^2} y$$

Being the moments of inertia $I = s^3/12$, identical with respect to *x* and *z* direction, the maximum stresses σ_x and σ_z, produced by M_x and M_z respectively, can be written as

$$\sigma_x = \frac{M_x s}{2I} \quad and \quad \sigma_z = \frac{M_z s}{2I}$$

The deformation ε_z, given by the first following formula, can be elaborated and set equal to zero, since the radial sides must stay radial.

$$\varepsilon_z = \frac{1}{E}(\sigma_z - \mu \sigma_x) = \frac{1}{E}\left(\frac{M_z s}{2I} - \mu \frac{M_x s}{2I}\right) = \frac{s}{2EI}(M_z - \mu M_x) = 0$$

where μ is the Poisson's ratio. This elaboration leads to find the following relation

$$M_z = \mu M_x$$

Now, ε_x can be written in function of M_x only

$$\varepsilon_x = \frac{1}{E}(\sigma_x - \mu \sigma_z) = \frac{1}{E}\left(\frac{M_x s}{2I} - \mu \frac{M_z s}{2I}\right) = \frac{M_x s}{2EI}(1 - \mu^2)$$

Given the last equality, it can be written:

$$\frac{d^2 y}{dx^2} = -\frac{\varepsilon_x}{s/2} = -\frac{M_x}{EI}(1 - \mu^2)$$

Twice deriving and combining it with the equation of $d^2 M_x / dx^2$ previously obtained, leads to

$$\begin{cases} \dfrac{d^4 y}{dx^4} = -\dfrac{d^2 M_x}{dx^2} \dfrac{(1 - \mu^2)}{EI} \\ \dfrac{d^2 M_x}{dx^2} = \dfrac{Es}{r^2} y \end{cases} \rightarrow \quad \frac{d^4 y}{dx^4} = -\frac{s(1 - \mu^2)}{r^2 I} y = -\frac{12(1 - \mu^2)}{r^2 s^2} y$$

where the last passage has been obtained substituting the moment of inertia I of the unitary width beam.

This last equation can be rewritten in function of a factor *β*

$$\frac{d^4y}{dx^4} = -4\beta^4 y$$

That, for the general value μ=0,3 of the steel, can be written as follow

$$\beta = \sqrt[4]{\frac{3(1-\mu^2)}{r^2 s^2}} = \frac{\sqrt[4]{3(1-\mu^2)}}{\sqrt{rs}} = \frac{1{,}285}{\sqrt{rs}} = \frac{1{,}817}{\sqrt{Ds}}$$

The general solution of the differential equation is

$$y = e^{-\beta x}(C_1 \sin\beta x + C_2 \cos\beta x) + e^{\beta x}(C_3 \sin\beta x + C_4 \cos\beta x)$$

with C_1, C_2, C_3 and C_4 integration constants. However, for *x=∞*, *y=0* follows, and therefore $C_3 = C_4 = 0$. Therefore, the general solution of the equation becomes

$$y = e^{-\beta x}(C_1 \sin\beta x + C_2 \cos\beta x)$$

This equation allows to calculate the behavior of both the shear force and the bending moment along the axis of the cylinder, once the values of C_1 and C_2 have been determined according to the boundary conditions. In fact, from the equations obtained during the discussion, the following one can be written

$$M_x = -\frac{Es^3}{12(1-\mu^2)}\frac{d^2y}{dx^2} \qquad with \qquad \frac{d^2y}{dx^2} = 2\beta^2 e^{-\beta x}(-C_1 \cos\beta x + C_2 \sin\beta x)$$

$$T = \frac{Es^3}{12(1-\mu^2)}\frac{d^3y}{dx^3} \qquad with \qquad \frac{d^3y}{dx^3} = 2\beta^3 e^{-\beta x}[(C_1 - C_2)\sin\beta x + (C_1 + C_2)\cos\beta x]$$

From here on, the value of $M_{x=0}$ will be named M_0, and the same will be done for $F_{x=0}$ and F_0. As boundary condition, $M_0 = 0$ can be now considered. It follows that the second derivative of *y* must be zero for $x = 0$, and $C_1 = 0$ can be found. Inserting $C_1 = 0$ in the third derivative of *y*, the following statement can be written

$$\frac{d^3y}{dx^3} = 2\beta^3 C_2$$

Furthermore, given that $T = F_0$ for $x = 0$, its expression can be rewritten as

$$T = \frac{Es^3}{12(1-\mu^2)}\frac{d^3y}{dx^3} = F_0 = \frac{Es^3}{6(1-\mu^2)}\beta^3 C_2$$

And it leads to find the value of the last constant C_2

$$C_2 = \frac{6(1-\mu^2)}{Es^3\beta^3}F_0$$

At this point, the moment can be written as

$$M_x = -\frac{F_0}{\beta}e^{-\beta x}\sin\beta x$$

$$\boxed{\frac{M_x}{-F_0 x} = \frac{e^{-\beta x} \sin \beta x}{\beta x}}$$

The product $(-F_0 x)$ represents the moment at distance *x* from the edge, if the elementary beam discussed earlier were not laterally constrained to the other elementary beams that make up the cylinder.

On the other hand, if $F_0 = 0$ is considered as boundary condition, the third derivative of *y* must be zero for $x = 0$, and therefore

$$C_1 + C_2 = 0$$

It follows that, collecting C_1, the expression of the moment M_x becomes

$$M_x = \frac{Es^3 C_1}{6(1-\mu^2)} \beta^2 e^{-\beta x} (\sin \beta x + \cos \beta x)$$

As already said, for $x = 0$ the notation $M_x = M_0$ is accepted, and consequently

$$C_1 = \frac{6M_0(1-\mu^2)}{Es^3\beta^2}$$

Inserting the value of C_1 in the last moment equation, it can be rewritten as

$$M_x = M_0 e^{-\beta x} (\sin \beta x + \cos \beta x)$$

$$\boxed{\frac{M_x}{M_0} = e^{-\beta x} (\sin \beta x + \cos \beta x)}$$

Where the value of βx is can be easily obtained from the previous definition of β.

$$\beta x = \frac{1,817}{\sqrt{Ds}} x$$

As can be seen, the two framed equations are functions of the parameter $x/\sqrt{Ds}$ only, which is present in *β*. The representative curves of these functions are shown in Figure 44 and they show the peculiar characteristic of the phenomenon. Indeed, it can be noticed the rapid damping of the bending moments, and therefore of the stresses, when moving from the edge of the cylinder inward. Moreover, the curves lead to the following considerations. Regarding the damping of the edge effect, the parameter $\sqrt{Ds}$ plays a crucial role: at a distance $x = \sqrt{Ds}$ from the edge, the moments present in the cylinder are already reduced to about 1/10 compared to those without the damping; while for $x = 1{,}5\,\sqrt{Ds}$ the edge effect becomes

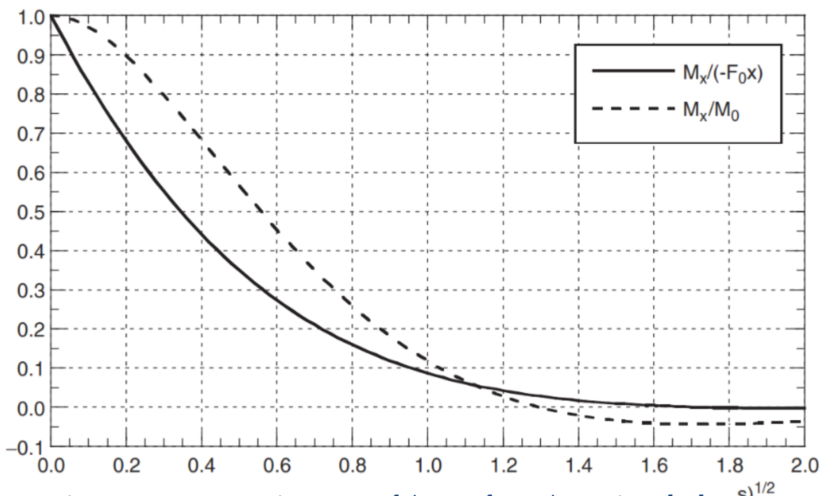


Figure 44: Representative curves of the two framed equations. [17]

almost insignificant. Note that if the thickness $s = 0{,}02D$, the parameter $\sqrt{Ds} = 0{,}14D$, and at a distance from the edge equal to $x = 1{,}5\,\sqrt{Ds} = 0{,}20D$ the edge effect has practically vanished. This has important implications as far as the plastic collaboration and the junctions between elements of different geometrical shape, e.g. cylinder and flat head or dished head. Indeed, in correspondence of the junction mentioned above there are internal forces and moments induced by the need to restore the congruence of deformations. These internal forces and moments at the edges of both elements cause stress peaks of relatively reduced extension due to the damping discussed above. As far as plastic collaboration is concerned, the conclusions resulting from the study of edge effects highlight the need to set limits to the extent of the collaborating area. In fact, it is intuitively clear that if deformations undergo rapid damping when moving away from peaks, the collaboration of too distant fibers could only occur at the expense of undesirably high deformations in correspondence of the peak. In conclusion, it can be stated that the extent of the collaborating area is correlated to the parameter $\sqrt{Ds}$.

Up to this point the discussion has focused only on the edge effects of the cylinder. Now it can move on the semisphere. In Figure 45 a generic parallel plane has been identified through the angle ϕ that the tangent to the sphere forms with the equatorial plane. By indicating with y the radial deflection generated by the forces F_0 and by the moments M_0, the variation of radius of the parallel is equal to $y \sin\varphi$. If r_p is the radius of the parallel plane, the deformation ε_t and a stress σ_t can be written as

$$\varepsilon_t = \frac{y}{r_p}\sin\varphi$$

$$\sigma_t = E\varepsilon_t = \frac{Ey}{r_p}\sin\varphi$$

On the other hand, if r is the radius of the sphere, $r_p = r\sin\varphi$, and the formulation obtained will be the same as the one analysed at the begin of the discussion

$$\sigma_t = \frac{Ey}{r}$$

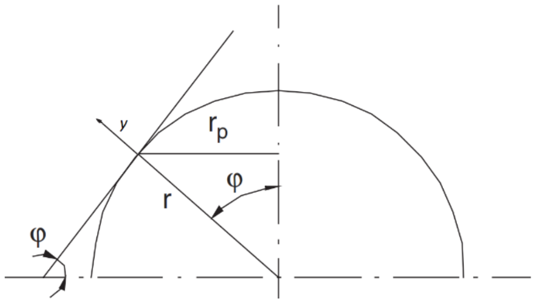


Figure 45: Generic parallel plane has been identified through the angle φ. [17]

Consequently, the forces F_t and the radial forces F_r do not depend on the position of the parallel and coincide with those of the cylinder of equal radius. The discussion about the cylinder is therefore equally valid for the semisphere. It can be argued that in the case of the cylinder has been assumed an infinite length, which does not hold in the case of a semisphere. However, the edge effects rapidly vanish and the discussion about the cylinder is true for the semisphere as well, with the only exception of very thick heads.

6.2.3. STRESS CONCENTRATION AROUND HOLES

Consider a hole with a radius r in a plate of ideally infinite size undergoing a uniform stress σ along the y-axis, as shown in Figure 46. The generic element at distance ρ from the center of the hole that is located at an angle θ with the y-axis, undergoes the following stresses:

$$\sigma_t = \frac{\sigma}{2}\left(1+\frac{r^2}{\rho^2}\right) - \frac{\sigma}{2}\left(1+\frac{3r^4}{\rho^4}\right)\cos 2\vartheta$$

$$\sigma_\rho = \frac{\sigma}{2}\left(1-\frac{r^2}{\rho^2}\right) + \frac{\sigma}{2}\left(1-\frac{4r^2}{\rho^2}+\frac{3r^4}{\rho^4}\right)\cos 2\vartheta$$

$$\tau = -\frac{\sigma}{2}\left(1+\frac{2r^2}{\rho^2}-\frac{3r^4}{\rho^4}\right)\sin 2\vartheta$$

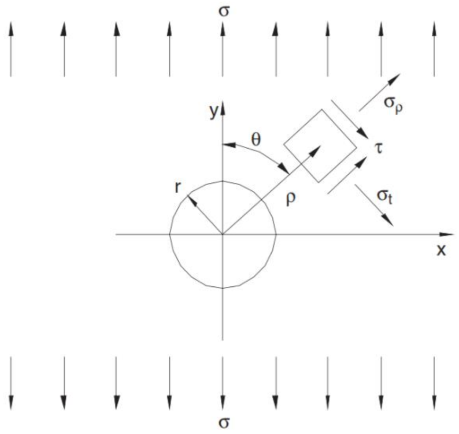


Figure 46: Generic element at distance ρ from center. [17]

Specifically, for:

- $\vartheta = \pi/2$, and hence along the x-axis

$$\sigma_{tx} = \frac{\sigma}{2}\left(2+\frac{r^2}{\rho^2}+\frac{3r^4}{\rho^4}\right)$$

$$\sigma_{\rho x} = \frac{3\sigma}{2}\left(\frac{r^2}{\rho^2}-\frac{r^4}{\rho^4}\right)$$

$$\tau_x = 0$$

- $\vartheta = 0$, and hence along the y-axis

$$\sigma_{ty} = \frac{\sigma}{2}\left(\frac{r^2}{\rho^2}-\frac{3r^4}{\rho^4}\right)$$

$$\sigma_{\rho y} = \frac{\sigma}{2}\left(2-\frac{5r^2}{\rho^2}+\frac{3r^4}{\rho^4}\right)$$

$$\tau_y = 0$$

Figure 47 (a) and (b) shows $\sigma_{tx}, \sigma_{\rho x}, \sigma_{ty}$and $\sigma_{\rho y}$. Notice that for $\rho = \infty$, $\sigma_{tx} = \sigma$, $\sigma_{\rho x} = 0$, $\sigma_{ty} = 0$ and $\sigma_{\rho y} = \sigma$. For $\rho = r$, therefore at the edge of the hole, can be determined that $\sigma_{tx} = 3\sigma$ and $\sigma_{ty} = -\sigma$. So, in the direction of the σ active on the plate, and in correspondence of the edge of the hole, the stress peak is equal to three times the stress in areas that are not influenced by the hole, and it rapidly decreases.

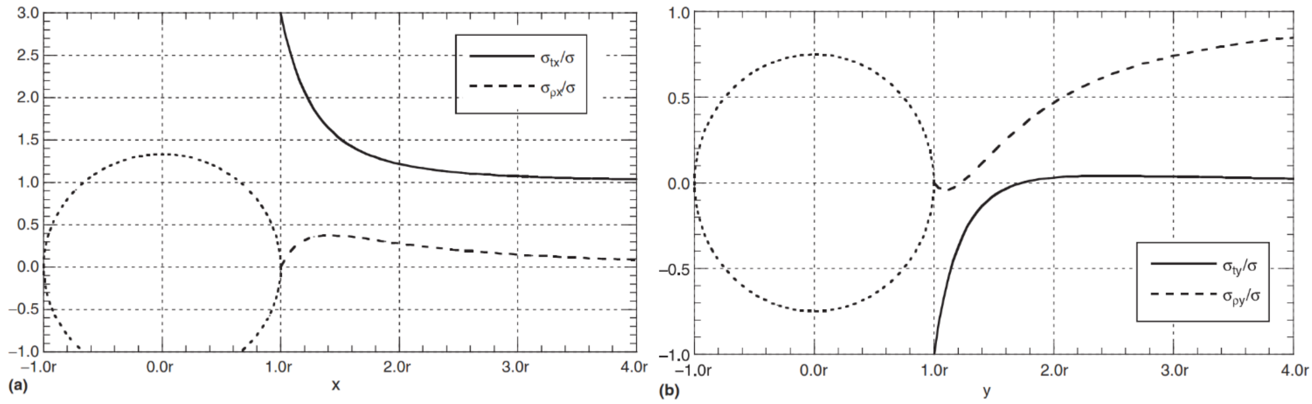


Figure 47: Graphical representation of $\sigma_{tx}, \sigma_{\rho x}, \sigma_{ty}$and $\sigma_{\rho y}$. [17]

If two identical stresses σ are active in the plate along both x- and y-axis, as shown in Figure 48, the following stresses can be determined

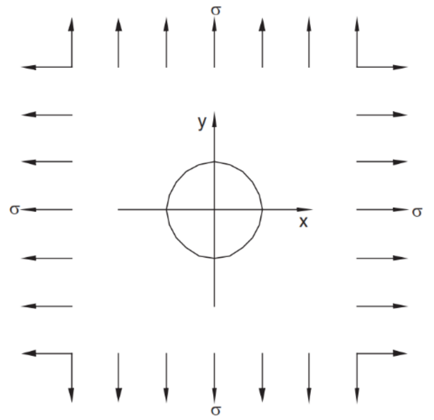


Figure 48: Plate with two identical stresses σ active along both the x- and y-axis. [17]

$$\sigma_{tx} = \sigma_{ty} = \sigma\left(1 + \frac{r^2}{\rho^2}\right)$$

$$\sigma_{\rho x} = \sigma_{\rho y} = \sigma\left(1 - \frac{r^2}{\rho^2}\right)$$

$$\tau_x = \tau_y = 0$$

They are represented in the Figure 49, and, in correspondence of the edge of the hole, their values are $\sigma_{tx} = \sigma_{ty} = 2\sigma$.

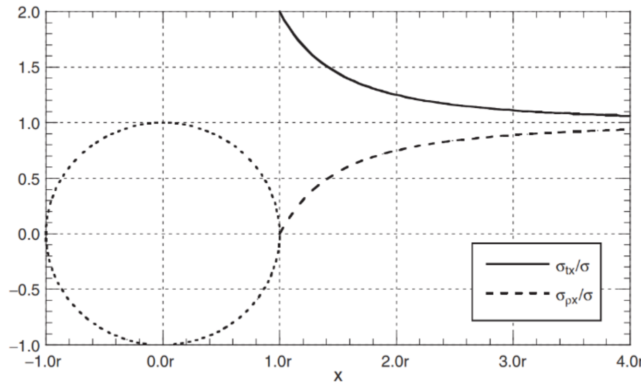


Figure 49: Graphical representation of $\sigma_{tx}, \sigma_{\rho x}, \sigma_{ty}$ and $\sigma_{\rho y}$ if two identical stresses σ are active in perpendicular directions. [17]

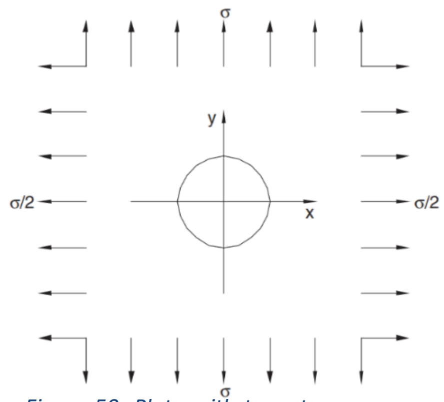


Figure 50: Plate with two stresses, one equal to σ along the y-axis and the other equal to $\sigma/2$ along the x-axis. [17]

Finally, if two stresses are on the plate, one equal to σ along the y-axis and the other equal to $\sigma/2$ along the x-axis, as shown in Figure 50, the values turn to be

$$\sigma_{tx} = \frac{\sigma}{4}\left(4 + \frac{3r^2}{\rho^2} + \frac{3r^4}{\rho^4}\right) \qquad \sigma_{ty} = \frac{\sigma}{4}\left(2 + \frac{3r^2}{\rho^2} - \frac{3r^4}{\rho^4}\right)$$

$$\sigma_{\rho x} = \frac{\sigma}{4}\left(2 + \frac{r^2}{\rho^2} - \frac{3r^4}{\rho^4}\right) \qquad \sigma_{\rho y} = \frac{\sigma}{2}\left(4 - \frac{7r^2}{\rho^2} + \frac{3r^4}{\rho^4}\right)$$

$$\tau_x = 0 \qquad \tau_y = 0$$

The pattern of behavior of such stresses is shown in Figure 51.

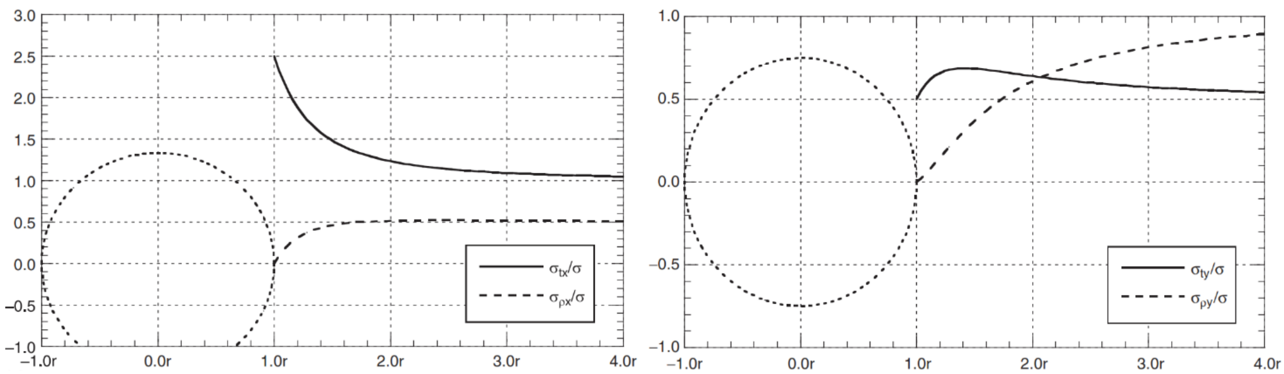


Figure 51: Graphical representation of $\sigma_{tx}, \sigma_{\rho x}, \sigma_{ty}$ and $\sigma_{\rho y}$ if two stresses σ and $\sigma/2$ are active in perpendicular directions. [17]

Notice that, in correspondence of the edge of the hole the stresses are $\sigma_{tx} = 2{,}5\sigma$ and $\sigma_{ty} = 0{,}5\sigma$.

Figure 48 indicatively represents the situation in correspondence of a hole on a spherical head, or at the center of a torospherical head under pressure. In fact, as already discussed, the meridian and circumferential stresses are equal in the sphere. A stress peak equal to twice the circumferential, or meridian, stress would be generated in correspondence of the hole on a spherical head.

Similarly, Figure 50 may describe the situation in correspondence of a hole in a cylinder, with the y-axis along the circumference and the x-axis along the axis of the cylinder. In fact, as shown in the previous chapter, the longitudinal stress in the cylinder is equal to half the hoop stress. Therefore, a stress peak twice and a half the hoop stress in the cylinder would occur in correspondence of the hole.

From this point of view, there would be no difference between a hole with a small or large diameter compared to the diameter of the cylinder or the sphere, with respect to the amount of increase in membrane stress in correspondence of the hole. Given that this peak stress is a local membrane stress, it is necessary to limit its value to $1{,}5f$ that corresponds to the yield strength. Therefore, the thickness of the cylinder needs to be locally increased, regardless of the diameter of the hole. Specifically, it can be observed that in the case of cylinder it would be necessary to increase the thickness and make it equal to 5/3 of the required thickness in the absence of holes. The hoop stress of reference σ would thus be equal to 3/5 of the basic allowable stress of the material. The value of the peak stress would therefore decrease to $2{,}5 \cdot {}^{3}\!/_{5} = 1{,}5f$, as required. The size of the reinforced area may be quite small. It can be noticed that for $\rho = 1{,}5r$ the value of the peak is already lower than $1{,}5\sigma$. Therefore, the reinforcement may be limited to a distance from the edge of the hole equal to $0{,}5r$, this notwithstanding the influence that local variations in thickness may exert on deformations and on the stress behavior. However, I can be noticed that increasing the thickness of the cylinder by 66% for a width equal to 0.5*r*, as required, the missing area due to the presence of the hole would not be compensated. In fact, the reinforcement area is equal to 0.5*r* × 0.66*s* = 0.33*rs,* where *s* is the thickness of the wall, while the corresponding area of half the hole is equal to *rs*.

In the case of a hole on a cylinder, a sphere, or at the center of a torospherical head, the curvature of the plate cannot be neglected. In fact, the peak at the hole's edge corresponds to a local increase in deformations that results in a local increase in radius greater than in the remaining component. The fibers around the hole go against a greater radial deflection in a similar way to what analyzed for the edge effects discussed in the previous chapter. Moreover, if a nozzle is welded in correspondence of a hole, the axial rigidity of the nipple counteracts the local increase in radius.

Experimental tests have shown that the area of the collaborating plate around the hole does not depend on its diameter, as the previous discussion about the flat plate would lead one to believe. However, similarly to the edge effects, being *D* the diameter of the cylinder or the sphere, and *s* its thickness, the area of the collaborating plate around the hole depend on $\sqrt{Ds}$. In addition, the diameter of the hole strongly influences the entity of the peaks. In the case of a small hole, the phenomenon is localized to a limited area. The increase in radius, which goes with a local increase in hoop membrane stress, is counteracted by the surrounding areas, that maintain their rigidity with minor changes versus the component without holes. If the hole is large, the cylinder or the sphere turn out to be weaker, and at the edge of the hole variations in radius that cause the stress peak are more likely. The widely accepted calculation criterion consists of compensating for the area of the hole by correlating the thickness of the reinforcement to the diameter of the hole. Even if this argument will be again mentioned during the study case, the whole discussion of the correlation is not proposed inside this thesis.

6.3. CYLINDERS UNDER INTERNAL PRESSURE

Three principal stresses are generated by internal pressure: a hoop stress σ_t, a radial stress σ_r, and a longitudinal stress σ_a. The latter is due to the thrust of pressure on the heads of the cylinder. The values of the stresses σ_t and σ_r, differently from the value of σ_a, are not constant through the cylinder wall. In the design phase it is therefore necessary to derive the ideal stress considering the stresses of the triaxial state. This procedure can be done by means of the theories of failure described in the section "Preliminary Considerations". Assuming that the ideal stress is equal to the basic allowable stress, an equation to compute the minimal required thickness can be obtained. In the first place, considering the semi-cylinder of unitary length shown in Figure 52, it may be useful recall the Mariotte's method. The resultant force along y it is obviously zero, while the one along x can be written as $F = pD_i$. Two identical forces equal to $F/2$ at the ends of the semi-cylinder have to be applied to balance this thrust. If the hoop stress in the cylinder is assumed as constant through the thickness, it can be expressed by means of the following formula:

$$\sigma_t = \frac{F/2}{s} = \frac{pD_i}{2s}$$

While, if the hoop stress is equal to the basic allowable stress *f*, the following formula, called Mariotte's formula, is obtained.

$$s = \frac{pD_i}{2f}$$

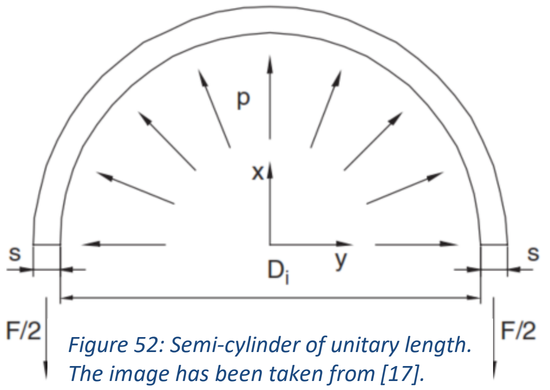


Figure 52: Semi-cylinder of unitary length. The image has been taken from [17].

However, given the assumption, it does not consider the variation of σ_t through the thickness, as well as the presence of the other two principal stresses σ_r and σ_a. Therefore, even if it is simply and practical, it cannot be used for the sizing of the cylinder. Anyhow, this procedure remains important because the value of σ_t obtained through this equation represents the hoop stress in a membrane. Indeed, considering the cylinder as a membrane reducing its thickness to a point, σ_t would no longer be a function of the radius, being the radius a unique value. Therefore, the obtained value is constant, and it corresponds to the average value, which is not dependent on the thickness.

Considering now the generic element shown in Figure 53. It has unitary dimension in the direction orthogonal to the figure itself, and it can be defined by means of the radius *r*, its thickness *dr* and the angle *dφ*. The stress σ_t, constant along the circumference, is exercised on sides A–B and C–D. Therefore, the equilibrium in this direction is assured in any case.

$$F_{AB} = F_{CD} = \sigma_t dr$$

The resulting, based on such two forces in the radial direction, is:

$$F_{AB,r} = 2F_{AB}\frac{d\varphi}{2} = \sigma_t dr d\varphi$$

Regarding the equilibrium in the radial direction, the presence of the force F_{AC} on the A–C side can be noticed.

$$F_{AC} = \sigma_r r d\varphi$$

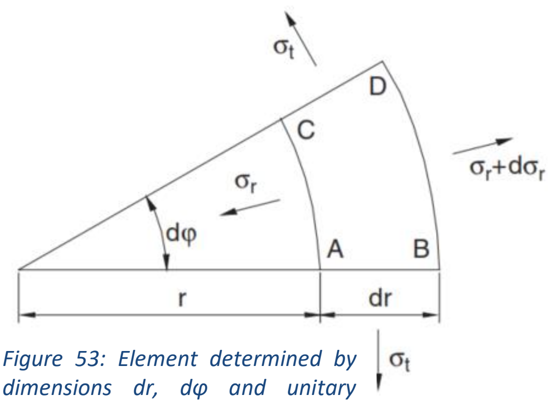


Figure 53: Element determined by dimensions dr, dφ and unitary thickness. [17]

Similarly, on the B–D side, a force can be developed neglecting the terms of higher order

$$F_{BD} = -(\sigma_r + d\sigma_r)(r + dr)d\varphi = -\sigma_r r d\varphi - \sigma_r dr d\varphi - r d\sigma_r d\varphi$$

The equation of equilibrium with reference to the radius must be expressed by the sum of the following forces. According to the previous equations, and with $d\varphi$ having a nonzero value, the formula can be rewritten as in the second passage.

$$F_{BD} + F_{AC} + F_{AB,r} = 0 \quad \rightarrow \quad -\sigma_r dr - r d\sigma_r + \sigma_t dr = 0$$

At the end, the equilibrium equation of the cylinder is

$$\boxed{\sigma_t - \sigma_r - r\frac{d\sigma_r}{dr} = 0}$$

Regarding the congruence of deformations, consider the circular ring of thickness *dr* shown in Figure 54. Due to the circumferential deformation ε_t the radii of circles *α* and *β* have a respective elongation Δr_α and Δr_β

$$\Delta r_\alpha = \varepsilon_t r \qquad and \qquad \Delta r_\beta = (\varepsilon_t + d\varepsilon_t)(r + dr\ )$$

To impose the congruence, the difference between these two elongations must correspond to the increment of thickness in the ring.

$$\Delta s = \varepsilon_r dr = \ \Delta r_\beta - \Delta r_\alpha = \varepsilon_t dr + d\varepsilon_t r$$

From this last equality it can be obtained the equation of congruence of the cylinder.

$$\boxed{\varepsilon_r - \varepsilon_t - r\frac{d\varepsilon_t}{dr} = 0}$$

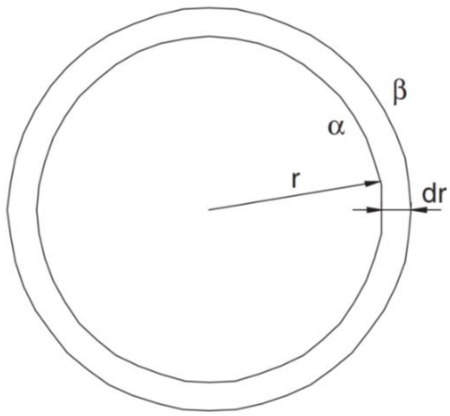


Figure 54: Circular rings of thickness dr. [17]

The expressions of the deformations obtained in the section “General Calculation Criteria”, where *E* was the normal modulus of elasticity and µ was the Poisson’s ratio, can be recalled

$$\varepsilon_t = \frac{1}{E}\left(\sigma_t - \mu(\sigma_r + \sigma_a)\right)$$

$$\varepsilon_r = \frac{1}{E}\left(\sigma_r - \mu(\sigma_t + \sigma_a)\right)$$

$$\varepsilon_a = \frac{1}{E}\left(\sigma_a - \mu(\sigma_t + \sigma_r)\right)$$

Inserting they in the previous obtained congruence equation of the cylinder, the formula in the first line can be obtained. Subsequently, a second formulation can be written starting from it.

$$\sigma_r - \sigma_t - r\frac{d\sigma_t}{dr} - \mu\left(\sigma_t - \sigma_r - r\frac{d\sigma_r}{dr}\right) + \mu r\frac{d\sigma_a}{dr} = 0$$

$$(1-\mu)\left(\sigma_t - \sigma_r - r\frac{d\sigma_r}{dr}\right) - r\frac{d\sigma_r}{dr} - r\frac{d\sigma_t}{dr} + \mu r\frac{d\sigma_a}{dr} = 0$$

Given that the part expressed inside the brackets of the previous equation represent the equilibrium equation of the cylinder, which is equal to zero, and the radius *r* is a non-zero value, it can be rewritten as

$$\frac{d\sigma_r}{dr}+\frac{d\sigma_t}{dr}=\mu\frac{d\sigma_a}{dr}$$

Deriving the third of the before recalled expressions of the deformations, and substituting the term $d\sigma_a/dr$ with the above achieved equation, the following expressions can be developed

$$\frac{d\varepsilon_a}{dr}=\frac{1}{E}\left[\frac{d\sigma_a}{dr}-\mu\left(\frac{d\sigma_t}{dr}+\frac{d\sigma_r}{dr}\right)\right] \quad \rightarrow \quad \frac{d\varepsilon_a}{dr}=\frac{1-\mu^2}{E}\frac{d\sigma_a}{dr}$$

For the sections of the cylinder to remain flat, ε_a must be constant and therefore the derivative of σ_a is zero, according to the last equation. The longitudinal stress σ_a is therefore constant, and the previously found equation becomes

$$\frac{d\sigma_r}{dr}+\frac{d\sigma_t}{dr}=0$$

As last passage, the equilibrium equation of the cylinder can be derived and introduced in the above equation

$$\frac{d\sigma_t}{dr}-\frac{d\sigma_r}{dr}-\frac{d\sigma_r}{dr}-r\frac{d^2\sigma_r}{dr^2}=0$$

$$\frac{d^2\sigma_r}{dr^2}=-\frac{3}{r}\frac{d\sigma_r}{dr}$$

Assuming that $y=d\sigma_r/dr$, it can be rewritten as a differential equation with separable variables.

$$y'=-\frac{3}{r}y$$

Which general integral, with Z and K constants, can be written as follow

$$\int\frac{dy}{y}=-\int\frac{3}{r}dr+Z \quad \rightarrow \quad \log_e y=-3\log_e r+Z=\log_e\frac{K}{r^3} \quad \rightarrow \quad y=\frac{d\sigma_r}{dr}=\frac{K}{r^3}$$

Finally, integrating the last equality and inserting the result in the above equations, being A and B constants, the expressions for σ_r and σ_t are obtained as follow.

$$\sigma_r=A-\frac{B}{r^2} \qquad and \qquad \sigma_t=A+\frac{B}{r^2}$$

Note that if $r=r_e$, with r_e the external radius, σ_r must be equal to zero, and therefore:

$$A=\frac{B}{r_e^2}, \qquad then \qquad \sigma_r=B\left(\frac{1}{r_e^2}-\frac{1}{r^2}\right)$$

On the other hand, if $r=r_i$, with r_i the inside radius, the radial compressive stress is equal to the pressure p, and therefore

$$-\mathrm{p}=B\left(\frac{1}{r_e^2}-\frac{1}{r_i^2}\right) \quad \rightarrow \quad B=\frac{pr_e^2}{\frac{r_e^2}{r_i^2}-1}$$

Assuming $a = r_e^2/r_i^2$ the three principal stresses can be finally written as follow. Please note that, as already said, σ_a is constant, and based on the thrust on the heads.

$$\sigma_t = \frac{p}{a^2 - 1}\left(1 + \frac{r_e^2}{r^2}\right)$$

$$\sigma_r = \frac{p}{a^2 - 1}\left(1 - \frac{r_e^2}{r^2}\right)$$

$$\sigma_a = \frac{\pi p r_i^2}{\pi\left(r_e^2 - r_i^2\right)} = \frac{p}{a^2 - 1}$$

These equations are known as Lamè's equations. Figure 55 and Figure 56 illustrate the diagrams of the three principal stresses and deformations, respectively. it is evident that σ_t reaches its maximum value in correspondence to the internal fibers. The same applies to the absolute value of σ_r, while the longitudinal stress has a value in between the two.

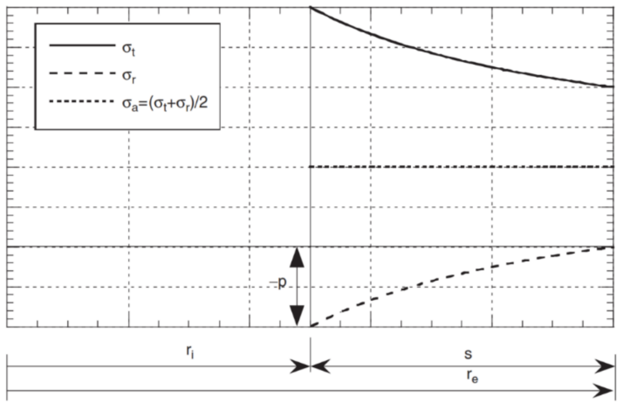


Figure 55: Stresses in the elastic condition. [17]

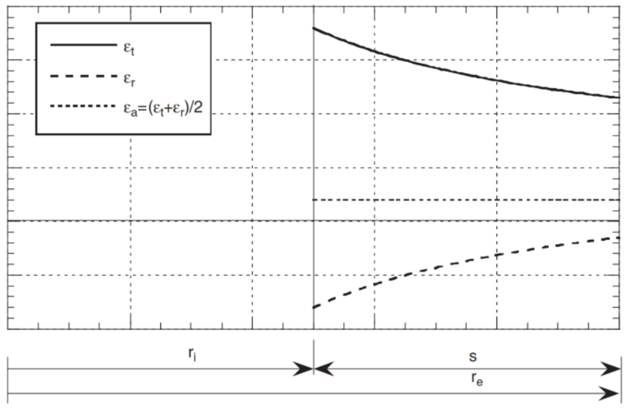


Figure 56: Deformations in the elastic condition. [17]

Therefore, the maximum value of the ideal stress, according to Guest, must be found in correspondence to the internal fibers, and it can be computed from σ_t and σ_r.

$$\sigma_{ti} = p\frac{a^2 + 1}{a^2 - 1} \quad and \quad \sigma_{ri} = -p$$

$$\sigma_{id(i)} = \sigma_{ti} - \sigma_{ri} = \frac{2a^2}{a^2 - 1}$$

As shown in section "Preliminary Considerations", there is general consensus that one should not look at the peak stress located in the internal fiber during the design, but instead take into account the possibility that other less stressed fibers may cooperate with the latter. According to this criterion, the calculation can be carried out in different ways. In this thesis only the approach which account for plastic collaboration in a streamlined and global way will be examined. However, the other method leads to a practically coincident equations which, even if σ_a is no more constant, slightly overestimate the thickness. In the analyzed method the danger is no longer linked to the maximum stress value, so at the yielding of the internal fiber, but rather to the whole plastic flow of the cylinder, which takes place when the ideal stress is equal to the yield strength everywhere. In other words, following Guest's theory the dangerous situation is represented by

$$\sigma_{id} = \sigma_t - \sigma_r = \sigma_s$$

where σ_s is the yield strength. Therefore, the equilibrium equation of the cylinder, and its integration, are

$$\sigma_s - r\frac{d\sigma_r}{dr} = 0 \quad \rightarrow \quad \sigma_r = \sigma_s \log_e r + \mathrm{A}$$

Where A is a constant which, recalling that $\sigma_r = 0$ if $r = r_e$, is equal to

$$A = -\sigma_s \log_e r_e$$

Inserting the value of A the equation of the radial stress can be written.

$$\sigma_r = \sigma_s \log_e \frac{r}{r_e}$$

On the other hand, for $r = r_i$, $\sigma_r = -p$, and therefore

$$p = \sigma_s \log_e \frac{r_e}{r_i} = \sigma_s \log_e a \quad \rightarrow \quad a = \frac{r_e}{r_i} = e^{p/\sigma_s}$$

This last equation means that, given a certain pressure *p* and a material having a yield strength equal to σ_s, if the ratio between outside and inside radii is equal to the value given by the exponential, the cylinder is completely plasticized. Of course, this approaches a danger condition and therefore a safety factor must be applied. The safety condition can be obtained by replacing σ_s with the allowable stress f.

$$a = \frac{r_e}{r_i} = e^{p/f}$$

The equation obtained allow to safely size the cylinder while taking into account the plastic collaboration. So, the minimum required thickness can be obtained from it

$$\boxed{s = \frac{e^{p/f} - 1}{2e^{p/f}} D_e}$$

Now, an alternative equation, called the average diameter equation can be founded. The average of σ_r can be computed from the extreme values, obtained for $r = r_i$ and $r = r_e$, and it results

$$\sigma_{rm} = -\frac{p}{2}$$

The average value of σ_t is simply the membrane stress σ mentioned at the beginning, thus recalling

$$\sigma_{tm} = \frac{pD_i}{2s} = \frac{pr_i}{r_e - r_i} = \frac{p}{a-1}$$

The ideal average stress that that represents the general membrane stress according to the section "Preliminary considerations – Verification criteria", will be given by

$$\sigma_{id(m)} = \sigma_{tm} - \sigma_{rm} = \frac{p}{a-1} + \frac{p}{2} = \frac{p}{2}\frac{a+1}{a-1}$$

With $\sigma_{id(m)} = f$ the following equality can be written

$$\frac{a+1}{a-1} = \frac{\frac{r_e}{r_i}+1}{\frac{r_e}{r_i}-1} = \frac{r_e+r_i}{r_e-r_i} = \frac{2\mathrm{f}}{\mathrm{p}}$$

The following equation, where D_m is the average diameter, can be now obtained

$$\frac{D_m}{s} = \frac{2\mathrm{f}}{\mathrm{p}} \quad \rightarrow \quad \boxed{s = \frac{pD_m}{2\mathrm{f}}}$$

This equation is the so-called average diameter equation, even if, it can also be rewritten considering the outside dimeter

$$p(D_e - s) = 2\mathrm{fs} \quad \rightarrow \quad \boxed{s = \frac{pD_e}{2\mathrm{f}+\mathrm{p}}}$$

This last equation is used in many national codes, e.g. in the ISO Code. It is more conservative than the Mariotte's formula seen at the begin, and therefore definitely acceptable. It is easier to use and results in walls only slightly thicker than strictly required. Until now it has been implicitly assumed that the cylinder does not have holes, or that the holes are completely compensated for, and that welding does not cause a reduction of the cylinder's resistance in the seams. Even though the influence of holes should be analyzed, and welded joints can generally be less resistant compared to the basic metal that constitutes the cylinder. It is therefore introduced a weld joint efficiency, here defined as *z,* which identifies the allowable stress at the weld joint as a function of the allowable stress of the basic metal through the relationship

$$f' = fz$$

Where f' is the allowable stress in the weld joint, and z is always less than or equal to one. Therefore, the equation of the thickness obtained from the outside diameter can be written in more general terms as follow.

$$s = \frac{pD_e}{2\mathrm{fz}+\mathrm{p}}$$

6.4. THERMAL STRESS OF CYLINDERS UNDER INTERNAL PRESSURE

If the wall of the cylinder is subjected to a thermal flux, stresses are created to re-establish the congruence of deformations. However, the congruence is perturbed by thermal dilations that vary throughout the wall. In fact, due to the heat transmission, the temperature of the metal is a function of the radius. The law of variations in temperature can be found via the following considerations.

Examining the cylinder of unitary length shown in Figure 57, the generic surface *S* corresponding to radius *r* is $S = 2\pi r$. The heat *q* that crosses this surface can be written as

$$q = \lambda S \frac{dT}{dr}$$

where dT/dr is the temperature gradient and λ is the thermal conductivity. Inserting the formula of the surface inside it, *dT* can be made explicit as:

$$dT = \frac{q}{2\pi\lambda}\frac{dr}{r}$$

Therefore, by integrating, the formula for the temperature is obtained. Being C a constant, it turns out to be

$$T = \frac{q}{2\pi\lambda}\log_e r + C$$

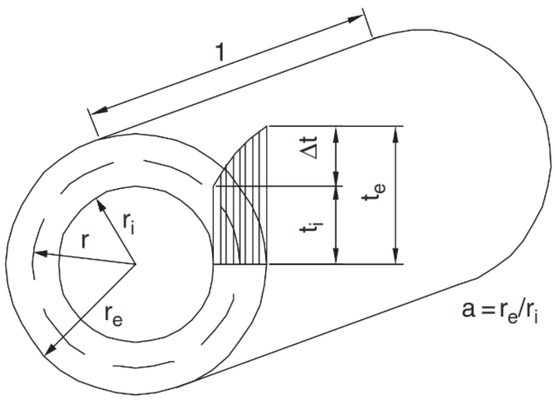


Figure 57: Cylinder of unitary length. [17]

By indicating with T_e and T_i the temperatures corresponding to r_e and r_i respectively, their difference ΔT can be expressed as:

$$\Delta T = T_e - T_i = \frac{q}{2\pi\lambda}\log_e \frac{r_e}{r_i}$$

As usual, $a = r_e/r_i$ is imposed, and the following formulation of the temperature gradient can be obtained

$$\frac{dT}{dr} = \frac{\Delta T}{r \log_e a}$$

To compute the stresses on the cylinder due to the effect of ΔT, the equations of equilibrium and congruence discussed in the previous chapter have to be recalled

$$\sigma_t - \sigma_r - r\frac{d\sigma_r}{dr} = 0 \quad \text{and} \quad \varepsilon_r - \varepsilon_t - r\frac{d\varepsilon_t}{dr} = 0$$

Contrariwise, the expressions of the deformations previously obtained in the section “General Calculation Criteria” can be now written in the following different way.

$$\varepsilon_t = \frac{1}{E}\big(\sigma_t - \mu(\sigma_r + \sigma_a)\big) + \alpha T$$

$$\varepsilon_r = \frac{1}{E}\big(\sigma_r - \mu(\sigma_t + \sigma_a)\big) + \alpha T$$

$$\varepsilon_a = \frac{1}{E}\big(\sigma_a - \mu(\sigma_t + \sigma_r)\big) + \alpha T$$

Inserting the expressions of the deformations, in which α represent the thermal expansion coefficient of the material, in the equation of congruence, the first formulation that follow can be obtained. The second one, instead, has been found using the equation of equilibrium.

$$-(1+\mu)\left(\sigma_t-\sigma_r-r\frac{d\sigma_r}{dr}\right)-r\frac{d\sigma_r}{dr}-r\frac{d\sigma_t}{dr}+\mu r\frac{d\sigma_a}{dr}-Er\alpha\frac{dT}{dr}=0 \rightarrow \frac{d(\sigma_t+\sigma_r)}{dr}=\mu\frac{d\sigma_a}{dr}-E\alpha\frac{dT}{dr}$$

Deriving the third expression of the deformations, and imposing $d\varepsilon_a/dr=0$, given that the sections of the cylinder orthogonal to the axis remain flat, the following expression can be developed

$$\frac{d\varepsilon_a}{dr}=\frac{1}{E}\left(\frac{d\sigma_a}{dr}-\mu\frac{d(\sigma_t+\sigma_r)}{dr}\right)+\alpha\frac{dT}{dr} \quad \rightarrow \quad \frac{d\sigma_a}{dr}=\mu\frac{d(\sigma_t+\sigma_r)}{dr}-E\alpha\frac{dT}{dr}$$

Inserting the last equation in the previous result, the following expression can be found

$$\frac{d(\sigma_t+\sigma_r)}{dr}=\mu^2\frac{d(\sigma_t+\sigma_r)}{dr}-E\alpha\mu\frac{dT}{dr}-E\alpha\frac{dT}{dr} \quad \rightarrow \quad \frac{d(\sigma_t+\sigma_r)}{dr}=-\frac{E\alpha}{1-\mu}\frac{dT}{dr}$$

Using the expression of the thermal gradient previously developed, and setting $K=\frac{E\alpha\Delta T}{2(1-\mu)}$, the last expression can be modified and integrated, with *A=constant*, as follow

$$\frac{d(\sigma_t+\sigma_r)}{dr}=-\frac{2K}{r\log_e a} \quad \rightarrow \quad \sigma_t+\sigma_r=-\frac{2K}{\log_e a}\log_e r+2A \rightarrow \quad \sigma_t=-\frac{2K}{\log_e a}\log_e r+2A-\sigma_r$$

Inserting the expression for σ_t inside the equations of equilibrium shown earlier and integrating

$$2\sigma_r-r\frac{d\sigma_r}{dr}+\frac{2K}{\log_e a}\log_e r-2A=0 \quad \rightarrow \quad \sigma_r=A-\frac{B}{r^2}+\frac{K}{2\log_e a}\left(1-2\log_e r\right)$$

where *B* is constant. For to $r=r_i$ and $r=r_e$, $\sigma_r=0$ have to be respected, and therefore

$$A-\frac{B}{r_e^2}+\frac{K}{2\log_e a}\left(1-2\log_e r_e\right)=0$$

$$A-\frac{B}{r_i^2}+\frac{K}{2\log_e a}\left(1-2\log_e r_i\right)=0$$

By subtracting the second of those equations from the first one, and inserting the obtained expression, the values for B and A can be found.

$$B=\left(\frac{1}{r_e^2}-\frac{1}{r_i^2}\right)+K=0 \quad \rightarrow \quad B=-\frac{K}{\frac{1}{r_e^2}-\frac{1}{r_i^2}}=\frac{Kr_e^2}{a^2-1}$$

$$A=K\frac{a^2}{a^2-1}-\frac{K}{2\log_e a}\left(1-2\log_e r_i\right)$$

Now, the value of the constants A and B can be inserted in the formulas derived so far, and new formulations of σ_r and σ_t can be found.

$$\sigma_r = A - \frac{B}{r^2} + \frac{K}{2\log_e a}\left(1 - 2\log_e r\right) \rightarrow \quad \sigma_r = K\left[\frac{a^2 - \left(\frac{r_e}{r}\right)^2}{a^2 - 1} - \frac{\log_e \frac{r}{r_i}}{\log_e a}\right]$$

$$\sigma_t = -\frac{2K}{\log_e a}\log_e r + 2A - \sigma_r \quad \rightarrow \quad \sigma_t = K\left[\frac{a^2 + \left(\frac{r_e}{r}\right)^2}{a^2 - 1} - \frac{1 + \log_e \frac{r}{r_i}}{\log_e a}\right]$$

The expression of σ_a, instead, can be obtained recalling a previous developed expression, and rewriting it by substituting the second term

$$\frac{d\sigma_a}{dr} = \mu\frac{d(\sigma_t + \sigma_r)}{dr} - E\alpha\frac{dT}{dr} = -\frac{1}{1-\mu}E\alpha\frac{dT}{dr} = -\frac{2K}{2\log_e a}$$

Integrating and calling C the constant, the formulation for the longitudinal stress is achieved

$$\sigma_a = -\frac{2K}{\log_e a}\log_e r + KC$$

On the other hand, the resultant of σ_a must be zero given the absence of external forces. Therefore, the value of C can be found.

$$2\pi\int_{r_i}^{r_e}\sigma_a r dr = 2\pi K\int_{r_i}^{r_e}\left(Cr - \frac{2r\log_e r}{\log_e a}\right)dr = 0$$

$$\int_{r_i}^{r_e}\left(Cr - \frac{2r\log_e r}{\log_e a}\right)dr = \left|\frac{Cr^2}{2} - \frac{r^2\log_e r - \frac{1}{2}r^2}{\log_e a}\right|_{r_i}^{r_e}$$

$$\frac{C}{2}\left(r_e^2 - r_i^2\right) = \frac{r_e^2\log_e r_e^2 - r_i^2\log_e r_i^2 - \frac{r_e^2}{2} + \frac{r_i^2}{2}}{\log_e a} \quad \rightarrow \quad C = \frac{2a^2}{a^2 - 1} + \frac{2\log_e r_i - 1}{\log_e a}$$

Inserting the value of C, the new formulation for σ_a is

$$\sigma_a = -\frac{2K}{\log_e a}\log_e r + KC \quad \rightarrow \quad \sigma_a = K\left[\frac{2a^2}{a^2 - 1} - \frac{1 + 2\log_e \frac{r}{r_i}}{\log_e a}\right]$$

By recalling the meaning of K, the final expressions for σ_t, σ_a and σ_r can be written

$$\sigma_t = \frac{E\alpha\Delta T}{2(1-\mu)}\left[\frac{a^2 + \left(\frac{r_e}{r}\right)^2}{a^2 - 1} - \frac{1 + \log_e \frac{r}{r_i}}{\log_e a}\right]$$

$$\sigma_a = \frac{E\alpha\Delta T}{2(1-\mu)}\left[\frac{2a^2}{a^2 - 1} - \frac{1 + 2\log_e \frac{r}{r_i}}{\log_e a}\right]$$

$$\sigma_r = \frac{E\alpha\Delta T}{2(1-\mu)}\left[\frac{a^2-\left(\frac{r_e}{r}\right)^2}{a^2-1} - \frac{\log_e \frac{r}{r_i}}{\log_e a}\right]$$

Specifically, in correspondence to the internal fiber $(r = r_i)$

$$\sigma_{t,i} = \sigma_{a,i} = \frac{E\alpha\Delta T}{2(1-\mu)}\left(\frac{2a^2}{a^2-1} - \frac{1}{\log_e a}\right)$$

$$\sigma_{r,i} = 0$$

In correspondence to the external fiber $(r = r_e)$

$$\sigma_{t,e} = \sigma_{a,e} = \frac{E\alpha\Delta T}{2(1-\mu)}\left(\frac{2}{a^2-1} - \frac{1}{\log_e a}\right)$$

$$\sigma_{r,\mathrm{e}} = 0$$

Figure 58 shows the three stresses assuming that ΔT is positive. Based on the following formula, this means that $T_e > T_i$, and that the thermal flux is centripetal.

$$\Delta T = T_e - T_i = \frac{q}{2\pi\lambda}\log_e \frac{r_e}{r_i}$$

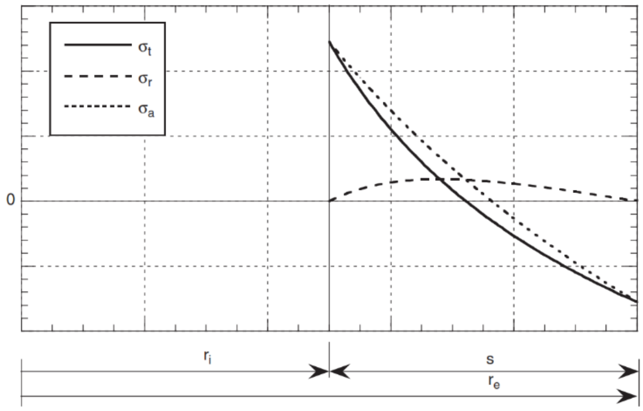


Figure 58: Three stresses σ_t, σ_a and σ_r for centripetal flux. [17]

Of course, the sign of the stresses changes when the thermal flux is centrifugal. It can be noticed that the maximum positive values of σ_t and σ_a occur in correspondence to the internal fiber if the flux is centripetal, and in correspondence to the external fiber if the flux is centrifugal. The simultaneous presence of internal pressure and thermal flux produces the maximum values for σ_t in correspondence to the internal fiber in the case of centripetal flux, and of the external fiber in the case of centrifugal flux, respectively. From this point of view, the centripetal thermal flux is more dangerous, because for the same ΔT the $\sigma_{t,i}$ caused by both pressure and centripetal flux is greater than the $\sigma_{t,e}$ caused by both the pressure and the centrifugal flux. In fact, recalling the result developed in the previous chapter, in the case of centripetal flux the total σ_{ti} can be written as

$$\sigma_{ti} = p\frac{a^2+1}{a^2-1} + \frac{E\alpha\Delta T}{2(1-\mu)}\left(\frac{2a^2}{a^2-1} - \frac{1}{\log_e a}\right)$$

whereas in the case of centrifugal flux, with negative ΔT, the total flux is given by

$$\sigma_{te} = p\frac{2}{a^2-1} + \frac{E\alpha\Delta T}{2(1-\mu)}\left(\frac{2}{a^2-1} - \frac{1}{\log_e a}\right)$$

Calling γ the highlighted part, with the positive sign for the centripetal flux and the negative for the centrifugal sign, last expressions can be written as follows:

$$\gamma = \pm \frac{E\alpha\Delta T}{2(1-\mu)}$$

$$\frac{\sigma_{ti}}{p} = \frac{a^2+1}{a^2-1} + \gamma\left(\frac{2a^2}{a^2-1} - \frac{1}{\log_e a}\right)$$

$$\frac{\sigma_{te}}{p} = \frac{2}{a^2-1} + \gamma\left(\frac{2}{a^2-1} - \frac{1}{\log_e a}\right)$$

The parameters σ_{ti}/p and σ_{te}/p derived from the equations above are graphically depicted in Figure 59 and Figure 60. The values for σ_{ti} and σ_{te} for the two types of flux show the greatest differences for large values of a, as well as for γ. Some portions of the curves have been dashed in Figure 60, since the stress σ_{te} does not show the highest absolute value under certain conditions. In fact, that portions are not significant for the identification of the most dangerous stress. Indeed, it may be the case that the longitudinal stress with negative sign is greater in absolute value than σ_{te} in correspondence to the internal fiber.

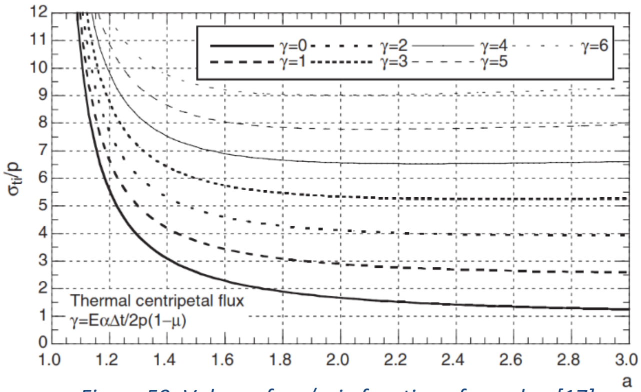


Figure 59: Values of σ_{ti}/p in function of a and γ. [17]

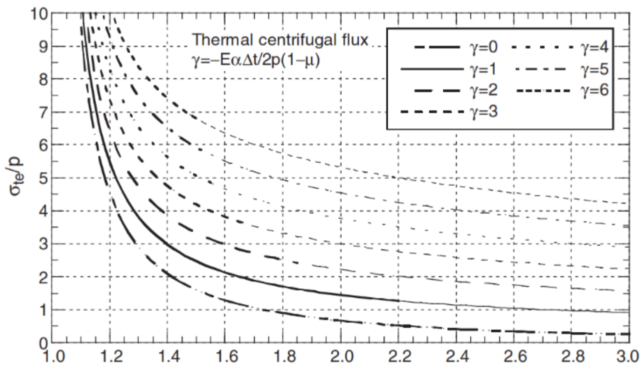


Figure 60: Values of σ_{te}/p in function of a and γ. [17]

It would be possible thoroughly analyze when the conditions required for this occur, however, this will not do in this document. As it is customary, steady-state conditions have been assumed, while the analysis of the phenomenon can be generalized for thermal transient conditions. These considerations would lead to observe the importance of having a gradual heating in order to avoid the onset of high stresses during transient.

A peculiar characteristic of stresses due to the thermal flux consists of having a zero-average value, since they are not generated to balance external forces, but only to re-establish the congruence of deformations. Based on the computation criterion that takes only the average values of the stresses into account and neglects peaks, the stresses generated by flux should not be taken into consideration. Such criterion is often, and rightfully, adopted when the values of ΔT are modest, and the cylinders are thin. Remember, however, that the presence of flux sometimes leads to an increase in peaks that cannot be ignored. In conclusion, it is worth to remember that such peaks, as well as the ones caused by pressure, are crucial whenever the working conditions of the vessel are such to require a fatigue analysis.

7. MECHANICAL DESIGN OF THE HEAT EXCHANGER: CASE STUDY

7.1. INTRODUCTION TO MECHANICAL DESIGN

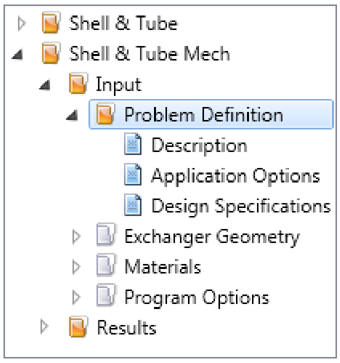
After the section of the theoretical concepts about mechanical design, it is now possible proceed with the second part of the study case. In order to develop the mechanical design of the steam generator, the program ASPEN Shell and Tubes Mechanical will be used. All the formulation and theories used by the program are taken from the designed calculation code, which in this case is the ASME code. However, the design procedure and computations recommended in the ASME codes are developed from the mechanical theories, some of which reported in the previous chapter. For this reason, in order to better understand the logical approach of the mechanical design, the theory has been reported before the study case. It is worth to observe that most of the mechanical equations to be solved during the design are simpler than the thermal one, which, instead, are usually of differential nature. This lead to require a more complex mathematical methods in order to solve the thermal equations, that in turn reflects in a higher cost of the licences for the thermal design programs.

Being the ASPEN Shell and Tubes Mechanical another section of the same program used for the thermal design, the procedure required to upload the configuration obtained with the previous passages is very simply. Indeed, selecting the correct section using the bar on top of the screen and following the instructions, the design developed with the thermal part and all the preliminary data of the configuration are uploaded. As for the thermal part, the mechanical one is divided in inputs and results, and new sections of the menu are created on the left of the program. As usual, the menu shows all the sections and the forms of the mechanical design, and it will be analysed in this chapter.

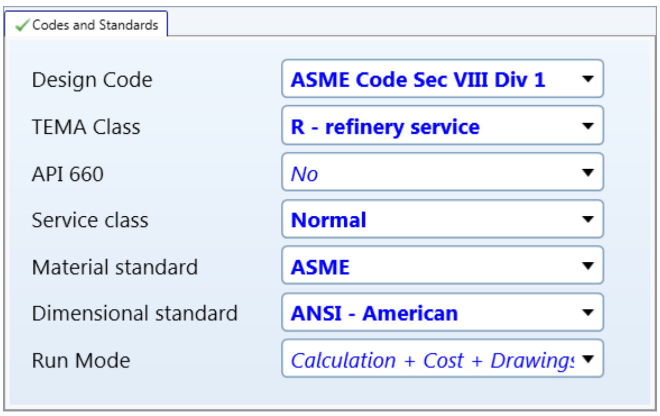
Finally, some new inputs and specifications, which were not previously required, are now used for the mechanical part. However, many clarifications and analysis will still be made in order to identify the necessary and appropriate parameters not provided by the customer. Concluding, it is worth remembering that is not required the vacuum working condition, so no vacuum design parameters will be indicated.

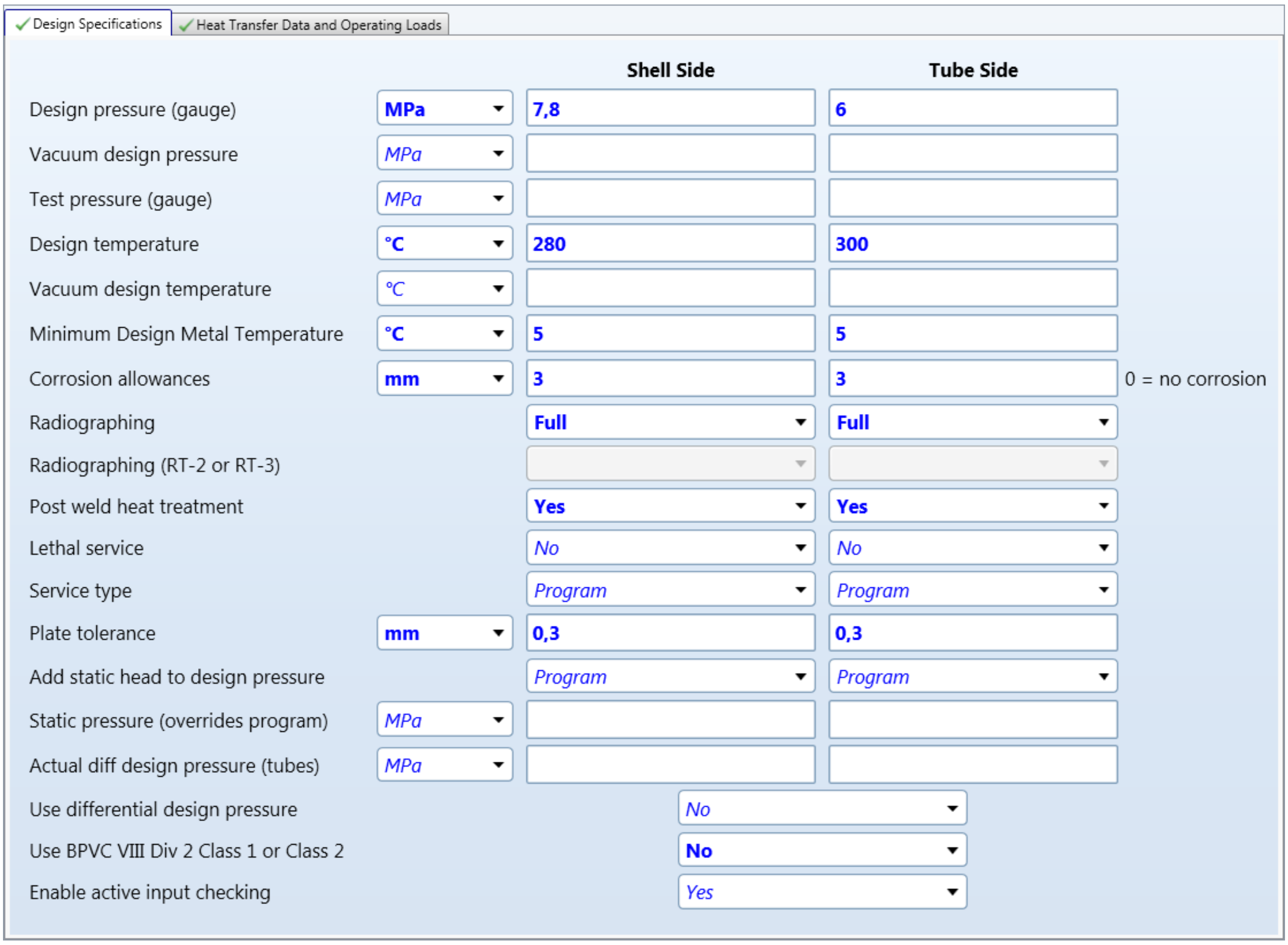
7.2. ANALYSIS AND IMPLEMENTATION OF THE INPUT DATA

7.2.1. PROBLEM DEFINITION



The first section of the input is the *"Problem Definition"*, which present three forms. The first, *"Description"*, is a module dedicated for the description of the project, as already seen in the thermal design. The second, *"Application Options"*, allow to insert the codes and the standards to be used. ASME Code Section VIII Division 1 must be inserted, and the TEMA class have to be defined. The TEMA standards identifies the heat exchanger in three classes in function of their use. The three sections are R, which is the one needed for the study case with intended use in the refinery service, C and B for the chemical and general service respectively. Notice that the requirements are always less restrictive from R to B, and the refinery service is the one in which a more complex approach is required. However the API 660, standards for the heat exchanger of the American Petroleum Institute, has not to be taken into account, given that was not in the client requirements. Regarding the service class is a parameter to declare the working conditions of the heat exchanger, and for the ASME the choice must be done between Normal, Lethal service or Low temperature. The inputs generally needed to design a heat exchanger's component are the temperature, the pressure, and the joint efficiency. The last one is a factor that can assume three different values in function of the designated mode of defect controls, and its aim is to increase the thickness of the piece with the increment of the probability to have defects. Indeed, its value is 1 in case of full radiography control, 0,85 if spot control are indicated or 0,7 if no control are done. Moreover, the ASME forces to make the full radiography, in the case in which the thickness overcome a certain value or in case of lethal service. Indeed, when the heat exchanger contains a fluid that has been classified as lethal by ASME, the lethal service has to be declared so that the program cannot accept an input of joint efficiency different than one. Another consequence that this option will dictate involves the Post Weld Heat Treatment, PWHT. Being the welds a thermal procedure that implies a modification of the crystal lattice, they leave a residual stress in the material. This residual stress can be removed by means of a stress relieving thermal treatment. The entity of these tensions depend by the material, hardest the material higher the residual stress, and from the thickness, that causes an increment of them. Some studies performed during the years have delineated the limits within the stress relief thermal treatment is mandatory or not. As introduced, these limits depends from the material and the thickness, but the treatment is always mandatory in case of lethal service. The low temperature class implies some complex obligations of materials to be used and heat treatments if the Minimum Design Metal Temperature, MDMT, is lower than -29°C. At the end, it can be resumed that the service class is a very important parameter because it influences the joint efficiency, so the welding inspections and the thickness of the material, and establishes the obligation of the thermal treatment. At the end of the sub-form the ASME and the ANSI-American standards have been implemented for the materials and dimensional standard respectively.





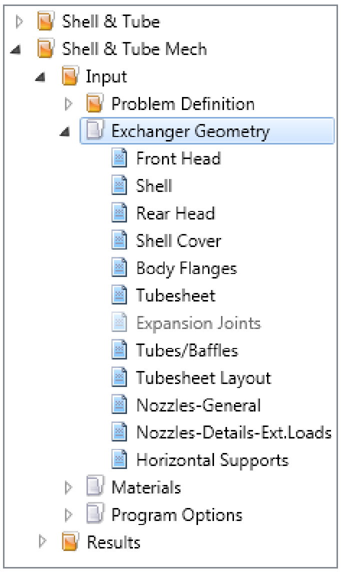
Design Specifications	Heat Transfer Data and Operating Loads		
		Shell Side	Tube Side
Design pressure (gauge)	MPa	7,8	6
Vacuum design pressure	MPa		
Test pressure (gauge)	MPa		
Design temperature	°C	280	300
Vacuum design temperature	°C		
Minimum Design Metal Temperature	°C	5	5
Corrosion allowances	mm	3	3 (0 = no corrosion)
Radiographing		Full	Full
Radiographing (RT-2 or RT-3)			
Post weld heat treatment		Yes	Yes
Lethal service		No	No
Service type		Program	Program
Plate tolerance	mm	0,3	0,3
Add static head to design pressure		Program	Program
Static pressure (overrides program)	MPa		
Actual diff design pressure (tubes)	MPa		
Use differential design pressure		No	
Use BPVC VIII Div 2 Class 1 or Class 2		No	
Enable active input checking		Yes	

The third form, "*Design Specifications*", starts with the project inputs design pressures and design temperatures, already provided in the thermal part. The vacuum design pressure must be inserted if the device has to perform under vacuum operation. In the input data there was indicated that it is not required the vacuum working condition, so no vacuum design parameters have to be inserted. Note that they are not mandatory fields to be filled, otherwise they would be highlighted in light blue and the red cross would appear on the icon of the form. The test pressure is the one at which the heat exchanger will be subjected during the final test. It does not correspond to the design pressure to compensate for the fact that the test will be made at room temperature. In order to take into account the higher temperature during the working condition, a higher test pressure will be determined based on the ASME code. The minimum test pressure of the water is done by $1{,}3 \cdot MAWP \cdot LSR$ where the MAWP is the Maximum Allowable Working Pressure of the shellside, or of the tubeside, and the LSR is the Lowest Stress Ratio. This last parameter is a conversion factor used to convert the ASME material stress value at the test temperature to the stress value at the design temperature. It is obtained dividing the σ_{amm}, the admissible stress at room temperature, by the $\sigma_{amm,T}$, that is the one at the design temperature. Attention must be paid in case in which the test pressure that will be really used is, for technological or process reasons, higher than the one computed. In that case the expected pressure must be inserted in order to make de design properly, or a further verification calculation must be performed. As already seen, all the values not inserted and not strictly necessary for the design will be computed by the program and provided in the mechanical design output. Because in the inlet datasheet the Minimum Design Metal Temperature, MDMT, of 5°C and a corrosion allowance of 3 millimeters have been provided, these values have been inserted. Contrariwise, if the MDMT would not be inserted, the program would provide it basing the computation on the material characteristics and thickness. Being this project in a Normal service class, any type of radiography can be chosen at the begin. However, the high pressure will probably lead to high metal thickness that, as previously discussed, will require the full radiography. The same logic was applied for the Post Welding Heat Treatment, PWHT. Indeed, the full radiography and the PWHT requirements had already been implemented. The plate tolerance represent the tolerance that will be used by the program to compute the minimum thickness of the plates for the design. Indeed, the supplier always provides the pieces that present the required tolerances or the general tolerance requirements dictated by the standard. In this case, the lower limit of 0,3 millimetres has been provided. The static head of the water contained in the shell, or eventually some addition due to an external device, can be implemented by the voices “add static head to design pressure” and “static pressure – override program”.

Proceeding, the actual difference design pressure is an important concept that allow to design the device for a difference in pressure between the shell and the tubeside, and provide a load procedure in which that pressure will not be overcome. Generally the components subjected to both the pressures, e.g. tubesheet and tubes, are designed twice using the shell and the tubeside one, and after the resulting greater thickness is assumed. However, they never have to withstand all the pressure for one side and nothing from the other. For this reason, in case in which expensive materials have to be used, it can be thought to design this pieces to withstand only the difference in pressure, and make this difference the maximum load that will be provided. In order to provide a better explanation of this design logic, an example of a pressure vessel that have to be designed with a shellside pressure of 7,8 MPa and a tubeside pressure of 6 MPa can be reported. Given that the heat exchanger in nominal conditions of working undergoes both the pressure together, the design can be made taking into account the only difference between them, increased for the safety, e.g. 2 MPa. This can be done because the compression stress given by both the pressure together is negligible with respect the stresses caused by the difference of them. This leads to mandatorily provide a load procedure which does not allow to occur more than 2 MPa of difference in pressure in every moment of the heat exchanger’s life.

Concluding, the last option to be chosen regards the difference between the Class 1 and Class 2 of the BPVC VIII division 2, which is represented by the admissible stress used for the design activity. Indeed, the Class 1 foreseen an admissible stress lower than Class 2, due to the fact that it has been computed with a higher safety factor. However, the Class 2 can be used only if the design will be approved from an American or Canadian engineer. Even if the design will be based on the BPVC VIII division 1 as before indicated, this choice is presented, and the program have automatically set no as parameter.

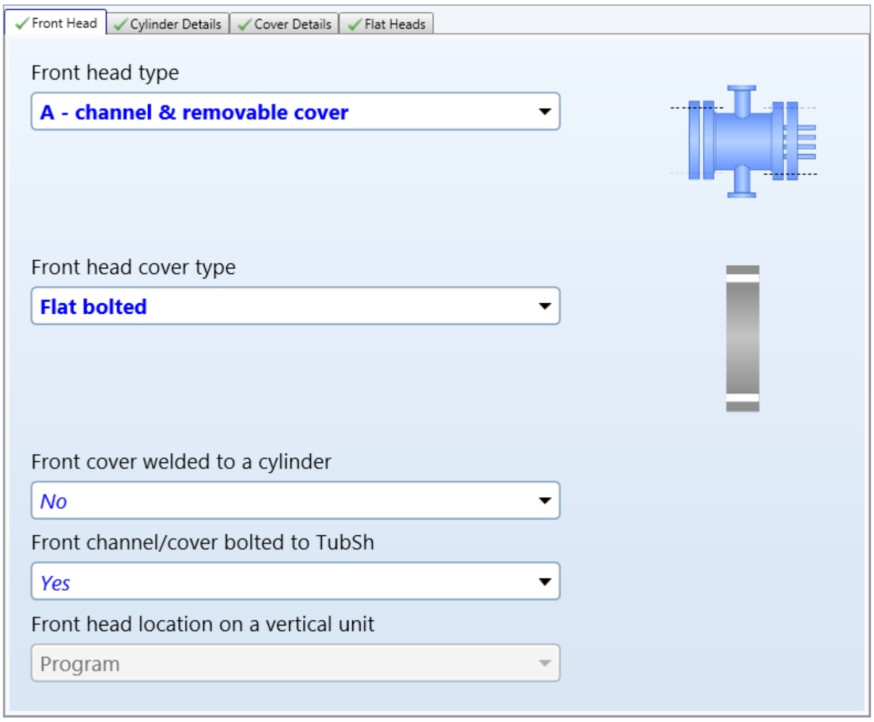
7.2.2. EXCHANGER GEOMETRY



The "Exchanger Geometry" is a very important set of forms which allows to modify the heat exchanger geometry. All the forms present here have been automatically filled with the results of the thermal design when the selected case has been uploaded. Special care must be taken if a modification of something impacting the thermal design have to be done. In such case, the thermal design have to be verified again and an iterative procedure starts. However, not all the variations would lead to strongly modify the heat exchanged, so, the necessity to make again the thermal design must be evaluated case by case. For example, the front head type can be changed without producing a variation in the thermal design, while the size of the nozzle or the passes of the tubes cannot be modified without update the results.

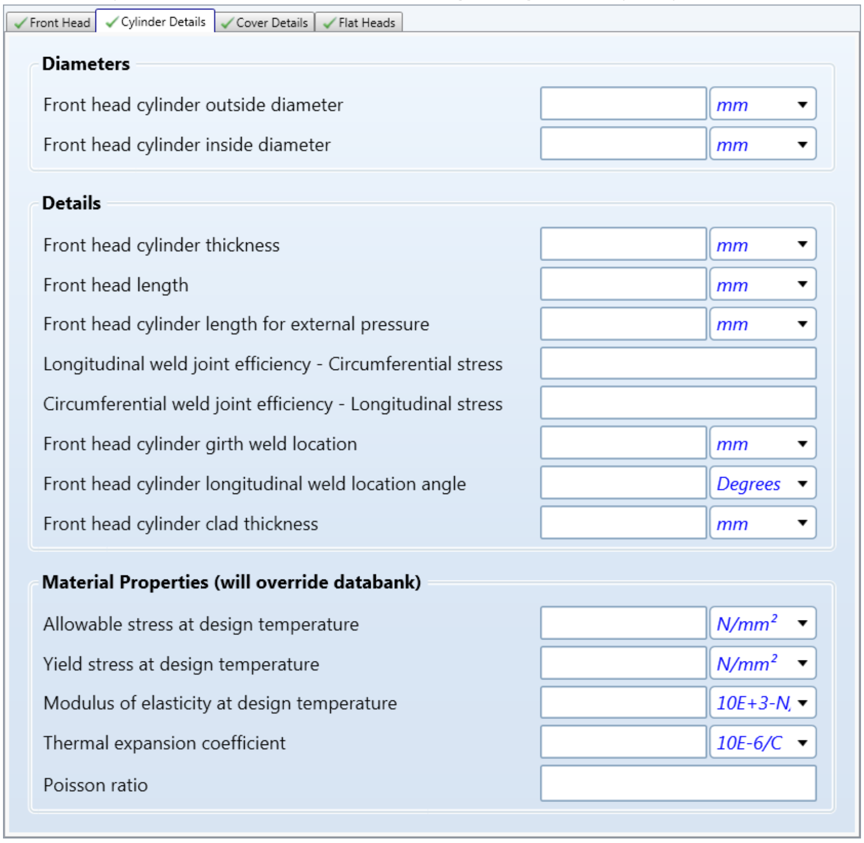
FRONT HEAD

The choice of front head type was already been commented inside "*Shell/Head/Flanges/Tubesheet*", in the "*Exchanger Geometry*" section of the thermal design. Indeed, the great advantage of the A type is the free access to the tubes without detach the whole channel, e.g. without detach the inlet and outlet pipes. For this application in which frequent cleaning operation will be done, it represents the correct choice, in spite of the higher cost with respect to a B channel. Since an A configuration has been chosen, the flat bolted front head cover type is mandatory, and it cannot be welded to a cylinder. Always for the same reason, it is correct to implement that the front channel is bolted to the tubesheet, since it is a characteristic of the A type. Generally these three inputs are automatically filled in by the program in function of the selected front head type. The last input, instead, is activated only in the case of the vertical devices. It allows to insert the location of the front head, which is always the one where the tubeside input is placed. In that case, the correct position have to be introduced because the possible presence of the hydrostatic head must be taken into account in the mechanical design. In this sub-form some generalities of the Front Head was presented again, even if most of them was already been discussed during the thermal design. In the following sub-forms the detailed inputs that can be inserted for the cylinder and the cover are introduced.



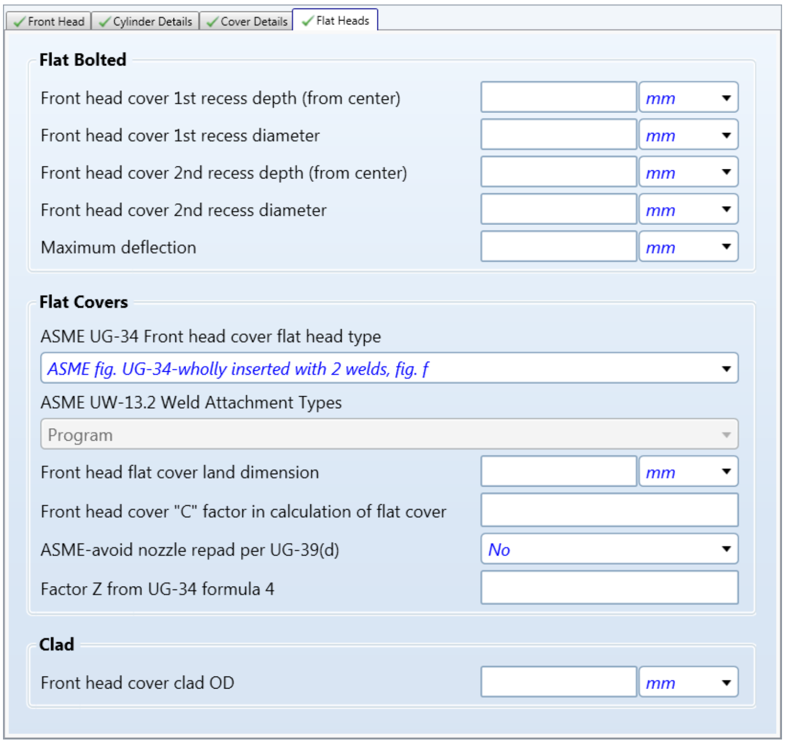
The next sub-form, called “*Cylinder Details*”, refers to the cylinder of the channel, and not to the kettle’s one. If both, inside and outside front head cylinder diameters, are not inserted, the program set the front head inside diameter equal to the inner shellside one, and compute the minimum required thickness. Notice that, as already said, the inner shellside is different from the kettle one. It follows that all the parameters contained in the details section can be implemented if some restriction are presents. However, if no special requirements have to be set, it is always better do not add constraints to the program in order to reduce the crash prospective. Despite this, other possible inputs are the length of the cylinder and the length for external pressure. The first is the simple length of the head cylinder, and it will be automatically computed in order to allow the assembly of all the component. However, a final check about the feasibility is a good habit. The second one is a value that refers to the devices that work under vacuum conditions, and so they must withstand the external pressure not balanced. As can be learned from scientific literature, the resulting thickness of shells under external pressure depends from its non-supported length. Differently, the length of the shell does not influence the thickness in case of design against internal pressure. The possibility to insert some stiffening rings allow to reduce the non-supported length, and so to decrease the required thickness against the external pressure. Indeed, if the thickness required for the vacuum working condition is higher than the normal one, the values can be matched adding some stiffening rings. This have not too much sense on the channel as much on the shell, but it could be necessary. Proceeding the joint efficiency for circumferential and longitudinal welds can be inserted. However, because the full radiography indication has already been inserted, the program will complete automatically these fields with the relatives correct values of one. The next parameters are useful only for the production of the technical drawing and for the cost analysis. They express the distance and the angle at which the girth weld and the longitudinal weld are located. However these values will be present only in the case in which the cylinder head is very long and have to be done from more pieces welded together or from a rolled plates.

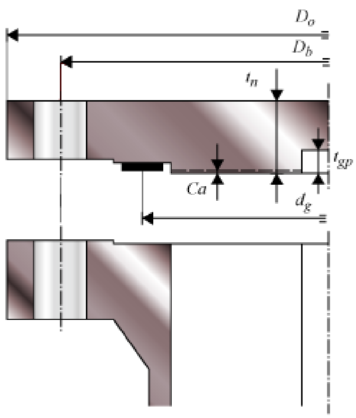
Sometime, due to the necessity to avoid corrosion, expensive materials like Inconel are required for the shellside. In order to avoid excessive cost, only a few millimetres of coating are generally deposited. These clad cannot be considered as thickness for the mechanical resistance of the shell. However, in order to prevent difficulties during the bundle insertion, it must be considered as encumbrance for the inside shell diameter. Typically, three millimetres of pure coating guaranteed by a chemical analysis must be applied, unless in cases in which they represent a limitation for the thermal exchange. Proceeding, the last part of inputs contains the data about the material of the cylinder. It is used if the material is not present in the program database and it must be manually implemented. If filled out, the values specified in the paragraph 7.2.3 will be overridden.

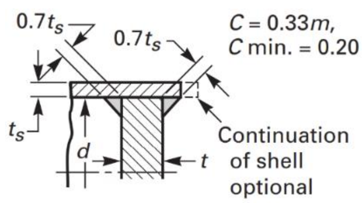


As already presented, the cover in this case must be a flat bolted, so the “*Cover Details*” sub-form has not to be compiled. Indeed, it refers only to other covers, e.g. torispherical and hemispherical, which can be used in a different type of heat exchangers. A very common one is the torispherical, in which a semi-ellipse is simulated by means of two curvature radii. The first is a large curvature radius present in the center of the cover, while the second is a smaller radius on the edge. The proportions between the two radii can change, leading to a family of possible torispherical covers. Returning to the sub-form an important input which is worth to be analysed is the percentage of thickness for forming allowance. The plate formed to obtain the cover undergoes a shrinkage, so the minimum thickness required from the mechanical design must be guarantee at the end of the procedure. The percent value of shrinkage must be inserted in this input. However, generally during the purchase of the formed cover the minimum thickness is specified, and this value can be left free. To report some common values, even if highly conservative, a shrinkage of 10% for the torispherical and a 15% for the hemispherical cover can be considered.

As previously assigned, the study case present a flat bolted head. For this reason the previous sub-form, “*Cover Details*”, has not to be considered, while the following, “*Flat Heads*”, is the important one. Given the TEMA Class R, the confined type of the flanges have to be mandatorily used. These joints present one or two recesses to accommodate a gasket, as in the upper figure on the right. D_0 and D_b represent the outside diameter of the cover and the bolt circle diameter respectively. The nominal thickness of the cover is t_n while C_a is the corrosion allowance. In the image there is also present the groove depth for the partition plate and the mean gasket diameter d_g. The program allow to insert manually the dept and the diameter for the first and second recess as well as the maximum deflection value. Notice that the first recess refers to the dimensions of the cavity which the gasket is located, while the second to the bolt one. The following entry allow to insert the flat head type regarding to the welds. The image below on the right shows the choice of the case study, the *f* type, which presents two welds and an associated factor *C=0,33m*, used to compute the thickness of the flat cover. The last two inputs do not have interest for the study case because they are a command to avoid the empirical reinforcement for the nozzle if it would be placed on the cover, and the factor Z, that it is used to compute the not circular covers.

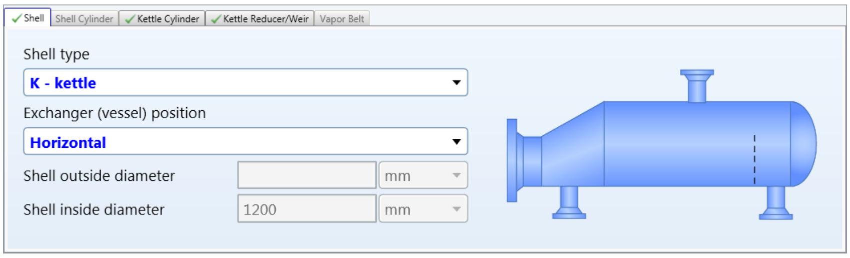




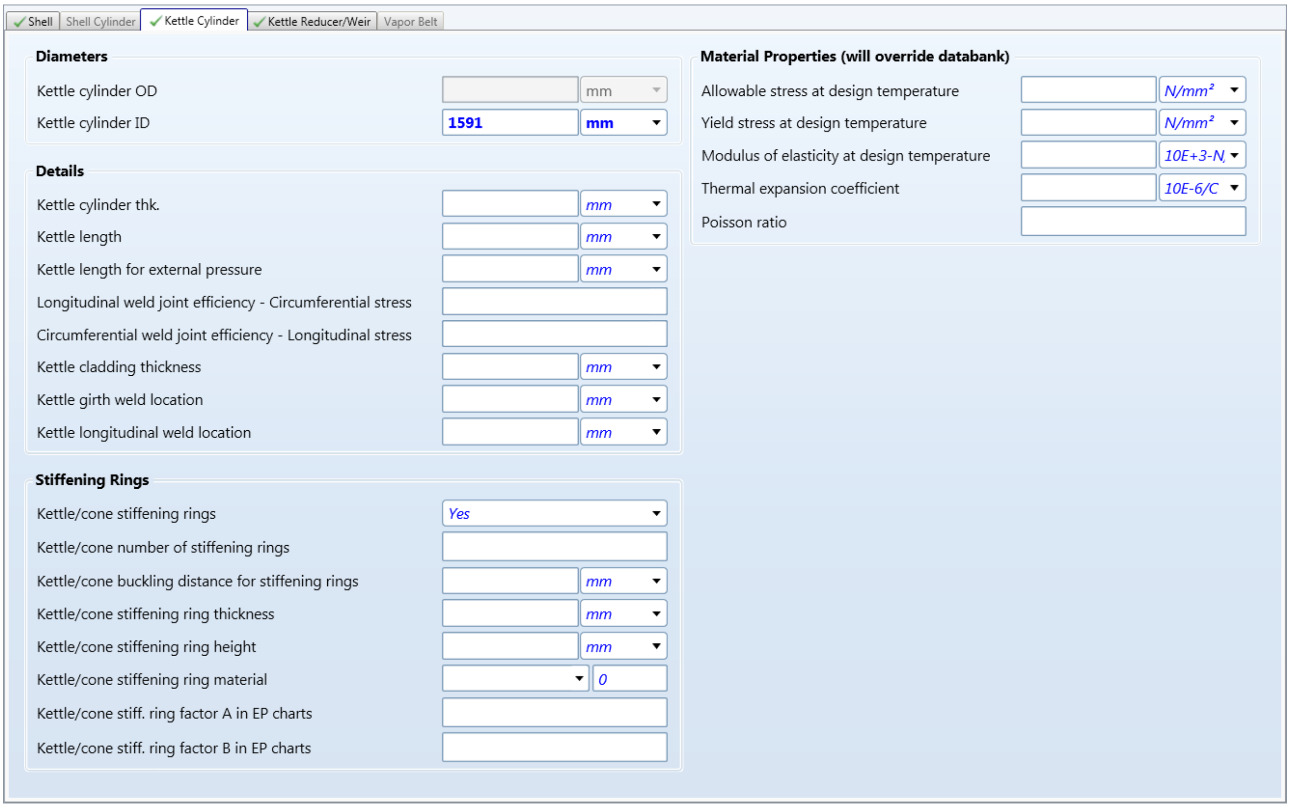


SHELL

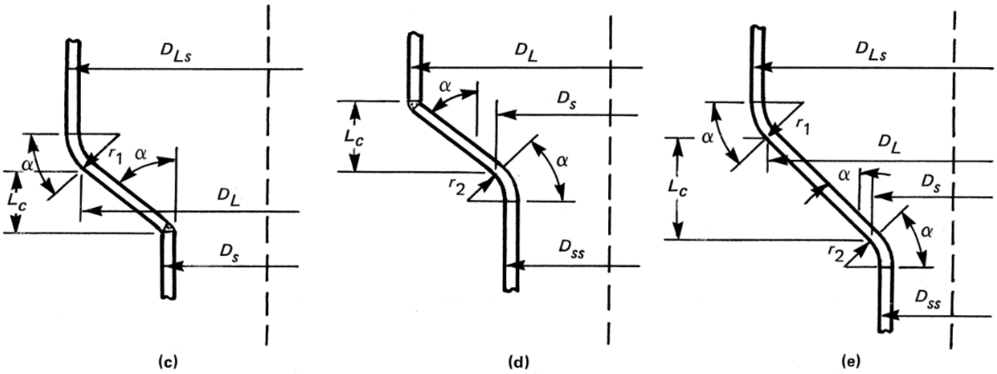
All the data, as the kettle shell type and the horizontal orientation, was already been inserted in the thermal calculation, and the mechanical one have automatically upload them. Please notice that, as already mentioned, the shell inlet and outlet diameters don't refer to the kettle diameter but to the one of the cylinder connected between the flange and the reducer.



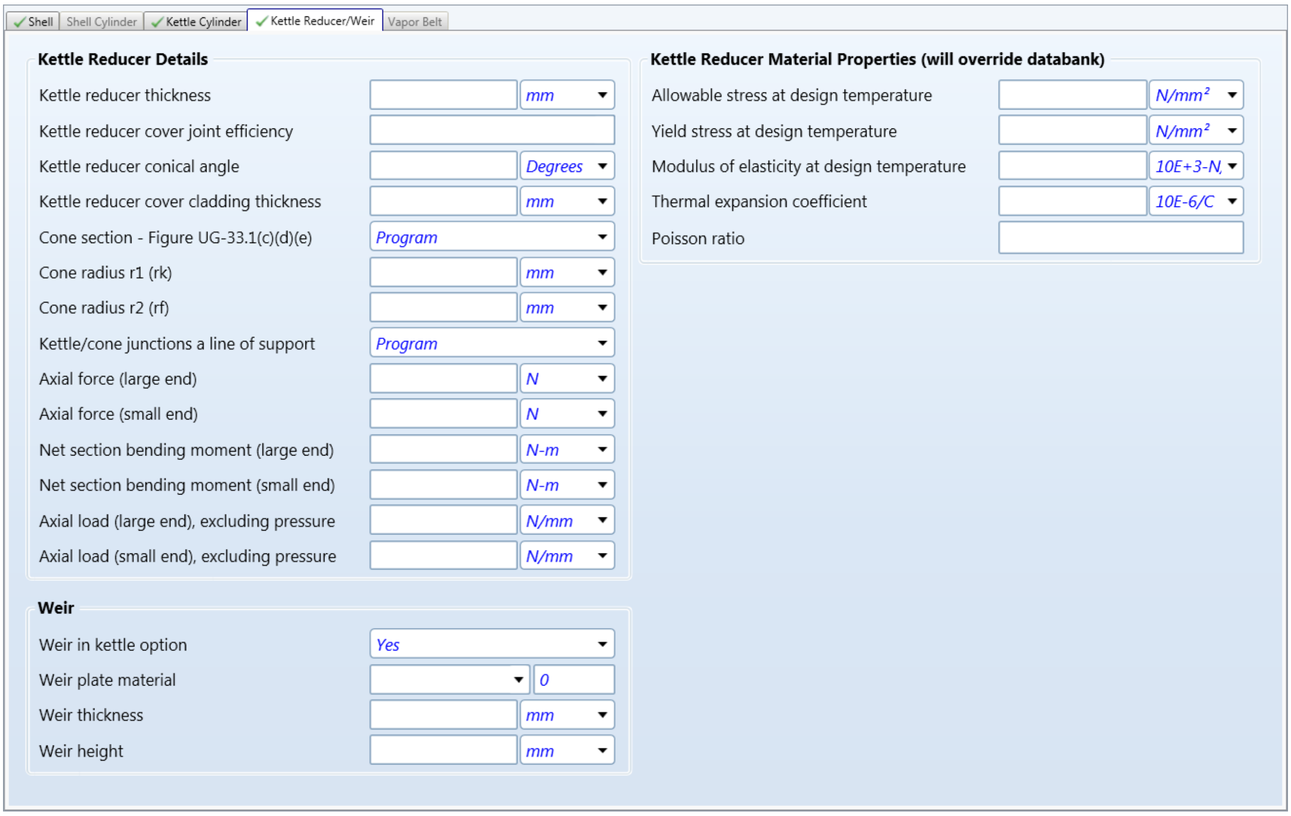
Referring to the image below, it can be noticed that also the kettle inner diameter has been automatically uploaded. Furthermore, the same input parameters already analyzed for the front head cylinder are now reported for the kettle. Attention must be paid since they refers to the kettle cylinder only and not to the cone, e.g. the kettle length. Differs, instead, the section of the stiffening rings, which in this case are not intended to add stiffness at the cylinder for the design against the external pressure. Given that the connections between the cone and the cylinders, for both the shellside and the kettle one, are two areas of high stress concentration and buckling risk, two possibilities can be adopted to avoid problems. The first is to form the cone in order to avoid the welds to be in the changes of slope. Making a knuckle and a flare on the cones, the welding joints will be a butt weld, avoiding problems but introducing a higher cost. The parameters of this solution can be inserted in the next sub-form which refers to the reducer. The second remedy, and less expensive solution, would be an increment in stiffening produced by placing some rings close to the joints. All the data to define them manually can be inserted here, however, as already said, it is important do not insert to much constraints in the first instance.



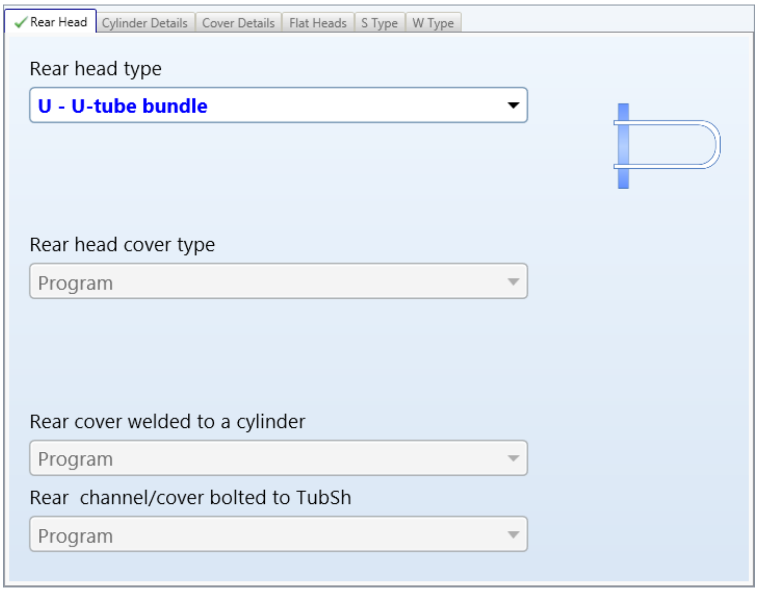
Focusing now on the reducer, the cone between the shellside cylinder and the kettle one, always the same parameters can be implemented. The diameters from both sides are constrained from the cylinders, while the geometrical parameters as the thickness, the angles and the radius have to be left empty. The input that refers to the cone section refers to the already presented discussion to avoid high stress concentration areas and buckling risk. Some possible solutions of formed cone with knuckle and flare are reported in the following image, which represent the figure UG-33.1(c)(d)(e) of the ASME Section VIII Division 1.



The possibility to control the flow by means of the weir method has already been discussed. If it has to be adopted, as in this case, the affirmative statement in the weir part of this sub-form has to be inserted. As all the devices present in the kettle, it will be considered in the weight and cost estimation of the whole heat exchanger.



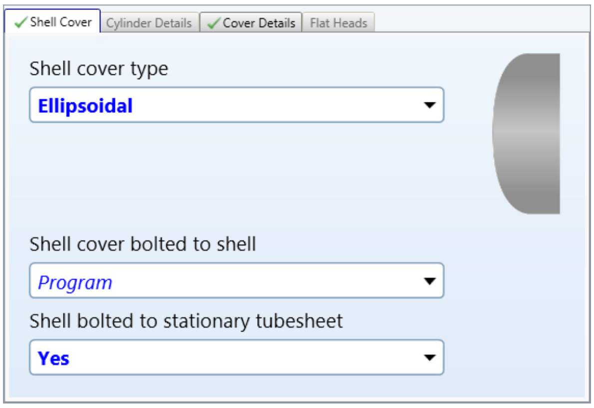
REAR HEAD



Various sub-forms are activated or disabled in function of the selected rear head. As already discussed, the U type is necessary for this application, and all the input parameters are practically turned off. In the other rear head cases, the inputs would be the same already seen for the front head. The only exceptions are the two sub-forms S and W type, dedicated at the floating heads. Notice that the W floating head is rarely used, but the S type can also be adopted with the kettle configuration as analysed in the next paragraph.

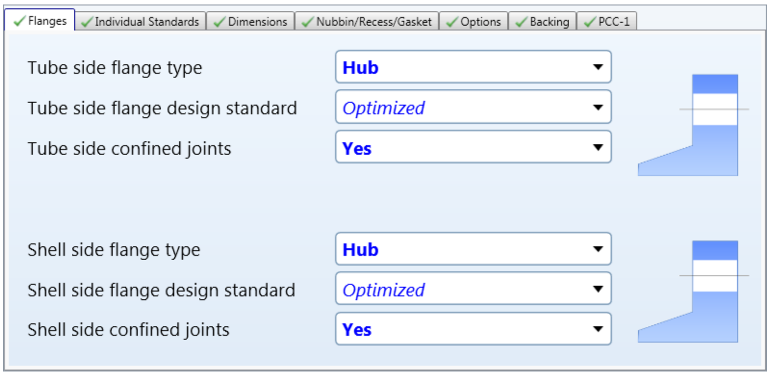
SHELL COVER

Talking about the shell cover, a lot of different situations could be analysed. For example, a very particular case would be the necessary to have a removable bundle in which the mechanical cleaning inside of the tubes must be ensured. In this situation the S floating rear head must be used, but it will be located inside the kettle cylinder. An ellipsoidal cover would be used as in the case of the U bundle, however it should be bolted to the shell rather than welded. This would allow the possibility to remove the shell cover in order to dismount of the inner S head during the bundle extraction. This example has been reported to demonstrate how this input should be carefully analysed case by case. The last choice proposed in this card is very important for the study case, because if not correctly inserted the program can generate an AKU heat exchanger without the removable bundle. Indeed, obviously the tubesheet cannot be welded to the shell if the bundle must be removable. The geometrical details of the shell cover can be inserted in the dedicated "Cover Details" sub-form if needed.

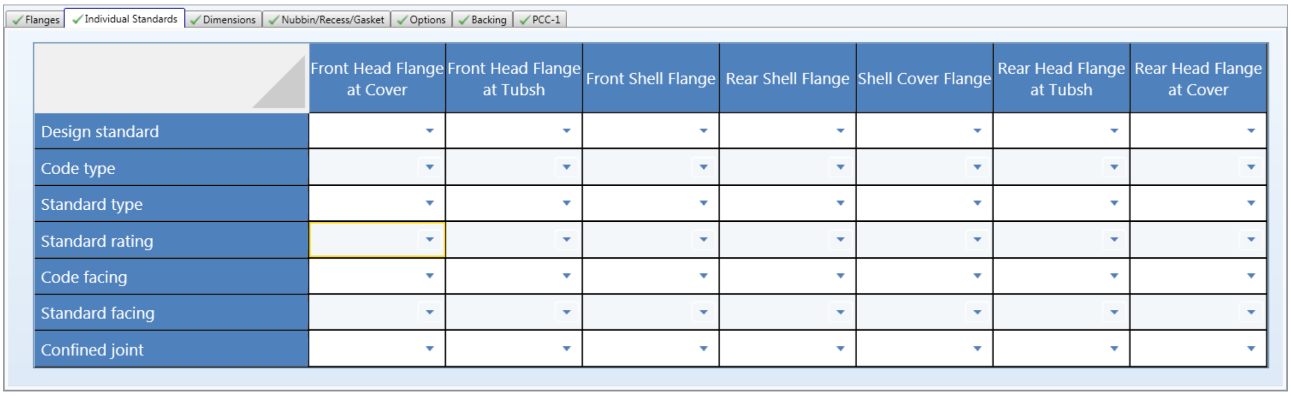


BODY FLANGES

The first thing to be noticed about the body flange section is that all the inputs present in it do not influence the thermal design, even if there they were already specified during the first part. Indeed, the Hub type of flanges was already set given the refinery service TEMA class that forces to use the butt welds. Optimized has been indicated for the design standard because the dimensions have not been determined yet, so the standard cannot be selected. The confined type of flanges also must be used, and they have already been commented. These settings apply to all the flanges of the tube and shellside present on the body, and not on the nozzles' flanges.

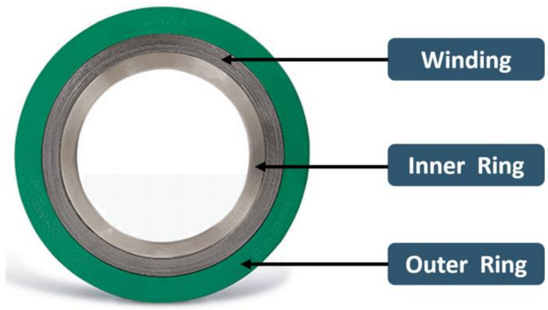


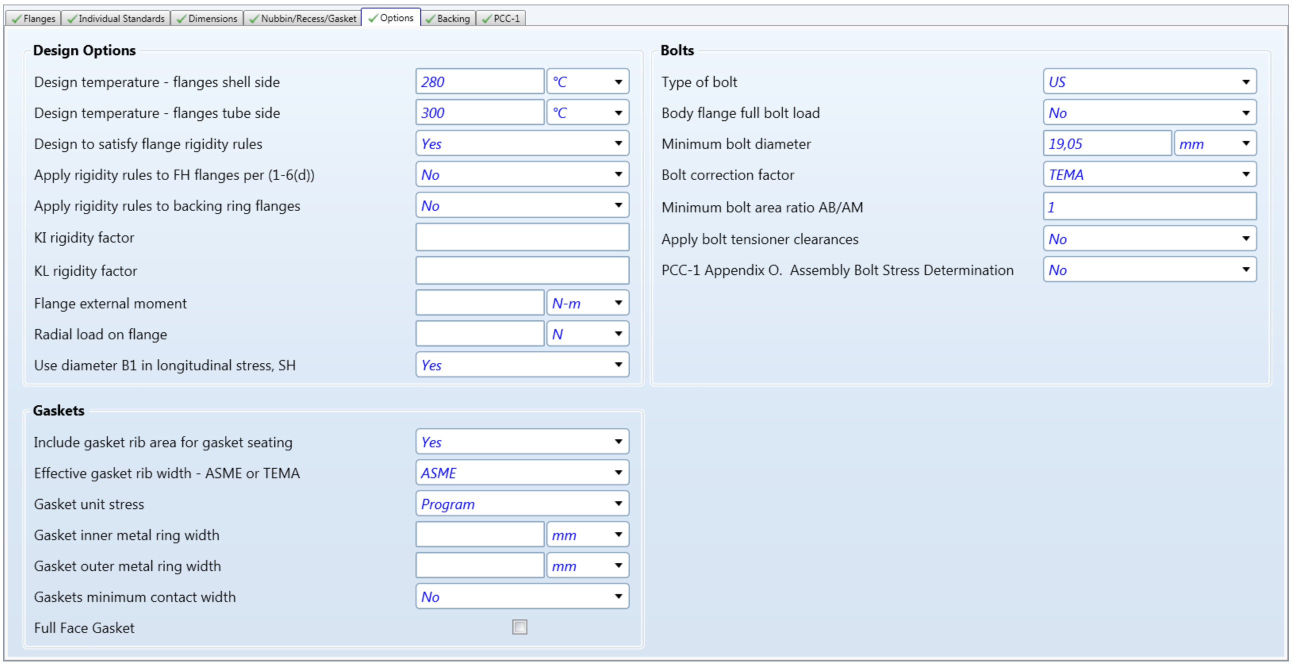
Proceeding, the "*Individual Standard*" sub-form presents a standard table of all the possible body flanges that can be present. In the analysed configuration there will be two flanges on the front head channel, the "Front Head Flange at Cover" and the "Front Head Flange at Tubesheet", and the one on the shell, the "Front Shell Flange". This table can be used to implement the characteristics of the individual flange if different from the general values expressed above. In this case all the flanges have the characteristics already delineated, and so no further specifications are required.



In the following two sub-forms all the geometrical details of the flanges, the characteristics of the material, of the nubbin and gaskets can be inserted by means of two tables. Some important parameters that still were not analysed are the *m* and *y* coefficients of the gaskets. The principal aim of the flanges is to counteract the forces, given by the internal pressure, that tend to open the joint and to provide the correct preload if not self-energizing gasket type are used. To design a flanged joint, two required bolting areas are found from the previously mentioned forces, and the major one is used. Indeed, the selected higher bolting area will be provided from a combination of number and size of bolts, taking into account that they must have enough space to be mounted and that more the bolt circle is enlarged, more the bending moment on the flange increases. The bolting area required to ensure the preloading is called gasket seating, while the force that must be provided can be determined from the coefficients *m* and *y*, given by the manufacturer, and from the gasket width. So, it is worth to observe that the gasket can have a principal role in the sizing of the bolts. As presented, the preload necessary for the correct behaviour of the gasket depends also from its width. However, this dimension cannot be reduced more than a certain limit in order to ensure the resistance of the gasket, that otherwise would break during the assembly. To overcome the problem of the high preload needed in the past, without reducing to much the width, the nubbin was used. It is a worked relief on the flange that, closing on the gasket, allow to reduce the effective area considered for the calculation of the preload, and allow to use wider and handy gasket. As discussed, in first instance it is better do not insert parameters that will produce some constraints at the calculation performed by the program. For this reason, and due the simplicity and the redundancy of the other inputs that would be present, the images of the sub-forms related to the topics just discussed have not been reported.

Analysing instead the "*Options*" sub-form, the first data reported are the design temperatures for the shellside and tubeside flanges, automatically uploaded from the thermal design. Just below these, an option to design the flanges according to the ASME rigidity rules can be activated. This allow to obtain stiffer components when the design loading conditions do not require too much in term of resistance, leading to a thin components. KI and KL are two factors used in the above-mentioned verification of the rigidity, and they refers to the integral and loose flange type respectively. Some standard values, KI=0,3 and KL=0,2, are provided from the ASME code. They will be automatically filled from the program, given the specification to follow the ASME code. However, these values are extremely precautionary, and the ASME code allow at the designer to insert personalized values if properly documented by technical verification. Changing section, a part of this sub-form is dedicated to the inputs required for some particular types of gaskets, e.g. the spiral wound gasket that can be seen in the image on the right. These device are delicate and difficult to handle, so an inner and outer rings provide the necessary stiffness and the centring simplicity. Proceeding, there is an option to activate the full-face gasket, which extends the gasket external diameter until the external diameter of the flange, and present the holes for the bolt passing through. In the last section the type of bolts, US in accordance with ASME standard, and the minimum bolts diameter can be selected. Finally, the clearance for the bolt tensioner can be foreseen. The last sub-forms have not been reported because not connected with the study case. However, to be thorough, the first one is "*Backing*", that allows to insert the input for the backing rings of the S type floating heads, while the second is "*PCC-1*", that refers to a good practice rules based on ASME for the bolt tightening.





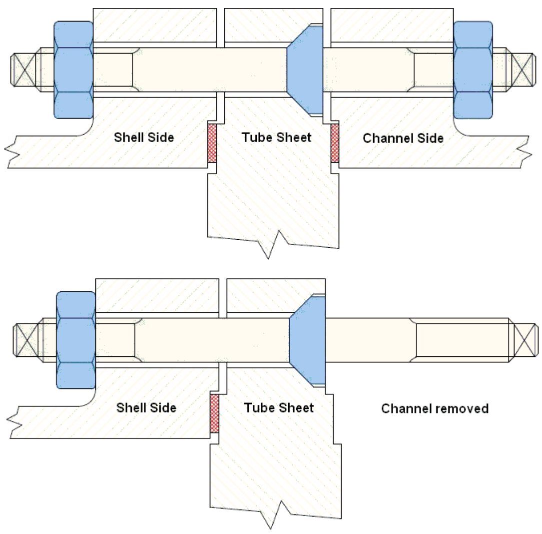
TUBESHEET

Moving on the tubesheet form, six sub-forms are present and now will be analysed. The first one is called "*Tubesheet*", as the general form, and an image of it is reported in the next page. At the begin, the tube to tubesheet joint type is recalled. It was already presented in the thermal design part, "*Exchanger Geometry – Shell/heads/Flanges/Tubesheet*", where the difference between the possible joint types was explained. However, most specific input have to be inserted here, defining completely the connections and guaranteeing a correct mechanical design. As already discussed, a *"Strength welding with expansion reliefs"* is required in this case. This is traduced here with the inputs "*e – Welded,* $a \geq 1{,}4t$*, and expanded*" and "*Full Strength*". They means that the tubes are connected to the tubesheet with a weld that ensure the mechanical resistance, being the coefficient $a \geq 1{,}4t$. However, a mechanical expansion, which acts as support only, is present and it is not enhanced with particular groves. The tubesheet extension type input allow to choose between three different tubesheet configurations, among which only the last two are interesting for the analysed application:

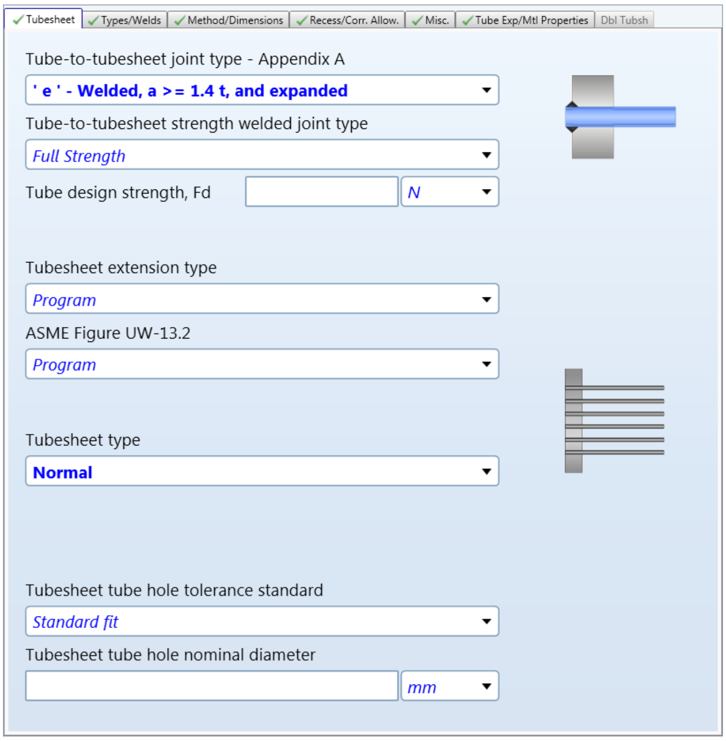
- Welded on both side: this configuration has to be used in the case of N head type only, and it expects a tubesheet welded on both sides, at the channel and at the shell. In the other cases, in which only one side can be welded on the shell, it is convenient to leave "Program" and specify the welding type in the following sub-form. In this way the tubesheet will be mounted on the channel by means of bolted joint, while on the other side, it will be welded to the shell.

- *Pinched* or *Not-bolted through*: the tubesheet is clamped between the flanges and the bolts are external. This allow to design the tubesheet exclusively to withstand the load provided by the pressure, and it is not influenced by the bolting preload, which acts only on the flanges. Generally it is cheaper than the full diameter configuration because it has the same thickness, or it is even thinner, and have a lower diameter. They are mostly used with smaller pressure devices given that generally, in these cases, the tubesheet required thickness would be imposed by the bolt preload. However, a strong disadvantage presented by this configuration is that when the channel have to be dismounted, also the tubesheet is detached, and the shellside have to be depressurized. Moreover, during the assembling, the whole bundle must be kept in position while tightening takes place, and this can represent a difficult operation.

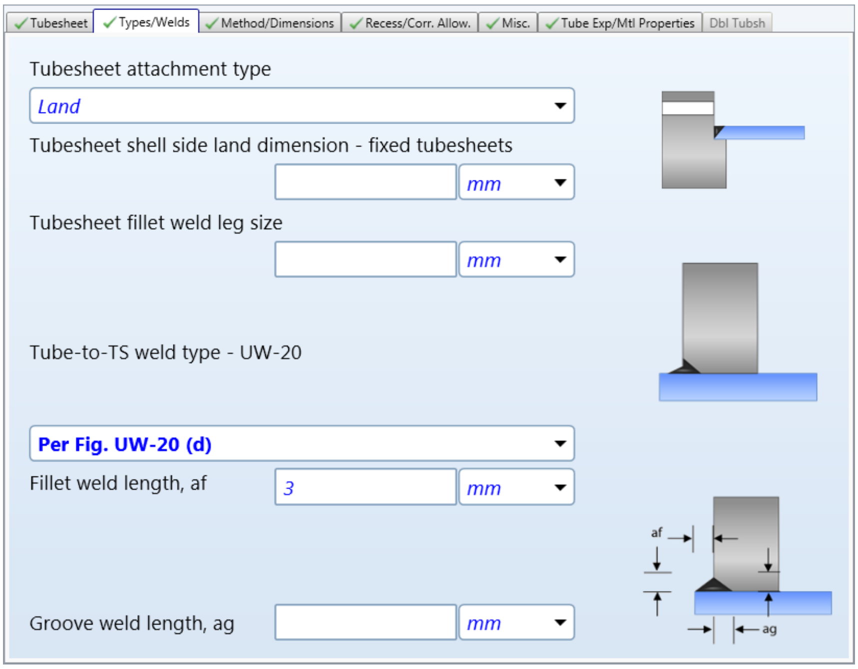
- *Full diameter* or *Bolted through:* the tubesheet is clamped between the flanges and the bolts pass through it. This requires the design of the tubesheet considering the load provided by the pressure and by the bolting preload. Indeed, if the load on the flanged extension, which is the tubesheet part clamped between the flanges and subjected to the bolting preload, requires a higher thickness to be withstood, the whole tubesheet have to be oversized, even if the part subjected to the pressure would require a lower thickness. This happens because the recess must be present to accommodate the gasket, but conceptually it will be made adding material in the center rather than machining it. Generally it is more expensive because it has at least the same thickness of the pinched one, and it has a larger diameter. However it can be used in concomitance with the collar bolts, which allow to dismount the channel without detach the tubesheet from the shell.

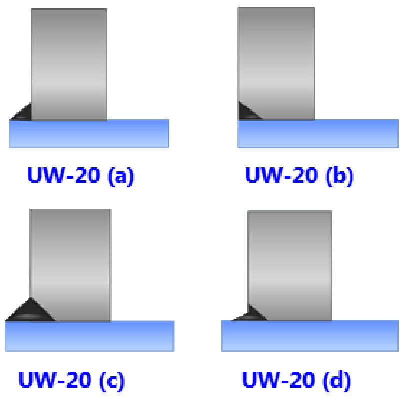


The input concerning to the figure UW-13.2 is not to be managed in this case because it refers only at the cases in which the tubesheet is welded to the shell. Furthermore, the tubesheet is a normal type, and not double, so it must be inserted as choice for the next parameter. The last input in this sub-form is the tolerance on the tubesheet holes, which can be "*Standard fit*" or "*Special close fit*". It slightly influences the thickness of the tubesheet, and the standard fit indicate an under tolerance of -0,10 millimetres, and an upper tolerance of +0,05 millimetres for the 96% of the holes. The remaining 4% can be in the upper tolerance of +0,25 millimetres. The special close fit is more restrictive and requires an under and upper tolerances of 0,05 millimetres. The special solution is sometime required when the pressure is high and with the ASME lethal service class.



In the "*Types/Welds*" sub-form the first three inputs allow to describe the way in which the tubesheet is welded to the shell, thing that has no meaning in this case. The default choice of the program is "Land" written in italic font, which means that it can be automatically changed during the computation. Similarly, the tube-to-tubesheet weld have to be specified in this sub-form. As already introduced in the "*Exchanger Geometry*" form of the thermal design, the tube can be fixed on the tubesheet in different ways. It follows the image of the main methods of welding configuration, from which the most common, the UW-20(d), has been selected.

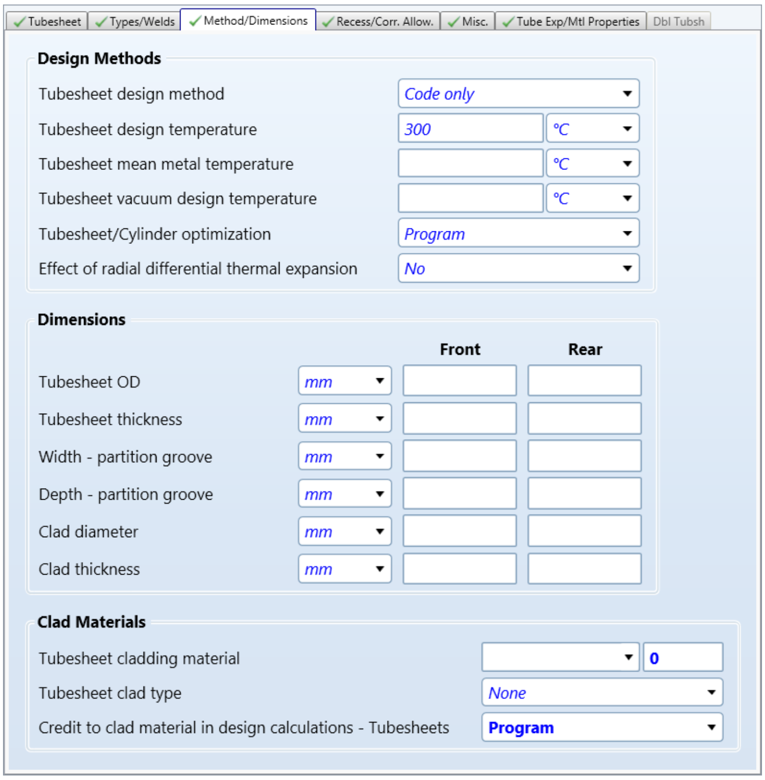




Moving on the next sub-form "*Method/Dimensions*", a section about the design methods can be noticed. Since the tubesheet design can be done following different approaches, a choice must be taken between:

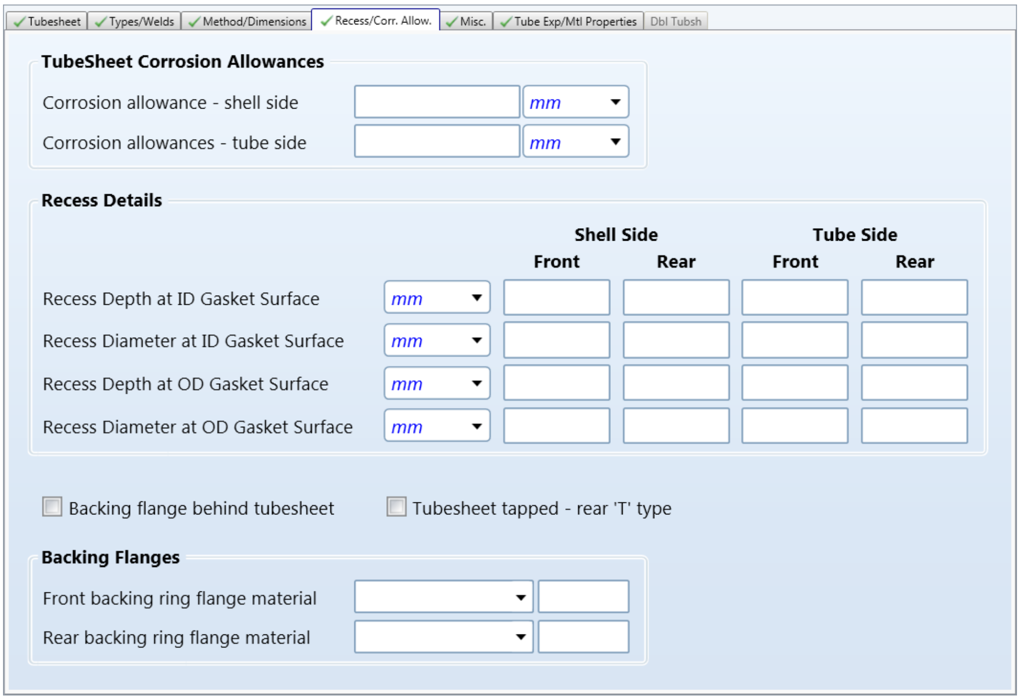
- *Code only*: the tubesheet design has to be done basing on the code rules only, ASME in this case. Please note that until 2005 the ASME did not contain the rules for the tubesheets design, and an external code or an internationally recognized standard, e.g. TEMA, had to be used.
- *TEMA only*: the tubesheet design has to be done basing on the TEMA rules. Since nowadays the ASME have inserted the design procedure for the tubesheet, the design rules contained in the TEMA have been moved in the appendix, and they have to be considered as good practicing.
- *Code/TEMA thicker or thinner*: this choice is used when the customer ask specifically for the TEMA verification. Indeed, even if the requirements are to design following the ASME code in concomitance with the TEMA rules as support, the TEMA no longer require the verification of the tubesheet. So the TEMA check nowadays represent something more than the ASME and TEMA requirements.

For the reason above described, the choice of the tubesheet design method can be "*Code only*". The tubesheet design temperature has been uploaded automatically from the thermal design since it is equal to the one of the tubeside. Instead, the tubesheet mean metal temperature is different from the tubeside and shellside ones, and has not been placed. This happened because it does not enter in the mechanical design calculations, since the U configuration of the tubes allow the thermal dilatation. The normal design of the tubesheet is done taking the higher thickness given by the worst loading condition. The two cases that are conceptually analysed are the ones resulting from the shellside or the tubeside subjected at the design pressure and temperature, and not counterbalanced on the other side. Indeed, the other ambient is considered as subjected at the normal conditions, or the vacuum if foreseen in the design conditions. This lead to consider an overestimated design condition due to the consideration of higher pressure and temperature, e.g. design versus the operative ones, and the neglection of the pressure and temperature effects in the other ambient that would counteract the gradients. The higher thickness given from one of these conditions would be assigned at the tubesheet. However, considering a case in which the thermal dilatation is prevented, the mean metal temperature of the components would have an important role, and the thickness resulting from this computation would be higher. The internal stress given from the thermal load and prevented thermal expansions, as can happen in a N head channel type, can be compensated in different ways. The first is to increase the thickness of the tubesheet, while the last two are to increase the shell or the channel thickness at welded junctions. Depending from the position of the stress concentration, one or a combination of these solutions can be required. This information may be provided in the input "*Tubesheet/Cylinder optimization*", where leaving the choice at the program means optimize the solution in function of the case. The following inputs allow to better specify these increases in thickness. However, the

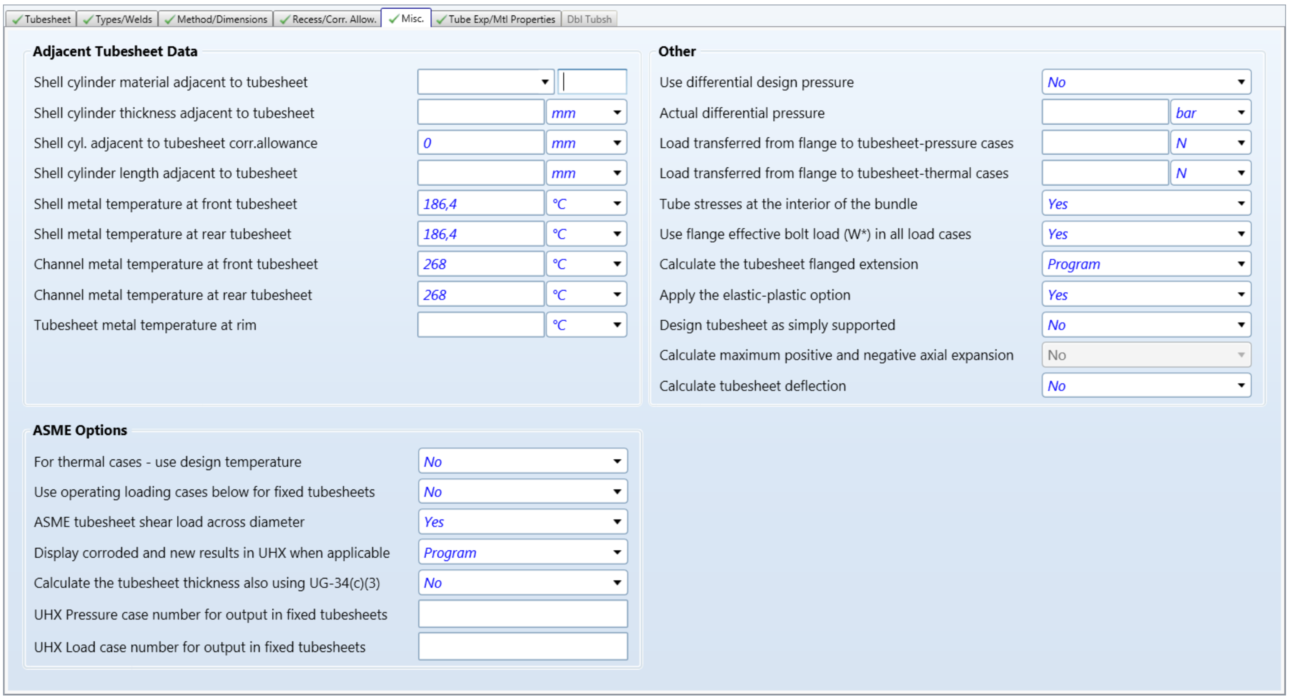


allowed thermal dilatation present in this case make these inputs unnecessary, and they will be left avoid in order to not provide wrong constraints. Notice that only as last chance a designer will insert an inconvenient component as an expansion joint. The two inputs that have not been analysed, since they are not foreseen in the datasheet, are the vacuum design temperature and the effect of radial differential thermal expansion.

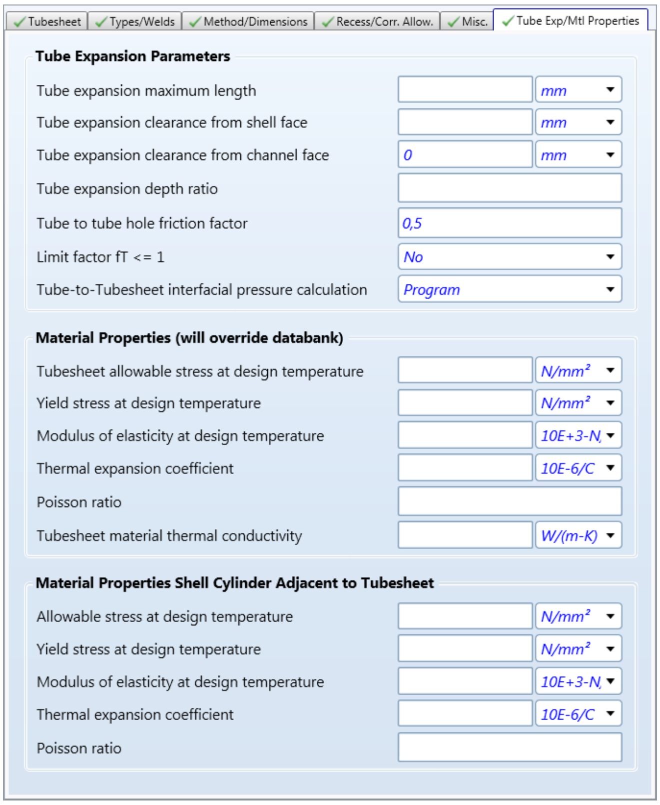
Changing sub-form, the first inputs presented in “*Recess/Corrosion Allowance*” are the tubesheet corrosion allowances in the shellside and tubeside. They have to be specified if they have different values from the general ones already inserted in “*Problem Definition - Design Specifications*”. If nothing is entered the values inserted in the general specifications will be automatically upload. In this case, three millimetres on each sides will be taken into account as already indicated. Proceeding, the recesses dimensions for the gasket accommodation, already discussed talking about the body flanges in the “*Front Head*” sub-form, can be specified for the tubesheet sides. They are represented here because the recesses are present even in the tubesheet, as counterpart of the flange’s ones. All the remaining inputs, e.g. backing flanges, are useful only in case of floating heads.



In the case in which the thermal dilatation would be prevented and the program cannot find a great solution automatically, some warning will be generated. In that case it is useful to insert manually some dimensional or technical constraints, e.g. one of the inputs contained in the “*Adjacent Tubesheet Data*” section of the “*Miscellaneous*” sub-form. The ASME code imposes that the design of the tubesheets, especially the cases in which the thermal expansion is prevented, must be based on the operative temperature, instead on the design ones. These because the thermal contribution on the sizing of the tubesheet is driven by the difference between the shellside and tubeside temperatures, which generally is higher for the operative temperatures. Reassuming, by default the ASME imposes to consider the operative temperature, however, it leaves the possibility to consider the design ones in case they will generate a greater gradient. All the rules to design the tubesheets in the ASME code are contained in the UHX, and the possible options can be selected in these sub-form, e.g. use the UG-34 (c)(3) option: for the thickness of the non-circular tubesheet. A lot of other option can be specified in the section “*Other”*. For example if the tubesheet design has to be made using the differential design pressure, or if consider an interior or an exterior tube of the bundle as reference for the thermal stress. Since the tubes have different thermal stress in function of their position, the interior tubes are generally used as reference, given that they have an higher thermal load. The values of the flanged extension thickness must be made explicit and known when the collar bolts have to be used, while the elastic-plastic option can be disabled, even if the tubesheet is the only part that can be designed by means of its adoption.

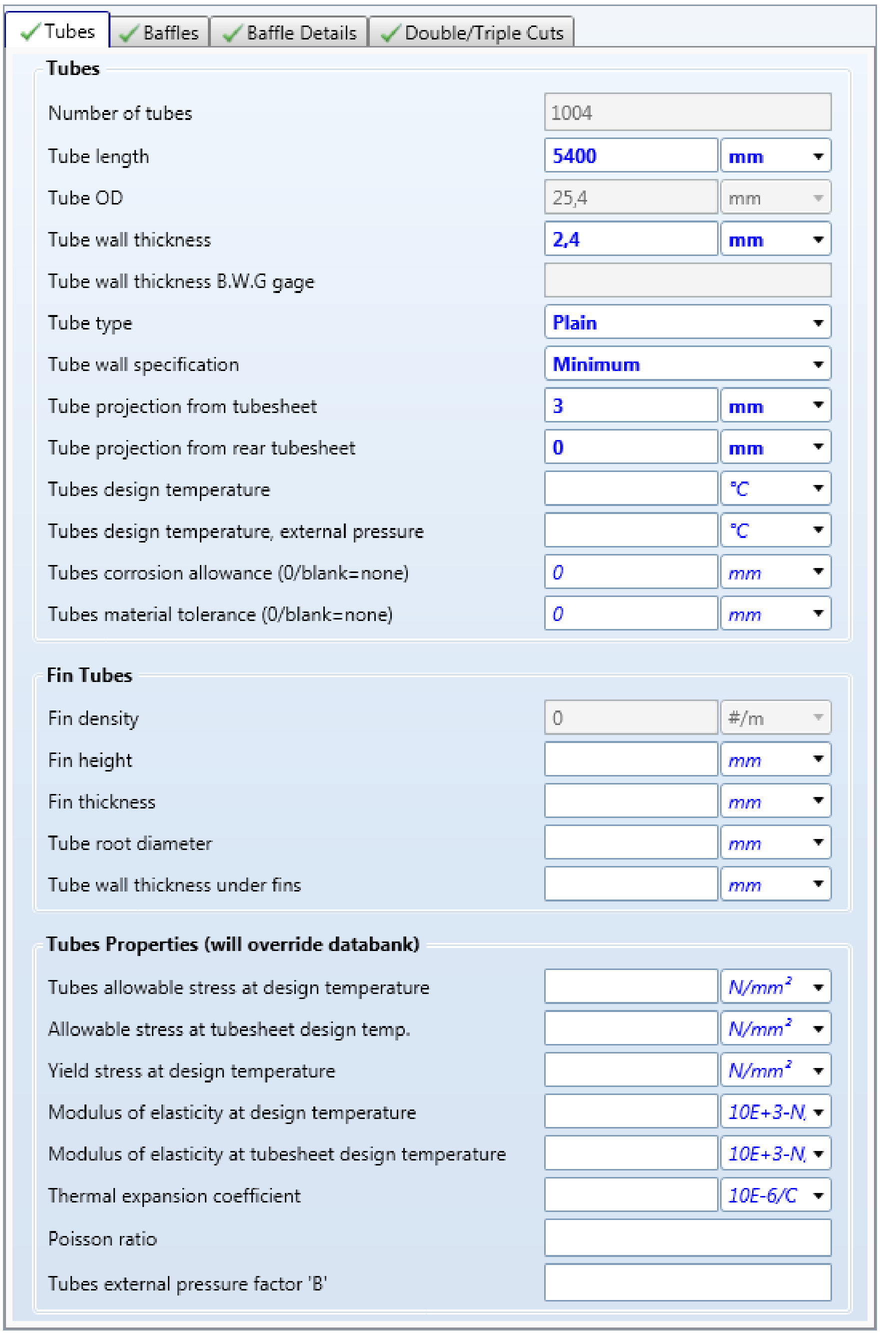


At the end of the tubesheet form, a last sub-form named “*Tube Expansion/Material Properties*” is present. Parameters regarding the expansion of the tubes in the bundle can be set here, e.g. the maximum length that they can reach. Finally, two sections dedicated at the material are displayed. The first is the same at those previously analysed, and it will override the data inserted in the material section that will be examined. The second one allow to insert the properties of the material adjacent to the tubesheet which is used where reinforcement is required. Even in this case, the properties inserted here will override the ones present in the general databank of the program.



TUBES/BAFFLES

In this part of the program, "*Tubes/Baffles*", as well as in the following one, "*Tubesheet Layout*", most of the inputs are locked because the variation of their values could modify the heat exchanger thermal design and the global configuration. They can be modified only if the user goes in the sub-form "*Tubesheet Layout*", present in the homonymous form, and sets "*Create a new layout*". In this case a new layout can be implemented, but it will not be taken automatically from the program. Indeed, all the data must be exported from the mechanical part and uploaded again in the thermal one, in order to update the results and the configuration. Given the necessity to make the mechanical design on the real heat exchanger configuration, these passages lead to a sort of iterative procedure. However, all the parameters here have been already explained and have been uploaded automatically from the thermal design. The only observation that can be done is on the corrosion allowance. Indeed, differently from all the other heat exchanger parts, the tubes generally are not oversized by the corrosion allowance, being the heat transfer in function of the thickness.



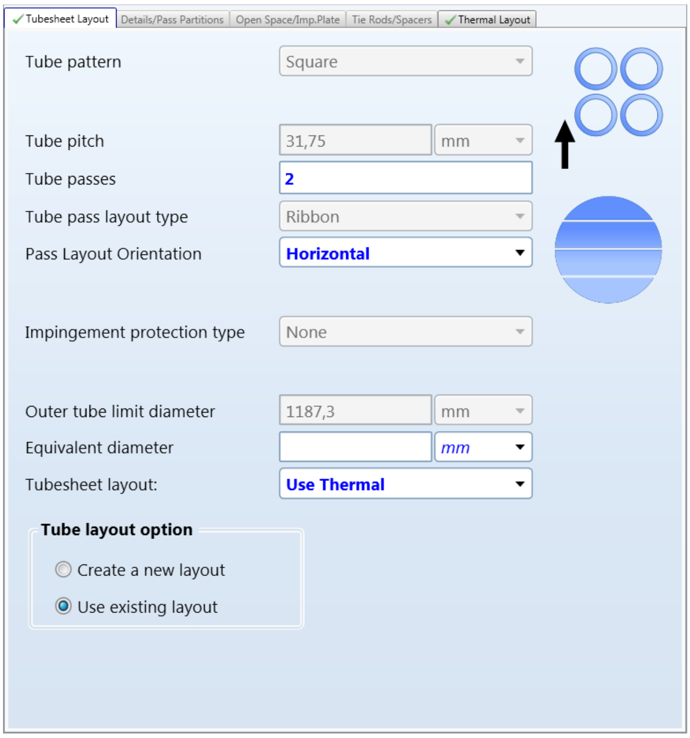
Regarding the second sub-form, "*Baffles*", only some inputs are consistent in this case. As a matter of fact, being the device a kettle, the baffles have a support function only, and the flow is not guided. The specification "*Unbaffled*", the number of supports, and the input regarding the ending one have to be set here. In the analysed application, three supports plus the ending one have been implemented as input. Some particular entries are the ones regarding the space around the heat exchanger required to remove the bundle. These inputs are not useful for the heat exchanger design, but they are present because the data will be provided to the program packs which allow the whole plant layout design. This is because the ASPEN is a suite compounded by a lot of program packs interacting one each other, not only intended to the mere heat exchanger design.

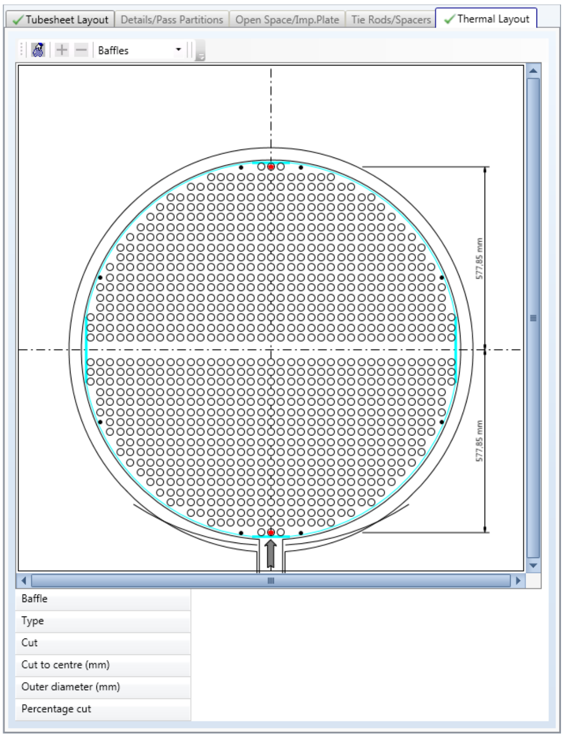
Same things happen in the sub-form "*Baffle Details*" in which no baffles are outlined. The tube unsupported spans, reported from the thermal design, are the only consistent parameters. They are three, the first is the unsupported span of the tubes between the supports, while the second and the third are the distances of the first and last baffle from the tubesheet and the U bend support respectively. If the heat exchanger is not a kettle, the baffle would have the purpose of deviating the flow. This means that the distances between the baffles would be a design parameter for the thermal computation, and they could not be modified unless the thermal design is done again. The only problems that the span length can introduce are the fluid elastic instability or an acoustic resonance problems. So, the distances between the supports are simply obtained dividing equally the length of the tubes.

As last sub-form the "*Double/Triple Cuts*" can be chosen. It allows to insert all the percentages of length cutting, needed to define the baffles in the case of double or triple segmental cuts, and to choose which segment is the starting one. These baffles have already been presented in the "*Generalities on heat exchanger*" chapter, however, being this part of the program pointless for the analysed case, they will not be thoroughly analysed in the thesis.

<u>TUBESHEET LAYOUT</u>

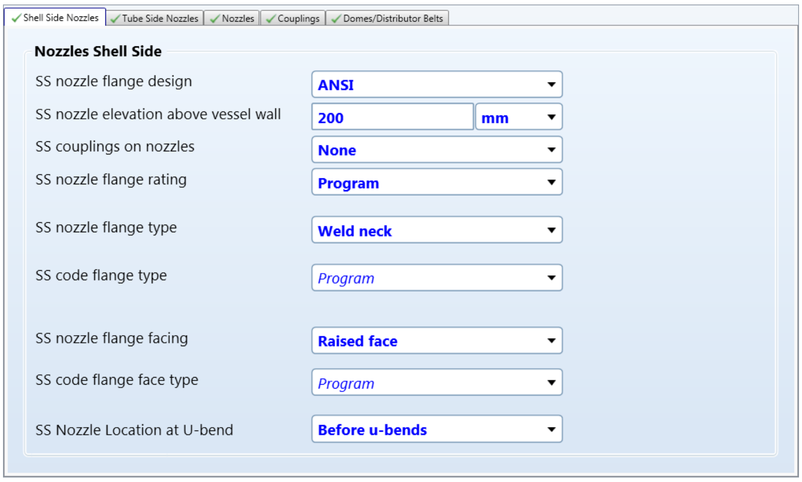
As already introduced, in "*Tubesheet Layout*" a lot of inputs are locked because depending on the layout generated in the thermal design, while the other inputs have already been analysed. Only the equivalent diameter, a service parameter used by the program to compute the shear stress, can be imposed. Proceeding, in the "*Thermal Layout*" sub-form, a first technical drawing of the support is shown. On it, the most dangerous tubes, analysed inside the "*Vibrational & Resonance Analysis*" form, have been outlined. Moreover, the square tube pattern, the ribbon and horizontal pass layout orientation and the tie rods holes can be seen. Notice that the horizontal pass layout determines the orientation of the U curvature of the bundle, which does not influence the heat exchange, but can be very important for the vibrational and resonance problems. At the end, the outer tube limit diameter, which cannot be modified, is shown in light blue in the image.



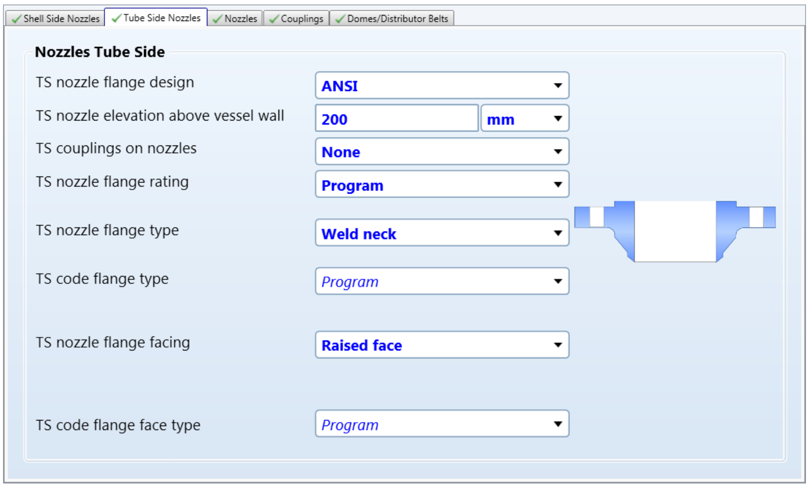


NOZZLES-GENERAL

In this set of sub-forms the parameters regarding the nozzles can be set. In the first two, "*Shell Side Nozzles*" and "*Tube Side Nozzles*", the general parameters can be selected. ANSI nozzle flange with an elevation of 200 mm above the vessel and without couplings have been chosen. The couplings, attachments for devices and sensors on the tube, were used in the past in a configuration of one or two per tube. Nowadays they are inserted only in particular cases and only if required. Talking about the flange design, an empirical method is the historical Taylor Forge, that highly overestimate the dimensions. While the DIN standards have developed a more scientific method for the flange design, the classical numerical method based on finite element analysis can always be used. However, if standard flanges have been selected, only the rating have to be verified. The ratings,

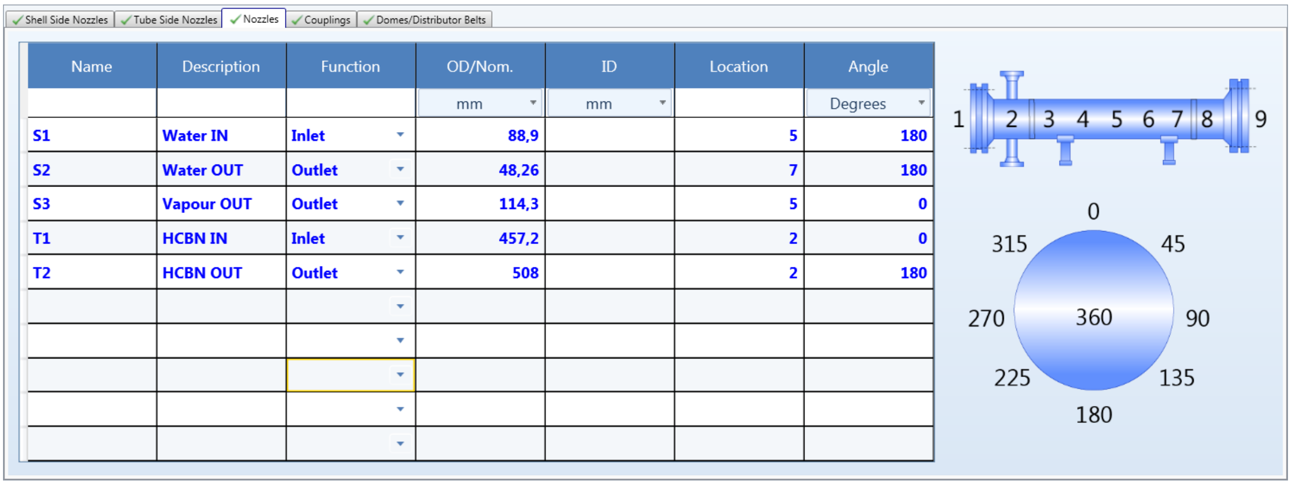


expressed in pounds [lb], represent the classes in which the flanges are classified in function of the working pressure and temperature, e.g. 150 lb, 300 lb; 600 lb; 900 lb; 1500 lb; 2500 lb. So, a flange is chosen from a table, or automatically by the program, in function of its working pressure and temperature. Conceptually the same process would have been done using the metric units of measure in the case of ISO standards. Furthermore, talking about the whole pipe and flange coupling, different methods can be used in order to obtain the reinforcement for the cut area of the hole. In this case the classic one is adopted, where a "*Weld neck*" type of joint is used and the required compensation is made with a reinforcing pad. Notice that in this case the flange can be of every type explained before, e.g. weld neck or slip on, while the "*Weld neck*" type refers to the shape of the joints, which have a bevel to reach the standard dimension of the pipe. The "*Long welding neck*" option instead does not present the bevel, allowing the pipe to be thicker, maintaining the same internal diameter. Both the flange and the non-standard pipe are obtained from the same forging piece, then welded at the end with the shell. If the long welding neck option does not allow enough increment of tube's area needed to satisfy the required reinforcement, the remaining area can be computed and added at the end of the tube, by means of a local increment in thickness. In this case, a "*Self-reinforcing*" coupling is obtained. Notice that, as already introduced, in some particular cases the reinforcing pad is not allowed, e.g. hydrogen service, and one of the other solutions have to be mandatorily chosen. The "*Code flange type*" is a parameter to be set in case of long welding neck or self-reinforcing nozzles, while the "*Flange facing*" can



be "*Raised Face*", "*Flat Face*" or with "*Tongue and Groove*". The most common choice is the raised face, which presents a prominence on the face to support the gasket, while the flat face are generally used in conjunction with the full diameter gaskets. The last parameter, "*Code flange face type*", represents the way in which the face of the flange have to be worked, e.g. plane or grooved.

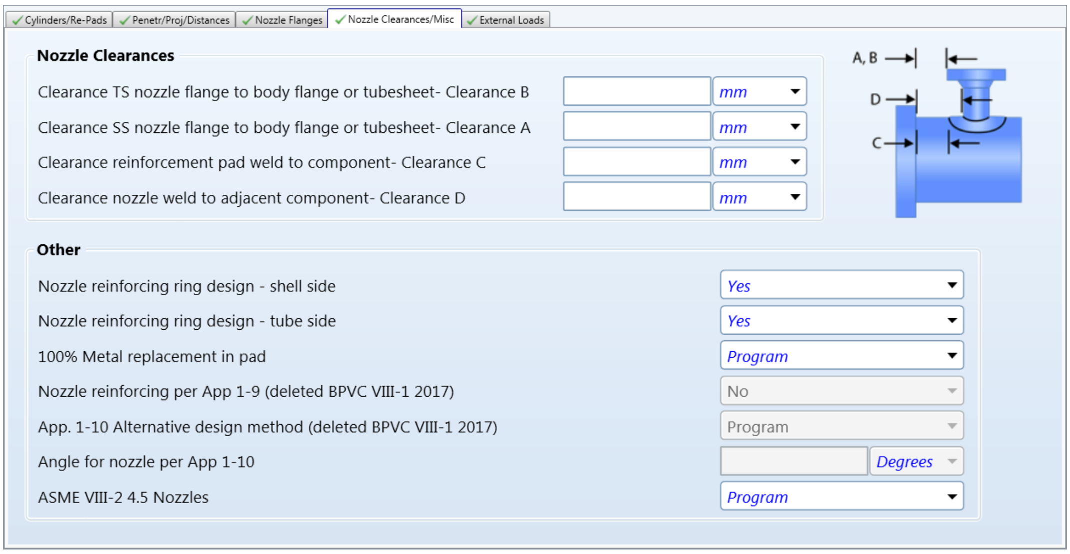
In the following sub-form, "*Nozzles*", the dimension of the single nozzle and the angle of position can be changed. These are the only parameters that would influence the thermal design of the device, e.g. changing the ρv^2 or the vibrational behaviour. Notice that the images on the right of the table serves as a guide for the location number and for the positioning angle. The name and description of every nozzles must be manually inserted, basing on the function, position and dimension. Finally it can be said that the coupling, the dome and the distributor belts settings cannot be modified due to the fact that they was not present in the application under exam.



Name	Description	Function	OD/Nom.	ID	Location	Angle
			mm	mm		Degrees
S1	Water IN	Inlet	88,9		5	180
S2	Water OUT	Outlet	48,26		7	180
S3	Vapour OUT	Outlet	114,3		5	0
T1	HCBN IN	Inlet	457,2		2	0
T2	HCBN OUT	Outlet	508		2	180

<u>*NOZZLES-DETAILS-EXTERNAL LOADS*</u>

In this form, all the tables presented allow to customize every single nozzle and flange. Indeed, if the previous section sets the general characteristics, here, all the geometrical dimensions, positions, as well as the joint efficiency and corrosion allowance can be specified for every nozzle. Also the dimensions of the reinforcing pads and the parameters of every nozzle's flanges, e.g. type, rating and face, can be implemented. However, in this section, as in some others already faced, it is not convenient insert values in the first attempt. Different is the case in which the device has some geometrical clearances to be observed, or some external loads acting on the nozzles. In that case, the input value have to be inserted using the last two sub-forms, named "*Nozzle clearances/Miscellaneous*" and "*External Loads*". The first has been reported below, and it presents the possibility to insert some other parameter. As examples can be reported the possibility to insert the nozzle reinforcing ring, previously named reinforcing pad, or "*100% Metal replacement in pad*" which allow to do not consider the corrosion in the reinforcing pad.



Nozzle Clearances

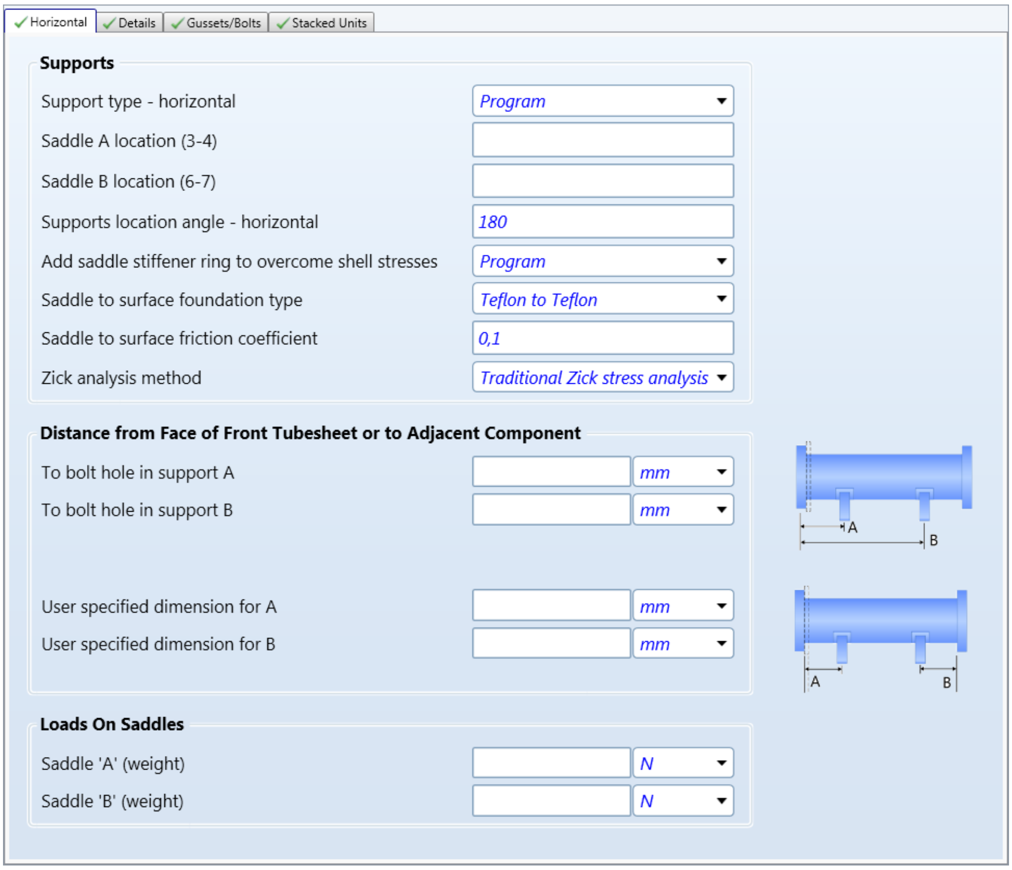
Clearance TS nozzle flange to body flange or tubesheet- Clearance B		mm
Clearance SS nozzle flange to body flange or tubesheet- Clearance A		mm
Clearance reinforcement pad weld to component- Clearance C		mm
Clearance nozzle weld to adjacent component- Clearance D		mm

Other

Nozzle reinforcing ring design - shell side	Yes
Nozzle reinforcing ring design - tube side	Yes
100% Metal replacement in pad	Program
Nozzle reinforcing per App 1-9 (deleted BPVC VIII-1 2017)	No
App. 1-10 Alternative design method (deleted BPVC VIII-1 2017)	Program
Angle for nozzle per App 1-10	Degrees
ASME VIII-2 4.5 Nozzles	Program

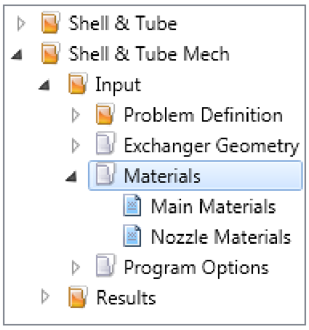
HORIZZONTAL SUPPORTS

The heat exchanger is generally supported by means of two saddles which allow to fix it at the ground, always leaving a degree of freedom for the thermal expansion, or to stack more device one to another. Location, distances and orientation, as well as the parameters for the friction of the contact with the ground can be specified. Given that the contact with the saddles induces a higher mechanical stress on the shell, particularly in the case in which the internal pressure is not to high and the thickness will not be too much, stiffening ring on the junctions are inserted. This will change the moment of inertia and increase the stiffness of the part subjected at the stress, which can be computed with the method of Zick. As last parameters, the total loads on the saddles can be inserted manually if other devices would be present on the heat exchanger, e.g. a device that discharge its weight on a nozzle.



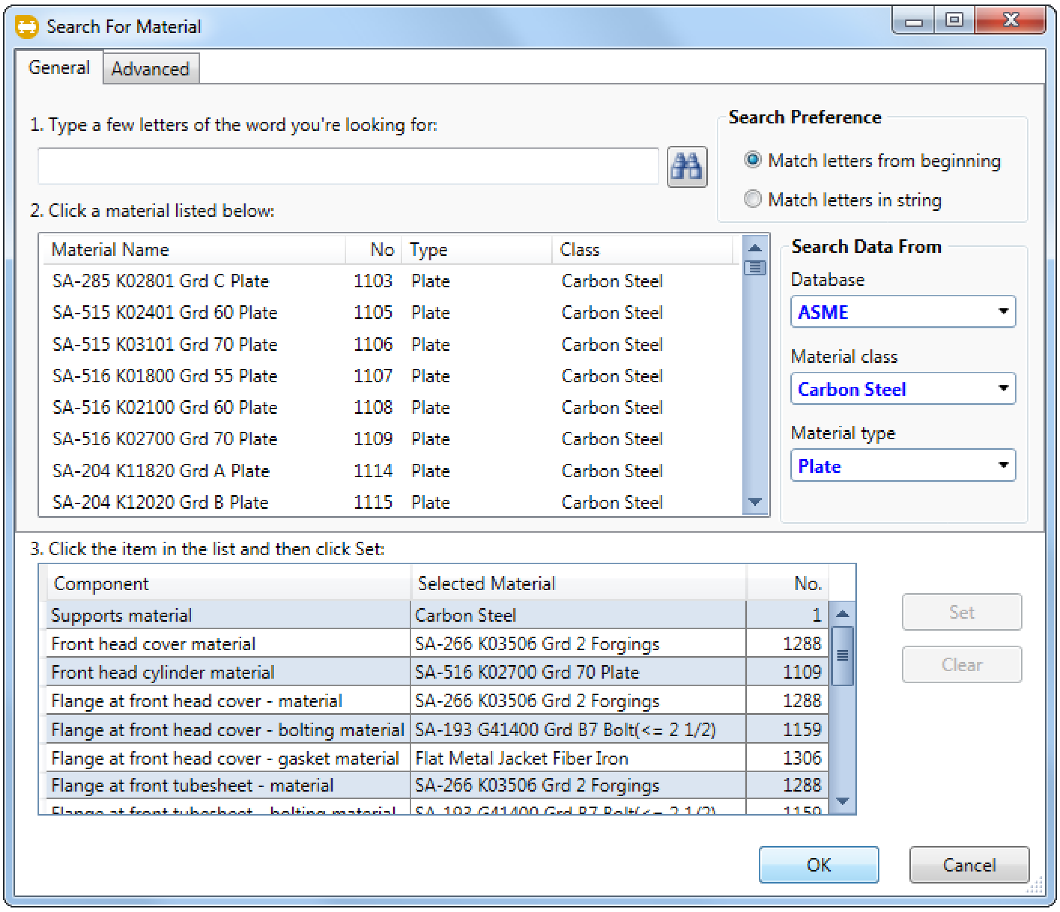
In the sub-forms "*Details*" and "*Gussets/Bolts*" the insertion of all the geometrical parameters of the parts that compound the saddles take places. Entering more in detail and providing some terminology, a support is formed by a *wear plate* welded to the shell and supported by some vertical internal rib which are indicated as *gussets*. The plate that cover the gussets is named *web*, and the one on which all is placed is the *base plate*. Last sub-form "*Stacked Units*" allow to choose if admit the stacking of more device whereas more than one would be present. However, the configuration on one only heat exchanger was an output provided from the thermal design.

7.2.3. MATERIALS



In the thermal design part only the class of the material, e.g. carbon steel, was implemented as input. Indeed, it was the only information required in that part in order to compute the thermal conductance, which, as delineated, it is also in function of the material class. In the mechanical design, instead, the material type and the grade, in concomitance with the standard specification of the used code have to be denote.

Starting with the first form "*Main Materials*", and clicking on the button "*Search Databank*", the correct type of material and grade can be found and assigned at every part of the heat exchanger. In order to make the research easier, the first section allows to type the name of the standard specification. The second section, instead, provide the list of all the materials contained in the selected code. Choosing a material type, e.g. Plate, a filter is applied to the whole list, and the correct specification can be selected among those who remain displayed. In the third part, the material chosen above can be assigned at every selected component of the heat exchanger by means of the "*Set*" button.



Please notice that the experience cover a high role in the selection of the correct standard specification. Indeed, all the specification should be opened, read, and evaluated in order to find the correct one. Have experience in this field means that the designer can easily determine the approximative range of the final thickness and made the correct choice of the specification at the first attempt. This allow to avoid incredible efforts and time spent to research the correct specification. Taking as example the regulation of the plates, a lot of them are available in the ASME code, which moreover insert also the low-alloyed steels in the carbon steel class. The standard specification generally used for the plates of not alloyed carbon steel, for the refinery service in-pressure applications, is the SA-516. It presents three options:

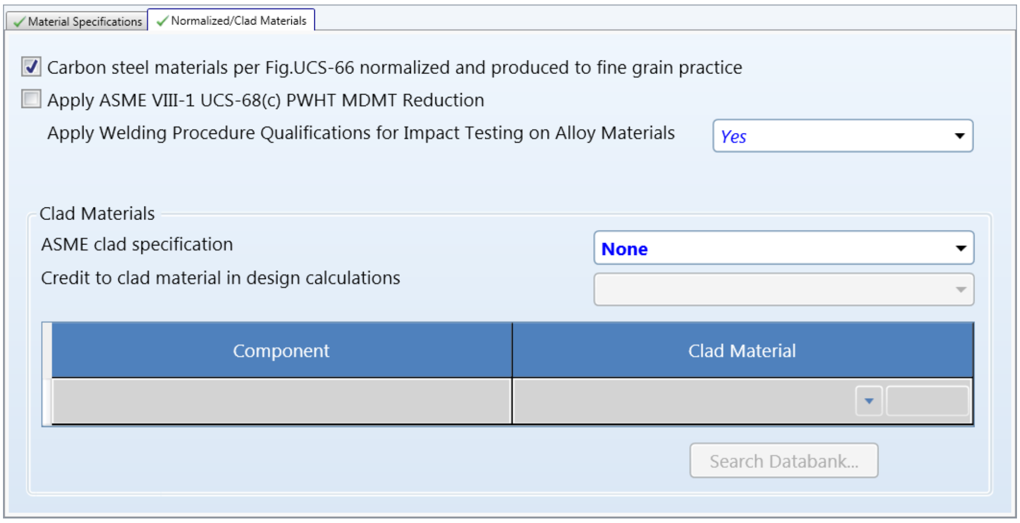
- SA-516 K01800 Grd 55 Plate
- SA-516 K02100 Grd 60 Plate
- SA-516 K02700 Grd 70 Plate

These three options differs from the grade, indicated with the number that represents the KSI, Kilopound per Square Inc, of the yield strength. So, increasing the grade, the yield strength of the material increase due to the high percentage of carbon inside of the steel. Again the experience can suggest that the grade 70 is the best because, even if the material is more expensive, there is a saving given by the less material required. However, if this concept is correct for a normal condition of working, the higher content of carbon present in the grade 70 makes it more prone to crack in working conditions in which high level of hydrogen, or hydrogen sulphide, is present. Those are the cases in which plates of grade 55 or 60 are used for the refinery service in-pressure applications. Returning on the program, it index all the materials, and the corresponding standard specification, with an internal number. All the selected materials chosen for the study case, and the relative internal numbers, can be seen in the following image.

Component	Material	
Supports material	Carbon Steel	1
Front head cover material	SA-266 K03506 Grd 2 Forgings	1288
Front head cylinder material	SA-516 K02700 Grd 70 Plate	1109
Flange at front head cover - material	SA-266 K03506 Grd 2 Forgings	1288
Flange at front head cover - bolting material	SA-193 G41400 Grd B7 Bolt(<= 2 1/2)	1159
Flange at front head cover - gasket material	Flat Metal Jacket Fiber Stainless Steel	1309
Flange at front tubesheet - material	SA-266 K03506 Grd 2 Forgings	1288
Flange at front tubesheet - bolting material	SA-193 G41400 Grd B7 Bolt(<= 2 1/2)	1159
Flange at front tubesheet - gasket material	Flat Metal Jacket Fiber Stainless Steel	1309
Shell cylinder material	SA-516 K02700 Grd 70 Plate	1109
Kettle cylinder material	SA-516 K02700 Grd 70 Plate	1109
kettle reducer cover material	SA-516 K02700 Grd 70 Plate	1109
Baffle material	SA-516 K02700 Grd 70 Plate	1109
Tube material	SA-179 K01200 Smls. tube	1401
Front tubesheet material	SA-266 K03506 Grd 2 Forgings	1288
Flange at front shell - material	SA-266 K03506 Grd 2 Forgings	1288
Flange at front shell - bolting material	SA-193 G41400 Grd B7 Bolt(<= 2 1/2)	1159
Flange at front shell - gasket material	Flat Metal Jacket Fiber Stainless Steel	1309
Shell cover material	SA-516 K02700 Grd 70 Plate	1109

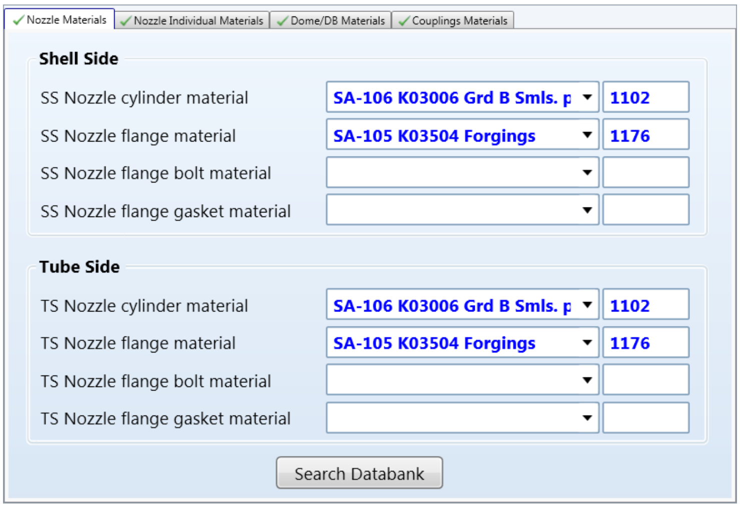
All the materials assigned was outlined by an ASME technical specification, given that it is the code used in the project. However, even if each codes define the materials using different technical standards, they are conceptually similar setting the minimum characteristics that the material must accomplish in order to be sell with a certain name.

It is now possible move on the second sub-form "*Normalized/Clad Materials*". Based on the provided technical standard, one or more heat treatments can be necessary if the thickness overcome a certain value. For example, the SA-516 Grd 70 foresees the mandatory normalization in case of plate thicker than 40mm, while they can be provided "as rolled" in the other cases. These heat treatment bring back the material grains to the original condition, before the rolling. This is particularly important for the carbon steel which will operate at medium or low temperature. In the ASME paragraph UCS-66, where UCS indicate a Carbon Steel requirement, the curves for every material allow to obtain the Rated Minimum Design Metal Temperature in function of the thickness. Conceptually, during the design activity the Rated MDMT of the material used must be lower or equal than the MDMT of the input data, 5°C in this case. As example, a material SA-516 of thickness of 50 mm would have a Rated MDMT of 20°C if not normalized or quenched and tempered. With the heat treatment, a Rated MDMT of -20°C would instead be shown. For this reason the first flag, which will take into account for a normalized material, have to be inserted. If the Rated MDMT would not be lower than the design MDMT, an impact test at that temperature is required. The second flag instead would implement the UCS-68(c), which allow to assume the limit temperature obtained from the UCS-66 curves decreased by 17°C. This decrement can be obtained only for the materials that belong at the P-Number 1 of ASME and if the Post Welding Heat Treatment was not required buy anyway performed, e.g. PWHT done on thickness lower than 32 mm. During this project the pressure and temperature will generate a thickness higher than 32 mm, and the PWHT has been inserted, so, the UCS-68(c) is not applicable.



Proceeding with the second form, “*Nozzle Materials*”, the ASME standards for the shellside and the tubeside nozzles material can be inserted. In the previous sections weld neck flange type have been implemented. This leads to the selection of a pipe material specification for the nozzle cylinder, and of a forging one for the flange. So, for both sides, the general specifications that have been chosen are:

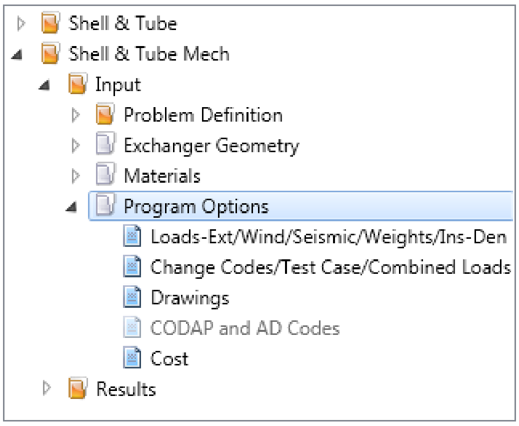
- SA-106 K03006 Grade B Seamless pipe
- SA-105 K03504 Forgings



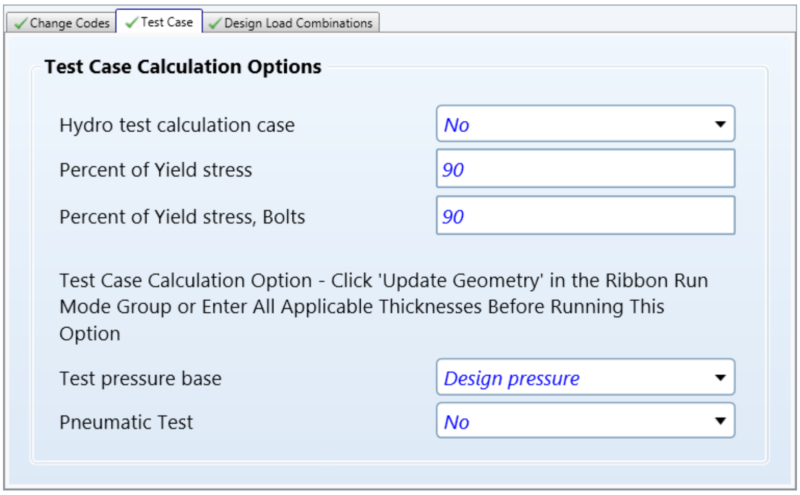
The flange bolt and the gasket material for these process nozzles are not to be implemented because they are generally provided with the pipe, and not with the heat exchanger. Only the bolts and gaskets of body flanges and the closed nozzle have to be chosen from the heat exchanger designer.

In the other sub-forms, the materials for the distributor belts and couplings could be chosen in the case in which they were inserted. “*Nozzle Individual Material*” instead allow to specify the material for every single part of every nozzles. This can be done by means of a table as already seen in other parts of the program which allow the data insertion of the individual parts.

7.2.4. PROGRAM OPTIONS



The program options of the mechanical design part will not be thoroughly analysed, as it had been done for the thermal part. During the pressure vessel design, different load types have to be taken into account, and these are the subjects to which the first form refers. Generally they are categorized in external loads, wind and seismic loads, earthquake loads, weight and insulation load. Common external loads to be considered are the thrusts given from thermal dilatation of the connection pipes. The best way to verify the resistance of the connection area in this case is to make a stress analysis with the loads provided from the pipe designer, or with the standard load estimated for the type of pipe. The Welding Research Council issues empirical methods, e.g. WRC 537, to verify the external loads, firstly developed to overcome the absence of the stress analysis method. However, given the possibility to provide fast result, they are still in use, even if the results are overestimated. However, attention must be paid because these methods have some applicability limits defined by the nozzle's tube to shell ratio and from their relative stiffness. If these limits are not respected, other methods, or the time expensive stress analysis, have to be evaluated. The wind loads should be taken into account for the pressure vessel designed for an outdoor working environment. However, given the commonly compact shape of the heat exchangers, this load only impacts on the supports size rather than on the shell design. Different would be the case in which a tall and wide device have to be designed, where the possibility of high moments induced by the wind have to be carefully considered. Similar considerations can be done for the seismic loads. The evaluation of these loads must be done using the different codes generally applied in the civil engineering field, e.g. ASCE, IBC o the Eurocodes. It is important to remark that every case can be different and can present some peculiar external loads to be considered. For example, due to the situation, it may happen that the weight of the snow or of the insulation have to be considered, as well as the external loads given from the sea or ground pressure if the device will be placed undersea or underground respectively. All these stress have been thoroughly considered in the real design of the heat exchanger associated at this study case. However, they would require a deep analysis of the situation and the development of many considerations not pertinent on the subject of this thesis. For this reason, the importance of these considerations have been outlined in this paragraph, but they will not be inserted explicitly.



A last important consideration to make is about the pressure of the water during the final test. As explained in the paragraph "*Problem Definition*", the minimum test pressure of the water is done by $1{,}3 \cdot MAWP \cdot LSR$ where the MAWP is the Maximum Allowable Working Pressure, and the LSR is the Lowest Stress Ratio. However, due to technical approximations or by a client requirements, the real value of the test can be higher than the minimum one explained above. If this happens, it means that a pressure higher than the minimum one computed by the program has been inserted in the field "*Test Pressure*" in "*Problem Definition – Design Specification*". In this case, a verification calculation have to be done using this pressure, which means to run one extra time the mechanical design inserting Yes in "*Hydro test calculation case*". Anyhow, being this pressure computed using the MAWP and highly overestimated with respect to the real pressure of working, but supported only for once time from the device, and at ambient temperature, a higher percent of the yield stress can be used for the calculation with respect to the one used in the design. The ASME code admit using the 90% of the theoretical yield stress as limit for this verification. Last but not least, the drawings to be produced as output have to be selected, and some technical parameter for the calculation of the cost can be implemented.

7.3. ITERATIVE CORRECTIONS AND RESULTS SECTION

Once that all the input parameters have been inserted, the button "*RUN*" can be pressed. In this phase, similarly to what has already been done for the thermal design, the program will solve all the formulas taken from the selected code of calculation and fill the output of the mechanical design. As for the input, even the result part of the program is compounded by several sections, each of which can be divided in forms and sub-forms. In the first section, *"Input Summary",* a resume of the input data is reported. Given the simplicity of the mechanical design results, represented mainly by the dimensions of every heat exchanger's parts, this section will not be analysed following the exact division proposed by the program. However, all the results can be found in the output tables reported in the next chapter, as well as the technical drawings and the calculation report can be seen it the dedicated appendices. In this first chapter, instead, the analysis of the warnings developed by the program for the first obtained result will be done. Moreover, in order to solve them, further implementations of the input data will be presented where necessary.

7.3.1. FIRST RESULT AND MODIFICATIONS

The "*Warning/Messages*" form, contained in the section "*Design Summary*", can be taken as reference for this procedure of final checks and modifications. Given that the software executes a mathematical approach to define the minimum thicknesses, the first thing that a designer have to do is to check the compatibility between the resulting dimensions and the pieces present on the market, or simply a conscious observation of the results. No input or result warnings have been generated, so the input section has been completed correctly and with all the mandatory data so that the program was able to calculate a result. However, some advisory and notes have been produced. Even if they do not compromise the design, they must be considered.

			Description
	Note	487	SHELL/KETTLE SMALL CYL REINF: Cone compression ring added for reinforcement. Please increase cone thickness if reinforcing ring not desired.
	Note	711	NOZZLE REINFORCEMENT T1: Nozzle cylinder thickness increased to satisfy reinforcing requirements.
	Note	711	NOZZLE REINFORCEMENT T2: Nozzle cylinder thickness increased to satisfy reinforcing requirements.
	Advisory	83	SHELL CYLINDER: ASME - Post Weld Heat Treatment may be required. See PWHT section in design summary of output.
	Advisory	83	SHELL COVER: ASME - Post Weld Heat Treatment may be required. See PWHT section in design summary of output.
	Advisory	83	ECCENTRIC REDUCER: ASME - Post Weld Heat Treatment may be required. See PWHT section in design summary of output.
	Advisory	83	KETTLE CYLINDER: ASME - Post Weld Heat Treatment may be required. See PWHT section in design summary of output.
	Note	909	SHELL COVER: The calculated fiber elongation for this component is 12,78 %, which exceeds the Max value of 5 %. Heat treatment per UG-79 may be required.

Starting from the two "*Notes 711*", they explain why the thicknesses of the two tubeside nozzles, T1 and T2, have been increased. As already presented, the compensation for the resistance decrement caused by the holes is achieved by an empirical method. Following the ASME code, this method implies to increment the transversal area present in the so-called reinforced limit in order to compensate the area of the shell that has been removed. With this purpose the program has increased the thicknesses of the cylinders that compound the nozzles. Checking the thicknesses of the plates from the correspondent drawing it can be noticed that the front head cylinder plate, which minimum thickness was been computed as 31 mm, must be set at 32 mm for market compatibility reasons. This should solve the "*Note 711*" because it results in an addition of compensation material for the nozzles T1 and T2. On the other hand, the thickness of the plate for the shell diameter can be set as the one of the starting plate from which the shell cover will be formed, e.g. 55 mm considering the 10% of shrinkage. In this way, no step will be present in the joint, and the same plate can be used for both parts. Notice than given the input provided at the program, the shell cover plate in the drawing is represented with the final thickness, and not with the one that will be necessary to purchase. To implement all the required modification, it is enough to go in the section correspondent at that input, and insert the value in the correct field. Running again the program with the new inserted constraints leads to a solution in which the two notes regarding the nozzles are no more present, and the plates thicknesses are acceptable.

The four "*Advisory 83*" show the possibility of the Post Weld Heat Treatment to be required for the shell cylinder and cover, as well as for the eccentric reducer and for the kettle cylinder. These advisory are only remainders, and given that the PWHT has already been requested in the input, they do not imply any further modification.

The "*Note 909*", instead, regards the after forming heat treatment of the shell cover. The ASME code impose always the heat treatment in case of hot forming, while, in case of cold forming, it is mandatory only if fiber elongation is higher than 5%. So, given the fiber elongation of 12,78%, this note is a reminder of this requirement if cold forming will be performed. No modification in the design parameters are required, but attention must be paid to comply with this requirement when the forming procedure will be done or subcontracted.

The last remaining note is the 487, which warns for the insertion of a cone compression ring. Due to the fact that, sometime, the compression ring leads to manufacturing problems given the lack of space for welding, it may not be desired. In this case, the plate thickness of the cone can be increased, from 58 mm to 60 mm, in order to intensify the stiffness of the cone and no longer make the ring necessary.

Last modifications before than the final mechanical design will be produced regard the nozzles and the reinforcing pads dimensions. Indeed, the standard pipes for the nozzles have to be selected in order to avoid further verifications with stress analysis methods. Given the outside diameters already inserted in "*Nozzles-General*" during the implementation of the inputs, the program have computed a possible solution for each pipe. However, standard pipes would be appreciated. In order to get this result, the inner diameter of each tube can be carefully chosen and inserted as input. Observing the minimum thickness of each tube computed by the program, a tube schedule that ensure a thickness higher than it can be chosen for each tube. This can be done because the standard tubes are organized in schedule which fix all the dimensions of the tubes in function of the outer diameter. It is now reported a table in which the results of this procedure can be seen.

NAME	DESCRIPTION	OUTER DIAMETER	MINIMUM THK.	SCHEDULE	THICKNESS	INNER DIAMETER
S1	Water IN	88,9 mm	9,19 mm	Schedule 160	11,13 mm	66,64 mm
S2	Water OUT	48,26 mm	7,11 mm	Schedule 160	7,12 mm	34,02 mm
S3	Vapour OUT	114,3 mm	9,66 mm	Schedule 120	11,13 mm	92,04 mm
T1	HCBN IN	457,2 mm	16,55 mm	Schedule 60	19,15 mm	418,90 mm
T2	HCBN OUT	508,0 mm	17,99 mm	Schedule 60	20,62 mm	466,76 mm

As before explained, the computed inner diameters can be inserted as input in the respective sub-form. In this way the dimensions of the pipes are fixed and they will be found on the market since they belong to standard schedules.

Name	Description	Function	OD/Nom.	ID	Location	Angle
			mm	mm		Degrees
S1	**Water IN**	**Inlet**	**88,9**	**66,64**	**5**	**180**
S2	**Water OUT**	**Outlet**	**48,26**	**34,02**	**7**	**180**
S3	**Vapour OUT**	**Outlet**	**114,3**	**92,04**	**5**	**0**
T1	**HCBN IN**	**Inlet**	**457,2**	**418,9**	**2**	**0**
T2	**HCBN OUT**	**Outlet**	**508**	**466,76**	**2**	**180**

Running the program with the addition of the inner diameters as constraints, the variations of thicknesses will lead to modifications on their respective reinforcing pad. Please notice that if the imposed thicknesses by the inner and outer diameters would not satisfy the minimum ones, a warning would be issued. The reinforcing pads can be analized and checked by means of the drawings. Particularly a pad of 31 mm of thickness can be increased to 32 mm given that a plate of that dimension has been already uses. This lead to a reduction of different material type to be purchased. In order to fix the thickness of that reinforcing pad, its value have to be inserted in the "*ExchangerGeometry - Nozzle Dettails*". Same procedure can be done for all the reinforcing pads with thickness different from the plates already used. Running for the last time, the final design of the heat exchanger is issued with the correspondant drawings, calculation report and the results that will be listed in the next chapter.

7.3.2. OUTPUT TABLES

DESIGN SUMMARY

Design Specifications

TEMAClass	R 1200/1583-5400 AKU	Shell Side	Tube Side	Tubesheets
Design Pressure	bar	78	60	
Static Head (included above)	bar			
Vacuum Design Pressure	bar			
Test Pressure	bar	101,4	78	
Design Temperature	°C	280	300	300
Average Metal Temperature	°C			
Min. Design Metal Temperature	°C	5	5	
Corrosion Allowance	mm	3	3	
Front Tubesheet Corrosion Allowance	mm	3	3	
Rear Tubesheet Corrosion Allowance	mm			
Radiography		Full	Full	
Number of Passes		1	2	
Nozzle Flange Rating		600	600	
Post Weld Heat Treatment		Yes	Yes	

Code:	ASME Section VIII Div.1 2017			TEMA 9th Edition			
Weights:	**Empty:**	35809	**Full:**	48299	**Bundle:**	10853	kgf
	Operating:	43038	**Material Standard:**			ASME	2017

Overall Dimensions

Major components			Overall length
Overall Front Head Assembly			1787 mm
Front Tubesheet			190 mm
Tubesheet Thickness	200	mm	
Tube Side Recess	5	mm	
Shell Side Recess	5	mm	
Welding Stub End(s)		mm	
Clad Thickness		mm	
Shell			6001 mm
Rear Tubesheet			mm
Tubesheet Thickness		mm	
Tube Side Recess		mm	
Shell Side Recess		mm	
Welding Stub End(s)		mm	
Clad Thickness		mm	
Overall Rear Head Assembly			mm
Overall Shell Cover Assembly			498 mm
Unit Overall Length			8476 mm

Component	Side of vessel	OD	Offset from shell CL	Elevation from shell CL	Distance from front tubesheet	Distance from front head nozzle
		mm	mm	mm	mm	mm
Nozzle S1	Shell	88,9		840	2855	3659
Nozzle S2	Shell	48,26		840	5211	6015
Nozzle S3	Shell	114,3		1047	2855	3659
Nozzle T1	Tube	457,2		832	804	
Nozzle T2	Tube	508		832	804	
Shell Support A				1430	1420	2224
Shell Support B				1430	5940	6744
Weir		1583			6188,5	6992,5
Center of Gravity					2180	2984

		Shell Side	Tube Side
Volume	m^3	8,8643	3,6346
Average fluid density	kg/m^3	942,03	99,7
Area	m^2	32,8	6,4
Insulation thickness	mm		
Insulation type			
Insulation density	kg/m^3	128,15	128,15
Insulation weight	kgf		
Insulation weight, extras	kgf		

Exchanger weight operating	kgf	43038
Exchanger weights empty	kgf	35809
Exchanger weights full	kgf	48299

Maximum Allowable Working Pressure

		Design Conditions			New and Cold		
Component	Side	Temp	Stress	MAWP	Temp	Stress	MAWP
		°C	N/mm^2	bar	°C	N/mm^2	bar
Front Hd Bolting At TS	S	300	172,37	78,03	21,11	172,37	78,03
Front Hd Bolting At Cov	T	300	172,37	60,01359	21,11	172,37	60,01359

***=Shell Side MAWP** **+=Tube Side MAWP**

		Design Pressure	Design MAWP	New and Cold MAWP
Shell side	bar	78	78,03	78,03
Tube side	bar	60	60,01359	60,01359

Minimum Design Metal Temperature for Impact Test Exemption

Controlling components (see details for potential compliance with UG-20(f))

Component	Curve	Temperature
		°C
Front Tubesheet	C	-9

Hydrostatic Test Pressure - UG-99

Side	Controlling Component	Material
Shell Side	Nozzle S1	SA-106 K03006 Grd B Smls. pipe
Tube Side	Nozzle T1	SA-106 K03006 Grd B Smls. pipe

	Hydrostatic	Design		Test		Stress
Side	Test Pressure	Temp	Stress	Temp	Stress	Ratio
	bar	°C	N/mm²	°C	N/mm²	
Shell Side	101,4	280	117,9	21,11	117,9	1
Tube Side	78	300	117,9	21,11	117,9	1

Post Weld Heat Treatment

Controlling Components

Component	Material	Actual thickness	Per ASME section	ASME material P number	ASME material group number
		mm			
Shell Cylinder	SA-516 K02700 Grd 70 Pla	40	UCS-56	1	2
Shell Cover	SA-516 K02700 Grd 70 Pla	50	UCS-56	1	2
Eccentric Reducer	SA-516 K02700 Grd 70 Pla	60	UCS-56	1	2
Kettle Cylinder	SA-516 K02700 Grd 70 Pla	55	UCS-56	1	2

VESSEL DIMENSIONS: CYLINDERS AND COVERS

		Front Head		Shell	Shell Cover	
		Cover	Cyl.	Cyl.	Cover	Tubes
Head type		Flat bolted			Ellipsoidal	
Outside diameter	mm	1472	1264	1280	1691	25,4
Calculated thk.	mm	170,78	30,85	39,01	49,22	0,81
TEMA minimum thk.	mm		12,7	12,7	12,7	
Actual thickness	mm	176	32	40	50	2,4
Radiography			Full	Full	Full	
Joint efficiency		1	1	1	1	1
Corrosion allowance	mm	3	3	3	3	
External pressure	bar					78
Length Ext.Press.	mm					5400
Maximum Ext.Press.	bar					97,68356
Minimum thk. Ext.Press.	mm					2,01
Max.length Ext.Press.	mm					16256

		Eccentric	Kettle
		Reducer	Cylinder
Head type		Cone	
Outside diameter	mm	1711	1693
Calculated thk.	mm	57,9	50,62
TEMA minimum thk.	mm	12,7	12,7
Actual thickness	mm	60	55
X-ray		Full	Full
Joint efficiency		1	1
Corrosion allowance	mm	3	3
Ext. pressure	bar		
Length ext. press.	mm	678	
Maximum pressure EP	bar		
Minimum thickness EP	mm		
Maximum length EP	mm		

VESSEL DIMENSIONS: NOZZLES – NOZZLE FLANGES

Nozzle description		Water IN	Water OUT	Vapour OUT	HCBN IN	HCBN OUT
Nozzle designator		S1	S2	S3	T1	T2
Vessel side		Shell	Shell	Shell	Tube	Tube
Outside diameter	mm	88,9	48,26	114,3	457,2	508
Inside diameter	mm	66,64	34,02	92,04	418,9	466,76
Calculated thickness	mm	6,89	5,27	7,77	16,55	17,99
Code minimum thk	mm	9,19	7,11	9,66	16,55	17,99
Actual thickness	mm	11,13	7,12	11,13	19,15	20,62
Nozzle weld leg	mm	8,13	4,12	8,13	16,15	17,62
Reinf.pad OD	mm	163		178	789	878
Reinf.pad thickness	mm	32		55	32	32
Reinf.pad weld leg	mm	20,07		25,78	17,49	17,53
Corrosion allowance	mm	3	3	3	3	3
External pressure	bar					
Length ext. press.	mm					
Nozzle projection	mm	200	200	200	200	200
Maximum ext. press.	bar					

Nozzle description		Water IN	Water OUT	Vapour OUT	HCBN IN	HCBN OUT
Nozzle designator		S1	S2	S3	T1	T2
Flange type		Weld neck	Weld neck	Weld neck	Weld neck	Weld neck
Nozzle flange design		ANSI	ANSI	ANSI	ANSI	ANSI
Flange rating		600	600	600	600	600
Flange OD	mm	209,55	155,57	273,05	742,95	812,8
Bolt circle	mm	168,28	114,3	215,9	654,05	723,9
Bolt diameter	mm	19,05	19,05	22,22	41,28	41,28
Bolt number		8	4	8	20	24
Gasket OD	mm	127	73,15	157,23	533,4	584,2
Gasket width	mm	9,65	9,65	12,7	22,35	25,4
Gasket thickness	mm	3,18	3,18	3,18	3,18	3,18
Flange calc. thk.	mm					
Flange actual thk.	mm	31,75	22,35	38,1	82,55	88,9
Flange height	mm	82,55	69,85	101,6	184,15	190,5
Lap jnt ring OD	mm					
Hub length	mm					
Hub slope						
Weld height	mm					

VESSEL DIMENSIONS: BODY FLANGES

		Front Head		Shell
		Cover	at TbSh	Front
Flange type		Hub	Hub	Hub
Flange design standard		Optimized	Optimized	Optimized
Flange OD	mm	1472	1536	1536
Bolt circle	mm	1389	1434	1434
Bolt diameter	mm	41,28	50,8	50,8
Bolt number		48	40	40
Gasket OD	mm	1263	1278	1278
Gasket width	mm	19	25	25
Gasket thk.	mm	3,18	3,18	3,18
Flange calc. thk.	mm	174	230	188
Flange overlay	mm			
Recess	mm	5	5	5
Flange act. thk.	mm	179	235	193
Lap jnt ring OD	mm			
Hub length	mm	48	48	60
Hub slope		0,175	0,1	0,225
Hub Thickness go	mm	32	32	40
Weld height	mm			

VESSEL DIMENSIONS: TUBESHEETS – TUBES – BAFFLES

		Front
Tubesheet diameter	mm	1536
TEMA minimum thickness	mm	25,4
TEMA bending thickness	mm	182,91
TEMA shear thickness	mm	81,74
TEMA flange extension thk	mm	
TEMA effective thickness	mm	182,91
Code flange extension thk	mm	
Code thickness	mm	194
Corrosion allowance - shell	mm	3
Shell side land or recess	mm	
Corrosion allowance - tube	mm	3
Recess	mm	5
Base thickness	mm	200
Clad thickness	mm	
Total thickness (base+clad)	mm	200
Weld leg size	mm	

Tube type		Plain
Tube OD	mm	25,4
Tube wall thickness	mm	2,4
Number of tubes		1004
Tube length	mm	5400
Tube pitch	mm	31,75
Tube pattern		90
Outer tube limit diameter	mm	1187,3
Number of tube passes		2
Tube projection from front tubesheet	mm	3
Tube projection from rear tubesheet	mm	
Tubesheet tube hole nominal diameter	mm	25,7

Tube to Tubesheet Joint: ' e ' - Welded, a >= 1.4 t, and expanded

Baffle type		Full supports
Baffle orientation		
Baffle outside diameter	mm	1193,65
Baffle thickness	mm	16
Baffle cut as percent of diameter		25
Baffle tube hole diameter	mm	25,8
Baffle number		3
Baffle spacing center-center	mm	1350
End length at front head	mm	1345
End length at rear head	mm	1355
Baffle spacing at front head	mm	1142
Baffle spacing at rear head	mm	1345

U-bend Details

Schedule number	Number of U's	U-bend diameter	U-bend length	U-tube length
		mm	mm	mm
1	36	76,2	119,69	10919,69
2	36	139,7	219,44	11019,44
3	36	203,2	319,19	11119,19
4	36	266,7	418,93	11218,93
5	36	330,2	518,68	11318,68
6	34	393,7	618,42	11418,42
7	34	457,2	718,17	11518,17
8	32	520,7	817,91	11617,91
9	32	584,2	917,66	11717,66
10	30	647,7	1017,4	11817,4
11	28	711,2	1117,15	11917,15
12	28	774,7	1216,9	12016,9
13	26	838,2	1316,64	12116,64
14	24	901,7	1416,39	12216,39
15	20	965,2	1516,13	12316,13
16	18	1028,7	1615,88	12415,88
17	12	1092,2	1715,62	12515,62
18	4	1155,7	1815,37	12615,37

Bundle Removal Space:

6152 mm

VESSEL DIMENSIONS: SUPPORTS

Dimensions

			Gussets		Hole Dimensions			Sad/shell	Wear
Thk.	Width	Depth	Number	Thk.	Size	To edge	C/C	Angle	Plt thk.
mm	mm	mm		mm	mm	mm	mm	Degrees	mm
40	1600	400	4	13	32	89	1066	134,8	20

	Saddle Distance from Tubesheet	
Exchanger weight	Saddle A	Saddle B
kgf	mm	mm
48299	1420	5940

Zick Stress Analysis

		Calculated		Allowed
		Saddle A	Saddle B	
Bending at saddle + pressure	N/mm²	68,32	65,52	136,41
Bending at midspan + pressure	N/mm²	67,45	66,25	136,41
Tangential shear stress - unstiffened	N/mm²	5,91		109,12
Tangential shear stress - stiffened	N/mm²		1,22	109,12
Circumferential stress at horn	N/mm²	-7,24	-0,65	-170,51
Ring compression in shell	N/mm²	5,39	1,09	104,51
Tangential shear stress in head	N/mm²		1,36	109,12
Head stress + pressure	N/mm²		134,01	170,51

8. CONCLUSIONS

As it has been introduced during some thesis' passages, the design and simulation programs perform the pre-set computations based on the indicated design code. Even if its use makes the computation phase easier, the only way to obtain, avoiding errors, the best configuration for the case under analysis is by a correct and careful management by the engineer. The first personal observation that is worth to do concerns the previously introduced approach the designer must have with the program. Often the problem is not represented by which input value have to be assigned, but rather by the necessity to do not insert too many constraints. Indeed, to insert an input means to add constraints to the configuration. This can lead to the impossibility to achieve a final design, to the issue of warning notifications or to the program crash. Furthermore, if this last option occurs and many inputs have been introduced, the designer might find hard to understand which value is causing the problem. As example, the heat exchanger of the study case has a removable U-shape bundle, in which there are not thermal expansions prevented. In this case the insertion of input parameters in a function which have the goal to balance the thermal stress will produce a constraint, which in turn will not allow to reach a possible configuration. It is therefore necessary to understand the possibilities of the program, the meaning of its functions, and which inputs are mandatory to obtain the desired configuration. Hence, the first computation phase of the program has to be launched with as few inputs as possible, and the needed constraints must be inserted gradually during the steps. This generates an iterative procedure, still allowing to maintain the control over the project activity.

Resuming the thermal study case, further observation and analysis can be pointed out. The configuration that has been chosen as the best one is named Design 6 in the study case. Among those analysed, it was the one which better satisfies the parameter of selection, as the heat exchanger cost, the area ratio and the feasibility of the device. Only the Design 4, which also satisfies the area ratio and the pressure loss constraints, could be seriously considered as an alternative option, being its estimated cost 5% lower than the Design 6. However, it presents some vibration problems which lead it to be discharged given that, during the design activity, the input data was already optimized for the cheapest solution. Indeed, a modification of the input parameters to make it compliant with the requirements would probably lead to an increment of the price for that solution. In the other cases, depending on the location in which resonance occurs, a modification on the supports, like increasing their number or the spacing between them, could solve the problem. Other solutions like using a different tube pattern, e.g. 45° instead of 90°, or modifications that change the fluid dynamic of the system can also be examined. However, given the pool boiling configuration with almost absent shellside flow, the adjustments of the baffles like the type and the cut orientation cannot be made. All these considerations lead to the definitive choice of the Design 6, which configuration presents a single AKU horizontal device as solution. As it has been indicated, for the shellside there are one input and two outputs, one for the steam and one to drain the non-vaporized water. For the HCBN in the tubeside, instead, only two nozzles are present, but the dimensions are much greater due to the higher mass flow rates. An acceptable number of 502 tubes, with length of 5400 mm and two passages are required. Being the bundle a U-type, the two passages are mandatory and the number of tubes lead to have 1004 holes in the tubesheet, while the total minimum straight length of the tube required to produce one fork turns to be *10800 mm*. This allows to define a gross surface exchange area of *464,6 m^2*, which at the net of the portion that does not effectively transfer heat, evolves into the effective heat exchange area of *449,4 m^2*. Note that being in this case the U-bend totally immersed, its area has been indicated as transferring and therefore it has not been subtracted. The required area, instead, can be found from the analysis of the heat transfer resistances for both conditions, clean and fouled. Once that all these data are available, the area ratio can be computed by means of the proportion between the effective and the required area. In particular, in the cleaned condition, a greatly overestimated area ratio of 2,36 can be computed by means of a required area of *190.5* m^2. In the fouled one, instead, an area ratio of 1,02 can be computed with a required area of *442,7* m^2. This means that this design has an oversized area of 2% in fouled condition, which is within the range of the minimum percentage of overestimation requested in the input parameters. Here, the high influence of the fouling can be seen. Indeed, in fouled state the required area is more than twice the one required in clean state. This happens because the fouling decreases the capacity to transfer heat in the device changing the effective values of the thermal conductance $h\ [W/m^2\ K]$, and so the overall heat transfer coefficient $U\ [W/m^2K]$. It is worth to recall that, as analysed during the thesis, all these aspects are in function of the fluid velocity. In order to enhance the heat transfer, a lower speed would be better. However, poor velocities lead to increase the fouling, while a too high speed can generate erosion or vibration problems. In the shellside, the velocity of the water is very low, but it can be acceptable due to the pool boiling configuration. In the tubeside, instead, the configuration leads to have *7 m/s*, which is higher than the minimum *2÷3 m/s* required for dirty fluid as the HCBN. On the other hand, as it was analysed in the "*Flow Analysis*" form, the values of ρv^2 obtained in the study case are far minor than the limit reported in the theoretical part, so no modifications on the nozzle diameters was required.

Comparing the mass flow rates present in the output with the design parameters inserted as input, it can be noticed that some calculation refinements have been done. In particular the mass flow rates divisions of the outing phases are slightly different due to the integration procedure. Despite that, they can be accepted being the variations in the order of 0,004% ÷ 0,005% with respect to the values indicated in the inputs. As demanded, the vaporization is practically complete, while the HCBN partially condenses during the path. The same minor alteration has been made on the outlet temperature of the steam. As already said, it was possible to choose the Design 6 because the computed pressure drops were lower than the maximum allowed. Indeed, the 0,12487 bar in the shellside is less than 0,3 bar, while in the tubeside 0,41924 bar is minor than 0,7 bar. Analysing more in detail the subdivisions of the pressure drop, it can be noticed that the higher contribution in the shellside is given from the outlet nozzle which is responsible for the *80,52%* of the whole value. In the tubeside, instead, the critical part is represented by the losses inside the tubes, which account for the *83,22%*.

Moreover, a brief summary of how the total thermal exchange of *5418,3 kW* is divided among the process can be discussed. In the tubeside the fluid is biphasic either at the inlet and at the outlet, but during the process it condenses providing heat. In particular, it can be seen from the results that the HCBN gas phase releases the *16,54%* of the total heat emitted, while the liquid the *27,2%*. The remaining *56,26%* is provided during the condensing process that transforms the vapour into liquid. In the shellside, where the heat is received, it can be seen that the *74,23%* of the total heat exchange is latent heat, while only the *25,8%* is used to bring the water at the boiling temperature. These are reasonable values since the aim of the device is to provide saturated vapour. Indeed, the steam will not be oversaturated, and the percentage of the heat absorbed by the vapour phase is nil. Another parameter which allows to observe the quality of the vapour at the outlet is the already explained entrainment ratio. Expressing the percentage of the outlet mass which can be in liquid phase, it links the quality of the vapour at the outlet with the difference between the shell diameter and the bundle one. The estimated diameters values for this configuration produce an entrainment fraction of 0,0001, which is much lower than the maximum limit of 0,02 prescribed for dry steam. Regarding the quality of the vapour at the top of the bundle, instead, it has to be as lower as possible, because the tubes must always be covered by water. The resulting value of 0,0172 can be accepted. If the quality of the vapour had been too high, an intervention on the process parameters would have been required, e.g. an increment of the water mass flow rate, or a decrement of the hot fluid one. Differently, if the problem had been represented by the entrainment fraction, an increment of the inner shell diameter must have been done.

In order to evaluate the configuration generated with the thermal design, two vibration & resonance analysis have been performed, and the results have been analysed. With both the analysis methods, HTFS and TEMA, either the fluid elastic instability and the evaluation of the resonance have not shown any problems. The assessment of the fluid elastic instability was done computing the ratio between the actual shellside flowrate and the critical flow rate which causes it. For all the points and for all the values of damping adopted in the computation, the resulting ratio was correctly far from 1. The acoustic resonance, instead, was investigated using the ratio between the tube frequency and the tube acoustic natural frequency. In the same way, this ratio must be as far as possible from 1. In a more detailed way, the program reports both values of the two riskiest tubes: the natural frequency F_n and the acoustic natural frequency F_a, that will cause resonance in motion and sonorous respectively. After the selection of the configuration, the rating mode was made available and modifications of the output parameters in a iterative way was made possible. The size of the nozzles have therefore been compared to the ASME standard diameters of the pipes on the market. Nothing has been changed except for the nozzle of the shellside vapour outlet, which was increased from 3,5 inches, which is not a very common dimension, to 4 inches. After the nozzle modification, the geometry configuration must be processed again. Notice that all the results reported in the thesis for the thermal design are the ones obtained in the first passage, and not in this last iteration. However, the modification of this nozzle have only produced some slightly improvements as a decrement of the pressure drop and of the ρv^2. It is worth to specify that the used input of the mechanical design was the last iteration of the thermal part, as it should be.

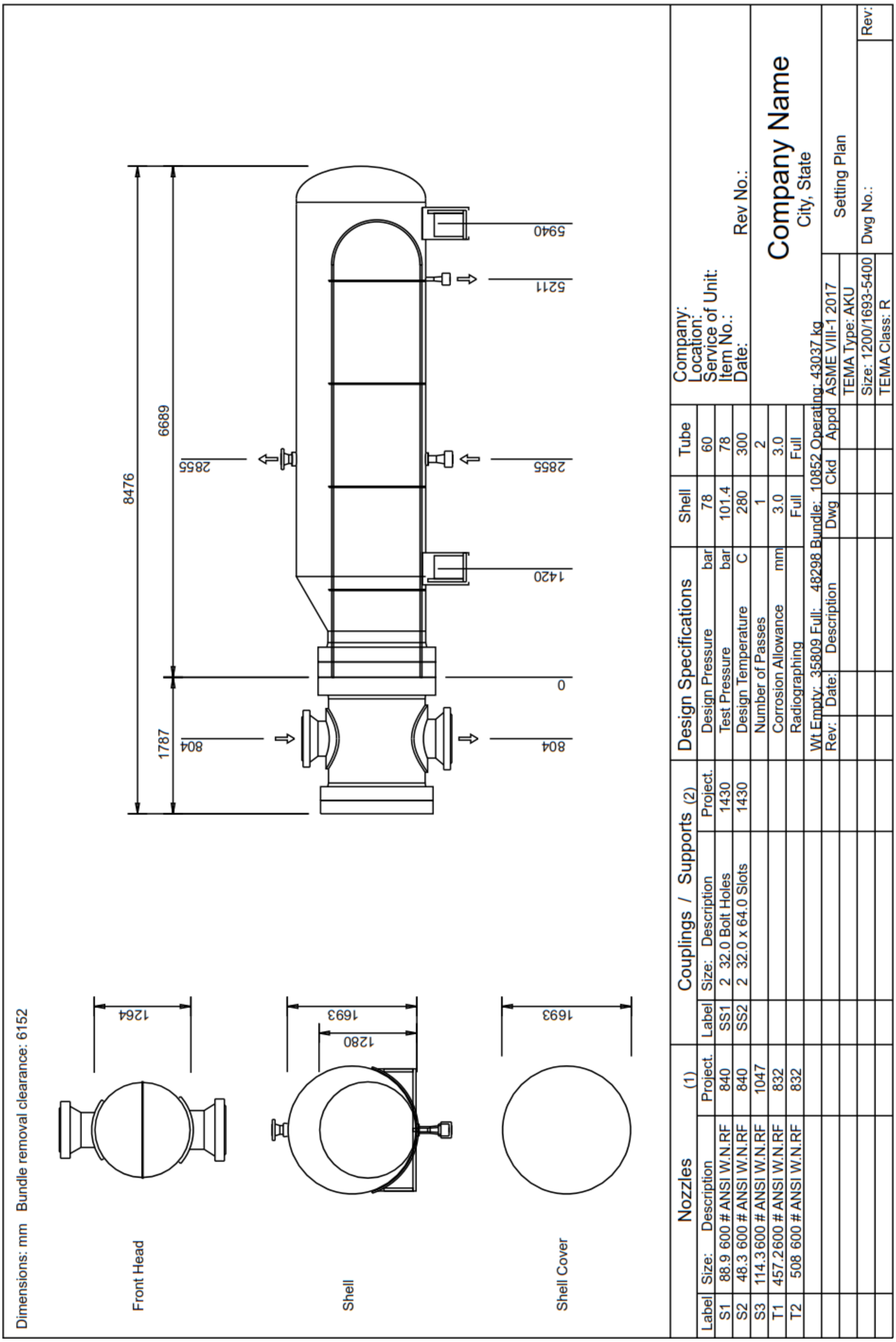
After the insertion of the still required inputs explained during the thesis, also the mechanical output has been made available. A first aspect that can be analysed in this final presentation is that, given the thicknesses over 40 mm, the normalization after rolling requested by the SA-516 Grd 70 plates have to be satisfied, as it was foreseen during the implementation of the inputs. This heat treatment brings back the material grains in the original condition, before the rolling. As explained, this is a particularly important factor because it allows to obtain, from the ASME paragraph UCS-66, a Rated Minimum Design Metal Temperature of -10°C for the higher thickness of 55 millimetres, which is lower than the 5°C of MDMT provided as input. Contrariwise, given that some parts in P number 1 material mandatorily require the PWHT because their thicknesses are higher than 32 millimetres, the UCS-68(c) would have been inapplicable, and an impact test at that temperature would have been required in order to further investigate the material resistance. Indeed, please note how, if the carbon steel had not been normalized, the Rated Minimum Design Metal Temperature obtained from the UCS-66 would have increased quickly with the thickness, so the impact test would have been required. During this test the specimens should have shown an amount of energy absorbed higher than the limits provided by the code in use, so as to certify that a ductile-brittle transition has not happened until that temperature.

Going back to the analysis, in the inputs and during the design process enhanced problems of corrosion or hydrogen embrittlement have not been indicated or foreseen. So, the canonical three millimeters of oversize, resulting from the 0,1 millimeters of average corrosion per year estimated for a not high-aggressive hydrocarbon on a lifetime of 30 years, have been inserted as corrosion allowance. However, unlike all the other heat exchanger parts, the tubes generally are not oversized, being the heat transfer in function of the thickness. Indeed, it is generally preferable to adopt more efficient materials, e.g. stainless steel, than to increase the thickness of the tubes. Totally different is the case of the tubesheet, which is in contact with fluids on both sides. As explained, in this case three millimeters have been added on both sides, resulting in 6 millimeters of oversize. Being this project in the ASME Normal service class, any mode of defect control could be designated. However, given that large thicknesses were foreseen, and the code forces to make the full radiography over a certain value, this control mode was already inserted from the beginning. This choice led to the possibility to adopt the joint efficiency equal to one in the thickness' formulas of the components, with the consequence that they have not been increased, given the total control of the welding joints and the impossible presence of not found defects. Still in the mechanical output the Maximum Allowable Working Pressures can be seen and compared with the respective design pressures. The MAWP of the shellside and of the tubeside are represented by the lowest MAWP of the single side-component. In this case, each side has the MAWP determined by the Front Head Bolting, which in particular is equal to *78,03 bar* for the shellside, and to *60,01 bar* for the tubeside. Notice that their values are really close to the design pressures of *78* and *60 bar* respectively. This situation is correct and indicates a well-done design, without waste of material. Indeed, these values already contain the necessary precautions, being the design pressure a large overestimation of the working one and the yield strength of the material assumed as the minimum indicated in the standard specification. Furthermore, the computation of the allowable stress is subjected at the safety factors of *1,5*, as indicated by the ASME section VIII Division I. The hydrostatic test pressures, used in the final test, are higher than the design ones to compensate the fact that the test will be made at room temperature. The resulting values, computed with the formula $1{,}3 \cdot MAWP \cdot LSR$ contained in the ASME code, are *101,4 bar* for the shellside, and *78,0 bar* for the tubeside. Notice that once that the $MAWP$ of the shellside and tubeside have been fixed, the Stress Ratio, value obtained dividing the admissible stress at room temperature by the one at the design temperature, being in function of the materials, is the only parameter that can change in the formula. The Lower Stress Ratio for both hydrostatic tests, the shellside and the tubeside ones, results to be 1. In each cases it is given from the material of the pipes, *SA-106 K03006 Grd B Seamless Pipe*, which generally degrades less than the other. Indeed, for the ASME, the lower ratio is always provided by the material which less suffers from the variation of the temperature, excluding the bolting materials whose admissible stress value practically doesn't change. Notice that, even if the vacuum working condition is not foreseen, the tubes of the bundle are designed also against the external pressure, and after the higher thickness is chosen. Even in this case, as well as for the tubesheet, it could be possible to adopt the differential pressure approach, providing that a loading procedure would be associated with the device. Finally, the discussion of the mechanical output can be concluded with the control on the component dimensions, which are all acceptable, and allow enough space to perform the welds during the production of the piece.

Now that the noteworthy results have been commented, it is possible to investigate the limitations of the study. During the thermal design the thickness of the tubesheet is roughly estimated, given that it influences the heat exchange. However, only the dimension evaluated in the mechanical design can be accepted as safe for the piece resistance. This value, even if moderately, is generally different than the one estimated, and leads to a slightly variation in the heat exchanged. In order to make a precise design, a verification of the thermal design with the new thickness of the tubesheet must be made reloading the results of the mechanical part in the thermal one. In case of a larger thickness difference and a small area ratio margin the new thermal design could result unacceptable. Please, note that during this procedure the configuration and the dimensions of the heat exchanger are set. However, it is worth to remember that in the study case the thermal design was overestimated of the 2% in the fouled conditions, and the variation of the tubesheet thickness was negligible. So even if slightly approximated, since it was the version before the rating mode and the refinement of the dimensions, the thermal design reported in the thesis results to be valid and truly representative.

The last remarks to conclude the discussion regard the technical drawings, reported in the Appendix I. Being automatically generated by the software, they are not enough accurate to be used during the production phase or to be provided to the client. However, they are an excellent starting point for a designer to produce the final version.

9. APPENDIX I: TECHNICAL DRAWINGS



Component			O.D.	Thk.	Materials of Construction
Nozzle	S1	Inlet	88.9	11.1	600 # ANSI W.N.RF Flg SA-105 on SA-106 K03006 Grd B Smls. pipe
Pad	S1	Inlet	163.0	32.0	SA-516 K02700 Grd 70 Plate
Nozzle	S2	Outlet	48.3	7.1	600 # ANSI W.N.RF Flg SA-105 on SA-106 K03006 Grd B Smls. pipe
Nozzle	S3	Outlet	114.3	11.1	600 # ANSI W.N.RF Flg SA-105 on SA-106 K03006 Grd B Smls. pipe
Pad	S3	Outlet	178.0	55.0	SA-516 K02700 Grd 70 Plate
Nozzle	T1	Inlet	457.2	19.2	600 # ANSI W.N.RF Flg SA-105 on SA-106 K03006 Grd B Smls. pipe
Pad	T1	Inlet	789.0	32.0	SA-516 K02700 Grd 70 Plate
Nozzle	T2	Outlet	508.0	20.6	600 # ANSI W.N.RF Flg SA-105 on SA-106 K03006 Grd B Smls. pipe
Pad	T2	Outlet	878.0	32.0	SA-516 K02700 Grd 70 Plate
Shell Cylinder			1280	40.0	SA-516 K02700 Grd 70 Plate
Eccentric Cylinder			1711	60.0	SA-516 K02700 Grd 70 Plate
Kettle Cylinder			1693	55.0	SA-516 K02700 Grd 70 Plate
Fr Hd Cylinder			1264	32.0	SA-516 K02700 Grd 70 Plate
Fr Hd Cover			1472	176.0	SA-266 K03506 Grd 2 Forgings (Flat Cover)
Shell Cover			1691	50.0	SA-516 K02700 Grd 70 Plate (Ellipsoidal Cover)
Front TubSh			1536	200.0	SA-266 K03506 Grd 2 Forgings
Fr Hd Flng TubSh			1536	235.0	SA-266 K03506 Grd 2 Forgings (Hub Flange)
Fr Hd Flng Cover			1472	179.0	SA-266 K03506 Grd 2 Forgings (Hub Flange)
Fr Shell Flng			1536	193.0	SA-266 K03506 Grd 2 Forgings (Hub Flange)
Fr Hd Gskt TubSh			1278	3.2	Flat Metal Jacket Fiber Stainless Steel (Periph. Width 25.0 mm)
Fr Hd Gskt Cover			1263	3.2	Flat Metal Jacket Fiber Stainless Steel (Periph. Width 19.0 mm)
Front Shell Gasket			1278	3.2	Flat Metal Jacket Fiber Stainless Steel (Periph. Width 25.0 mm)
Fr Hd Blts TubSh			51		SA-193 G41400 Grd B7 Bolt (40 Bolts on 1434.0 mm B.C.)
Fr Hd Blts Cover			41		SA-193 G41400 Grd B7 Bolt (48 Bolts on 1389.0 mm B.C.)
Tubes			25.40	2.4	SA-179 K01200 Smls. tube (Plain Tubes)
Baffles			1194	16.0	SA-516 K02700 Grd 70 Plate
Shell Supports				40.0	SA-285 K02801 Grd C Plate

Dimensions: mm

Design Specifications

		Shell Side	Tube Side
Design Pressure	bar	78	60
Test Pressure	bar	101.4	78
Design Temperature	C	280	300
Number of Passes		1	2
Corrosion Allowance	mm	3.0	3.0
Radiographing		Full	Full

TEMA Type: AKU Size: 1200/1693-5400 Area: 433 m2

Tube Type: Plain # Holes: 1004 Length: St 5400 mm

Layout: 31.75 mm (90) Tube-Ts Joint: Exp/Str.Welded

Baffle Type: (Full Baffle) Cut: 25% No: 3

Baffle Spacing (C-C): 1350 Inlet: 1142 mm

Impingement Protection: None

Code: ASME VIII-1 2017 TEMA Class: R

Wt Empty: 35809 Full: 48299 Bundle: 10853 Operating: 43038

Drawn By: Ckd By: Apvd By:

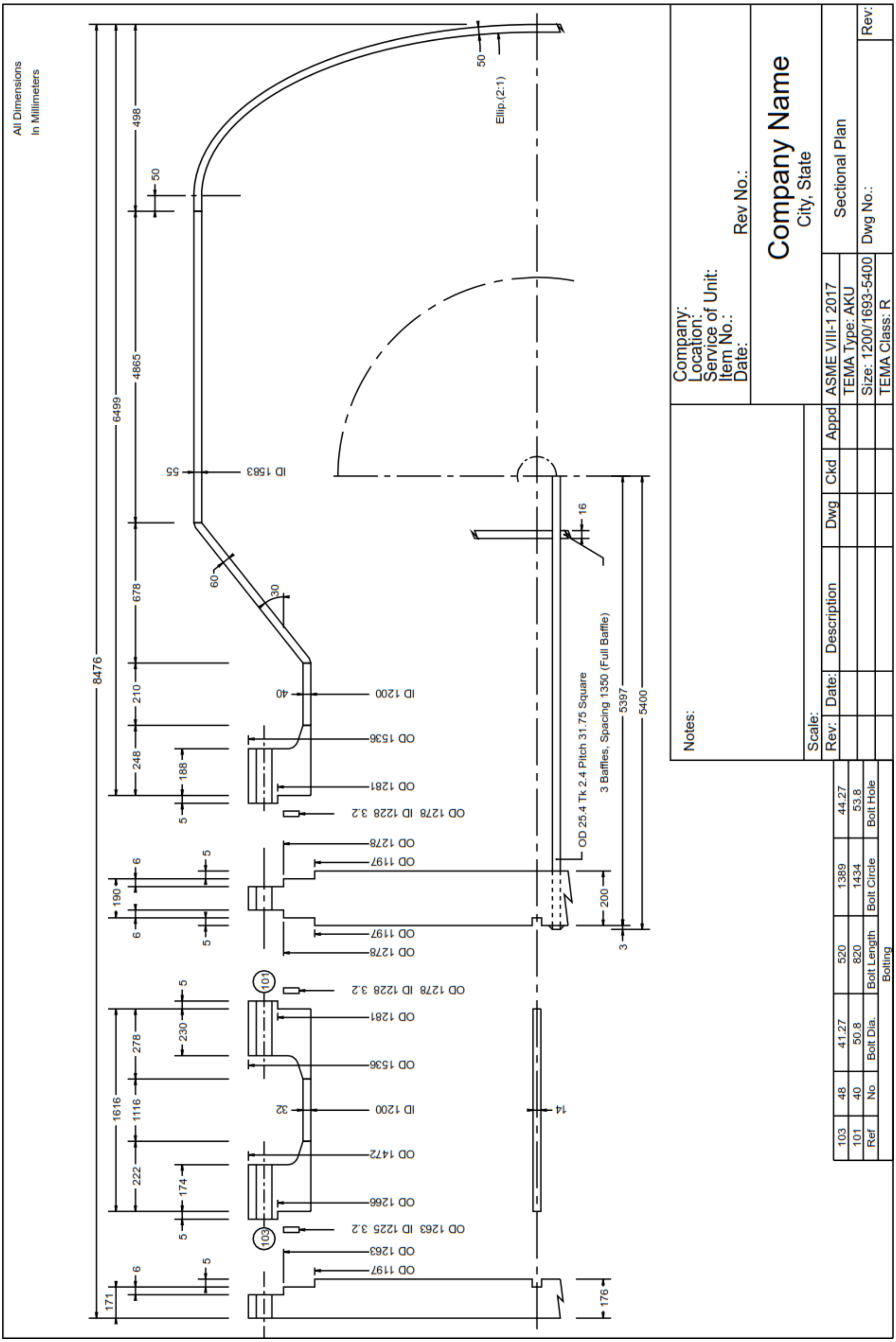
Dwg No.: Rev: Date:

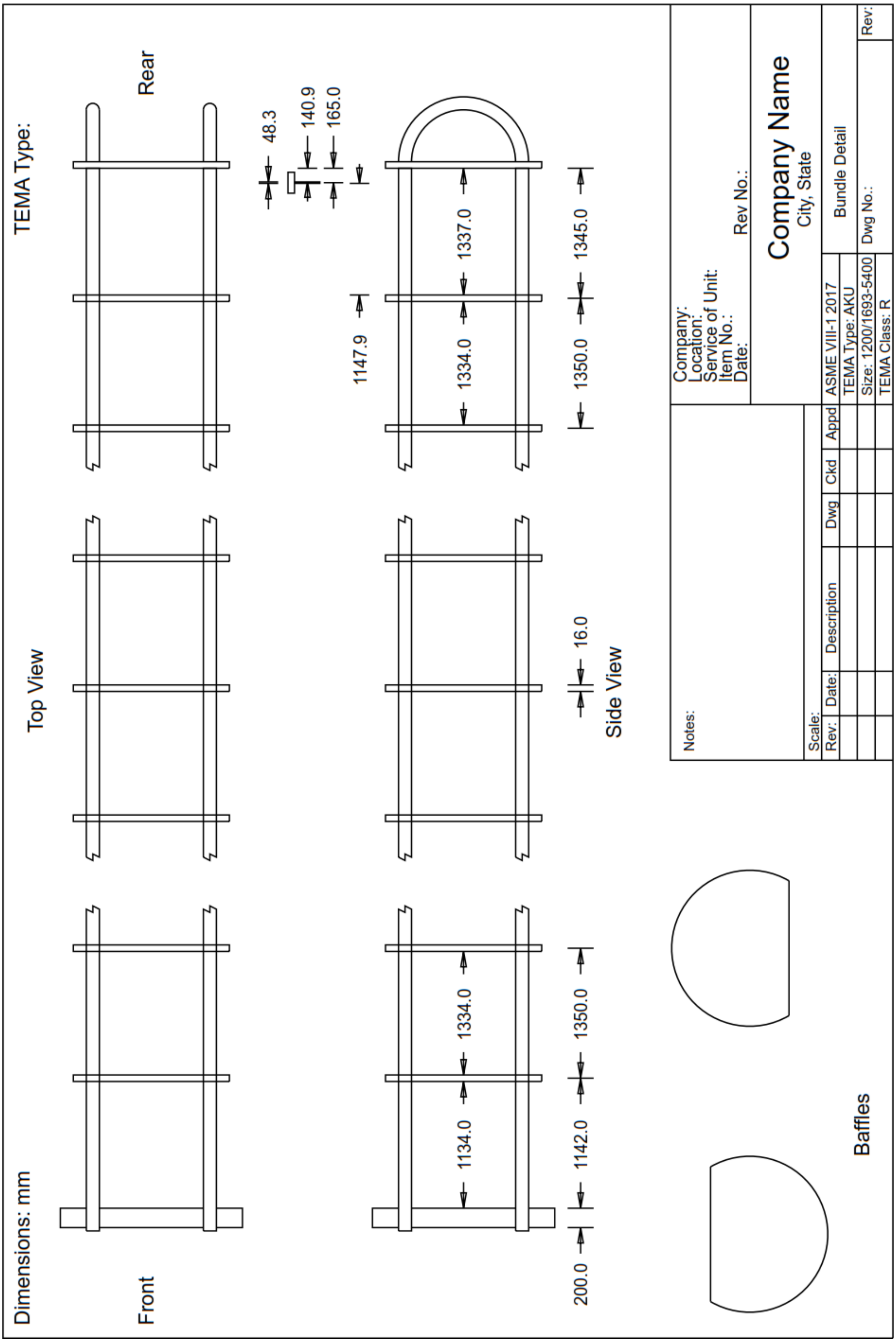
Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

Bill Of Material

Dwg No.: Rev:


All Dimensions
In Millimeters
Ellip.(2:1)
ID 1583
ID 1200
OD 1536
OD 1281
OD 1278 ID 1228 3.2
OD 1278
OD 1197
OD 1472
OD 1266
OD 1263 ID 1225 3.2
OD 1263
OD 25.4 Tk 2.4 Pitch 31.75 Square
3 Baffles, Spacing 1350 (Full Baffle)
8476
6499
498
4865
678
210
248
188
190
200
5397
5400
1616
1116
278
222
230
174
171
176
101
103
Notes:
Company:
Location:
Service of Unit:
Item No.:
Date:
Rev No.:
Company Name
City, State
Sectional Plan
Dwg No.:
Rev:
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Scale:
Rev: | Date: | Description | Dwg | Ckd | Appd
Bolting
Ref | No | Bolt Dia. | Bolt Length | Bolt Circle | Bolt Hole
101 | 40 | 50.8 | 820 | 1434 | 53.8
103 | 48 | 41.27 | 520 | 1389 | 44.27


TEMA Type:
Dimensions: mm
Rear
Front
Top View
Side View
48.3
140.9
165.0
1147.9
1337.0
1334.0
1345.0
1350.0
16.0
1334.0
1350.0
1134.0
1142.0
200.0
Baffles
Company:
Location:
Service of Unit:
Item No.:
Date:
Rev No.:
Company Name
City, State
Bundle Detail
Dwg No.:
Rev:
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Notes:
Scale:
Rev: Date: Description Dwg Ckd Appd

Row	Holes
36	4
35	12
34	18
33	20
32	24
31	26
30	28
29	28
28	30
27	32
26	32
25	34
24	34
23	36
22	36
21	36
20	36
19	36
18	36
17	36
16	36
15	36
14	36
13	34
12	34
11	32
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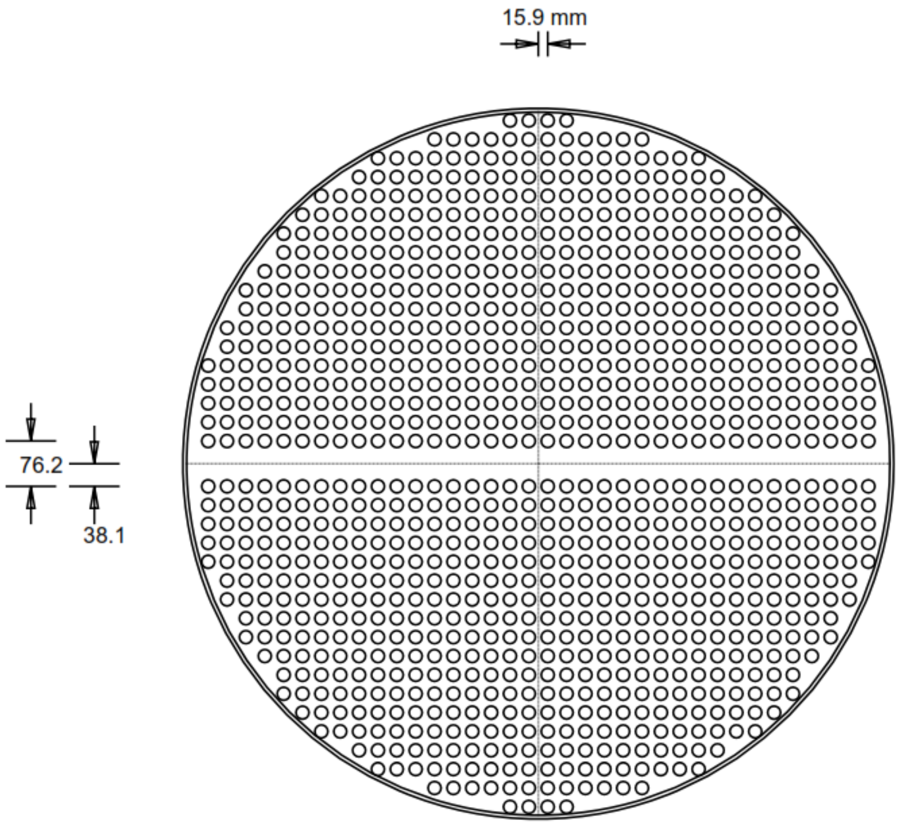
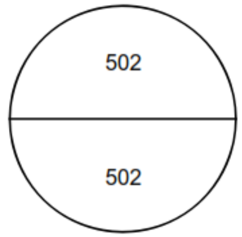
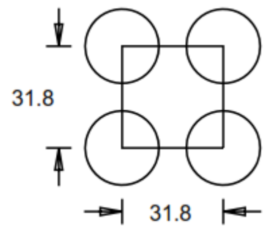
8	28
7	28
6	26
5	24
4	20
3	18
2	12
1	4
	1004



Shell ID	1200 mm
O.T.L.	1187.3 mm





Design Specifications		Tie Rod Locations
Number of Tube Holes	1004	
Tube Outside Diameter	25.4 mm	
Tube Pitch	31.8 mm	
Tube Pattern	Square	
Tube Passes	2	
Number of Tie Rods	0	
Tie Rod Diameter	12.7 mm	
Baffle Diameter	1193.7 mm	
Baffle Type	Full Supports	
Baffle Cut	0%	
Tube Thickness	2.4 mm	

Notes:

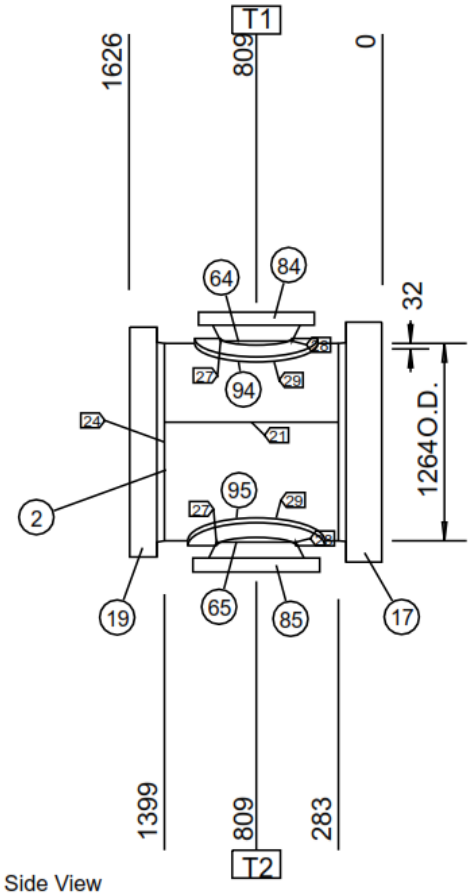
Scale:

Rev:	Date:	Description	Dwg	Ckd	Appd

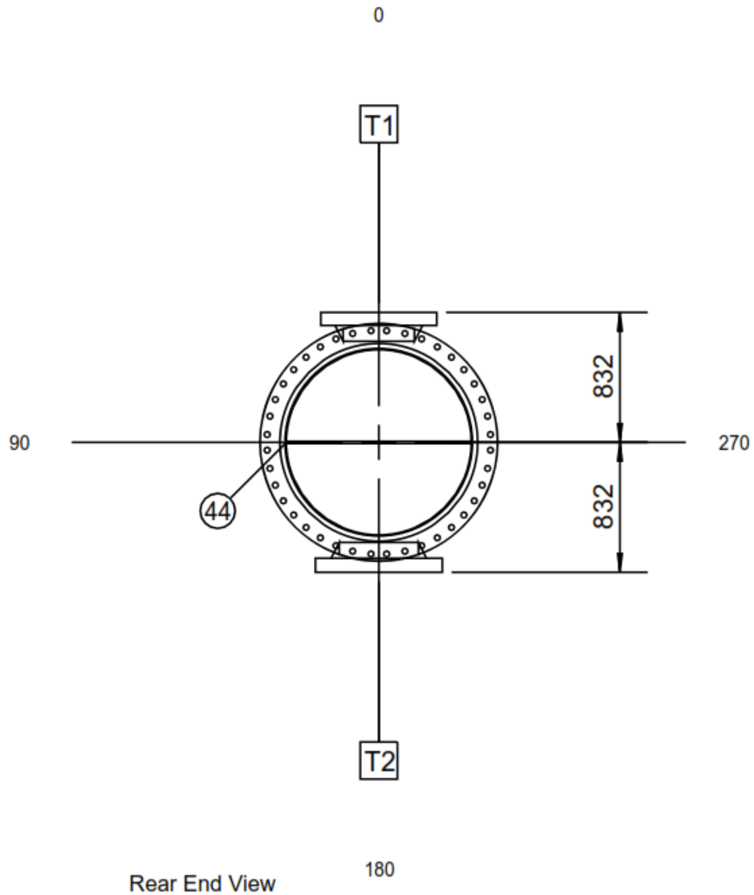
Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

ASME VIII-1 2017	Tube Layout	
TEMA Type: AKU		
Size: 1200/1693-5400	Dwg No.:	Rev:
TEMA Class: R		



Side View



Rear End View

Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines
Weld joint details per drawing 20

Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

Scale: 1:47

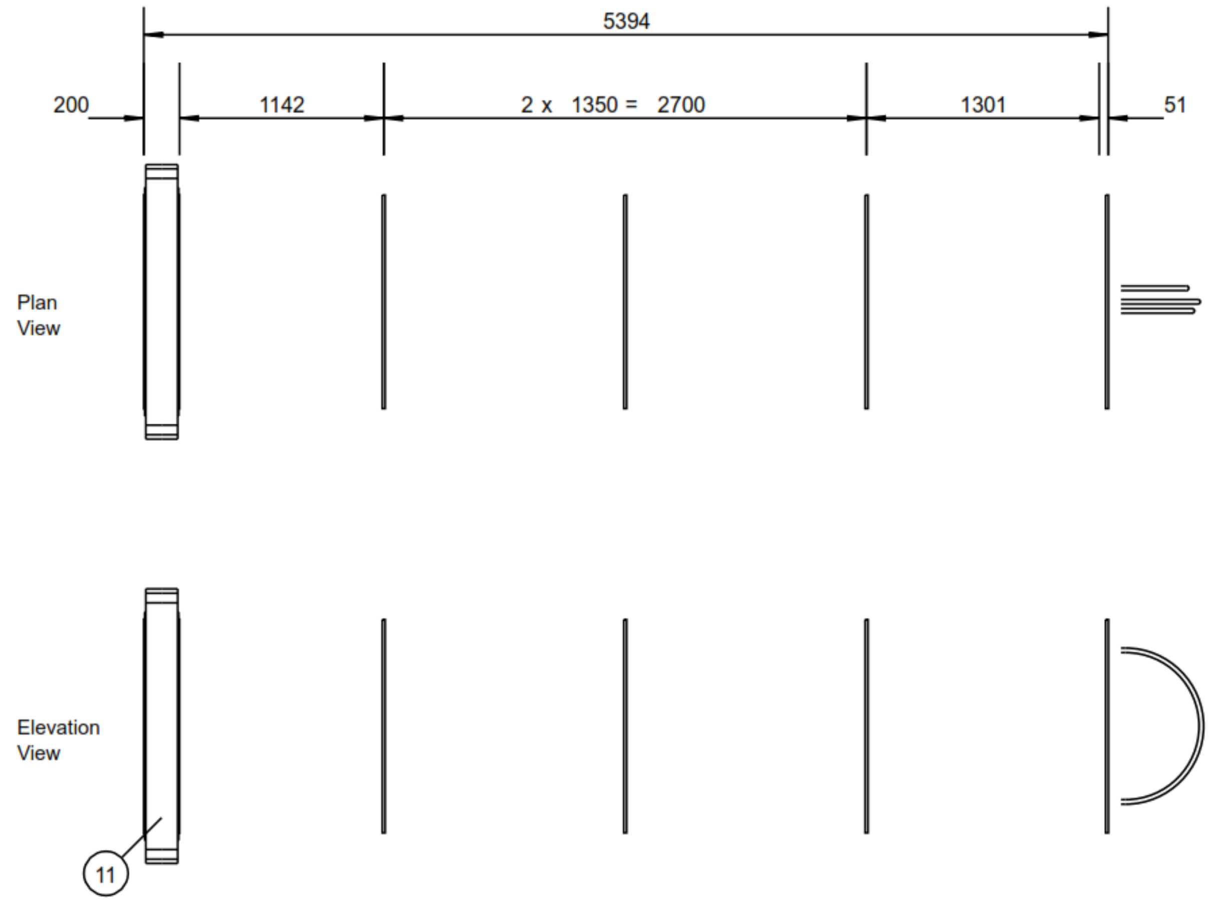
Rev:	Date:	Description	Dwg	Ckd	Appd

ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R

Front Head Detai

Dwg No.:

Rev:



Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines
Weld joint details per drawing 20

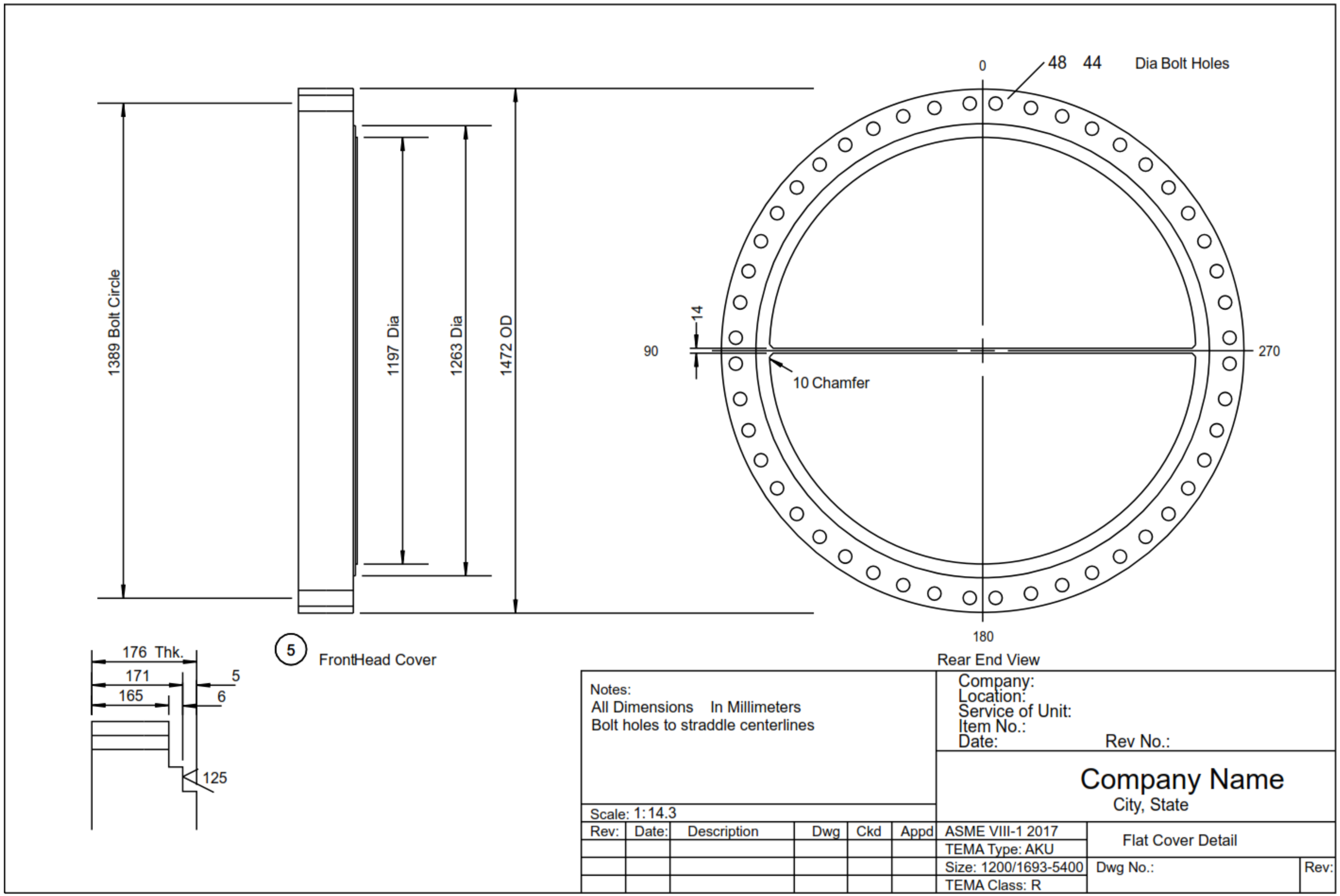
Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

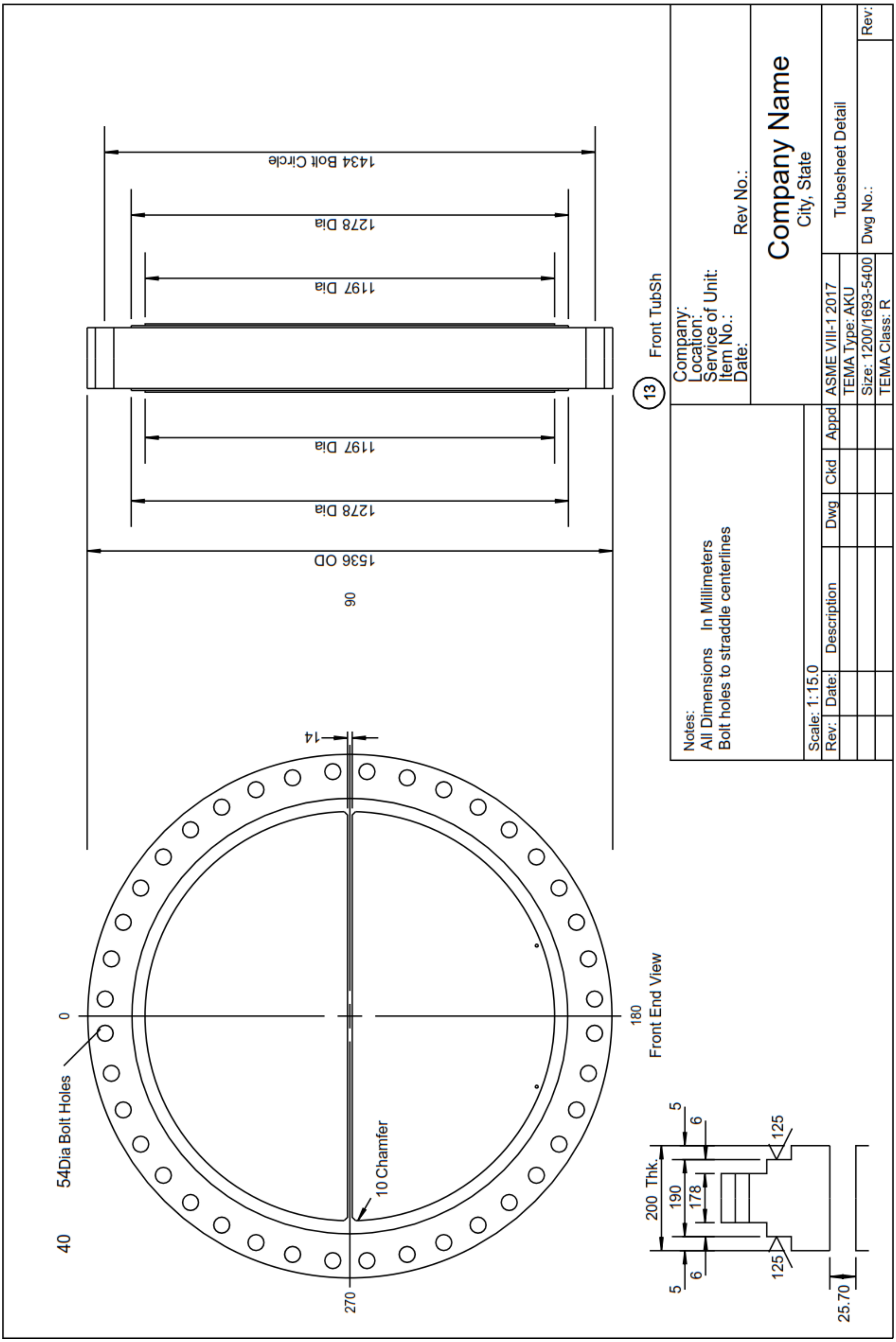
Company Name
City, State

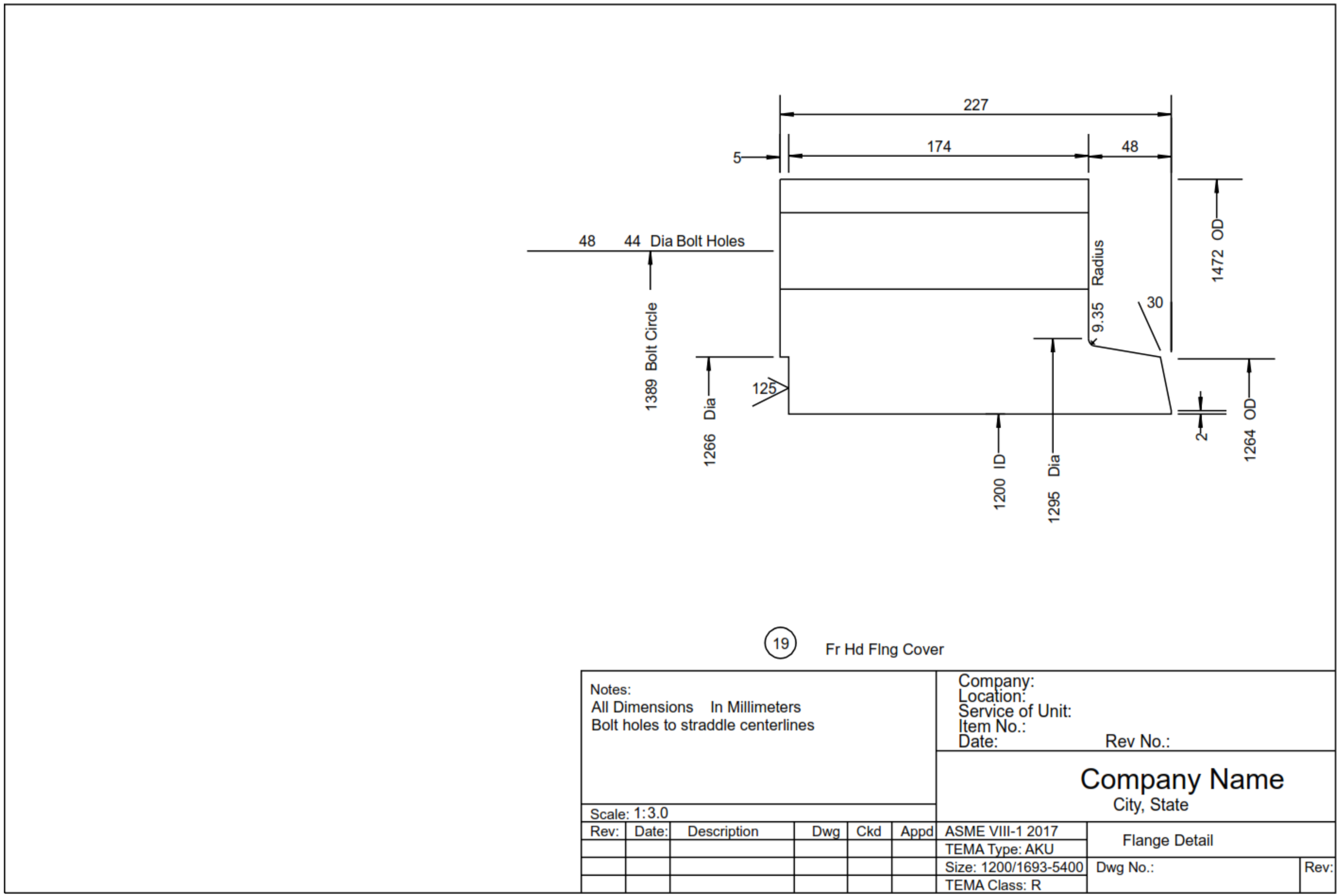
Scale: 1:41

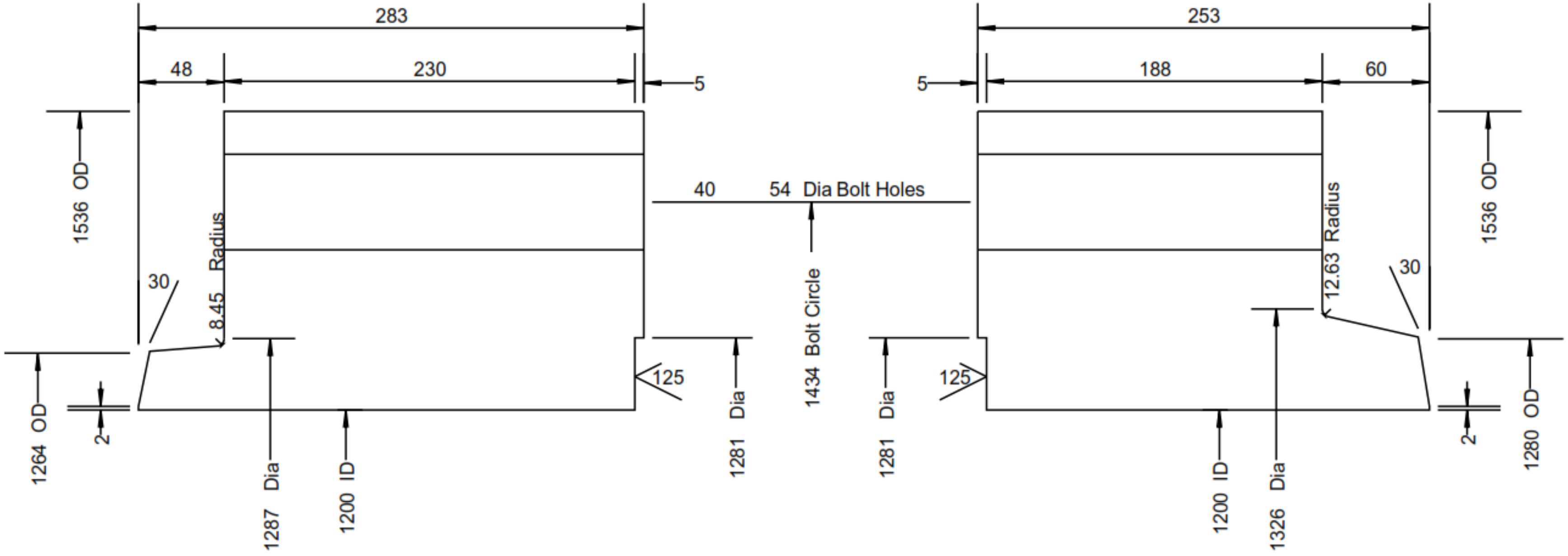
Rev:	Date:	Description	Dwg	Ckd	Appd

ASME VIII-1 2017	Bundle Detail	
TEMA Type: AKU		
Size: 1200/1693-5400	Dwg No.:	Rev:
TEMA Class: R		


48 44 Dia Bolt Holes
0
90
270
180
14
10 Chamfer
Rear End View
1389 Bolt Circle
1197 Dia
1263 Dia
1472 OD
5
FrontHead Cover
176 Thk.
171
165
5
6
125
Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines
Scale: 1:14.3
Rev: Date: Description Dwg Ckd Appd
Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:
Company Name
City, State
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Flat Cover Detail
Dwg No.:
Rev:


1434 Bolt Circle
1278 Dia
1197 Dia
1197 Dia
1278 Dia
1536 OD
90
14
0
180
270
40
54Dia Bolt Holes
Front End View
10 Chamfer
200 Thk.
190
178
5
6
125
125
6
5
25.70
13
Front TubSh
Company:
Location:
Service of Unit:
Item No.:
Date:
Rev No.:
Company Name
City, State
Tubesheet Detail
Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines
Scale: 1:15.0
Rev:
Date:
Description
Dwg
Ckd
Appd
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Dwg No.:
Rev:


227
5
174
48
48
44 Dia Bolt Holes
1389 Bolt Circle
1266 Dia
125
9.35 Radius
30
1472 OD
1200 ID
1295 Dia
2
1264 OD
19
Fr Hd Flng Cover
Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines
Scale: 1:3.0
Rev: Date: Description Dwg Ckd Appd
Company:
Location:
Service of Unit:
Item No.:
Date:
Rev No.:
Company Name
City, State
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Flange Detail
Dwg No.:
Rev:



17 Fr Hd Flng TubSh

21 Fr Shell Flng

Notes:
All Dimensions In Millimeters
Bolt holes to straddle centerlines

Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

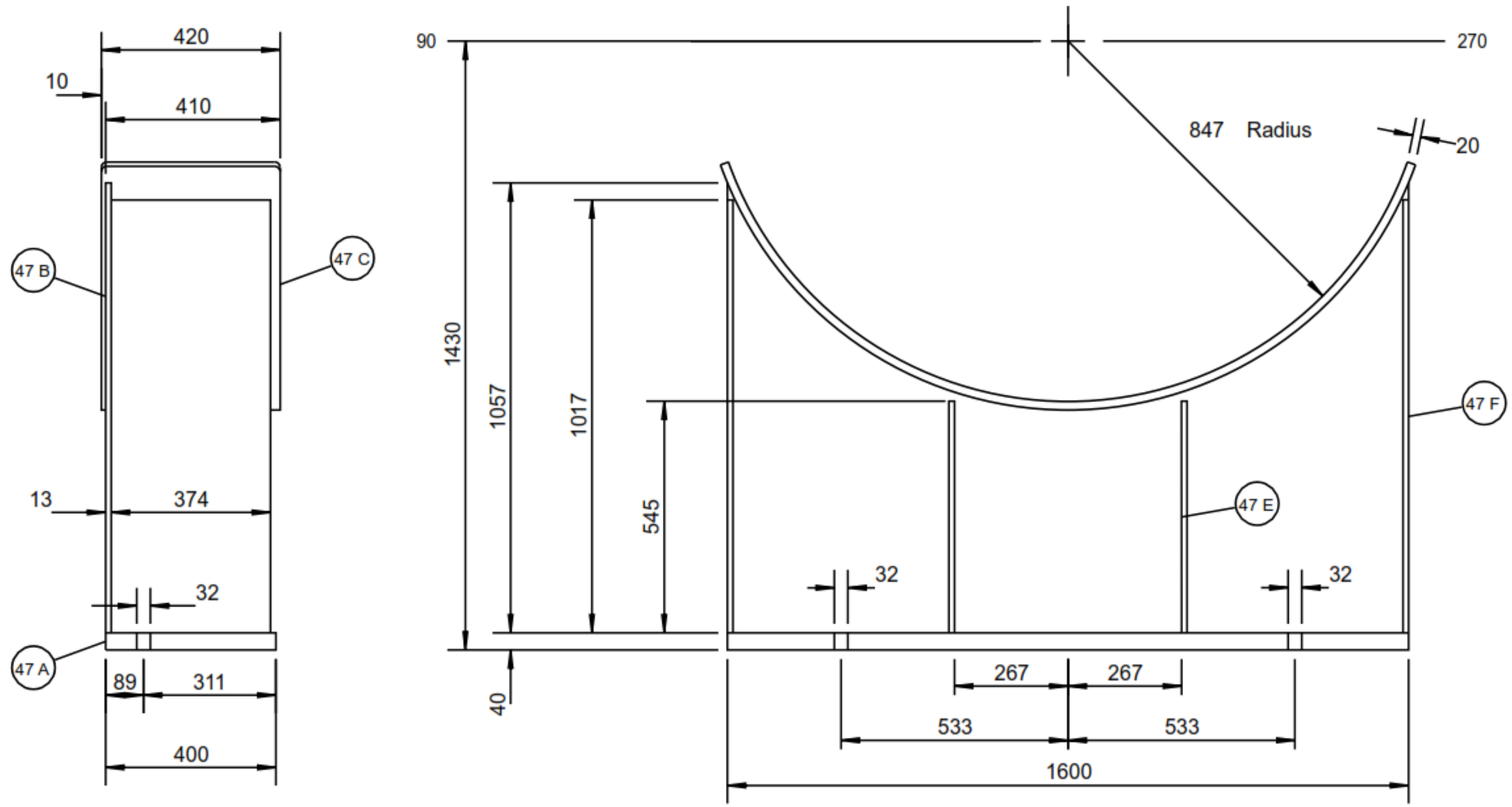
Scale: 1:3.7

Rev:	Date:	Description	Dwg	Ckd	Appd

ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R

Flange Detail

Dwg No.: Rev:



Notes:
All Dimensions In Millimeters

Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

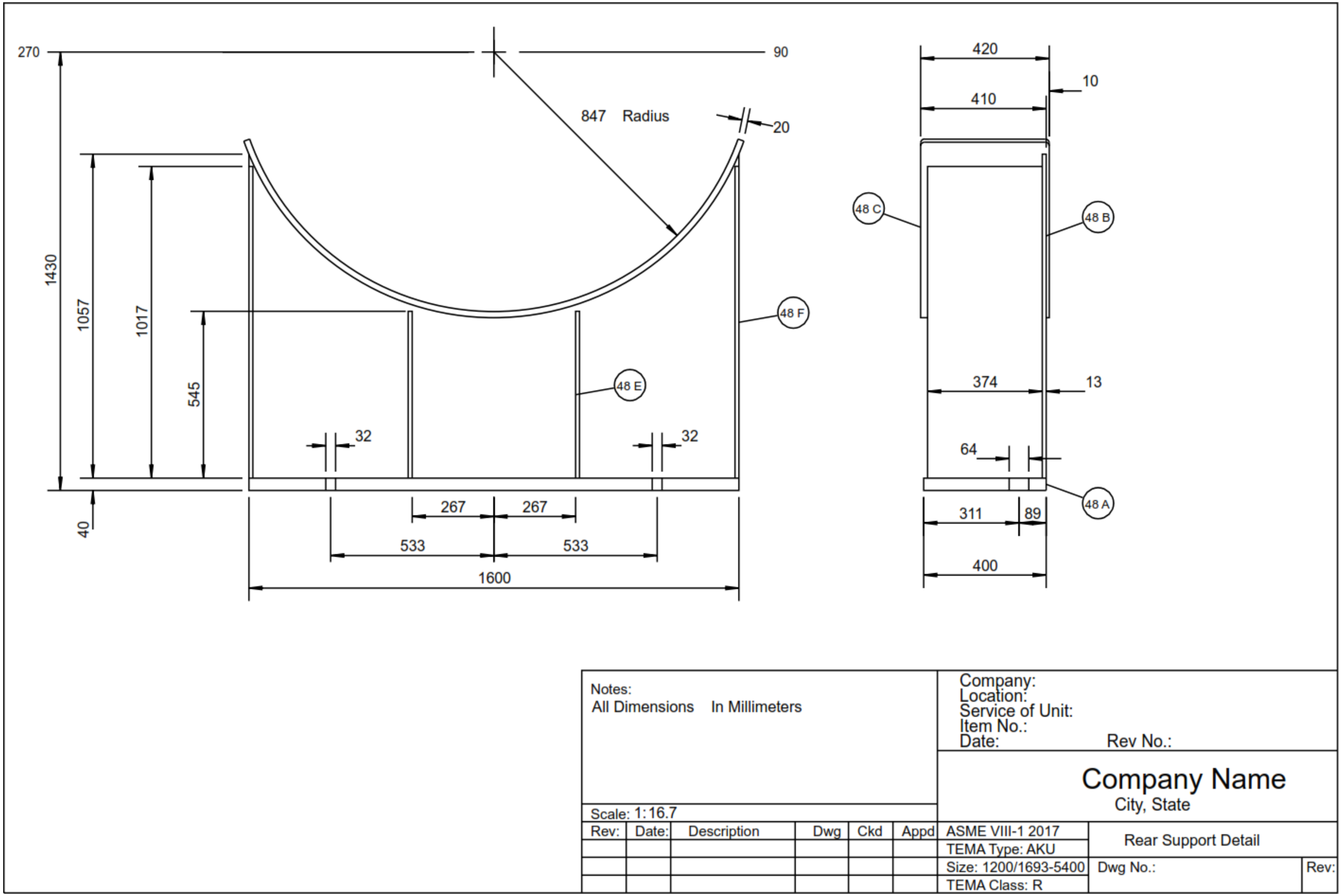
Scale: 1:16.7

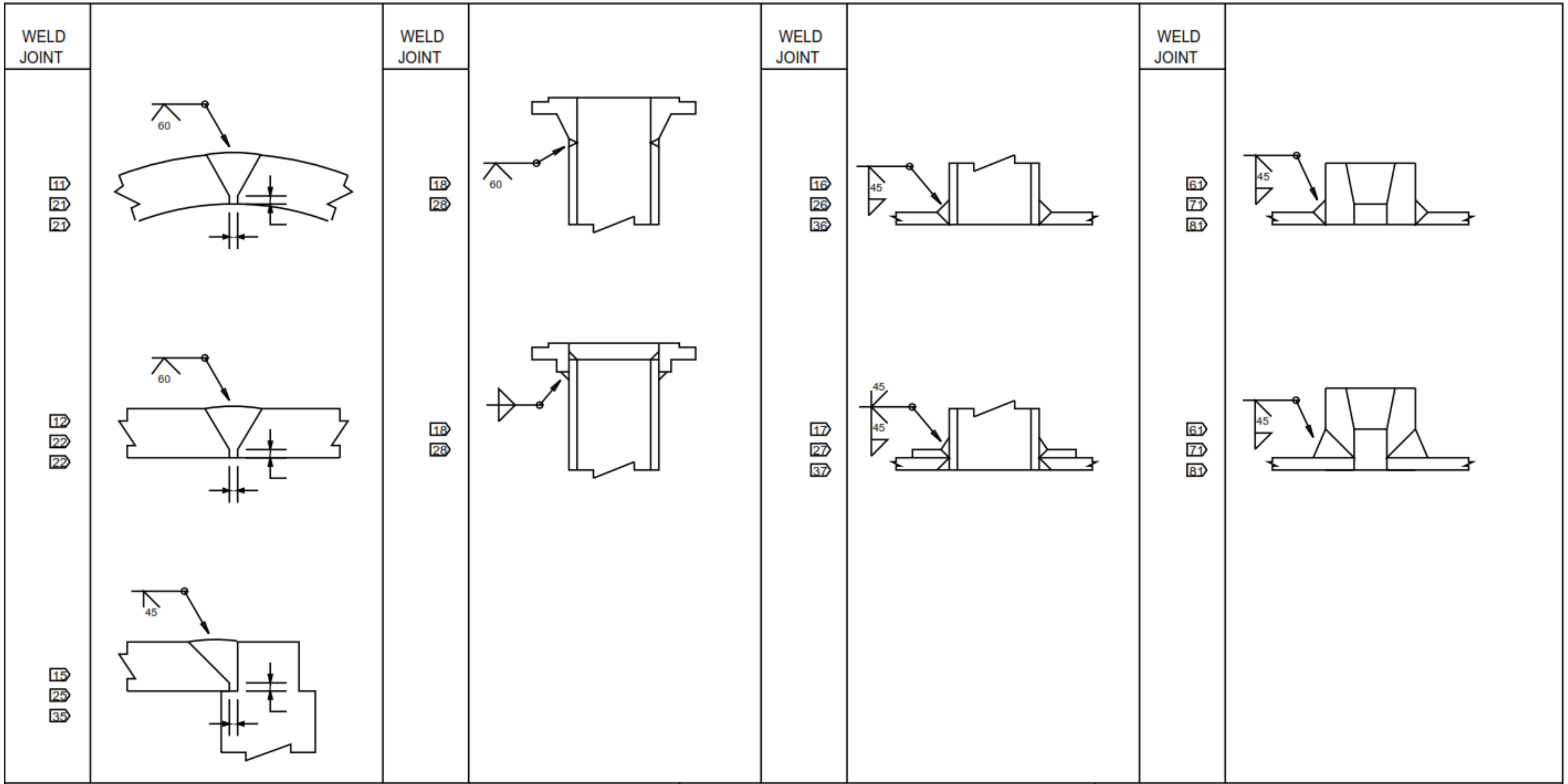
Rev:	Date:	Description	Dwg	Ckd	Appd

ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R

Front Support Detail

Dwg No.:
Rev:


270
90
847 Radius
20
1430
1057
1017
545
40
32
32
267
267
533
533
1600
48 E
48 F
420
410
10
48 C
48 B
374
13
64
48 A
311
89
400
Notes:
All Dimensions In Millimeters
Scale: 1:16.7
Rev: Date: Description Dwg Ckd Appd
Company:
Location:
Service of Unit:
Item No.:
Date:
Rev No.:
Company Name
City, State
ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R
Rear Support Detail
Dwg No.:
Rev:



Notes:
All Dimensions In Millimeters
Welding in accordance with Code

Company:
Location:
Service of Unit:
Item No.:
Date: Rev No.:

Company Name
City, State

Scale:

Rev:	Date:	Description	Dwg	Ckd	Appd

ASME VIII-1 2017
TEMA Type: AKU
Size: 1200/1693-5400
TEMA Class: R

Weld Joint Detail
Dwg No.:
Rev:

10. APPENDIX II: CALCULATION REPORT

Component: Shell Cylinder

```
ASME Section VIII-1 2017 UG-27 Thickness of Shells under Int. Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-516 K02700 Grd 70 Plate
Design pressure       P = 7.8 N/mm2     Design temperature   T = 280 C
Radiography             = Full          Joint eff.circ str.  E = 1
Design stress         S = 136.406 N/mm2 Joint eff.long str.  E = 1
Design stress, long   S = 136.406 N/mm2 Min thk. UG-16(b) tmin = 4.89 mm
Inside corr.allow.  CAI = 3 mm          Outside corr. all. CAO = 0 mm
Material tolerance  Tol = 0.3 mm        TEMA min. thickness tm = 12.7 mm
Outside diameter     OD = 1280 mm       Corroded radius     IR = 603 mm
Required wall thickness of the cylinder , greater of:
  Circumferential stress
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 39.01 mm    UG-27(c)(1)
  Longitudinal stress
            t = (P*IR / (2*S*E+0.4*P))+cai+cao+tol = 20.35 mm    UG-27(c)(2)
Actual wall thickness of cylinder:            tnom = 40 mm
  (Required wall tks. for nozzle attachments, E=1    ,  tri = 36.01 mm  )
```

Component: Front Head Cylinder

```
ASME Section VIII-1 2017 UG-27 Thickness of Shells under Int. Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-516 K02700 Grd 70 Plate
Design pressure       P = 6 N/mm2       Design temperature   T = 300 C
Radiography             = Full          Joint eff.circ str.  E = 1
Design stress         S = 134.917 N/mm2 Joint eff.long str.  E = 1
Design stress, long   S = 134.917 N/mm2 Min thk. UG-16(b) tmin = 4.89 mm
Inside corr.allow.  CAI = 3 mm          Outside corr. all. CAO = 0 mm
Material tolerance  Tol = 0.3 mm        TEMA min. thickness tm = 12.7 mm
Outside diameter     OD = 1264 mm       Corroded radius     IR = 603 mm
Required wall thickness of the cylinder , greater of:
  Circumferential stress
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 30.85 mm    UG-27(c)(1)
  Longitudinal stress
            t = (P*IR / (2*S*E+0.4*P))+cai+cao+tol = 16.59 mm    UG-27(c)(2)
Actual wall thickness of cylinder:            tnom = 32 mm
  (Required wall tks. for nozzle attachments, E=1    ,  tri = 27.85 mm  )
```

Component: Shell Cover

```
ASME Section VIII-1 2017 UG-32 Formed Heads, and Sections,
                                Pressure on Concave Side
--- Calculations --- Ellipsoidal Cover Internal Pressure with t/L >= 0.002
Material:  SA-516 K02700 Grd 70 Plate
Design pressure       P = 7.8 N/mm2      Design temperature  T = 280 C
Radiography             = Full           Joint efficiency    E = 1
Design stress         S = 136.406 N/mm2  TEMA min. thk      tm = 12.7 mm
                                         Min thk UG-16(b) tmin = 4.59 mm
Inside corr.all.    CAI = 3 mm           Outside corr.all. CAO = 0 mm
Major/minor rat.   D/2h = 2.0            Forming tolerance Tol = 0 mm
Corroded min. thk     t = 45.92 mm       Equiv.dish radius   L = 1437.3 mm
tnom-CAI-CAO-Tol     ts = 46.7 mm        Ratio ts/L       ts/L = 0.03249
K =0.1667*(2+(D/2h)**2) = 1.0            Material tol.     Tol = 0.3 mm
Outside diameter     OD = 1691 mm        Corroded diameter  ID = 1597 mm
Required wall thickness of the cover:
            t = (P*ID*K / (2*S*E-0.2*P))+cai+cao+tol = 49.22 mm   App. 1-4(c)
Actual wall thickness of cover:                 tnom = 50 mm
   (Required wall tks. for nozzle attachments, E=1    ,  tri = 46.22 mm  )
   (If opening & reinf. are within 80% of head diameter, tri = 41.63 mm  )
```

Component: Eccentric Reducer

```
ASME Section VIII-1 2017 UG-32 Formed Heads and Sections,
                                Pressure on Concave Side
--- Calculations --- Conical Cover Internal Pressure
Material:  SA-516 K02700 Grd 70 Plate
Design pressure       P = 7.8 N/mm2     Design temperature   T = 280 C
Radiography             = Full          Joint efficiency     E = 1
Design stress         S = 136.406 N/mm2 TEMA min. thickness tm = 12.7 mm
Inside corr.allow.  CAI = 3 mm          Outside corr. all. CAO = 0 mm
                                        Material tolerance Tol = 0.3 mm
Large outside dia.   Do = 1711 mm       Large inside dia.   ID = 1597 mm
Cone length           L = 678 mm        Conical angle    Alpha = 30 Deg
   Required wall thickness of the conical cover:
        t = (P*ID/(2*COS(Alpha)*(S*E-0.6*P))+cai+cao+tol = 57.9 mm   App.1-4(e)
   Actual wall thickness of cover:                    ta = 60 mm
```

Component: Eccentric Reducer

```
ASME Section VIII-1 2017 App.1-5 Rules for Conical Reducer Sect. and
                                   Conical Heads Under Internal Pressure
--- Design conditions
Design pressure     P = 7.8 N/mm2       Design temperature T = 280 C
Inside cor.allow. CAI = 3 mm            Outside corr.all.CAO = 0 mm
Large cylinder:                         Small cylinder:
SA-516 K02700 Grd 70 Plate              SA-516 K02700 Grd 70 Plate
Design stress      Ss = 136.41 N/mm2    Design stress     Ss = 136.41 N/mm2
Joint efficiency   E1 = 1               Joint efficiency  E1 = 1
Mod. of elasticity Es = 185110 N/mm2    Mod. of elast.    Es = 185110 N/mm2
Min len 2.0*SQRT(R*t) = 406.52 mm       Min len 1.4*SQRT(R*t)= 209.12 mm
Actual length         = 4865 mm         Actual length =       210 mm
Cone material: SA-516 K02700 Grd 70 Plate
Design stress      Sc = 136.4 N/mm2     Mod. of elast.    Ec = 185110 N/mm2
Joint efficiency   E2 = 1               Halfapex angle Alpha = 30
Reinforcement ring material: ---
Design stress  Sr = -              Mod. of elasticity    Er = -
                                        Large cyl.          Small cyl.
Inside corroded Radius             RL = 794.5 mm         RS = 603 mm
Axial force                        FL = 0 N              FS = 0 N
Bending moment                     ML = 0 N*m            MS = 0 N*m
f = F/(2*pi*R)+-M/(pi*R**2)
Axial load (wind+dead loads)       f1 = 0 N/mm           f2 = 0 N/mm
(P*RL /2.)+f1 & (P*RS/2.)+f2       QL = 3098.55 N/mm     QS = 2351.7 N/mm
Cylinder corroded thickness        ts = 52.0 mm          ts = 37.0 mm
Minimum required thk. of cylinder   t = 47.3 mm           t = 35.7 mm
Thickness of cone                  tc = 56.7 mm          tc = 56.7 mm
Minimum required thk. of cone      tr = 54.6 mm          tr = 41.2 mm
Pressure/stress ratio         P/Ss*E1 = 0.0572      P/Ss*E1 = 0.0572
Angle for reinforcement (deg)  DeltaL = 30           DeltaS = 20.29
Reinforcement required when Delta is less than Alpha
```

Component: Eccentric Reducer

```
Cross sectional area requirement cone to large cylinder junction:
Cone to cyl. factor (reinf. ring on cyl.) yL = Sl*Es           yL = 25250152
Reinforcement factor (minimum=1) kL = yL/Sr*Er      Calculated kL = 1
                                                       Minimum kL = 1
Area required for reinforcement element:
    ArL = (kL*QL*RL/(Sl*E1))*(1-DeltaL/Alpha)*tan(Alpha)      ArL = 0 mm2
Excess area available in large cylinder:
    AeL = (ts-t)*(RL*ts)**0.5+(tc-tr)*(RL*tc/Cos Alpha)**0.5  AeL = 1366.9 mm2
        *** Enough area available - reinforcing not required per area rules ***
Reinforcing element width:     w = -
Reinforcing element thickness: t = -
Reinforcing element area required = ArL - AeL = -
Reinforcing element area: Ar = w * t       Ar = -
Reinforcing element within this dist.from junction SQRT(RL*ts)    = -
Centroid reinf. elem. within this dist. junction 0.25*SQRT(RL*ts) = -
Cross sectional area requirement cone to small cylinder junction:
Cone to cyl. factor (reinf. ring on cyl.) yS = Ss*Es           yS = 25250152
Reinforcement factor (minimum=1) kS = yS/Sr*Er      Calculated kS = 1
                                                       Minimum kS = 1
Area required for reinforcement element:
   ArS = (kS*QS*RS/(Ss*E1)) * (1-DeltaS/Alpha) * tan(Alpha)   ArS = 1942.3 mm2
Excess area available in small cylinder:
   AeS = 0.78*(RS*ts)**0.5 * ( (ts-t)+(tc-tr)/Cos Alpha )     AeS = 2188.1 mm2
        *** Enough area available - reinforcing not required per area rules ***
Reinforcing element width:     w = -
Reinforcing element thickness: t = -
Reinforcing element area required = ArS - AeS = -
Reinforcing element area: Ar = w * t       Ar = -
                                                                               -
Reinforcing element within this dist.from junction SQRT(Rs*ts)    = -
Centroid reinf. elem. within this dist. junction 0.25*SQRT(Rs*ts) = -
```

Component: Kettle Cylinder

```
ASME Section VIII-1 2017 UG-27 Thickness of Shells under Int. Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-516 K02700 Grd 70 Plate
Design pressure       P = 7.8 N/mm2     Design temperature   T = 280 C
Radiography             = Full          Joint eff.circ str.  E = 1
Design stress         S = 136.406 N/mm2 Joint eff.long str.  E = 1
Design stress, long   S = 136.406 N/mm2 Min thk. UG-16(b) tmin = 4.89 mm
Inside corr.allow.  CAI = 3 mm          Outside corr. all. CAO = 0 mm
Material tolerance  Tol = 0.3 mm        TEMA min. thickness tm = 12.7 mm
Outside diameter     OD = 1693 mm       Corroded radius     OR = 846.5 mm
Required wall thickness of the cylinder , greater of:
  Circumferential stress
              t = (P*OR / (S*E+0.4*P))+cai+cao+tol = 50.62 mm    APP.1-1(A)
  Longitudinal stress
            t = (P*IR / (2*S*E+0.4*P))+cai+cao+tol = 25.76 mm    UG-27(c)(2)
Actual wall thickness of cylinder:            tnom = 55 mm
  (Required wall tks. for nozzle attachments, E=1    ,  tri = 47.62 mm  )
```

Component: Tubes

```
ASME Section VIII-1 2017 UG-27 Thickness of Shells under Int. Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-179 K01200 Smls. tube
Design pressure       P = 6 N/mm2       Design temperature   T = 300 C
Radiography             =  -            Joint eff.circ str.  E = 1
Design stress         S = 91.893 N/mm2  Joint eff.long str.  E = -
Design stress, long   S = -             Min thk. UG-16(b) tmin = -
Inside corr.allow.  CAI = 0 mm          Outside corr. all. CAO = 0 mm
Material tolerance  Tol = 0 mm          TEMA min. thickness tm = 0 mm
Outside diameter     OD = 25.4 mm       Corroded radius     OR = 12.7 mm
Required wall thickness of the cylinder , greater of:
  Circumferential stress
              t = (P*OR / (S*E+0.4*P))+cai+cao+tol = 0.81 mm     APP.1-1(A)
  Longitudinal stress
            t = (P*IR / (2*S*E+0.4*P))+cai+cao+tol = -           UG-27(c)(2)
Actual wall thickness of cylinder:            tnom = 2.4 mm
  (Required wall tks. for nozzle attachments, E=-    ,  tri = -         )
TEMA RCB-2.31 U-Bend Requirements - Minimum tube wall thk in the bent portion
   to = t1 * (1 + do / (4*R)) + c       to = 2.34 mm
Min. Code wall thk   t1 = 2.01 mm       Outside diameter    do = 25.4 mm
Min. Code wall thickness:               Corrosion allowance  c = 0 mm
     Internal press t1i = 0.81 mm       External pressure   t1e= 2.01 mm
Min. mean bend radius R = 38.1 mm
ASME Section VIII-1 2017 UG-28 Thickness of Shells under Ext. Pressure
--- Calculations --- Cylinder External Pressure
Material:  SA-179 K01200 Smls. tube
Design pressure      PE = 7.8 N/mm2      Design temperature  T = 300 C
Inside corr. allow. CAI = 0 mm           Corrosion allow.  CAO = 0 mm
Radiography             = -              Material tol.     Tol = 0 mm
Cyl. outside dia.    Do = 25.4 mm        Cylinder length EP  L = 5400 mm
                                         Max length EP    Lmax = 16256 mm
Nominal thickness  tnom = 2.4 mm         (tnom-CAI-CAO-Tol)  t = 2.4 mm
L/Do ratio          Ldo = 212.6          Do/t              Dot = 10.58
(2*S) or (0.9*yield) SE = -              Mod. of elasticity ME = 184255 N/mm2
A factor SII-D-FigG   A = 0.010377       B factor CS-1       B = 77.54
Max allowed external pressure: Pa = 4*B / (3*Dot)              = 9.7684 N/mm2
Actual external design pressure:                            PE = 7.8 N/mm2
   (Required cyl. tks. for nozzle attachments at PE, tre = 2 mm      )
```

Component: Tube-to-Tubesheet Welds

```
ASME Section VIII Div.1 2017  UW-20 Tube-To-Tubesheet Welds
--- Calculations ---        Fig UW-20.1 Sketch (d) Full Strength
Tubesheet material:  SA-266 K03506 Grd 2 Forgings
Tubesheet clad mtl:  -
Tubes material:      SA-179 K01200 Smls. tube
Allowable stress TubS St = 129.18 N/mm2 All. stress tubes    Sa = 91.893 N/mm2
Allowable stress weld Sw = 91.893 N/mm2 Tube OD              do = 25.4 mm
Tubes yield stress   Syt = 139.715 N/mm2Tube thickness        t = 2.4 mm
Design temperature TubSh = 300 C        Design temp. tubes      = 300 C
Appendix A weld size   a = 3.36 mm
Fillet weld leg       af = 3 mm         Groove weld leg      ag = 1.62 mm
Minimum length ac  acmin = 2.92 mm      Total length ac = af+ag = 4.62 mm
Fillet weld strength = Ff = 0.55*Pi*af*(do+0.67*af)*Sw       Ff = 13057 N
Groove weld strength = Fg = 0.85*Pi*ag*(do+0.67*ag)*Sw       Fg = 10529 N
Tube strength          Ft = Pi * t * (do - t) * Sa           Ft = 15936 N
Design Strength        Fd                                    Fd = 15936 N
Fillet weld strength, Ff = min (Ff, Ft)                      Ff = 13057 N
Groove weld strength, Fg = min (Fg, Ft)                      Fg = 10529 N
Weld strength factor fw = Sa / Sw                            fw = 1
Ratio fd = Fd / Ft                                           fd = 1
Ratio ff = 1 - Fg / (fd * Ft)                                ff = 0.339
Minimum required length of the weld leg(s), ar
ar = SQRT((0.75*do)**2 + 2.73*t*(do-t)*fw*fd*ff) - 0.75*do   ar = 1.3 mm
UW-18(d) - Allowable load on fillet/groove welds       Weld Leg = 4.62 mm
                Allowable Load = PI * do * Weld Leg * Sw * 0.55 = 18633 N
Maximum Allowable Axial Loads, Lmax
  Pressure only = LmaxP = Ft                                LmaxP = 15936 N
  Other loads   = LmaxO = 2*Ft                              LmaxO = 31872 N
  Total weld throat dimension = 3.27 mm
```

Component: Front Pass Partition

```
Pass Partition Plate Max. Allowed Pressure Differential (TEMA 2007 RCB-9.132)
Pass plate material: SA-516 K02700 Grd 70 Plate
  Thickness            t = 14 mm        Pressure drop  qa = 0.419 bar
  TEMA min thk      tmin = 12.7 mm
  Corrosion allowance  c = 0 mm         Minimum thickness, tm
  Design stress        S = 134.92 N/mm2 tm = b*SQRT((qa*B)/(1.5*S)) + c
Max. allowable pressure drop:  q = (1.5*S*((t-c)/b)**2)/B = see table below
```

Sides fixed	Dim a mm	Dim b mm	a/b	B factor	q N/mm2	tm mm	Selected
a & b	1616	1200	1.347	0.468	0.0589	11.8	
a	1616	1200	1.347	0.48	0.0574	12	
b	1616	1200	1.347	0.578	0.0477	13.1	*

Component: Front Head Flng At Cov

```
ASME Section VIII-1 2017 ------- App.2 Bolted Flange With Ring Type Gaskets
Flange type: Integral tapered hub - ASME fig.2-4(6)
Flange material: SA-266 K03506 Grd 2 Forgings
Int. design pressure  PI = 6 N/mm2      Design temperature     T = 300 C
Ext. design pressure  PE = 0 N/mm2      B1 = B+g1 or B+go     B1 = -
Inside corr. allow   CAI = 3 mm         Outside corr. all.   CAO = 0 mm
Stress (operating)   SFO = 129.18 N/mm2 Stress (atmos.)      SFA = 137.9 N/mm2
Outside diameter       A = 1472 mm      Inside spherical rad.  L = -
Inside diameter        B = 1206 mm      Hub thickness         g1 = 37.4 mm
Bolt circle diameter   C = 1389 mm      Hub tks. at attach.   go = 29 mm
Mean gasket diameter   G = 1247.47 mm   Weld leg/hub length    h = 48 mm
Hub to bolt circle     R = 54.1 mm      Bolt circle to OD      E = 41.5 mm
Flange thickness       t = 174 mm
Overlay thickness     OL = -
Gasket material: Flat Metal Jacket Fiber Stainless Steel
Gasket outside dia.  ODG = 1263 mm      Gasket width           N = 19 mm
Gasket thickness     tks = 3.17 mm      Gasket eff. width      b = 7.77 mm
Gasket seating stress  y = 62.05 N/mm2  Gasket factor          m = 3.750
Gasket rib seat. str. yr = 62.05 N/mm2  Gasket rib factor     mr = 3.750
Gasket unit stress    Sg = 120.77 N/mm2 W = 0 mm        factor f = 0 mm
Gasket rib length    Rib = 1225 mm      Seating width         bo = 9.5 mm
Gasket rib eff width  Br = 4.76 mm       (Table 2-5.2  facing 1a/1b  Col. II )
Bolt material: SA-193 G41400 Grd B7 Bolt(<= 2 1/2)
Bolt diameter         db = 41.27 mm     No. of bolts           n = 48
Bolt root area      Area = 1083.87 mm2  Sg = Ab*Sa/((Pi/4)*((do-f)**2-di**2))
Bsmax = 2*db+6*t/(m+0.5)                Actual bolt spacing   Bs = 90.9 mm
Max bolt spacing   BsMax = 328.2 mm     Min bolt spacing   BsMin = 88.9 mm
Cf = SQRT(Bs/Bsmax)   Cf = 0.526        Cf used               Cf = 1
Stress (operating)    SB = 172.37 N/mm2 Stress (atmos.)       SA = 172.37 N/mm2
Bolting calculations:
  Joint-contact compr. load  HP = 6.2832*b*G*PI*m+2*Br*mr*PI*RIB=    1632281 N
  Hydrostatic end force       H = 0.7854*G*G*PI                =    7333303 N
  Hydrostatic end force       H = 0.7854*G*G*PE                =          0 N
Operating conditions:
  Min. calc. bolt load      WM1 = HP+H                         =    8965584 N
  Min. used bolt load       WM1 = max of 2 mating flanges      =    8965584 N
Bolting up conditions:
Minimum bolt load           WM2 = 3.1416*b*G*y+Br*yr*RIB       =    2250832 N
Min. used bolt load         WM2 = max of 2 mating flanges      =    2250832 N
Required bolt area           AM = WM2/SA or WM1/SB             =  52013.9 mm2
Available bolt area          AB = No.Bolt*Area                 =  52025.7 mm2
Ratio of bolt areas       AB/AM = 1
Design bolt load              W = 0.5*(AM+AB)*SA               =    8966599 N
Minimum gasket width       NMIN = AB*SA/(6.283*y*G)            =     18.44 mm
Gasket compression stress  Gcst = AB*SA/((Pi*G*N)+(Br*RIB))    = 111.68 N/mm2
```

Loads: **Integral Flange Calculations**

```
Operating conditions:
  Hydrostatic end load       HD = 0.785*B*B*PI    =   6853884 N
  Hydrostatic end load       HDe= 0.785*B*B*PE    =         0 N
  Gasket load                HG = WM1-H           =   1632281 N
  Result. hydrostatic force  HT = H-HD            =    479419 N
  Result. hydrostatic force  HTe= He-HDe          =         0 N
Bolting up conditions:
  Gasket load                HG = W               =   8966599 N
Operating conditions:
  Hydrostatic lever arm      hd = R+0.5*g1        = 72.8 mm
  Gasket load lever arm      hg = (C-G)/2         = 70.77 mm
  Result. hydro. lever arm   ht = (R+g1+hg)/2     = 81.13 mm
Bolting up conditions:
  Gasket load lever arm      hg = (C-G)/2         = 70.77 mm
Operating conditions:
  Hydrostatic moment         MD = HD*hd           =     498963 N*m
  Gasket moment              MG = HG*hg           =     115512 N*m
  Result. hydro. moment      MT = HT*ht           =      38897 N*m
  Total operating moment    MOP = MD+MG+MT        =     653371 N*m
                  MOPe = HDe(hd-hg)+HTe(ht-hg)  =          0 N*m
Bolting up conditions:
  Bolt up moment           MATM = W*hg            =     634539 N*m
  Effective bolt moment      MB = MATM*SFO/SFA    =     594436 N*m
Total moment                 MO = MOP or MB       =     653371 N*m
Bolt spacing correction       M = MO*Cf           =     653371 N*m
 (TEMA 2007 RCB-11.23) Cf= 1
Flange shape constants:
       K = A/B           = 1.2206        ho = SQ(B*g0)    = 187.0134
       T = Fig.2-7.1     = 1.8306      h/ho = h/ho        = 0.2567
       Z = Fig.2-7.1     = 5.0835         F = Fig.2-7.2   = 0.8921
       Y = Fig.2-7.1     = 9.8551         V = Fig.2-7.3   = 0.4349
       U = Fig.2-7.1     = 10.8298        f = Fig.2-7.6   = 1.0
   g1/g0 = g1/g0         = 1.2897         e = F/ho        = 0.0048
       t =               = 174 mm
       d = U*ho*g0*g0/V  = 3916780    Alpha = t*e+1.0     = 1.83
    Beta = 1.333*t*e+1.0 = 2.1064     Gamma = Alpha/T     = 0.9997
   Delta = t*t*t/d       = 1.345     Lambda = Gamma+Delta = 2.3447
Stress calculations:                                   Allowable stress:
Long. hub     SH = (f*M)/(Lambda*g1**2*B)  =165.19 N/mm2 1.5*SFO =193.77 N/mm2
Radial        SR = Beta*M/(Lambda*t**2*B)  = 16.08 N/mm2     SFO =129.18 N/mm2
Tangential   ST1 = M*Y/(t**2*B)-(Z*SR)     = 94.63 N/mm2     SFO =129.18 N/mm2
  (greater)  ST2 = (SH+SR)/2 or (SH+ST1)/2 =   129 N/mm2     SFO =129.18 N/mm2
```

Component: Channel Cover

```
ASME Section VIII-1 2017 ------- UG-34 Unstayed Flat Heads and Covers
--- Calculations --- Channel Cover Internal Pressure
Material:  SA-266 K03506 Grd 2 Forgings
Design pressure        P = 6 N/mm2      Design temperature     T = 300 C
Stress (operating)    SO = 129.18 N/mm2 Stress (atmos.)       SA = 137.9 N/mm2
Corr. allowance      CAI = 3 mm
Outside diameter      OD = 1472 mm
Bolt circle diameter   C = 1389 mm
Mean gasket diameter   G = 1247.47 mm
Mod. of elasticity    ME = 183373 N/mm2
Required thickness     t = 170.78 mm    Nominal thickness     tn = 176 mm
Gasket load lever arm         hg = (C-G)/2         = 70.77 mm
ASME Section VIII-1 2017 ------- UG-34 Unstayed Flat Heads and Covers
--- Calculations --- Channel Cover Internal Pressure
Operating bolt load   Wo = 8965584 N    Factor 'C'             C = 0.3
Gasket seating load   Wb = 8966599 N    Joint efficiency       E = 1
Available bolt area   Ab = 52025.7 mm2  Factor x per UG-39(d)(2) = 1
Bolt stress (oper.)   Sb = 172.37 N/mm2 Nominal diameter      ND = 1200 mm
Code required cover thickness:
   Operating:  t = G*sqrt(x*(C*P/SO*E+1.9*Wo*hg/(SO*E*G**3)))    = 170.78 mm
   Bolting:    t = G*sqrt(x*(1.9*Wb*hg/(SA*E*G**3)))             = 83.72 mm
TEMA 2007 RCB-9.21 (cover deflection)
        Y = G*(0.0435*G**3*P+0.5*Sb*Ab*hg)/(ME*tn**3)            = 1.13 mm
     Ymax = 0.03 in. (0.076 mm) or ND/800 or User specified      = 1.5 mm
        t = (G*(0.0435*G**3*P+0.5*Sb*Ab*hg)/(ME*Ymax))**.333     = 155.18 mm
Thicknesses do NOT include corrosion or recesses.
```

Component: Front Head Flng At Cov

```
ASME Section VIII Div.1 2017, Appendix 2, 2-14 Flange Rigidity
--- Calculations ---
Operating moment,    Mo = 653371 N*m    Gasket seat. moment Ma = 634539 N*m
Factor VI            VI = 0.4349        Factor L             L = 2.3447
Mod. elast.design T  Ed = 183373 N/mm2  Mod.elast.atm. temp Ea = 201327 N/mm2
Thickness g0         g0 = 29 mm         Factor h0           h0 = 187 mm
Factor KI            KI = 0.3           Factor KL           KL = 0.2
Corrosion allowance  ca = 3 mm          Factor K             K = 1.2206
Thickness, T          T = 174 mm
Flange Rigidity
  Loose type flanges without hubs and optional flanges designed as loose type
    Gasket seating  J = 109.4 * Ma / (E * T ** 3 * Ln(K) * KL) = -
    Operating       J = 109.4 * Mo / (E * T ** 3 * Ln(K) * KL) = -
  Integral type flanges and optional type flanges designed as integral and
  Loose type flanges with hubs
    Gasket seating  J = 52.14 * Ma * VI/ (L*E*G0**2 * ho * KI) = 0.646
    Operating       J = 52.14 * Mo * VI/ (L*E*G0**2 * ho * KI) = 0.7303
  ASME appendix 2 calculation of hub thickness 'go' as a cylinder
Design pressure       P = 6 N/mm2       Allowable stress     S = 129.18 N/mm2
Outside radius       OR = -             Inside radius       IR = 603 mm
Joint efficiency      E = 1             Corr.Allow or OL     c = 3 mm
                                        Material tolerance tol = 0 mm
Min hub thk / small end = P*IR / (S*E - 0.6*P)+c+Tol                UG-27(c)(1)
                        = 31.81 mm
Hub thk / small end     = 32 mm
New thickness 'go       = 32 mm         New thickness 'g1'     = 40.4 mm
Corroded thickness 'go' = 29 mm         Corroded thk  'g1'     = 37.4 mm
```

Component: Front Head Flng At TS

```
ASME Section VIII-1 2017 ------- App.2 Bolted Flange With Ring Type Gaskets
Flange type: Integral tapered hub - ASME fig.2-4(6)
Flange material: SA-266 K03506 Grd 2 Forgings
Int. design pressure  PI = 6 N/mm2      Design temperature     T = 300 C
Ext. design pressure  PE = 0 N/mm2      B1 = B+g1 or B+go     B1 = -
Inside corr. allow   CAI = 3 mm         Outside corr. all.   CAO = 0 mm
Stress (operating)   SFO = 129.18 N/mm2 Stress (atmos.)      SFA = 137.9 N/mm2
Outside diameter       A = 1536 mm      Inside spherical rad.  L = -
Inside diameter        B = 1206 mm      Hub thickness         g1 = 33.8 mm
Bolt circle diameter   C = 1434 mm      Hub tks. at attach.   go = 29 mm
Mean gasket diameter   G = 1260.18 mm   Weld leg/hub length    h = 48 mm
Hub to bolt circle     R = 80.2 mm      Bolt circle to OD      E = 51 mm
Flange thickness       t = 230 mm
Overlay thickness     OL = -
Gasket material: Flat Metal Jacket Fiber Stainless Steel
Gasket outside dia.  ODG = 1278 mm      Gasket width           N = 25 mm
Gasket thickness     tks = 3.17 mm      Gasket eff. width      b = 8.91 mm
Gasket seating stress  y = 62.05 N/mm2  Gasket factor          m = 3.750
Gasket rib seat. str. yr = 62.05 N/mm2  Gasket rib factor     mr = 3.750
Gasket unit stress    Sg = 119.87 N/mm2                 factor f = 0 mm
Gasket rib length    Rib = 1228 mm      Seating width         bo = 12.5 mm
Gasket rib eff width  Br = 4.76 mm       (Table 2-5.2  facing 1a/1b  Col. II )
Bolt material: SA-193 G41400 Grd B7 Bolt(<= 2 1/2)
Bolt diameter         db = 50.8 mm      No. of bolts           n = 40
Bolt root area      Area = 1710.96 mm2  Sg = Ab*Sa/((Pi/4)*((do-f)**2-di**2))
Bsmax = 2*db+6*t/(m+0.5)                Actual bolt spacing   Bs = 112.6 mm
Max bolt spacing   BsMax = 426.3 mm     Min bolt spacing   BsMin = 107.9 mm
Cf = SQRT(Bs/Bsmax)   Cf = 0.514        Cf used               Cf = 1
Stress (operating)    SB = 172.37 N/mm2 Stress (atmos.)       SA = 172.37 N/mm2
Bolting calculations:
  Joint-contact compr. load  HP = 6.2832*b*G*PI*m+2*Br*mr*PI*RIB=    1850400 N
  Hydrostatic end force       H = 0.7854*G*G*PI                =    7483561 N
  Hydrostatic end force       H = 0.7854*G*G*PE                =          0 N
Operating conditions:
  Min. calc. bolt load      WM1 = HP+H                         =    9333961 N
  Min. used bolt load       WM1 = max of 2 mating flanges      =   11792021 N
Bolting up conditions:
Minimum bolt load           WM2 = 3.1416*b*G*y+Br*yr*RIB       =    2551607 N
Min. used bolt load         WM2 = max of 2 mating flanges      =    2551607 N
Required bolt area           AM = WM2/SA or WM1/SB             =  68411.5 mm2
Available bolt area          AB = No.Bolt*Area                 =  68438.6 mm2
Ratio of bolt areas       AB/AM = 1
Design bolt load              W = 0.5*(AM+AB)*SA               =   11794352 N
Minimum gasket width       NMIN = AB*SA/(6.283*y*G)            =     24.01 mm
Gasket compression stress  Gcst = AB*SA/((Pi*G*N)+(Br*RIB))    = 112.54 N/mm2
```

```
Loads:                        Integral Flange Calculations

Operating conditions:
  Hydrostatic end load       HD = 0.785*B*B*PI    =   6853884 N
  Hydrostatic end load       HDe= 0.785*B*B*PE    =         0 N
  Gasket load                HG = WM1-H           =   4308460 N
  Result. hydrostatic force  HT = H-HD            =    629677 N
  Result. hydrostatic force  HTe= He-HDe          =         0 N
Bolting up conditions:
  Gasket load                HG = W               =  11794352 N
Operating conditions:
  Hydrostatic lever arm      hd = R+0.5*g1        = 97.1 mm
  Gasket load lever arm      hg = (C-G)/2         = 86.91 mm
  Result. hydro. lever arm   ht = (R+g1+hg)/2     = 100.45 mm
Bolting up conditions:
  Gasket load lever arm      hg = (C-G)/2         = 86.91 mm
Operating conditions:
  Hydrostatic moment         MD = HD*hd           =     665512 N*m
  Gasket moment              MG = HG*hg           =     374445 N*m
  Result. hydro. moment      MT = HT*ht           =      63254 N*m
  Total operating moment    MOP = MD+MG+MT        =    1103211 N*m
                  MOPe = HDe(hd-hg)+HTe(ht-hg)  =          0 N*m
Bolting up conditions:
  Bolt up moment           MATM = W*hg            =    1025039 N*m
  Effective bolt moment      MB = MATM*SFO/SFA    =     960257 N*m
Total moment                 MO = MOP or MB       =    1103211 N*m
Bolt spacing correction       M = MO*Cf           =    1103211 N*m
 (TEMA 2007 RCB-11.23) Cf= 1
Flange shape constants:
       K = A/B           = 1.2736        ho = SQ(B*g0)    = 187.0134
       T = Fig.2-7.1     = 1.8086      h/ho = h/ho        = 0.2567
       Z = Fig.2-7.1     = 4.2147         F = Fig.2-7.2   = 0.8979
       Y = Fig.2-7.1     = 8.1651         V = Fig.2-7.3   = 0.4744
       U = Fig.2-7.1     = 8.9726         f = Fig.2-7.6   = 1.0
   g1/g0 = g1/g0         = 1.1655         e = F/ho        = 0.0048
       t =               = 230 mm
       d = U*ho*g0*g0/V  = 2974446    Alpha = t*e+1.0     = 2.1043
    Beta = 1.333*t*e+1.0 = 2.472      Gamma = Alpha/T     = 1.1635
   Delta = t*t*t/d       = 4.0905    Lambda = Gamma+Delta = 5.254
Stress calculations:                                     Allowable stress:
Long. hub     SH = (f*M)/(Lambda*g1**2*B)  = 152.4 N/mm2 1.5*SFO =193.77 N/mm2
Radial        SR = Beta*M/(Lambda*t**2*B)  =  8.14 N/mm2     SFO =129.18 N/mm2
Tangential   ST1 = M*Y/(t**2*B)-(Z*SR)     = 106.9 N/mm2     SFO =129.18 N/mm2
  (greater)  ST2 = (SH+SR)/2 or (SH+ST1)/2 =   129 N/mm2     SFO =129.18 N/mm2
```

Component: Front Head Flng At TS

```
ASME Section VIII Div.1 2017, Appendix 2, 2-14 Flange Rigidity
--- Calculations ---
Operating moment,    Mo = 1103211 N*m   Gasket seat. moment Ma = 1025039 N*m
Factor VI            VI = 0.4744        Factor L             L = 5.254
Mod. elast.design T  Ed = 183373 N/mm2  Mod.elast.atm. temp Ea = 201327 N/mm2
Thickness g0         g0 = 29 mm         Factor h0           h0 = 187 mm
Factor KI            KI = 0.3           Factor KL           KL = 0.2
Corrosion allowance  ca = 3 mm          Factor K             K = 1.2736
Thickness, T          T = 230 mm
Flange Rigidity
  Loose type flanges without hubs and optional flanges designed as loose type
    Gasket seating  J = 109.4 * Ma / (E * T ** 3 * Ln(K) * KL) = -
    Operating       J = 109.4 * Mo / (E * T ** 3 * Ln(K) * KL) = -
  Integral type flanges and optional type flanges designed as integral and
  Loose type flanges with hubs
    Gasket seating  J = 52.14 * Ma * VI/ (L*E*G0**2 * ho * KI) = 0.5081
    Operating       J = 52.14 * Mo * VI/ (L*E*G0**2 * ho * KI) = 0.6003
  ASME appendix 2 calculation of hub thickness 'go' as a cylinder
Design pressure       P = 6 N/mm2       Allowable stress     S = 129.18 N/mm2
Outside radius       OR = -             Inside radius       IR = 603 mm
Joint efficiency      E = 1             Corr.Allow or OL     c = 3 mm
                                        Material tolerance tol = 0 mm
Min hub thk / small end = P*IR / (S*E - 0.6*P)+c+Tol                UG-27(c)(1)
                        = 31.81 mm
Hub thk / small end     = 32 mm
New thickness 'go       = 32 mm         New thickness 'g1'     = 36.8 mm
Corroded thickness 'go' = 29 mm         Corroded thk  'g1'     = 33.8 mm
```

Component: Front Shell Flng

```
ASME Section VIII-1 2017 ------- App.2 Bolted Flange With Ring Type Gaskets
Flange type: Integral tapered hub - ASME fig.2-4(6)
Flange material: SA-266 K03506 Grd 2 Forgings
Int. design pressure  PI = 7.8 N/mm2    Design temperature     T = 280 C
Ext. design pressure  PE = 0 N/mm2      B1 = B+g1 or B+go     B1 = -
Inside corr. allow   CAI = 3 mm         Outside corr. all.   CAO = 0 mm
Stress (operating)   SFO = 132.16 N/mm2 Stress (atmos.)      SFA = 137.9 N/mm2
Outside diameter       A = 1536 mm      Inside spherical rad.  L = -
Inside diameter        B = 1206 mm      Hub thickness         g1 = 50.5 mm
Bolt circle diameter   C = 1434 mm      Hub tks. at attach.   go = 37 mm
Mean gasket diameter   G = 1260.18 mm   Weld leg/hub length    h = 60 mm
Hub to bolt circle     R = 63.5 mm      Bolt circle to OD      E = 51 mm
Flange thickness       t = 188 mm
Overlay thickness     OL = -
Gasket material: Flat Metal Jacket Fiber Stainless Steel
Gasket outside dia.  ODG = 1278 mm      Gasket width           N = 25 mm
Gasket thickness     tks = 3.17 mm      Gasket eff. width      b = 8.91 mm
Gasket seating stress  y = 62.05 N/mm2  Gasket factor          m = 3.750
Gasket rib seat. str. yr = 62.05 N/mm2  Gasket rib factor     mr = 3.750
Gasket unit stress    Sg = 119.87 N/mm2                 factor f = 0 mm
Gasket rib length    Rib = 0 mm         Seating width         bo = 12.5 mm
Gasket rib eff width  Br = 0 mm          (Table 2-5.2  facing 1a/1b  Col. II )
Bolt material: SA-193 G41400 Grd B7 Bolt(<= 2 1/2)
Bolt diameter         db = 50.8 mm      No. of bolts           n = 40
Bolt root area      Area = 1710.96 mm2  Sg = Ab*Sa/((Pi/4)*((do-f)**2-di**2))
Bsmax = 2*db+6*t/(m+0.5)                Actual bolt spacing   Bs = 112.6 mm
Max bolt spacing   BsMax = 367 mm       Min bolt spacing   BsMin = 107.9 mm
Cf = SQRT(Bs/Bsmax)   Cf = 0.554        Cf used               Cf = 1
Stress (operating)    SB = 172.37 N/mm2 Stress (atmos.)       SA = 172.37 N/mm2
Bolting calculations:
  Joint-contact compr. load  HP = 6.2832*b*G*PI*m+2*Br*mr*PI*RIB=    2063392 N
  Hydrostatic end force       H = 0.7854*G*G*PI                =    9728629 N
  Hydrostatic end force       H = 0.7854*G*G*PE                =          0 N
Operating conditions:
  Min. calc. bolt load      WM1 = HP+H                         =   11792021 N
  Min. used bolt load       WM1 = max of 2 mating flanges      =   11792021 N
Bolting up conditions:
Minimum bolt load           WM2 = 3.1416*b*G*y+Br*yr*RIB       =    2188700 N
Min. used bolt load         WM2 = max of 2 mating flanges      =    2551607 N
Required bolt area           AM = WM2/SA or WM1/SB             =  68411.5 mm2
Available bolt area          AB = No.Bolt*Area                 =  68438.6 mm2
Ratio of bolt areas       AB/AM = 1
Design bolt load              W = 0.5*(AM+AB)*SA               =   11794352 N
Minimum gasket width       NMIN = AB*SA/(6.283*y*G)            =     24.01 mm
Gasket compression stress  Gcst = AB*SA/((Pi*G*N)+(Br*RIB))    = 119.19 N/mm2
```

Loads: **Integral Flange Calculations**

```
Operating conditions:
  Hydrostatic end load       HD = 0.785*B*B*PI    =   8910049 N
  Hydrostatic end load       HDe= 0.785*B*B*PE    =         0 N
  Gasket load                HG = WM1-H           =   2063392 N
  Result. hydrostatic force  HT = H-HD            =    818580 N
  Result. hydrostatic force  HTe= He-HDe          =         0 N
Bolting up conditions:
  Gasket load                HG = W               =  11794352 N
Operating conditions:
  Hydrostatic lever arm      hd = R+0.5*g1        = 88.75 mm
  Gasket load lever arm      hg = (C-G)/2         = 86.91 mm
  Result. hydro. lever arm   ht = (R+g1+hg)/2     = 100.45 mm
Bolting up conditions:
  Gasket load lever arm      hg = (C-G)/2         = 86.91 mm
Operating conditions:
  Hydrostatic moment         MD = HD*hd           =     790767 N*m
  Gasket moment              MG = HG*hg           =     179328 N*m
  Result. hydro. moment      MT = HT*ht           =      82230 N*m
  Total operating moment    MOP = MD+MG+MT        =    1052325 N*m
                   MOPe = HDe(hd-hg)+HTe(ht-hg)  =          0 N*m
Bolting up conditions:
  Bolt up moment           MATM = W*hg            =    1025039 N*m
  Effective bolt moment      MB = MATM*SFO/SFA    =     982397 N*m
Total moment                 MO = MOP or MB       =    1052325 N*m
Bolt spacing correction       M = MO*Cf           =    1052325 N*m
 (TEMA 2007 RCB-11.23) Cf= 1
Flange shape constants:
       K = A/B           = 1.2736        ho = SQ(B*g0)    = 211.2392
       T = Fig.2-7.1     = 1.8086      h/ho = h/ho        = 0.284
       Z = Fig.2-7.1     = 4.2147         F = Fig.2-7.2   = 0.886
       Y = Fig.2-7.1     = 8.1651         V = Fig.2-7.3   = 0.4063
       U = Fig.2-7.1     = 8.9726         f = Fig.2-7.6   = 1.0
   g1/g0 = g1/g0         = 1.3649         e = F/ho        = 0.0042
       t =               = 188 mm
       d = U*ho*g0*g0/V  = 6386791    Alpha = t*e+1.0     = 1.7886
    Beta = 1.333*t*e+1.0 = 2.0512     Gamma = Alpha/T     = 0.9889
   Delta = t*t*t/d       = 1.0404    Lambda = Gamma+Delta = 2.0293
Stress calculations:                                     Allowable stress:
Long. hub     SH = (f*M)/(Lambda*g1**2*B)  = 168.6 N/mm2 1.5*SFO =198.24 N/mm2
Radial        SR = Beta*M/(Lambda*t**2*B)  = 24.95 N/mm2     SFO =132.16 N/mm2
Tangential   ST1 = M*Y/(t**2*B)-(Z*SR)     = 96.41 N/mm2     SFO =132.16 N/mm2
  (greater)  ST2 = (SH+SR)/2 or (SH+ST1)/2 =   132 N/mm2     SFO =132.16 N/mm2
```

Component: Front Shell Flng

```
ASME Section VIII Div.1 2017, Appendix 2, 2-14 Flange Rigidity
--- Calculations ---
Operating moment,    Mo = 1052325 N*m   Gasket seat. moment Ma = 1025039 N*m
Factor VI            VI = 0.4063        Factor L             L = 2.0293
Mod. elast.design T  Ed = 185110 N/mm2  Mod.elast.atm. temp Ea = 201327 N/mm2
Thickness g0         g0 = 37 mm         Factor h0           h0 = 211.2 mm
Factor KI            KI = 0.3           Factor KL           KL = 0.2
Corrosion allowance  ca = 3 mm          Factor K             K = 1.2736
Thickness, T          T = 188 mm
Flange Rigidity
  Loose type flanges without hubs and optional flanges designed as loose type
    Gasket seating  J = 109.4 * Ma / (E * T ** 3 * Ln(K) * KL) = -
    Operating       J = 109.4 * Mo / (E * T ** 3 * Ln(K) * KL) = -
  Integral type flanges and optional type flanges designed as integral and
  Loose type flanges with hubs
    Gasket seating  J = 52.14 * Ma * VI/ (L*E*G0**2 * ho * KI) = 0.6126
    Operating       J = 52.14 * Mo * VI/ (L*E*G0**2 * ho * KI) = 0.684
  ASME appendix 2 calculation of hub thickness 'go' as a cylinder
Design pressure       P = 7.8 N/mm2     Allowable stress     S = 132.16 N/mm2
Outside radius       OR = -             Inside radius       IR = 603 mm
Joint efficiency      E = 1             Corr.Allow or OL     c = 3 mm
                                        Material tolerance tol = 0 mm
Min hub thk / small end = P*IR / (S*E - 0.6*P)+c+Tol                UG-27(c)(1)
                        = 39.9 mm
Hub thk / small end     = 40 mm
New thickness 'go       = 40 mm         New thickness 'g1'     = 53.5 mm
Corroded thickness 'go' = 37 mm         Corroded thk  'g1'     = 50.5 mm
```

Component: Front Tubesheet

Tubesheet Details - ASME VIII-1 2017 - UHX - U-tube Construction

Materials of construction   Fig UHX-12.1 U-Tube Tubesheet configuration (d)

Tubesheet:	SA-266 K03506 Grd 2 Forgings
Tubes:	SA-179 K01200 Smls. tube
Shell:	SA-516 K02700 Grd 70 Plate
Channel:	SA-516 K02700 Grd 70 Plate

Design conditions		Shell side	Tube side	Tubes	Tubesheet
Design pressure	N/mm2	7.8	6		
Vacuum	N/mm2	-	-		
Design temperature	C	280	300	300	300

All.stress tubesheet	S = 129.18 N/mm2	All.stress tubes	St = 91.893 N/mm2
All.stress shell	Ss = 136.406 N/mm2	All.stress channel	Sc = 134.917 N/mm2
Yield stress shell	Sys = 209.021 N/mm2	Yield stress chann.	Syc = 204.305 N/mm2
Mod.of elas.tubesheet	E = 183373 N/mm2	Mod.of elas. tubes	Et = 184255 N/mm2
All.str.tubes at T	Stt = 91.893 N/mm2	Mod.of E.tubes at T	Ett = 184255 N/mm2
Mod.of elas.shell	Es = 185110 N/mm2	Mod.of elas. channel	Ec = 183373 N/mm2
Poisson Ratio shell	vs = 0.3	Poisson ratio chan.	vc = 0.3
Shell diameter	Ds = 1206 mm	Channel diameter	Dc = 1206 mm
Shell thickness	ts = 37 mm	Channel thickness	tc = 29 mm

Tube OD dt = 25.4 mm Tube thickness tt = 2.4 mm

```
Number of tube holes Nt = 1004          Tube pitch            p = 31.75 mm
Outer tube limit     Do = 1187.3 mm     Outer tube radius    ro = 580.95 mm
Tube expan. ratio   rho = 0.985         Tube expanded len.  ltx = 191 mm
Gasket Gs diameter   Gs = 1260.18 mm    Gasket Gc diameter   Gc = 1260.18 mm
Center distance      UL = 76.2 mm       Gasket G diameter     G = 1260.18 mm
Tubesheet cor.all.   ct = 3 mm          Pass groove depth   hgt = 5 mm
Tubesheet cor.all.   cs = 3 mm          Pass groove depth   hgs = 0 mm
Tubes     cor.all.    c = -             Effective groove depth:
h'g = MAX[(hgt-ct),(0)]                                     h'g = 2 mm
Bolt circle diameter  C = 1434 mm       Bolt load            W* = 11792021 N
Shell bolt load    Wm1s = 11792021 N    Channel bolt load  Wm1c = 9333961 N
Tubesheet diameter    A = 1536 mm       DL = (4*Ap/Cp)       DL = 873.35 mm
Tubesheet thickness   h = 194 mm        Actual tubesheet thk ha = 200 mm
UHX-12.5.1 Step 1. Determnie Do, Mu, Mu* and h'g from UHX-11.5.1
Basic ligamente efficiency, mu = (p - dt) / p                mu = 0.2
Effective tube hole diameter  d* = dt-2*tt*(Et/E)*(St/S)*rho    = 22.02 mm
                (maximum of)  d* = dt-2*tt                      = 20.6 mm
                                                             d* = 22.02 mm
Pass lane area limit                                     4*Do*p = 150787.11 mm2
Actual pass lane area, AL                                    AL = 90472.26 mm2
Effective tube pitch = p/SQRT(1-(4*MIN[AL,4*Do*p]/Pi*Do**2)) p* = 33.13 mm
Effective ligament efficiency, mu* = (p*-d*)/p*             mu* = 0.3353
UHX-12.5.2 Step 2. Calculate diameter ratios rhos and rhoc.  For each loading
   case, calculate moment Mts due to pressures Ps and Pt acting on the
   unperforated tubesheet rim.
   Rhos = Gs / Do     Rhos = 1.0614      Rhoc = Gc / Do    Rhoc = 1.0614
   MTS = (Do**2/16)*((rhos-1)*(rhos**2+1)*Ps - (rhoc-1)*(rhoc**2+1)*Pt)
   Load case                        1          2          3          4
   Shell Pressure, N/mm2            0         7.8        7.8         0
   Tubes Pressure, N/mm2            6          0          6          0
   Moment MTS,  N*mm/mm         -69005      89707      20702         0
UHX-12.5.3 Step 3. Calculate h/p.  If rho changes, recalculate d* and mu* from
   figure UHX-11.5.1.  Determin E*/E and v* relative to h/p from UHX-11.5.2.
   Determine E*/E and v* from Fig UHX-11.4                  h/p = 6.2992
             Ratio E*/E = 0.3822                      Factor v* = 0.3237
```

Tubesheet Details - ASME VIII-1 2017 - UHX - U-tube Construction

UHX-12.5.5 Step 5. Calculate diameter ratio K and coefficient F

K = A/Do   K = 1.2937

F = ((1-v*)/E*)*E*Ln(K))   F = 0.4557

Load case	1	2	3	4
Factor F	0.46	0.46	0.46	0.46

UHX-12.5.6 Step 6. For each loading case, calculate moment M* acting on the unperforated tubesheet rim.

M* = Mts + ((Gc-Gs)/ (2*Pi*Do)) * W*

Load case	1	2	3	4
Effective W, N	9333961	11792021	11792021	0
Moment M*, N*mm/mm	-69005	89707	20702	0

Component: Front Tubesheet

Tubesheet Details - ASME VIII-1 2017 - UHX - U-tube Construction

UHX-12.5.7 Step 7. For each loading case, calculate the maximum bending moments acting on the tubesheet at the periphery Mp and at the center Mo

At the periphery:

Mp = (M* - (Do**2/32) * F * (Ps - Pt)) / (1 + F)

At the center:

Mo = Mp + (Do**2/64) * (3+v*) * (Ps-pt)

M = MAX[|Mp|,|Mo|]

Load case	1	2	3	4
Mp, N*mm/mm	35340	-45942	-10602	0
Mo, N*mm/mm	-403914	525088	121174	0
M , N*mm/mm	403914	525088	121174	0

UHX-12.5.8 Step 8. For each loading case, calculate the tubesheet bending

stress sigma.

sigma = 6 * M / (mu* * (h - h'g)**2)

Load case	1	2	3	4
sigma, N/mm2	196.05	254.86	58.81	0
Allowable stress, N/mm2	258.36	258.36	258.36	258.36
Min tubesheet thk, mm	169.25	192.7	93.61	2

UHX-12.5.9 Step 9. For each loading case, calculate the average shear stress in the tubesheet at the outer edge of the perforated region.

Tau = 1/(4*mu)*(Diam/h)*|Ps-Pt|

Load case	1	2	3	4
|Ps-Pt|, N/mm2	6	7.8	1.8	0
3.2*S*Mu*h/Do	13.51	13.51	13.51	13.51
Diam = DL or Do, mm	1187.3	1187.3	1187.3	1187.3
Tau, N/mm2	45.9	59.67	13.77	0
Allowable stress, N/mm2	103.34	103.34	103.34	103.34
Min tubesheet thickness, mm	86.17	112.02	25.85	0

Component: Nozzle S1

```
ASME VIII-1 2017 UG-27 Thickness of Cylinders under Internal Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-106 K03006 Grd B Smls. pipe
Design pressure      P = 7.8 N/mm2      Design temperature   T = 280 C
Radiography            = Full           Joint efficiency     E = 1
Design stress        S = 117.9 N/mm2
Inside corr.allow. cai = 3 mm           Outside corr. all. cao = 0 mm
Material tolerance tol = 1.391 mm       Minimum thickness tmin = 9.19 mm
Outside diameter    OD = 88.9 mm        Corroded radius     IR = 36.32 mm
- Minimum thickness t(UG-45) = max(ta,tb,UG-16(b)):
    - ta = internal pressure:
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 6.89 mm     UG-27(c)(1)
    - ta = external pressure+cai+cao+tol         t = 0 mm
    - tb = min(tb3,max(tb1,tb2,UG-16(b))         t = 9.19 mm
           tb3 = Table UG-45+cai+cao+tol  = 9.19 mm
             max(tb1,tb2,UG-16(b))      t = 52.01 mm
             - tb1+cai+cao+tol      = 52.01 mm
             - UG-16(b)+cai+cao+tol = 5.98 mm
             - tb2+cai+cao+tol      = 0 mm
                        Minimum thickness:               tmin = 9.19 mm
                        Nominal thickness:               tnom = 11.13 mm
```

Component: Reinforcement Nozzle S1

```
ASME Section VIII-1 2017 UG-37  Reinforcement Required for Openings in
                                Shells and Formed Heads
--- Design Conditions:
Int. design pressure PI = 7.8 N/mm2     Ext. design press.  PE = 0 N/mm2
Design temperature    T = 280 C         Fig.UW-16.1 Sketch (h)
Vessel material: SA-516 K02700 Grd 70 Plate
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Vessel design stress Sv = 136.41 N/mm2  Joint efficiency     E = 1
Vessel outside dia   Do = 1693 mm       Corroded radius     OR = 846.5 mm
Nominal thickness  tnom = 55 mm         Reinforcement limit lp = 96.45 mm
Req. tks. int.pres.  tr = 47.62 mm      Req. tks.ext.pres. tre = 0 mm
Corroded thickness    t = 51.7 mm       Reinf. efficiency   E1 = 1.0
Attachment Material: SA-106 K03006 Grd B Smls. pipe
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Nozzle design stress Sn = 117.9 N/mm2   Joint efficiency     E = 1
Nozzle outside dia. Don = 88.9 mm       Corroded radius     IR = 36.32 mm
Nominal thickness  tnom = 11.13 mm      Reinforcement limit ln = 52.33 mm
Req.tks. int.pres.  trn = 2.8 mm        Req.tks.ext.pres. trne = 0 mm
Corroded thickness   tn = 8.13 mm       Nozzle Projection   ha = 0 mm
                                        Nozzle Proj. used    h = 0 mm
Reinforcement element material: SA-516 K02700 Grd 70 Plate
Limit of reinf.      Dp = 163 mm        Nominal thickness   te = 32 mm
Outside diameter        = 163 mm        Design stress       Se = 136.41 N/mm2
Minimum weld size tmin  = 19.05 mm      Weld leg (1/2*tmin)    = 20.07 mm
Weld throat (1/2*tmin)  = 9.52 mm       Weld throat (1/2*tmin) = 14.05 mm
Weld throat tw   (min)  = 5.69 mm       Weld throat tw         = 5.69 mm
Weld throat tc   (min)  = 5.69 mm       Weld throat tc         = 5.69 mm
    smaller |6.35 mm   |                Weld leg tw            = 8.13 mm
 tc   of    |0.7 * tmin|                Weld leg tc            = 8.13 mm
Outward nozzle weld   L1 = 8.13 mm      fr1 = Sn/Sv              = 0.8643
Outer element weld    L2 = 20.07 mm     fr2 = Sn/Sv              = 0.8643
Inward nozzle weld    L3 = 0 mm         fr3 = Sn/Sv or Se/Sv     = 0.8643
Inward nozzle weld new   = 0 mm         fr4 = Se/Sv              = 1.0
Corroded int.proj.thk ti = 0 mm
```

Corroded inside diameter d = 72.64 mm

```
Vessel wall length available for reinforcement    2*Lp-d = 120.26 mm
Plane correction factor (Fig.UG-37)                    F = 1
Offset distance from centerline                     doff = 0 mm
Reinforcement areas (internal pressure condition) ASME 2017 UG-37
A1 = Vessel wall.  Larger of:
 |(2*Lp-d)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 481.38 mm2
 |2*(t+tn)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 478.93 mm2
                                                       A1 = 481.38 mm2
A2 = Nozzle wall outward  | 5*(tn-trn)*fr2*t   | = 1190.4 mm2
     Smaller of:  | 2*(tn-trn)*(2.5*tn+te)*fr2 | = 481.92 mm2
                                                       A2 = 481.92 mm2
A3 = Nozzle wall inward   | 5*t*ti*fr2  | = 0 mm2
     Smallest of:         | 5*ti*ti*fr2 | = 0 mm2
                          | 2*h*ti*fr2  | = 0 mm2
                                                       A3 = 0 mm2
A41 = Outward nozzle weld = (L1**2)*fr3 = 57.13 mm2
A42 = Outer element weld  = (L2**2)*fr4 = 402.74 mm2
A43 = Inward nozzle weld  = (L3**2)*fr2 = 0 mm2
                                                       A4 = 459.87 mm2
JE = pad joint efficiency = 1
A5 = Reinforcement pad Area = (Dp-d-2*tn)*te*fr4*JE    A5 = 2371.2 mm2
Aa = Area Available = A1+A2+A3+A4+A5                   Aa = 3794.37 mm2
A = Area required   = (d*tr*F)+2*tn*tr*F*(1-fr1)        A = 3564.34 mm2
```

Nozzle attachment weld loads - UG-41 - Strength of reinforcement

```
  -
Total weld load (UG-41(b)(2))
  W = (A-A1+2*tn*fr1*(E1*t-F*tr))*Sv                         W = 428352 N
Weld load for strength path 1-1 (UG-41(b)(1)
  W(1-1) = (A2+A5+A41+A42)*Sv                           W(1-1) = 451911 N
Weld load for strength path 2-2 (UG-41(b)(1)
  W(2-2) = (A2+A3+A41+A43+2*tn*t*fr1)*Sv                W(2-2) = 172641 N
Weld load for strength path 3-3 (UG-41(b)(1)
  W(3-3) = (A2+A3+A5+A41+A42+A43+2*tn*t*fr1)*Sv         W(3-3) = 551023 N
Reinforcing element strength = A5 * Se                         = 323446 N
Nozzle attachment weld loads - ASME 2017 UG-41-Strength of reinforcement
 Unit stresses - UW15(c)and UG-45(c)
    Inner fillet weld shear            =   57.77 N/mm2
    Outer fillet weld shear            =   66.84 N/mm2
    Groove weld tension                =   87.25 N/mm2
    Groove weld shear                  =   70.74 N/mm2
    Nozzle wall shear                  =   82.53 N/mm2
 Strength of connection elements
    Inner fillet weld shear            =   65555 N
    Nozzle wall shear                  =   85086 N
    Groove weld tension                =   629563 N
    Outer fillet weld shear            =   343266 N
 Possible paths of failure
    1-1        85086 + 343266          =   428352 N
    2-2        65555 + 629563          =   695118 N
    3-3       629563 + 343266          =   972829 N
 Welds strong enough if path greater than the smaller of W or W(path)
                            Path 1-1 > W or W11
                            428352 N > 428352 N                  OK
                            Path 2-2 > W or W22
                            695118 N > 172641 N                  OK
                            Path 3-3 > W or W33
                            972829 N > 428352 N                  OK
```

Component: Nozzle S2

```
ASME VIII-1 2017 UG-27 Thickness of Cylinders under Internal Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-106 K03006 Grd B Smls. pipe
Design pressure      P = 7.8 N/mm2      Design temperature   T = 280 C
Radiography            = Full           Joint efficiency     E = 1
Design stress        S = 117.9 N/mm2
Inside corr.allow. cai = 3 mm           Outside corr. all. cao = 0 mm
Material tolerance tol = 0.892 mm       Minimum thickness tmin = 7.11 mm
Outside diameter    OD = 48.26 mm       Corroded radius     IR = 20.01 mm
- Minimum thickness t(UG-45) = max(ta,tb,UG-16(b)):
    - ta = internal pressure:
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 5.27 mm     UG-27(c)(1)
    - ta = external pressure+cai+cao+tol         t = 0 mm
    - tb = min(tb3,max(tb1,tb2,UG-16(b))         t = 7.11 mm
           tb3 = Table UG-45+cai+cao+tol  = 7.11 mm
             max(tb1,tb2,UG-16(b))      t = 51.51 mm
             - tb1+cai+cao+tol      = 51.51 mm
             - UG-16(b)+cai+cao+tol = 5.48 mm
             - tb2+cai+cao+tol      = 0 mm
                        Minimum thickness:               tmin = 7.11 mm
                        Nominal thickness:               tnom = 7.12 mm
```

Corroded inside diameter d = 40.02 mm

```
Vessel wall length available for reinforcement    2*Lp-d = 112.24 mm
Plane correction factor (Fig.UG-37)                    F = 1
Offset distance from centerline                     doff = 0 mm
Reinforcement areas (internal pressure condition) ASME 2017 UG-37
A1 = Vessel wall.  Larger of:
 |(2*Lp-d)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 453.11 mm2
 |2*(t+tn)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 450.66 mm2
                                                            A1 = 453.11 mm2
A2 = Nozzle wall outward  | 5*(tn-trn)*fr2*t   | = 545.5 mm2
     Smaller of:          | 5*(tn-trn)*fr2*tn  | = 43.47 mm2
                                                            A2 = 43.47 mm2
A3 = Nozzle wall inward   | 5*t*ti*fr2  | = 0 mm2
     Smallest of:         | 5*ti*ti*fr2 | = 0 mm2
                          | 2*h*ti*fr2  | = 0 mm2
                                                            A3 = 0 mm2
A41 = Outward nozzle weld = (L1**2)*fr3 = 14.67 mm2
A42 = Outer element weld  = (L2**2)*fr4 = 0 mm2
A43 = Inward nozzle weld  = (L3**2)*fr2 = 0 mm2
                                                            A4 = 14.67 mm2
JE = pad joint efficiency = 1
A5 = Reinforcement pad Area = (Dp-d-2*tn)*te*fr4*JE         A5 = 0 mm2
Aa = Area Available = A1+A2+A3+A4+A5                        Aa = 511.25 mm2
A = Area required   = (d*tr*F)+2*tn*tr*F*(1-fr1)             A = 1959.08 mm2
Per UG-36(c)(3)(a), this opening does NOT require additional reinforcement
                    other than that inherent in the construction.
Nozzle attachment weld loads per UG-41 not required per UW-15(b).
```

Component: Nozzle S3

```
ASME VIII-1 2017 UG-27 Thickness of Cylinders under Internal Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-106 K03006 Grd B Smls. pipe
Design pressure      P = 7.8 N/mm2      Design temperature   T = 280 C
Radiography            = Full           Joint efficiency     E = 1
Design stress        S = 117.9 N/mm2
Inside corr.allow. cai = 3 mm           Outside corr. all. cao = 0 mm
Material tolerance tol = 1.391 mm       Minimum thickness tmin = 9.66 mm
Outside diameter    OD = 114.3 mm       Corroded radius     IR = 49.02 mm
- Minimum thickness t(UG-45) = max(ta,tb,UG-16(b)):
    - ta = internal pressure:
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 7.77 mm     UG-27(c)(1)
    - ta = external pressure+cai+cao+tol         t = 0 mm
    - tb = min(tb3,max(tb1,tb2,UG-16(b))         t = 9.66 mm
           tb3 = Table UG-45+cai+cao+tol  = 9.66 mm
             max(tb1,tb2,UG-16(b))      t = 52.01 mm
             - tb1+cai+cao+tol      = 52.01 mm
             - UG-16(b)+cai+cao+tol = 5.98 mm
             - tb2+cai+cao+tol      = 0 mm
                        Minimum thickness:                tmin = 9.66 mm
                        Nominal thickness:                tnom = 11.13 mm
```

Component: Reinforcement Nozzle S3

```
ASME Section VIII-1 2017 UG-37  Reinforcement Required for Openings in
                                Shells and Formed Heads
--- Design Conditions:
Int. design pressure PI = 7.8 N/mm2     Ext. design press.  PE = 0 N/mm2
Design temperature    T = 280 C         Fig.UW-16.1 Sketch (h)
Vessel material: SA-516 K02700 Grd 70 Plate
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Vessel design stress Sv = 136.41 N/mm2  Joint efficiency     E = 1
Vessel outside dia   Do = 1693 mm       Corroded radius     OR = 846.5 mm
Nominal thickness  tnom = 55 mm         Reinforcement limit lp = 109.15 mm
Req. tks. int.pres.  tr = 47.62 mm      Req. tks.ext.pres. tre = 0 mm
Corroded thickness    t = 51.7 mm       Reinf. efficiency   E1 = 1.0
Attachment Material: SA-106 K03006 Grd B Smls. pipe
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Nozzle design stress Sn = 117.9 N/mm2   Joint efficiency     E = 1
Nozzle outside dia. Don = 114.3 mm      Corroded radius     IR = 49.02 mm
Nominal thickness  tnom = 11.13 mm      Reinforcement limit ln = 75.33 mm
Req.tks. int.pres.  trn = 3.68 mm       Req.tks.ext.pres. trne = 0 mm
Corroded thickness   tn = 8.13 mm       Nozzle Projection   ha = 0 mm
                                        Nozzle Proj. used    h = 0 mm
Reinforcement element material: SA-516 K02700 Grd 70 Plate
Limit of reinf.      Dp = 178 mm        Nominal thickness   te = 55 mm
Outside diameter        = 178 mm        Design stress       Se = 136.41 N/mm2
Minimum weld size tmin  = 19.05 mm      Weld leg (1/2*tmin)    = 25.78 mm
Weld throat (1/2*tmin)  = 9.52 mm       Weld throat (1/2*tmin) = 18.04 mm
Weld throat tw   (min)  = 5.69 mm       Weld throat tw         = 5.69 mm
Weld throat tc   (min)  = 5.69 mm       Weld throat tc         = 5.69 mm
    smaller |6.35 mm   |                Weld leg tw            = 8.13 mm
 tc   of    |0.7 * tmin|                Weld leg tc            = 8.13 mm
Outward nozzle weld   L1 = 8.13 mm      fr1 = Sn/Sv              = 0.8643
Outer element weld    L2 = 25.78 mm     fr2 = Sn/Sv              = 0.8643
Inward nozzle weld    L3 = 0 mm         fr3 = Sn/Sv or Se/Sv     = 0.8643
Inward nozzle weld new   = 0 mm         fr4 = Se/Sv              = 1.0
Corroded int.proj.thk ti = 0 mm
```

Corroded inside diameter d = 98.04 mm

```
Vessel wall length available for reinforcement    2*Lp-d = 120.26 mm
Plane correction factor (Fig.UG-37)                    F = 1
Offset distance from centerline                     doff = 0 mm
Reinforcement areas (internal pressure condition) ASME 2017 UG-37
A1 = Vessel wall.  Larger of:
 |(2*Lp-d)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 481.38 mm2
 |2*(t+tn)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 478.93 mm2
                                                             A1 = 481.38 mm2
A2 = Nozzle wall outward  | 5*(tn-trn)*fr2*t   | = 994.92 mm2
     Smaller of:  | 2*(tn-trn)*(2.5*tn+te)*fr2 | = 579.82 mm2
                                                             A2 = 579.82 mm2
A3 = Nozzle wall inward   | 5*t*ti*fr2  | = 0 mm2
     Smallest of:         | 5*ti*ti*fr2 | = 0 mm2
                          | 2*h*ti*fr2  | = 0 mm2
                                                             A3 = 0 mm2
A41 = Outward nozzle weld = (L1**2)*fr3 = 57.13 mm2
A42 = Outer element weld  = (L2**2)*fr4 = 664.52 mm2
A43 = Inward nozzle weld  = (L3**2)*fr2 = 0 mm2
                                                             A4 = 721.65 mm2
JE = pad joint efficiency = 1
A5 = Reinforcement pad Area = (Dp-d-2*tn)*te*fr4*JE         A5 = 3503.5 mm2
Aa = Area Available = A1+A2+A3+A4+A5                         Aa = 5286.34 mm2
A = Area required   = (d*tr*F)+2*tn*tr*F*(1-fr1)             A = 4773.95 mm2
```

Nozzle attachment weld loads - UG-41 - Strength of reinforcement

-

```
Total weld load (UG-41(b)(2))
  W = (A-A1+2*tn*fr1*(E1*t-F*tr))*Sv                         W = 593350 N
Weld load for strength path 1-1 (UG-41(b)(1)
  W(1-1) = (A2+A5+A41+A42)*Sv                           W(1-1) = 655426 N
Weld load for strength path 2-2 (UG-41(b)(1)
  W(2-2) = (A2+A3+A41+A43+2*tn*t*fr1)*Sv                W(2-2) = 185996 N
Weld load for strength path 3-3 (UG-41(b)(1)
  W(3-3) = (A2+A3+A5+A41+A42+A43+2*tn*t*fr1)*Sv         W(3-3) = 754538 N
Reinforcing element strength = A5 * Se                         = 477898 N
Nozzle attachment weld loads - ASME 2017 UG-41-Strength of reinforcement
 Unit stresses - UW15(c)and UG-45(c)
    Inner fillet weld shear             =   57.77 N/mm2
    Outer fillet weld shear             =   66.84 N/mm2
    Groove weld tension                 =   87.25 N/mm2
    Groove weld shear                   =   70.74 N/mm2
    Nozzle wall shear                   =   82.53 N/mm2
 Strength of connection elements
    Inner fillet weld shear             =   84285 N
    Nozzle wall shear                   =   111843 N
    Groove weld tension                 =   809438 N
    Outer fillet weld shear             =   481507 N
 Possible paths of failure
    1-1       111843 + 481507           =   593350 N
    2-2        84285 + 809438           =   893723 N
    3-3       809438 + 481507           =   1290945 N
 Welds strong enough if path greater than the smaller of W or W(path)
                           Path 1-1 > W or W11
                           593350 N > 593350 N                  OK
                           Path 2-2 > W or W22
                           893723 N > 185996 N                  OK
                           Path 3-3 > W or W33
                          1290945 N > 593350 N                  OK
```

Component: Nozzle T1

```
ASME VIII-1 2017 UG-27 Thickness of Cylinders under Internal Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-106 K03006 Grd B Smls. pipe
Design pressure      P = 6 N/mm2        Design temperature   T = 300 C
Radiography            = Full           Joint efficiency     E = 1
Design stress        S = 117.9 N/mm2
Inside corr.allow. cai = 3 mm           Outside corr. all. cao = 0 mm
Material tolerance tol = 2.394 mm       Minimum thickness tmin = 16.55 mm
Outside diameter    OD = 457.2 mm       Corroded radius     IR = 212.45 mm
- Minimum thickness t(UG-45) = max(ta,tb,UG-16(b)):
    - ta = internal pressure:
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 16.55 mm    UG-27(c)(1)
    - ta = external pressure+cai+cao+tol         t = 0 mm
    - tb = min(tb3,max(tb1,tb2,UG-16(b))         t = 13.73 mm
           tb3 = Table UG-45+cai+cao+tol  = 13.73 mm
             max(tb1,tb2,UG-16(b))      t = 33.25 mm
             - tb1+cai+cao+tol      = 33.25 mm
             - UG-16(b)+cai+cao+tol = 6.98 mm
             - tb2+cai+cao+tol      = 0 mm
                        Minimum thickness:               tmin = 16.55 mm
                        Nominal thickness:               tnom = 19.15 mm
```

Component: Reinforcement Nozzle T1

```
ASME Section VIII-1 2017 UG-37  Reinforcement Required for Openings in
                                Shells and Formed Heads
--- Design Conditions:
Int. design pressure PI = 6 N/mm2       Ext. design press.  PE = 0 N/mm2
Design temperature    T = 300 C         Fig.UW-16.1 Sketch (h)
Vessel material: SA-516 K02700 Grd 70 Plate
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Vessel design stress Sv = 134.92 N/mm2  Joint efficiency     E = 1
Vessel outside dia   Do = 1264 mm       Corroded radius     IR = 603 mm
Nominal thickness  tnom = 32 mm         Reinforcement limit lp = 424.9 mm
Req. tks. int.pres.  tr = 27.85 mm      Req. tks.ext.pres. tre = 0 mm
Corroded thickness    t = 28.7 mm       Reinf. efficiency   E1 = 1.0
Attachment Material: SA-106 K03006 Grd B Smls. pipe
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Nozzle design stress Sn = 117.9 N/mm2   Joint efficiency     E = 1
Nozzle outside dia. Don = 457.2 mm      Corroded radius     IR = 212.45 mm
Nominal thickness  tnom = 19.15 mm      Reinforcement limit ln = 71.75 mm
Req.tks. int.pres.  trn = 11.45 mm      Req.tks.ext.pres. trne = 0 mm
Corroded thickness   tn = 16.15 mm      Nozzle Projection   ha = 0 mm
                                        Nozzle Proj. used    h = 0 mm
Reinforcement element material: SA-516 K02700 Grd 70 Plate
Limit of reinf.      Dp = 789 mm        Nominal thickness   te = 32 mm
Outside diameter        = 789 mm        Design stress       Se = 134.92 N/mm2
Minimum weld size tmin  = 19.05 mm      Weld leg (1/2*tmin)    = 17.49 mm
Weld throat (1/2*tmin)  = 9.52 mm       Weld throat (1/2*tmin) = 12.24 mm
Weld throat tw   (min)  = 11.31 mm      Weld throat tw         = 11.31 mm
Weld throat tc   (min)  = 6.35 mm       Weld throat tc         = 6.35 mm
    smaller |6.35 mm   |                Weld leg tw            = 16.15 mm
 tc   of    |0.7 * tmin|                Weld leg tc            = 9.07 mm
Outward nozzle weld   L1 = 16.15 mm     fr1 = Sn/Sv              = 0.8739
Outer element weld    L2 = 17.49 mm     fr2 = Sn/Sv              = 0.8739
Inward nozzle weld    L3 = 0 mm         fr3 = Sn/Sv or Se/Sv     = 0.8739
Inward nozzle weld new   = 0 mm         fr4 = Se/Sv              = 1.0
Corroded int.proj.thk ti = 0 mm
```

Corroded inside diameter **d = 424.9 mm**

```
Vessel wall length available for reinforcement    2*Lp-d = 424.9 mm
Plane correction factor (Fig.UG-37)                    F = 1
Offset distance from centerline                     doff = 0 mm
Reinforcement areas (internal pressure condition) ASME 2017 UG-37
A1 = Vessel wall.  Larger of:
 |(2*Lp-d)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 356.97 mm2
 |2*(t+tn)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 72.63 mm2
                                                            A1 = 356.97 mm2
A2 = Nozzle wall outward  | 5*(tn-trn)*fr2*t   | = 589.11 mm2
     Smaller of:  | 2*(tn-trn)*(2.5*tn+te)*fr2 | = 594.24 mm2
                                                            A2 = 589.11 mm2
A3 = Nozzle wall inward   | 5*t*ti*fr2  | = 0 mm2
     Smallest of:         | 5*ti*ti*fr2 | = 0 mm2
                          | 2*h*ti*fr2  | = 0 mm2
                                                            A3 = 0 mm2
A41 = Outward nozzle weld = (L1**2)*fr3 = 227.93 mm2
A42 = Outer element weld  = (L2**2)*fr4 = 305.99 mm2
A43 = Inward nozzle weld  = (L3**2)*fr2 = 0 mm2
                                                            A4 = 533.92 mm2
JE = pad joint efficiency = 1
A5 = Reinforcement pad Area = (Dp-d-2*tn)*te*fr4*JE         A5 = 10617.6 mm2
Aa = Area Available = A1+A2+A3+A4+A5                        Aa = 12097.6 mm2
A = Area required   = (d*tr*F)+2*tn*tr*F*(1-fr1)             A = 11947.67 mm2
```

Nozzle attachment weld loads - UG-41 - Strength of reinforcement

-

```
Total weld load (UG-41(b)(2))
  W = (A-A1+2*tn*fr1*(E1*t-F*tr))*Sv                      W = 1567007 N
Weld load for strength path 1-1 (UG-41(b)(1)
  W(1-1) = (A2+A5+A41+A42)*Sv                        W(1-1) = 1584006 N
Weld load for strength path 2-2 (UG-41(b)(1)
  W(2-2) = (A2+A3+A41+A43+2*tn*t*fr1)*Sv             W(2-2) = 219527 N
Weld load for strength path 3-3 (UG-41(b)(1)
  W(3-3) = (A2+A3+A5+A41+A42+A43+2*tn*t*fr1)*Sv      W(3-3) = 1693300 N
Reinforcing element strength = A5 * Se                        = 1432490 N
Nozzle attachment weld loads - ASME 2017 UG-41-Strength of reinforcement
 Unit stresses - UW15(c)and UG-45(c)
    Inner fillet weld shear           =   57.77 N/mm2
    Outer fillet weld shear           =   66.11 N/mm2
    Groove weld tension               =   87.25 N/mm2
    Groove weld shear                 =   70.74 N/mm2
    Nozzle wall shear                 =   82.53 N/mm2
 Strength of connection elements
    Inner fillet weld shear           =   669715 N
    Nozzle wall shear                 =   922940 N
    Groove weld tension               =   1797359 N
    Outer fillet weld shear           =   1432491 N
 Possible paths of failure
    1-1       922940 + 1432491        =   2355431 N
    2-2       669715 + 1797359        =   2467074 N
    3-3      1797359 + 1432491        =   3229850 N
 Welds strong enough if path greater than the smaller of W or W(path)
                           Path 1-1 > W or W11
                          2355431 N > 1567007 N                 OK
                           Path 2-2 > W or W22
                          2467074 N > 219527 N                  OK
                           Path 3-3 > W or W33
                          3229850 N > 1567007 N                 OK
```

Component: Nozzle T2

```
ASME VIII-1 2017 UG-27 Thickness of Cylinders under Internal Pressure
--- Calculations --- Cylinder Internal Pressure
Material:  SA-106 K03006 Grd B Smls. pipe
Design pressure      P = 6 N/mm2        Design temperature   T = 300 C
Radiography            = Full           Joint efficiency     E = 1
Design stress        S = 117.9 N/mm2
Inside corr.allow. cai = 3 mm           Outside corr. all. cao = 0 mm
Material tolerance tol = 2.577 mm       Minimum thickness tmin = 17.99 mm
Outside diameter    OD = 508 mm         Corroded radius     IR = 236.38 mm
- Minimum thickness t(UG-45) = max(ta,tb,UG-16(b)):
    - ta = internal pressure:
              t = (P*IR / (S*E-0.6*P))+cai+cao+tol = 17.99 mm    UG-27(c)(1)
    - ta = external pressure+cai+cao+tol         t = 0 mm
    - tb = min(tb3,max(tb1,tb2,UG-16(b))         t = 13.91 mm
           tb3 = Table UG-45+cai+cao+tol  = 13.91 mm
             max(tb1,tb2,UG-16(b))      t = 33.43 mm
             - tb1+cai+cao+tol      = 33.43 mm
             - UG-16(b)+cai+cao+tol = 7.16 mm
             - tb2+cai+cao+tol      = 0 mm
                        Minimum thickness:               tmin = 17.99 mm
                        Nominal thickness:               tnom = 20.62 mm
```

Component: Reinforcement Nozzle T2

```
ASME Section VIII-1 2017 UG-37  Reinforcement Required for Openings in
                                Shells and Formed Heads
--- Design Conditions:
Int. design pressure PI = 6 N/mm2       Ext. design press.  PE = 0 N/mm2
Design temperature    T = 300 C         Fig.UW-16.1 Sketch (h)
Vessel material: SA-516 K02700 Grd 70 Plate
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Vessel design stress Sv = 134.92 N/mm2  Joint efficiency     E = 1
Vessel outside dia   Do = 1264 mm       Corroded radius     IR = 603 mm
Nominal thickness  tnom = 32 mm         Reinforcement limit lp = 472.76 mm
Req. tks. int.pres.  tr = 27.85 mm      Req. tks.ext.pres. tre = 0 mm
Corroded thickness    t = 28.7 mm       Reinf. efficiency   E1 = 1.0
Attachment Material: SA-106 K03006 Grd B Smls. pipe
Inside corr. allow. CAI = 3 mm          Outside corr.allow.CAO = 0 mm
Nozzle design stress Sn = 117.9 N/mm2   Joint efficiency     E = 1
Nozzle outside dia. Don = 508 mm        Corroded radius     IR = 236.38 mm
Nominal thickness  tnom = 20.62 mm      Reinforcement limit ln = 71.75 mm
Req.tks. int.pres.  trn = 12.71 mm      Req.tks.ext.pres. trne = 0 mm
Corroded thickness   tn = 17.62 mm      Nozzle Projection   ha = 0 mm
                                        Nozzle Proj. used    h = 0 mm
Reinforcement element material: SA-516 K02700 Grd 70 Plate
Limit of reinf.      Dp = 878 mm        Nominal thickness   te = 32 mm
Outside diameter        = 878 mm        Design stress       Se = 134.92 N/mm2
Minimum weld size tmin  = 19.05 mm      Weld leg (1/2*tmin)    = 17.53 mm
Weld throat (1/2*tmin)  = 9.52 mm       Weld throat (1/2*tmin) = 12.27 mm
Weld throat tw   (min)  = 12.33 mm      Weld throat tw         = 12.33 mm
Weld throat tc   (min)  = 6.35 mm       Weld throat tc         = 6.35 mm
    smaller |6.35 mm   |                Weld leg tw            = 17.62 mm
 tc   of    |0.7 * tmin|                Weld leg tc            = 9.07 mm
Outward nozzle weld   L1 = 17.62 mm     fr1 = Sn/Sv              = 0.8739
Outer element weld    L2 = 17.53 mm     fr2 = Sn/Sv              = 0.8739
Inward nozzle weld    L3 = 0 mm         fr3 = Sn/Sv or Se/Sv     = 0.8739
Inward nozzle weld new   = 0 mm         fr4 = Se/Sv              = 1.0
Corroded int.proj.thk ti = 0 mm
```

Corroded inside diameter **d = 472.76 mm**

```
Vessel wall length available for reinforcement    2*Lp-d = 472.76 mm
Plane correction factor (Fig.UG-37)                    F = 1
Offset distance from centerline                     doff = 0 mm
Reinforcement areas (internal pressure condition) ASME 2017 UG-37
A1 = Vessel wall.  Larger of:
 |(2*Lp-d)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 397.26 mm2
 |2*(t+tn)*(E1*t-F*tr)-2*tn*(E1*t-F*tr)*(1-fr1)| = 74.81 mm2
                                                            A1 = 397.26 mm2
A2 = Nozzle wall outward  | 5*(tn-trn)*fr2*t   | = 615.92 mm2
     Smaller of:  | 2*(tn-trn)*(2.5*tn+te)*fr2 | = 652.84 mm2
                                                            A2 = 615.92 mm2
A3 = Nozzle wall inward   | 5*t*ti*fr2  | = 0 mm2
     Smallest of:         | 5*ti*ti*fr2 | = 0 mm2
                          | 2*h*ti*fr2  | = 0 mm2
                                                            A3 = 0 mm2
A41 = Outward nozzle weld = (L1**2)*fr3 = 271.31 mm2
A42 = Outer element weld  = (L2**2)*fr4 = 307.27 mm2
A43 = Inward nozzle weld  = (L3**2)*fr2 = 0 mm2
                                                            A4 = 578.58 mm2
JE = pad joint efficiency = 1
A5 = Reinforcement pad Area = (Dp-d-2*tn)*te*fr4*JE         A5 = 11840 mm2
Aa = Area Available = A1+A2+A3+A4+A5                        Aa = 13431.76 mm2
A = Area required   = (d*tr*F)+2*tn*tr*F*(1-fr1)             A = 13290.98 mm2
```

Nozzle attachment weld loads - UG-41 - Strength of reinforcement

Total weld load (UG-41(b)(2))

W = (A-A1+2*tn*fr1*(E1*t-F*tr))*Sv W = 1743102 N

Weld load for strength path 1-1 (UG-41(b)(1)

W(1-1) = (A2+A5+A41+A42)*Sv W(1-1) = 1758571 N

Weld load for strength path 2-2 (UG-41(b)(1)

W(2-2) = (A2+A3+A41+A43+2*tn*t*fr1)*Sv W(2-2) = 238945 N

Weld load for strength path 3-3 (UG-41(b)(1)

W(3-3) = (A2+A3+A5+A41+A42+A43+2*tn*t*fr1)*Sv W(3-3) = 1877814 N

Reinforcing element strength = A5 * Se = 1597413 N

Nozzle attachment weld loads - ASME 2017 UG-41-Strength of reinforcement

Unit stresses - UW15(c)and UG-45(c)

Inner fillet weld shear	=	57.77 N/mm2
Outer fillet weld shear	=	66.11 N/mm2
Groove weld tension	=	87.25 N/mm2
Groove weld shear	=	70.74 N/mm2
Nozzle wall shear	=	82.53 N/mm2

Strength of connection elements

Inner fillet weld shear	=	811859 N
Nozzle wall shear	=	1119571 N
Groove weld tension	=	1997065 N
Outer fillet weld shear	=	1597413 N

Possible paths of failure

1-1	1119571 + 1597413	=	2716984 N
2-2	811859 + 1997065	=	2808924 N
3-3	1997065 + 1597413	=	3594478 N

Welds strong enough if path greater than the smaller of W or W(path)

Path 1-1 > W or W11

2716984 N > 1743102 N OK

Path 2-2 > W or W22

2808924 N > 238945 N OK

Path 3-3 > W or W33

3594478 N > 1743102 N OK

Component: Nozzle Flange Details

Flange, Gasket and Bolting Details

Dimensional data mm

Nozzle	Flg Type	Flg Dia.(*)	Flg Rating	Neck tks	Flg tks	Bolt Cir.	Gaskets O.D.	Gaskets Width	Bolts No	Bolts Dia.
S1	ANSI WN	88.90	600	11.13	31.75	168.27	127.00	9.65	8	19.05
S2	ANSI WN	48.26	600	7.12	22.35	114.30	73.15	9.65	4	19.05
S3	ANSI WN	114.30	600	11.13	38.10	215.90	157.23	12.70	8	22.23
T1	ANSI WN	457.20	600	19.15	82.55	654.05	533.40	22.35	20	41.27
T2	ANSI WN	508.00	600	20.62	88.90	723.90	584.20	25.40	24	41.27

* Dia. = Nozzle O.D. if standard flange

= Flange O.D. if non-standard flange

Horizontal Vessels on Saddles

```
Saddle material: SA-285 K02801 Grd C Plate
Wear plate mtl:  SA-516 K02700 Grd 70 Plate
Shell mean radius      R = 820.5 mm     Total force  Q = W + WS = 473667 N
Weight used            W = 473667 N     From wind/seismic    WS = 0 N
Shell length           L = 5801 mm      Angle alpha       Alpha = 1.8669    rad
Angle beta          Beta = 112.6  deg   Angle delta       Delta = 1.504     rad
Vessel thickness      TS = 55 mm        Wear plate tks.       w = 20 mm
Vessel corr.allowance CA = 3 mm         Vessel thk TS+w-CA  tsw = 72 mm
Saddle depth           b = 400 mm       Effective depth      b1 = 722.2 mm
Wear plate width      bw = 420 mm       Angle theta       Theta = 134.8     deg
Pressure               P = 7.8 N/mm2    Inside diameter      ID = 1589 mm
Joint efficiency      JE = 1            Head joint eff.     JEH = 1
Front head thickness  TH = 176 mm       Rear head thickness  TH = 50 mm
Head diameter          D = 1597 mm      Head corr.allowance CAH = 3 mm
b1 = b+1.56*(R*(TS-CA))**0.5
                                                    Saddle A      Saddle B
Loads on saddles            Q =                     394024         79643     N
Distance from ref. point    A =                       967           314     mm
Head length                 H =                        0            498     mm
Ratio A/R                 A/R =                     1.1785        0.3827
Bending moment factor      K7 =                     0.0414        0.0104
Shell pressure stress                    PS = P*R/(2*tsw) = 65.51 N/mm2
Fr. head press. stress   = P*D+0.2*P*(TH-CAH)/(2*(TH-CAH)) = ---
Re. head press. stress   = P*D+0.2*P*(TH-CAH)/(2*(TH-CAH)) = 133.3 N/mm2
Alpha = Pi-(Pi/180)*(Theta/2+Beta/20)   Delta = (Pi/180)*(5*Theta/12+30)
S11 = (3*Q*L/(Pi*(TS-CA)*R**2)) S12 = 1-(1-A/L+(R**2-H**2)/(2*A*L))/(1+4*H/3*L)
S13 = Pi*(Sin(Delta)/Delta-Cos(Delta))                   S13/S14=1 if shell is
S14 = Delta+Sin(Delta)*Cos(Delta)-2*Sin(Delta)**2/Delta            stiffened
Stresses in N/mm2                                *** Saddles ***
Bending stress at saddle + pressure                A         B     Allowable
  S1 = S11*(4*A/L)*S12*S13/S14 + PS              68.32     65.52    136.41
Bending stress at midspan + pressure
  S21 = (1+2*(R**2-H**2)/L**2)/(1+4*H/(3*L))
  S2  = S11*(S21-4*A/L) + PS                     67.45     66.25    136.41
Tangential shear in shell (unstiffened)
  S41 = Sin(Alpha)/
      (Pi-Alpha+Sin(Alpha)*Cos(Alpha))
  S42 = L-H-2*A/(L+H)
  S4  = (Q/R*(TS-CA))*S42*S41                    5.91        0      109.12
```

Tangential shear in shell (stiffened)

```
  S61 = Sin(Alpha) * Cos(Alpha)
  S62 = (Sin(Alpha)/Pi)*(Alpha-S61)/
        (Pi-Alpha+S61)
  S6  = (Q/(R*(TS-CA)))*S62                  0       1.22     109.12
Circumferential stress at horn, N/mm2
  S71 = 12 * Q * K7 * R/(L *tsw**2)
  S72 = 4 * tsw * b1
  S7  = -Q/S72 - S71                        -7.24    -0.65    -170.51
Ring compression in shell over saddle, N/mm2
  S91 = 1+Cos(Alpha)/
      (Pi-Alpha+Sin(Alpha)*Cos(Alpha))
  S92 = (TS-CA)*(B+1.56*(R*(TS-CA))**0.5)
  S9  = (Q/S92) * S91                       5.39     1.09     104.51
Tangential shear stresses in head, N/mm2
  S51 = Sin(Alpha) * Cos(Alpha)
  S52 = (Sin(Alpha)/Pi)*(Alpha-S51)/
          (Pi-Alpha+S51)
  S5  = (Q/(12*R*(TH-CAH)))*S52              0       1.36     109.12
Head stresses, N/mm2
  S81 = 3*Q/(8*12*R*(TH-CAH))
  S82 = (Sin(Alpha))**2
  S83 = Pi-Alpha+Sin(Alpha)*Cos(Alpha)
  S8  = S81*(S82/S83)                        0       0.71
Head stresses + pressure, N/mm2
  S8 + PH                                    0     134.01     170.51
```

Saddle Geometry Verification

```
Saddle width        A = 1600 mm         Saddle depth       F = 400 mm
Rib thickness       J = 13 mm           Web thickness     tw = 13 mm
Base plate thk     tb = 40 mm           Wear plate thk   tra = 20 mm
Min saddle height   h = 583.5 mm        Corr.Shell thk    ts = 52 mm
Number of ribs      n = 4               Vessel CL to base  B = 1430 mm
Angle alpha,rad alpha = 1.8669          Angle beta, rad beta = 1.9651
Angle theta,rad theta = 2.3529          Rib depth          G = 400 mm
Yield saddle mtl   fy = 164.76 N/mm2    Inside radius     Ri = 3 mm
                                        Outside radius    Ro = 846.5 mm
Max. load/saddle   Qm = 394024 N        Friction factor   mu = 0.1
     Saddle to foundation surface type: Teflon to Teflon
Expansion load     mu*Qm = FL1 = 39402 N
Max wind or seismic load = Fws = 0 N
Bundle pull load         = Fbl = -
Maximum horizontal load = FL = MAX[FL1,Fws,Fbl]           FL = 39402 N
Saddle coefficient K1
K1 = (1+Cos(beta)-0.5*(Sin(beta))**2)/
        (Pi-Beta+Sin(Beta)*Cos(Beta))                     K1 = 0.2307
Saddle splitting force, fh = K1*Qm                        fh = 90916 N
Cross sectional area of saddle, As                        As = 30530 mm2
Web tension stress, sigmaT = fh/As                    sigmaT = 2.98 N/mm2
Max web tension stress, sigmaTmax = 0.6*fy         sigmaTmax = 98.85 N/mm2
Distance centroid of saddle to base plate, d
  d = h - C1                                               d = 181 mm
Web moment M = fh*d                                        M = 16451514 N*mm
Saddle centroid, C1                                       C1 = 402.5 mm
Saddle moment of inertia, I                                I =  0.3650E+10 mm4
Web bending stress, fbweb = M*C1/I                     fbweb = 1.81 N/mm2
                fbweb <= fbMax = 0.66*fy            fbwebMax = 109.84 N/mm2
Base plate with center web
  Base plate area,  Ab = A*F                              Ab = 640000 mm2
  Bearing pressure, Bp = Q/Ab                             Bp = 0.62 N/mm2
  Base plate moment, M = Q*F/8                             M = 19701208 N*mm
  Section modulus, Z = A*tb**2/6                           Z = 426666.66 mm3
  Base plate bending stress, fbc = M/Z                   fbc = 46.17 N/mm2
                  fbc <= fbcMax = 0.66*fy             fbcMax = 109.84 N/mm2
```

Base plate with offset web

```
  Distance edge base plate to web, d2                      d2 = 13 mm
  Weld leg size, base to web, ww                           ww = 13 mm
  Web length, Lw = A - 2*j                                 Lw = 1574 mm
  Ribs length, Lr = 2*(G-tw)+2*G                           Lr = 1574 mm
  Overall length, L = Lw + Lr                               L = 3148 mm
  Unit load, fu = Q/L    N/mm                              fu = 125.17
  Distance l1 = d2+tw+ww+tb                                l1 = 79 mm
  Distance l2 = F - l1                                     l2 = 321 mm
  Linear load, Omega = Fu/(l1+0.5*l2)                   Omega = 0.52 N/mm2
  Base plate linear moment, M = (omega*l2**2)/6             M = 8975 N
  Base plate bending stress, fbo = 6*M/tb**2              fbo = 33.66 N/mm2
                fbo <= fboMax = 0.66*fy                fboMax = 109.84 N/mm2
  fb = MAX(fbweb,fbc,fbo)                                  fb = 46.17 N/mm2
  Minimum depth of saddle at top, Gtm
  Gtm=SQRT((5.012*Fl/(J*(n-1)*Fb))*
           (h+(A/1.96)*(1-Sin(Alpha))))                   Gtm = 168.9 mm
  Actual depth of saddle at top, Gt                        Gt = 400 mm
Minimum wear plate width, H = Gt + 1.56*SQRT(Ri*ts)         H = 419.5 mm
Actual wear plate width, Ha                                Ha = 420 mm
Minimum wear plate thickness, tr = (Ha-Gt)**2/(2.43*Ro)    tr = 0.2 mm
Actual wear plate thickness, tra                          tra = 20 mm
Anchor bolts
Bolt material: SA-325 Carbon steel Bolt
Longitudinal load, QL = 13353 N         Operating load, Qo   = 211035 N
Bolt diameter, d    d = 29 mm           Bolt area         at = 660.5 mm2
Bolt mtl allow st  Sb = 139.3 N/mm2     Number of bolts    N = 2
Bolt mtl yield st  Sy = 558.5 N/mm2     Yield str factor  yf = 0.4
If Qo>QL, no uplift occurs.    Uplift load per bolt: QL-Qo/N = -
Shear load/bolt = FL/N = 19701 N        Allow force Sy*yf*at = 147554 N
  Bolt transverse load
  Maximum transverse load (seismic or wind), Ftr          Ftr = 0 N
  Bolt transverse moment, MtransB = Ftr * B           MtransB = 0 N*mm
  Critical bolt distance, e = MtransB / Q                   e = 0 mm
  If e < A/6, no uplift occurs                            A/6 = 266.7 mm
                                   Uplift bolt tension force = -
```

Inside ribs/web design

```
  Saddle rib width   Gb = 400 mm        Base length        e = 533.3 mm
  Pressure area F*e  Ap = 213333 mm2    Axial load Bp*Ap   P = 131341 N
  Area rib and web   Ar = 12133 mm2     Comp. str P/Ar    fa = 10.82 N/mm2
  Mom.iner.J*Gb**3/12 I = 69333336 mm4  Rad gyr SQRT(I/Ar) r = 75.6 mm
  Compression dist.  l2 = 586.6 mm      Slender ratio K*l2/r = 15.5
  Unit force Fl/2*A  fu = 12.31 N       Moment fu*l2*e     M = 3852245 N*mm
  Moment arm C2      C2 = 200 mm        Bend.str. M*C2/I  fb = 11.11 N/mm2
  Max. comp.stress   Fa = 141.62 N/mm2  Max.bending str.  Fb = 109.84 N/mm2
  K end connection coeff= 2
  Combined stress (must be less than one)      fa/Fa + fb/Fb = 0.18
Outside ribs/web design
  Press.area 0.5F*e  Ap = 106667 mm2    Axial load Bp*Ap   P = 65671 N
  Area rib and web   Ar = 8667 mm2      Comp. str P/Ar    fa = 7.58 N/mm2
  Compression dist.  l1 = 1113.3 mm     Slender ratio K*l1/r = 29.5
  Unit force Fl/2*A  fu = 12.31 N       Moment 0.5*fu*l1*e M = 3655563 N*mm
  Moment arm C1      C1 = 200 mm        Bend.str. M*C1/I  fb = 10.54 N/mm2
  Max. comp.stress   Fa = 137.49 N/mm2  Max.bending str.  Fb = 109.84 N/mm2
  Combined stress (must be less than one)      fa/Fa + fb/Fb = 0.15

Wind loads - ANSI/SEI/IBC-2009/ASCE 7-10
Equipment Risk Category - wind design                   = II          (1.5.1)
Vessel outside diameter, OD                          OD = 1280 mm
Vessel effective length, EL                          EL = 8476 mm
Vessel effective diameter, EOD                      EOD = 1753.6 mm
Effective wind area, Af = EOD*EL                     Af = 15 m2
Velocity pressure exposure, Kz                       Kz = 0.85       (29.3.1)
Topographic factor, Kzt                             Kzt = 1            (26.8)
Directionality factor, Kd                            Kd = 1            (26.6)
Wind speed, Vkm/h                                     V = 160.9        (26.5)
Velocity pressure, qz,     N/m2
    qz = 0.04727*Kz*Kzt*Kd*V**2                      qz = 1040.64    (29.3-1)
Gust effect factor, G                                 G = 0.85         (26.9)
Force coefficient, Cf                                Cf = 1            (29.5)
Wind force, F                          F = qz*G*Cf*Af = 13147 N     (29.5-1)
User entered wind force, F                            F = -
Moment arm, L                                         L = 1.046 m
Overturning moment, OM,   m-N                  OM = F*L = 13758
Seismic Loads - ANSI/SEI/IBC-2009/ASCE 7-10
Equipment Risk Category - earthquake design             = II          (1.5.1)
Equipment earthquake site class                         = B          (11.4.2)
Response modification factor, Rp                     Rp = 3          (13.3.1)
Earthquake importance factor, Ip                     Ip = 1          (13.3.1)
Vessel amplification factor, Ap                      Ap = 1          (13.3.1)
Mapped MCEr,5% damped spectral resp acceleration     Ss = 0.5        (11.4.1)
Mapped MCEr,5% damped spectral resp acceleration,1s  S1 = 0.2        (11.4.1)
Site coefficient Fa - Table 11.4-1                   Fa = 1          (11.4.3)
```

```
Site coefficient Fv - Table 11.4-2                    Fv = 1           (11.4.3)
Adjusted MCEr,5% damped spectral resp acc    Sms = Fa*Ss = 0.5         (11.4.3)
Adjusted MCEr,5% damped spectral resp acc,1s Sm1 = Fv*S1 = 0.2         (11.4.3)
Design,MCEr,5% damped spectral resp acc    Sds=(2/3)*Sms = 0.333       (11.4.4)
Design,MCEr,5% damped spectral resp acc,1s Sd1=(2/3)*Sm1 = 0.133       (11.4.4)
Height ratio z/h                                     z/h = 0.5         (13.3.1)
Weight of vessel, operating, Wo                       Wo = 422069 N
Vertical seismic force, Fv = 0.2*Sds*Wo               Fv = 28138 N     (13.3.1)
Horizontal seismic force, Fp
  Fp =0.4*Ap*Sds*Wp*(1+2*(z/h))/(Rp/Ip)               Fp = 37517 N     (13.3-1)
Max horizontal force, Fpmax = 1.6*Sds*Ip*Wo        Fpmax = 225104 N    (13.3-2)
Min horizontal force, Fpmin = 0.3*Sds*Ip*Wo        Fpmin = 42207 N     (13.3-3)
Horizontal seismic design force, F                     F = 42207 N
User entered horizontal seismic design force, F        F = -
Moment arm, L                                          L = 1.43 m
Overturning moment, OM,   m-N                   OM = F*L = 60356
```

Wind and Seismic Loads - Effect on Saddles

```
Dist.between saddles                                  Ls = 4.52 m
Saddle width                                           E = 1.6 m
Horizontal seismic design force                        F = 42207 N
Projected area of vessel          Af=PI*EOD**2/4      Af = 2.415 m2
Longitudinal wind force           Fl=Af*Cf*G*qz       Fl = 2136 N
Longitudinal seismic load         Qls=F*L/Ls         Qls = 13353 N
Longitudinal wind load            Qlw=Fl*L/Ls        Qlw = 676 N
Projected area of vessel          Af=EOD*EL           Af = 14.863 m2
Transversal wind force            Ft=0.5*Af*Cf*G*qz   Ft = 6574 N
Transverse seismic load           Qts=3*F*L/E        Qts = 113167 N
Transverse wind load              Qtw=3*Ft*L/E       Qtw = 17625 N
Seismic load (compression) Qs1 = MAX[Qls,Qts]+Fv     Qs1 = 141305 N
Seismic load (tension)     Qs2 = MAX[Qls,Qts]-Fv     Qs2 = 85029 N
Maximum seismic load        Qs = MAX[Qs1,Qs2]         Qs = 141305 N
Maximum wind load           Qw = MAX[Qlw,Qtw]         Qw = 17625 N
Maximum load                 Q = max(Qs,Qw)            Q = 141305 N
  Seismic and wind loads have NOT been applied to supports design.
```

Weights, surface area, Insulation

		Shell side	Tube side
Volume	m3	8.86	3.63
Volume - Operating	m3	7.29	3.63
Avg. fluid density	kg/m3	942.03	99.7
Fluid weight	kg	6866	362
Surface area	m2	32.82	6.42
Insulation thk	mm	-	-
Insulation type		-	-
Insulation density	kg/m3	-	-
Insulation weight	kg	-	-
Weight, seals and jackets		-	-
Weight ext.piping	kg	-	-

Total surface Area	39.23 m2
Weight of Accessories	-
Bundle weight	10853 kg
Empty weight	35809 kg
Full weight (test)	48299 kg
Operating weight	43038 kg

Tube Heat Transfer Surface Area

Tube Bundle	416.36 m2
Front Head Tubesheet	16.02 m2
Rear Head Tubesheet	0 m2
Rear Head (after Full Diameter Support)	0.8 m2
U-Bend	32.93 m2

Maximum Allowable Working Pressures

* = Shell Side MAWP   + = Tube Side MAWP

Component	Side	--Design conditions-- Temp	Stress	MAWP	---- New and cold --- Temp	Stress	MAWP
		C	N/mm2	bar	C	N/mm2	bar
Shell Cylinder	S	280	136.41	80.09	21.1	137.9	88.39
Front Head Cylinder	T	300	134.92	62.43	21.1	137.9	71.26
Front Head Cover	T	300	129.18	60.16	21.1	137.9	68.03
Shell Cover	S	280	136.41	79.82	21.1	137.9	85.81
Front Tubesheet	S	300	129.18	80.73	21.1	137.9	91.58
Front Tubesheet	T	300	129.18	80.73	21.1	137.9	91.58
Front Head Flng At TS	T	300	129.18	76.04	21.1	137.9	81.84
Front Head Flng At Cov	T	300	129.18	61.32	21.1	137.9	67.87
Front Shell Flng	S	280	132.16	78.29	21.1	137.9	84.24
Tubes	T	300	91.89	187.86	21.1	92.39	188.87
Nozzle S1	S	280	117.9	232.66	21.1	117.9	328.07
Nozzle S2	S	280	117.9	216.06	21.1	117.9	394.44
Nozzle S3	S	280	117.9	177.84	21.1	117.9	249.01
Nozzle T1	T	300	117.9	85.72	21.1	117.9	102.19
Nozzle T2	T	300	117.9	84.12	21.1	117.9	98.93
Nozzle Flng S1	S	280	132.16	81.35	21.1	137.9	102.05
Nozzle Flng S2	S	280	132.16	81.35	21.1	137.9	102.05
Nozzle Flng S3	S	280	132.16	81.35	21.1	137.9	102.05
Nozzle Flng T1	T	300	129.18	79.61	21.1	137.9	102.05
Nozzle Flng T2	T	300	129.18	79.61	21.1	137.9	102.05
Nozzle Reinforcement S1	S	280	136.41	79.17	21.1	137.9	85.45
Nozzle Reinforcement S3	S	280	136.41	81.52	21.1	137.9	87.87
Nozzle Reinforcement T1	T	300	134.92	60.83	21.1	137.9	66.07
Nozzle Reinforcement T2	T	300	134.92	60.83	21.1	137.9	65.93
Front Hd Bolting At TS	T	300	172.37	75.83	21.1	172.37	75.83
Front Hd Bolting At TS	S	300	172.37	78.03*	21.1	172.37	78.03*
Front Hd Bolting At Cov	T	300	172.37	60.01+	21.1	172.37	60.01+
Eccentric Reducer	S	280	136.41	81.31	21.1	137.9	86.36
Kettle Cylinder	S	280	136.41	85.40	21.1	137.9	91.99
Nozzle Flng Bolting S1	S	280	172.37	81.35	21.1	172.37	102.05
Nozzle Flng Bolting S2	S	280	172.37	81.35	21.1	172.37	102.05
Nozzle Flng Bolting S3	S	280	172.37	81.35	21.1	172.37	102.05
Nozzle Flng Bolting T1	T	300	172.37	79.61	21.1	172.37	102.05
Nozzle Flng Bolting T2	T	300	172.37	79.61	21.1	172.37	102.05

Minimum Design Metal Temperature for Impact Test Exemption

* Indicates the controlling components + Indicates compliance with UG-20(f)

Component	Curve	Temp		*******	UCS-66.1	*******	
		C		Ratio	Reduction	Temperature	
Shell Cylinder	D	-24		0.97	1	-25	
Front Head Cylinder	D	-29		0.96	2	-31	
Front Head Cover	C	-6		0.88	6	-12	
Shell Cover	D	-20		-	-	-	
Front Tubesheet	C	-3		0.88	6	-9	*
Front Head Flng At TS	C	-14		0.93	3	-17	
Front Head Flng At Cov	C	-14		0.90	5	-19	
Front Shell Flng	C	-8		0.93	3	-11	
Tubes	C	-48		0.34	77	-125	
Front Head Partitions	D	-46		0.94	3	-49	
Nozzle S1	C	-39		0.42	49	-88	
Nozzle S2	C	-48		0.52	28	-76	
Nozzle S3	C	-39		0.55	26	-65	
Nozzle T1	C	-26	+	0.83	9	-35	
Nozzle T2	C	-24	+	0.84	8	-32	
Nozzle Flng S1	-	-28		0.76	13	-41	
Nozzle Flng S2	-	-28		0.76	13	-41	
Nozzle Flng S3	-	-28		0.76	13	-41	
Nozzle Flng T1	-	-28		0.59	22	-50	
Nozzle Flng T2	-	-28		0.59	22	-50	
Nozzle Reinforcement S1	D	-29		0.96	2	-31	
Nozzle Reinforcement S3	D	-18		0.96	2	-20	
Nozzle Reinforcement T1	D	-29		0.96	2	-31	
Nozzle Reinforcement T2	D	-29		0.92	4	-33	
Front Hd Bolting At TS	A	-48		-	-	-	
Front Hd Bolting At Cov	A	-48		-	-	-	
Eccentric Reducer	D	-16		0.96	2	-18	
Kettle Cylinder	D	-18		0.92	4	-22	
Nozzle Flng Bolting S1	A	-48		-	-	-	
Nozzle Flng Bolting S2	A	-48		-	-	-	
Nozzle Flng Bolting S3	A	-48		-	-	-	
Nozzle Flng Bolting T1	A	-48		-	-	-	
Nozzle Flng Bolting T2	A	-48		-	-	-	

Minimum Design Metal Temperature for Impact Test Exemption

Ratio = smaller of [tr*je/(tn-corr),MDP/MAP]   UCS-66(b)(1)(b)

Component	Ratio	tr	je	tn	corr	Ratio	MDP	MAP
Shell Cylinder	0.97	35.71	1	39.7	3	-	-	-
Front Head Cylinder	0.96	27.55	1	31.7	3	-	-	-
Front Head Cover	0.97	167.78	1	176	3	0.88	6.02	6.8
Shell Cover	0.98	45.92	1	49.7	3	-	-	-
Front Tubesheet	1.00	193.21	1	197	3	0.88	8.07	9.16
Front Head Flng At TS	-	-	-	-	-	0.93	7.6	8.18
Front Head Flng At Cov	-	-	-	-	-	0.90	6.13	6.79
Front Shell Flng	-	-	-	-	-	0.93	7.83	8.42
Tubes	0.34	0.81	1	2.4	0	-	-	-
Front Head Partitions	0.94	13.13	1	14	0	-	-	-
Nozzle S1	0.42	2.8	1	9.74	3	-	-	-
Nozzle S2	0.52	1.68	1	6.23	3	-	-	-
Nozzle S3	0.55	3.68	1	9.74	3	-	-	-
Nozzle T1	0.83	11.45	1	16.76	3	-	-	-
Nozzle T2	0.84	12.71	1	18.04	3	-	-	-
Nozzle Flng S1	-	-	-	-	-	0.76	7.8	10.2
Nozzle Flng S2	-	-	-	-	-	0.76	7.8	10.2
Nozzle Flng S3	-	-	-	-	-	0.76	7.8	10.2
Nozzle Flng T1	-	-	-	-	-	0.59	6	10.2
Nozzle Flng T2	-	-	-	-	-	0.59	6	10.2
Nozzle Reinforcement S1	-	-	-	-	-	-	-	-
Nozzle Reinforcement S3	-	-	-	-	-	-	-	-
Nozzle Reinforcement T1	-	-	-	-	-	-	-	-
Nozzle Reinforcement T2	-	-	-	-	-	-	-	-
Front Hd Bolting At TS	-	-	-	-	-	-	-	-
Front Hd Bolting At Cov	-	-	-	-	-	-	-	-
Eccentric Reducer	0.96	54.6	1	59.7	3	-	-	-
Kettle Cylinder	0.92	47.32	1	54.7	3	-	-	-
Nozzle Flng Bolting S1	-	-	-	-	-	-	-	-
Nozzle Flng Bolting S2	-	-	-	-	-	-	-	-
Nozzle Flng Bolting S3	-	-	-	-	-	-	-	-
Nozzle Flng Bolting T1	-	-	-	-	-	-	-	-
Nozzle Flng Bolting T2	-	-	-	-	-	-	-	-

Hydrostatic Test Pressure - ASME VIII-1 2017 UG-99 Factor: 1.3

Shell Side: 101.4 bar Tube Side: 78 bar

Component	Material	Side	Design Temp C	Stress N/mm2	Test Stress N/mm2	Stress Ratio
Shell Cylinder	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Front Head Cylinder	SA-516 K02700 Grd 70 Plate	T	300	134.92	137.9	1.0221
Front Head Cover	SA-266 K03506 Grd 2 Forgin	T	300	129.18	137.9	1.0675
Shell Cover	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Front Tubesheet	SA-266 K03506 Grd 2 Forgin	S	300	129.18	137.9	1.0675
Front Tubesheet	SA-266 K03506 Grd 2 Forgin	T	300	129.18	137.9	1.0675
Front Head Flng At TS	SA-266 K03506 Grd 2 Forgin	T	300	129.18	137.9	1.0675
Front Head Flng At Cov	SA-266 K03506 Grd 2 Forgin	T	300	129.18	137.9	1.0675
Front Shell Flng	SA-266 K03506 Grd 2 Forgin	S	280	132.16	137.9	1.0434
Tubes	SA-179 K01200 Smls. tube	T	300	91.89	92.39	1.0054
Nozzle S1	SA-106 K03006 Grd B Smls.	S	280	117.9	117.9	1
Nozzle S2	SA-106 K03006 Grd B Smls.	S	280	117.9	117.9	1
Nozzle S3	SA-106 K03006 Grd B Smls.	S	280	117.9	117.9	1
Nozzle T1	SA-106 K03006 Grd B Smls.	T	300	117.9	117.9	1
Nozzle T2	SA-106 K03006 Grd B Smls.	T	300	117.9	117.9	1
Nozzle Flng S1	SA-105 K03504 Forgings	S	280	132.16	137.9	1.0434
Nozzle Flng S2	SA-105 K03504 Forgings	S	280	132.16	137.9	1.0434
Nozzle Flng S3	SA-105 K03504 Forgings	S	280	132.16	137.9	1.0434
Nozzle Flng T1	SA-105 K03504 Forgings	T	300	129.18	137.9	1.0675
Nozzle Flng T2	SA-105 K03504 Forgings	T	300	129.18	137.9	1.0675
Nozzle Reinforcement S1	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Nozzle Reinforcement S3	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Nozzle Reinforcement T1	SA-516 K02700 Grd 70 Plate	T	300	134.92	137.9	1.0221
Nozzle Reinforcement T2	SA-516 K02700 Grd 70 Plate	T	300	134.92	137.9	1.0221
Front Hd Bolting At TS	SA-193 G41400 Grd B7 Bolt(	T	300	172.37	172.37	-
Front Hd Bolting At Cov	SA-193 G41400 Grd B7 Bolt(	T	300	172.37	172.37	-
Eccentric Reducer	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Kettle Cylinder	SA-516 K02700 Grd 70 Plate	S	280	136.41	137.9	1.0109
Nozzle Flng Bolting S1	SA-193 G41400 Grd B7 Bolt(	S	280	172.37	172.37	-
Nozzle Flng Bolting S2	SA-193 G41400 Grd B7 Bolt(	S	280	172.37	172.37	-
Nozzle Flng Bolting S3	SA-193 G41400 Grd B7 Bolt(	S	280	172.37	172.37	-
Nozzle Flng Bolting T1	SA-193 G41400 Grd B7 Bolt(	T	300	172.37	172.37	-
Nozzle Flng Bolting T2	SA-193 G41400 Grd B7 Bolt(	T	300	172.37	172.37	-

Hydrostatic Test Pressure - UG-99 - Bolting exception

Component	Material	1.3*LSR*All.Stress N/mm2	90% Yield N/mm2
Front Hd Bolting At TS	SA-193 G41400 Grd B7 Bolt(	224	651.55
Nozzle Flng Bolting S1	SA-193 G41400 Grd B7 Bolt(	224	651.55

***** Extreme Fiber Elongation - UG-79 *****

Cylinders formed from plate, ef = (50t/Rf)(1-Rf/Ro) (Ro = infinity)

Double curvature parts (heads) ef = (75t/Rf)(1-Rf/Ro) (Ro = infinity)

Component	Material * governing geometry *	thk,t	Radius,Rf	Elong,e	
		mm	mm	%	Max
Shell Cylinder	SA-516 K02700 Grd 70 Plate	40	620	3.13	5
Front Head Cylinder	SA-516 K02700 Grd 70 Plate	32	616	2.53	5
Shell Cover	SA-516 K02700 Grd 70 Plate	50	293.49	12.78	5 *

* Warning - Heat treating may be required per UG-79

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