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RAPHAEL ***(ReActor for Process heat, Hydrogen And ELectricity generation)***

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RAPHAEL IP – DELIVERABLE D-CT1.6

Synthesis on Design studies for IHX, isolation valves and coolers

Executive summary

This document gives the overview of the design studies carried out so far on both current options of either plate-type or helical-tube bundle Intermediate Heat Exchanger, and on the Water/Helium Cooler. It has been deemed preferable to include the contributions of the WP-CT1 Partners as they have been issued, and not to try to merge them and derive conclusions, which would have been premature at this early stage of the design, e.g., neither option of the IHX can be chosen conclusively as reference or backup design, pending the necessary information. As for the contents, the review of relevant experience has been carried out as the first step of the design approach in both contributions of CEA and AMEC. Sizing of the gasket-free, welded-plate IHX has been carried out by CEA, basing on available heat transfer correlations, however at the reduced 850°C steady-state inlet temperature of the primary Helium, under consideration of the creep strength of the proven nickel alloy 617 as the construction material. Assessment of the thermal stresses during transients will have to wait for the results of an IHX mock up test, scheduled for early next year, in a Helium test rig facility. The AMEC report, besides general design requirements and the review of the UK relevant experience in the field of heat transfer equipment, provides the rationale of the several candidate materials for development for the upper limit of the operating temperature range of VHTR/RAPHAEL. Finally, AREVA NP have preliminary sized the helical-tube IHX at the nominal 950°C upper temperature limit, focusing on the pressure loss requirement as the limiting factor. For a power removal duty of 610 MW, the IHX unit power has been chosen as 152.5 MW giving four units. IHX and reactor pressure vessels are connected by means of co-axial cross ducts. In the general arrangement, each IHX-pressure vessel assembly occupies a quadrant. The AREVA NP reports illustrates also the about 1.2 MW water/helium plate-type cooler of the blower, which is almost the same water/helium cooler type proposed by CEA, however with graphite gaskets. This choice has to be confirmed for validation by testing.

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Note notes

This document includes the contributions of AMEC, AREVA NP-F and CEA

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Contents

1. INTRODUCTION

This document collects the contributing reports of the WP-CT1 Partners on the Intermediate Heat Exchanger and the Water/Helium Cooler. It has been deemed preferable for this first issue, namely, to include the contributions as they have been prepared, and not to try to merge them and derive conclusions, which would have been premature at this early stage of the design. Thus, this document is a review of the activities carried out so far, rather than a synthesis report, topics of which such as Objectives, Options and Hypotheses of the study, Experience from previous facilities, Study approach, Description of the component, Preliminary thermal-hydraulics, Design Code and mechanical sizing, References, which would have deserved dedicated chapters, are preliminarily dealt with within the three reports: Chapter 2 written by CEA, Chapter 3 written by AMEC and Chapter 4 written by AREVA NP.

The isolation valves and topics peculiar to the synthesis report such as the conclusion of the study, the proposed reference and backup designs, further studies and required R&D activities, which cannot be treated conclusively to date, will be included in the next revision of this document, most likely after completion of testing on a mockup of the plate-type IHX and on a small-size, welded-plate water/helium cooler, scheduled for early next year.

2. IHX AND COOLER LITERATURE REVIEW INCLUDING DESIGN STUDY (BY CEA)

This document is a part of the deliverable D-CT1.7 - Design study for IHX, isolation valves and cooler done within the RAPHAEL FP6 Project.
This work is the CEA contribution to which AREVA NP France and Germany parts have to be added.
The objective of this Work package is to develop the key technologies for heat exchangers, isolation valves and the reactor vessel of a modern HTR and VHTR with indirect cycle.
The key issue for the intermediate heat exchanger (IHX) is relative to the compactness and the applied loads on the component (temperature and pressure difference).
The development of such a heat exchanger is a real challenge as no present technology can meet the anticipated requirements in term of heat exchange performances, pressure drops, mechanical resistance…. A feasibility study is proposed for two types of concepts: plate and tube concepts.
Studies of tube and plate concepts are respectively carried out by AREVA NP and CEA.
Thermal hydraulic and thermo-mechanical analyses will be carried out to assess the performances of the IHX plate concept.

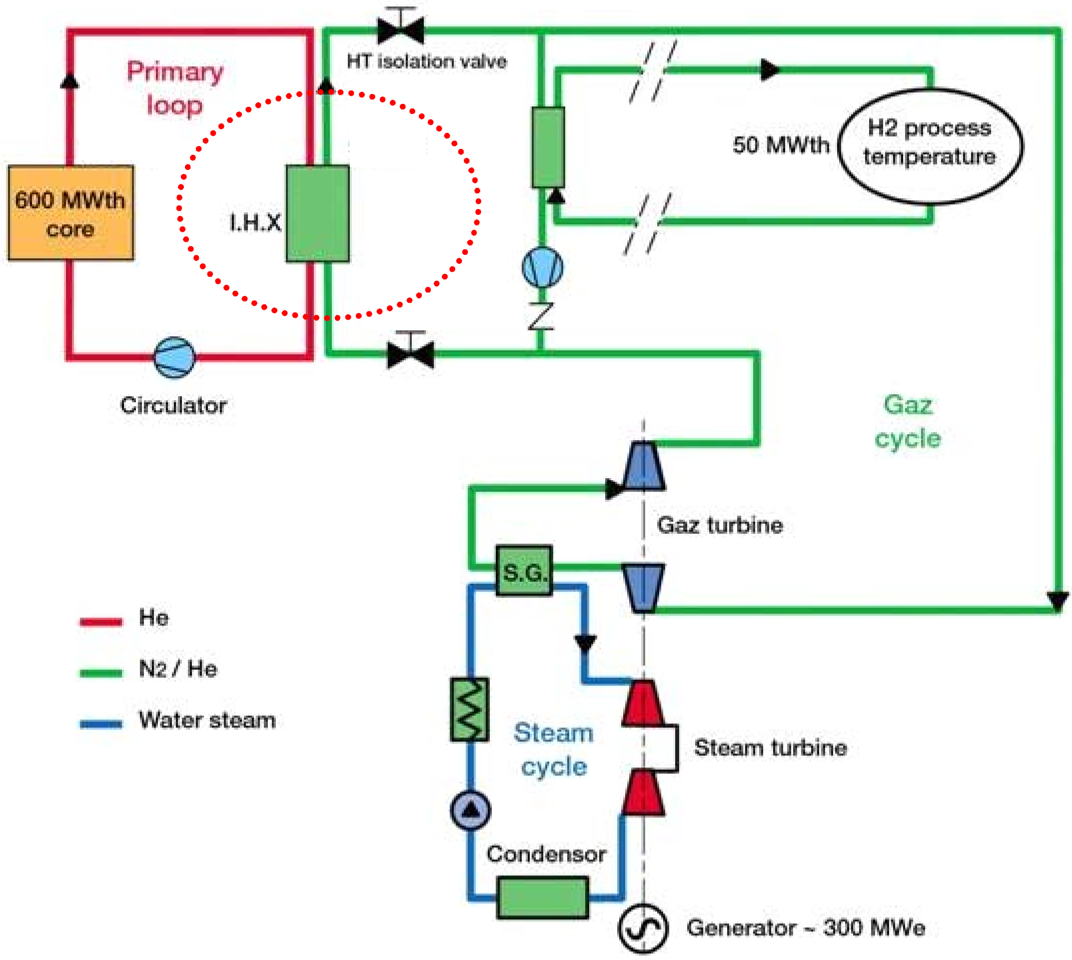
The helium/water heat exchangers used as coolers are to be studied for VHTR. The key issues for these heat exchangers are relative to safety and compactness. For direct cycle, the design must be also compatible with potential heat applications like desalination or district heating using the energy extracted by the water circuit. For indirect cycle, the coolers could be used to get low temperature (lower than 80°C) of the motor of the main blower.

High performances hot helium valves could be also necessary on the secondary pipes to isolate the reactor in case of failure of the IHX. A design study is proposed based on past experience in Germany and also on a technical survey of present technologies (e.g. AGR).

2.1 INTRODUCTION

New concepts based on High Temperature gas cooled Reactors (HTRs and VHTRs) with indirect cycles gas turbine have become potentially interesting for future power generation. These concepts, theoretically able to provide higher efficiency than a classical steam cycle, need high efficiency components to reach high level of performances.
Designed for electricity production and heat for hydrogen massive production, high working temperatures are required. Heat exchangers have to resist to high level of temperatures, to thermal gradients and to important pressures. These working conditions

generates important level of stresses inside the materials and furthermore during transients cases.
Figure 1 reminds the VHTR indirect cycle.



Two potential technologies are studied to fulfil to IHX specifications:

- Tubular concept which generates quite significant volume but which has already been experimented by the past (typically German and English gas cooled reactors)
- Plate concept which is more compact than previous one but certainly more sensitive to thermal gradients.

Thus, no directly available compact technologies exist since they are often used for cryogenic, air cooling and automotive systems which are far from the HTR or VHTR requirements. CEA contribution is limited to compact plate technology.

After reminding working conditions of the IHX under the indirect cycle, a limited review of potential plate technologies available on the market is done with main physical phenomena generated by the concerned geometry. The third and fourth parts are dedicated to preliminary thermal hydraulic sizing of the IHX and of the cooler based on available correlations.

2.2 IHX WORKING CONDITIONS

	Primary (Hot side)		Secondary (Cold side)
Fluid	Helium		Helium
Mass flow rate (kg/s)	210		210
Inlet Temperature (°C)	950		340
Outlet Temperature (°C)	390		900
Thermal efficiency (%)		90	
Inlet Pressure (MPa)	5		5
Allowed pressure drops	2% Pnom. (1.5% core)		2% Pnom. (1.5% core)

Table 1: RAPHAEL nominal working conditions for an IHX plate technology

2.3 IHX POTENTIAL TECHNOLOGIES

2.3.1 Classification of heat exchangers

Heat exchangers could be classified in many different ways such as according to transfer processes, number of fluids, surface compactness, flow arrangements, heat transfer mechanisms, type of fluids (gas–gas, gas–liquid, liquid–liquid, gas–two-phase, liquid–two-phase, etc.) and industry. Heat exchangers can also be classified according to the construction type (Figure 2). Refer to Shah and Mueller [2] for further details.

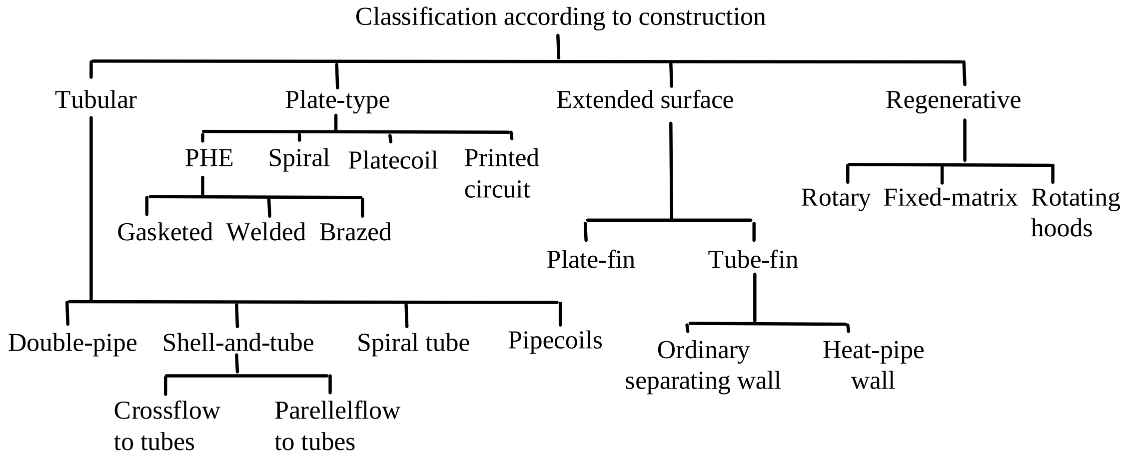


Fig. 2 Heat exchangers classification (Shah et Mueller [2])

Taking into account following needs: high temperature, high pressure and compactness lead to eliminate some technologies described previously. Finally, only plate concepts without gaskets are kept and the next part of document only focuses on Plate Heat Exchangers such as Printed Circuit Heat exchanger and Plate Fin Exchanger.

These technologies can also be divided in 2 categories:
Primary surface Heat exchanger: stack of corrugated or non corrugated plates which give alternatively a hot and a cold channel. The exchange surface is given by the developed area of the plates.
Secondary surface Heat exchanger: similar to primary surface HX but folded thin plates are inserted within the channels generating fins. Fins increase the exchange surface and may also promote turbulence depending on their shape. Since they are brazed on the plate, they also have a mechanical function.

2.3.2 Description of potential heat exchangers for VHTR application

2.3.2.1 Printed Circuit Heat Exchangers

PCHEs are constructed from flat metal plates into which fluid flow channels are chemically milled. The chemical milling technique is similar to that employed for etching electrical printed circuits, and hence is capable of producing fluid circuits of unlimited variety and complexity. The channel may be straight or wavy.

The major benefits of this technology are:

- Very compact heat exchangers;
- Core design is readily tailored for containment of exceptionally high pressure. Design pressures in excess of 500 bars are feasible;
- The all-metal diffusion bonded and welded construction of PCHEs minimise design temperature constraints. Materials such as austenitic stainless steels are suitable for temperatures ranging from cryogenic to 800°C.
- High efficiency heat exchanger;
- Used to very harsh working conditions (off-shore applications).

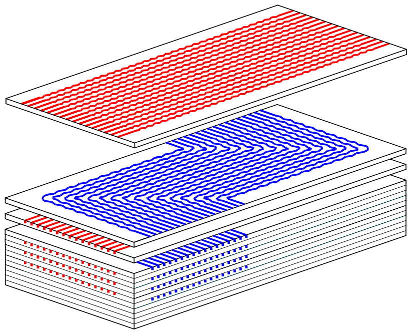
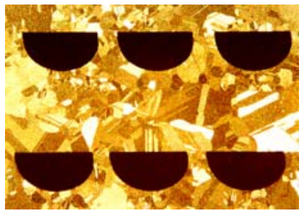


Plate stacking prior to diffusion bonding



Section through diffusion bonded core

Fig. 3 *HEATRIC concept – courtesy of HEATRIC*

The milled plates are then stacked and diffusion bonded together to form strong, compact, all-metal heat exchange cores (see figure 3). The diffusion bonding is a "solid-state joining" process entailing pressing metal surfaces together at temperatures below the melting point, thereby promoting grain growth between the surfaces. Under carefully controlled conditions, diffusion bonded joints reach parent metal strength, and stacks of milled plates are converted into solid blocks containing the fluid flow passages. The blocks are then welded together to form the complete heat exchange core for the thermal duty. Finally, fluid headers and nozzles are welded to the core in order to direct the fluids to the appropriate sets of passages. No gaskets or brazing material - potential sources of leakage, fluid incompatibility and temperature limitations - are required for exchanger assembly.
Classically, the hydraulic diameters are less than 1 mm which leads to laminar flow regimes on both sides. Moreover, the channels can be shaped (straight or wavy) in order to generate turbulence inside.
This technology seems to be very promising in terms of high pressure / temperature resistance and compactness. Furthermore, the diffusion bounding process leads to a single-metal heat exchanger, without secondary-metal supply.
Within the HTRE program, such a technology was experimentally tested giving good results in term of thermal effectiveness (around 95%) and in term of resistance during thermal cycling.

2.3.2.2 Plate Fin Heat Exchangers

The secondary surface heat exchangers are also called plate-fin heat exchangers (figure 4). They offer high process integration possibilities (12 simultaneous different streams and more in one single heat exchanger) and high effectiveness under close temperature approaches (1 to 2 °C) in large variety of geometrical configurations or applications (cryogenic, air cooling and automotive systems).

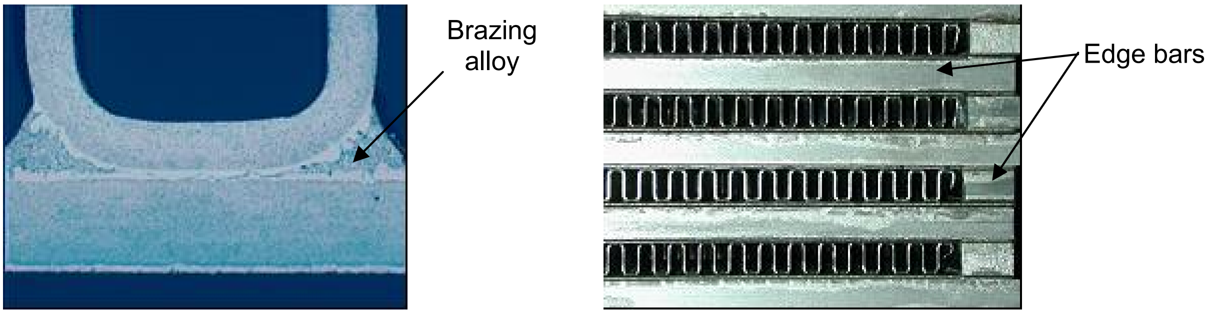
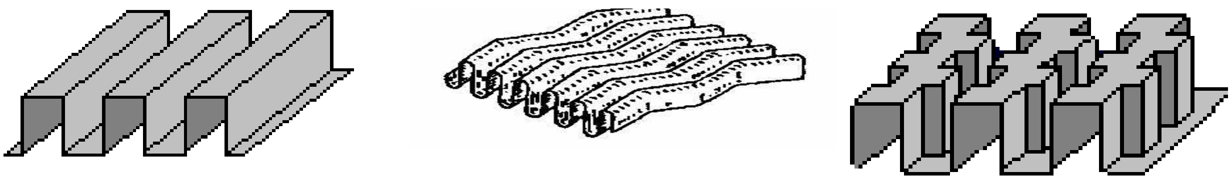


Fig. 4Plate-fin heat exchangers – courtesy of NORDON



(a) plain *(b) wavy* *(c) Offset strip*

Fig. 5 *Different types of fins used in compact Plate Fin Heat Exchangers*

A plate fin heat exchanger consists of fins (corrugated sheets) separated by parting sheets (flat plates), forming passages which are closed by bars, with openings for the inlet and outlet of fluids. The heat exchangers are manufactured in the following way:

- The production of fins consists in passing a band of metal through a special press;
- Prior to stacking, fins, separating sheets and bars are degreased, acid etched, detergent washed, rinsed and oven dried. Once stacked, the assembly is then brazed in a vacuum furnace.
- A brazed core is then fitted with headers and nozzles.

These heat exchangers can provide secondary surface up to 90% of the overall heat transfer surface. Several types of fins are available (figure 5), and the selection depends essentially on the application. For air flows, louvered fins are extensively used, while for process applications (single and two-phase) plain or offset strip fins are used.

This technology of heat exchangers is mainly characterised by the following factors:

- Single sheet equipped with metal inserts (fins) between 2 fluids;
- Generally, very thin sheet and fins: reduction of size and weight, high developed surface;
- Use of brazing alloy;
- No limitation of pressure difference between the fluids since the fins brazing offers high mechanical resistance;
- Manufacturers: INGERSOLL-RAND (USA), NORDON (France), MITSUBISHI (Japan)…

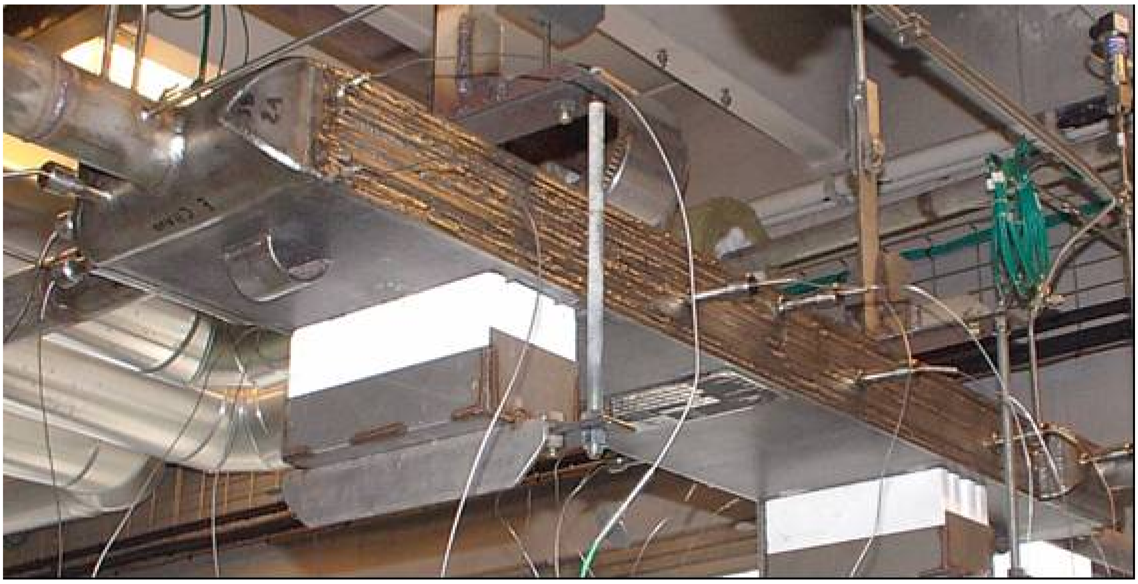


Fig. 6 Tested plate fins mock up within the HTRE project

2.4 PHYSICS OF FLOW

2.4.1 Wavy channels (used in PCHE)

In this section, the current understanding of the physics of flow and heat transfer in PCHE is described. This technology allows forming channels which can be straight or wavy (herringbone pattern). In this case, the heat transfer surface is convoluted such that the flow passage geometry does not allow the boundary layer growth. The Reynolds number for this plate heat exchanger is commonly defined with the hydraulic diameter (four times the channel volume divided by the total heat transfer surface area).

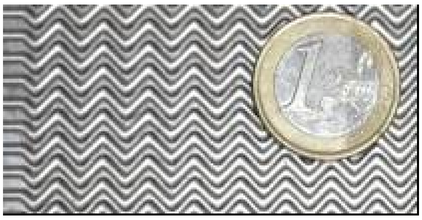


Fig. 7 Wavy channel

Compared to straight ducts, wavy geometries (see Figure 7) provide little advantage at low Reynolds numbers, and maximum advantage at transitional Reynolds numbers. However, at higher Reynolds numbers, periodic shedding of transverse vortices increases the Nusselt number with a considerable increase in the friction factor. The following are important flow mechanisms associated with wavy channels:

1. At low Re < 200, steady recirculation zones form in the troughs of the wavy passages and heat transfer is not enhanced. For higher Reynolds numbers, the free shear layer becomes unstable; vortices roll up and are conveyed downstream, thus enhancing the heat transfer.
2. Transition to turbulence occurs at Re ~ 1200, depending on the geometry. It appears that chaotic advection may contribute to the heat transfer in the transitional Reynolds number range.
3. For Reynolds numbers of 4000 and over, the flow is fully turbulent with very high pressure drops (figure 8). Thus, wavy channels provide higher heat transfer rates than plain channels, but with higher pressure drops. Ali & Ramadhyani [3] and Gschwind *et al*. [4] found that a stream wise Görtler-like vortex system forms in the transitional Reynolds number range. Although the impact on heat transfer and pressure drop is not completely clear, such a vortex system is known to increase heat transfer.

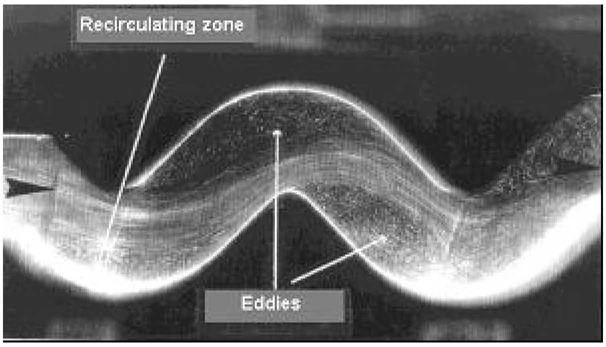


Fig. 8 Physical phenomena inside a wavy channel [5]

Wavy passages clearly offer heat transfer enhancement over plain channel passages; however, they do not offer a performance advantage (heat transfer relative to the pressure drop) over interrupted passages (OSF).

Available correlations:

For the thermal-hydraulic correlations, one needs first to define the flow regime, classically based on the Reynolds number $\mathrm{Re} = (\rho\, U\, D_h)/\mu$ where U is the bulk flow velocity (m.s^{-1}), ρ the fluid density (kg.m^{-3}), μ the dynamic viscosity (Pa.s) and D_h the hydraulic diameter given by $D_h = 4\,S\,/\,P$ with S the cross section (m^2) and P the wetted perimeter (m). Depending on the Reynolds number, three various flow regimes exist: laminar ($\mathrm{Re} \le 2300$), transitional ($2300 \le \mathrm{Re} \le 10000$) or turbulent ($\mathrm{Re} \ge 10000$). Pressure drops ΔP are computed with $\Delta P = \Lambda\,(L\,/\,Dh)(\rho\, U^2\,/\,2)$ where L is the length of the duct (m) and Λ the Darcy factor (or $f = \Lambda/4$ with *f*: Fanning factor).

For a <u>straight rectangular channel</u>, the following correlations allow to estimate <u>the Darcy coefficient</u>:

- $\mathrm{Re} \le 2\,300$: Shah & Bhatti (1987) [6] proposed for $0 < \alpha^* < 1$, $\alpha^* = 2b/2a$ with $2b$ the channel height (m) and $2a$ the channel width (m).

$$\Lambda = \begin{bmatrix} 96\,(1 - 1.3553\alpha * + 1.9467\alpha *^2 - 1.7012\alpha *^3 + \\ 0.9564\alpha *^4 - 0.2537\alpha *^5) \end{bmatrix} / \mathrm{Re}$$

- $5\,000 \leq \mathrm{Re} \leq 10^7$: Shah & Bhatti (1987) proposed the following correlation for $0 < \alpha^* < 1$:

$$\Lambda = 4\,(1{,}0875 - 0{,}1125\,\alpha^*)\;f_c$$

Where $f_c = \left(1.7372\,\ln\left(\dfrac{\mathrm{Re}}{1.964\ln \mathrm{Re} - 3.8215}\right)\right)^{-2}$

<u>For the thermal point of view,</u>

- $0 \leq \mathrm{Re} \leq 2300$: In the laminar flow regime, the heat transfer coefficient is a function of the heating conditions (fixed wall temperature or fixed heat flux). For the IHX condition, since it is a fully counter flow, the relevant condition is the fixed heat flux (see Figure 9).

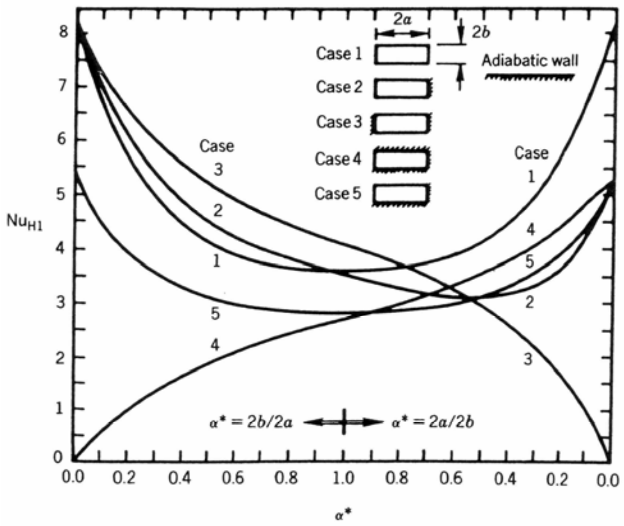


Fig. 9 Nusselt number according to the duct aspect ratio, and the heating process, for uniform heat flux

- $10^4 \leq \mathrm{Re} \leq 10^6$: The well-known Colburn correlation could be used:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}$$

<u>For a wavy channel</u> (herringbone pattern), no general correlation exists. However, Hesselgreaves (2001) [7] proposed a relation which takes into account the wavy geometrical parameters *b*, the channel width, *p* the wavy pitch length and *w* the wave height (see Figure 7).

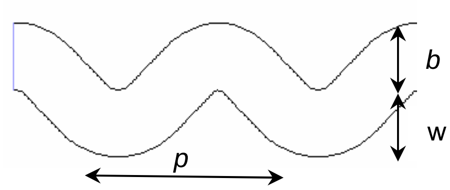


Fig. 10 Parameters for the wavy duct correlation.

The available correlations according to the flow regime are the following:

- $10^4 < \mathrm{Re} < 10^5$:

$$f = 4.8\,\mathrm{Re}^{-0.36}\left(\frac{2b}{p}\right)^{1.5} \quad ; \quad j = 0.4\,\mathrm{Re}^{-0.36}\left(\frac{2b}{p}\right)^{0.75}$$ with j the Colburn factor $j = \dfrac{Nu}{\mathrm{Re}\,\mathrm{Pr}^{1/3}}$

- $600 < \mathrm{Re} < 2300$:

$$j = 0.4\,\mathrm{Re}^{-0.4}\left(\frac{2b}{p}\right)^{0.75} \quad ; \quad f = 5\,j$$

According to the author, this relation predicts the experimental data quite well for $w/p = 0.25$

2.4.2 Offset Strip Fins

In this section, the current understanding of the physics of flow and heat transfer in plate-fins heat exchanger are described. Depending of the fins type (figure 5), the flow passage could be interrupted or uninterrupted. However, since a high thermal efficiency is required (> 90%), the offset strip fins should be recommended to fit to the specifications. So the uninterrupted flow passage will not be described here.

For interrupted flow passage, the most common geometry is the offset strip fin in figure 11. Here the fin surface is broken into a number of small sections. For each section, a new leading edge is encountered, and thus a new boundary layer development begins, and is then abruptly disrupted at the end of the fin offset length.

The objective for such flow passages is not to allow the boundary layers to thicken thus resulting in high heat transfer coefficients associated with thin boundary layers. However, the interruptions create the wake region and self-sustained flow unsteadiness (Figure 12). Separation, recirculation and reattachment are important flow features in most interrupted heat exchanger geometries. Consider, for example, the flow at the leading edge of a fin of finite thickness. The flow typically encounters such a leading edge at the heat exchanger inlet or at the start of new fins, offset strips or louvers. For most Reynolds numbers, a geometric flow separation will occur at the leading edge because the flow cannot turn the sharp corner of the fin as shown in figure 12. Downstream from the leading edge, the flow reattaches to the fin. The fluid between the separating streamline and the fin surface is recirculating. This region is called a separation bubble or recirculation zone. Within the recirculation zone, a relatively slow-moving fluid flows in a large eddy pattern. The boundary between the separation bubble and the separated flow (along the separation streamline) consists of a free-shear layer. Since free shear layers are highly unstable, velocity fluctuations develop in the free shear layer downstream from the separation point.

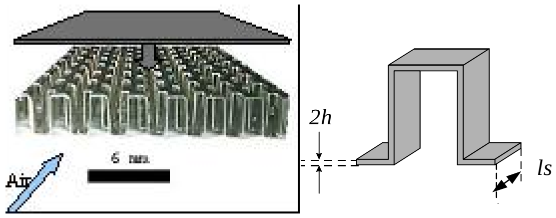


Fig. 11 Offset strip fins geometry

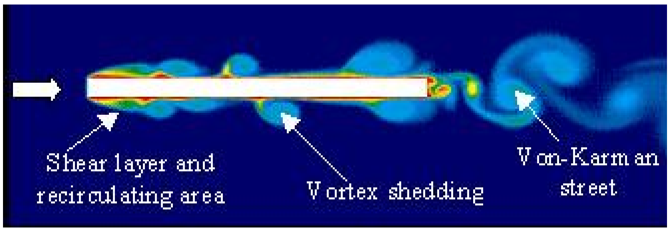


Fig. 12 Physical phenomena around a fin [8]

These perturbations are conveyed downstream to the reattachment region, and there they result in an increased heat transfer. The fin surface in contact with the recirculation zone is subject to lower heat transfer because of the lower fluid velocities and the thermal isolation associated with the recirculation eddy. The separation bubble increases the form drag, and thus usually represents an increase in pumping power with no corresponding gain in heat transfer. If the flow does not reattach to the surface from which it separates, a wake results.

A free shear layer is also manifested in the wake region at the trailing edge of a fin element. Depending on the Reynolds number and geometry, the wake from the upstream fins can have a profound impact on the downstream fin elements. The highly unstable wake can promote strong mixing that destroys the boundary layers from the upstream fins causing downstream heat transfer enhancement. However, at low Reynolds numbers, or for very close stream wise spacing of fin elements, the shear layers might not be destroyed or the next fin element might be embedded in the wake of an upstream fin element. In such cases, the low velocity and near-fin temperature of the wake will have a detrimental effect on the downstream heat transfer.

At low Reynolds numbers (less than 400), flow through the offset strip fin geometry is laminar and nearly steady, and the boundary layer effects dominate the heat transfer and friction. For intermediate Reynolds numbers (roughly 400 < Re < 1000), the flow remains laminar, but unsteadiness and vortex shedding become important. For example, at Re = 850, boundary layer restarting causes roughly a 40% increase in heat transfer over the plain channel with vortex shedding causing an additional 40% increase. Unfortunately, there is a commensurate increase in the pressure drop due to boundary layer restarting and vortex shedding. For Reynolds numbers greater than 1000, the flow becomes turbulent in the array, and chaotic advection may be important in the low Reynolds number turbulent regime. An increase with a factor of 2 or 3 in heat transfer and pressure drop over plain fins can be obtained as a result of the turbulent mixing. The important variables affecting the wake region identified are the strip length, the fin spacing and the fin thickness. The fin spacing and the strip length are responsible for the boundary layer interactions and wake dissipation; and the fin thickness introduces form drag and also affects the heat transfer performance. Higher aspect ratio, shorter strip lengths and thinner fins are found to provide higher heat transfer coefficients and friction factor.

Available correlations:

For the thermal-hydraulic correlations, one needs first to define the flow regime, classically based on the Reynolds number $\mathrm{Re} = (\rho\, U\, D_h)/\mu$ where U bulk flow velocity (m.s^{-1}), ρ the fluid density (kg.m^{-3}), μ the dynamic viscosity (Pa.s) and D_h the hydraulic diameter (m). For the OSF geometry, the geometrical parameters are given on figure 13.

w fin width
l_S fin length
h_S fin height
e fin thickness

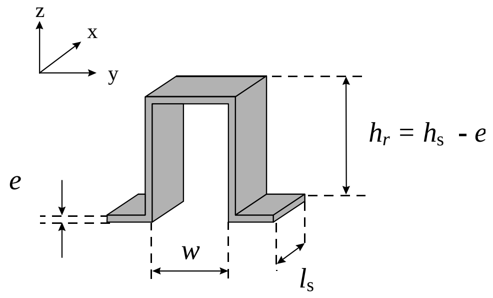


Fig. 13 Parameters for the OSF correlation

Figure 1 : The friction factor f (Fanning factor) is linked to the pressure drops ΔP per unit length L (length of the heat exchanger) by $f = (\Delta P\, D_{\mathrm{H}}) \Big/ \left[4L \left(\frac{1}{2} \rho U_0^2 \right) \right]$

Figure 2 :
The heat transfer coefficient j (Colburn factor) is linked to the temperature difference over a heat transfer surface by $j = \mathrm{Nu}_{\mathrm{D_H}} / \mathrm{Re}_{\mathrm{D_H}} \mathrm{Pr}^{1/3}$ where the Nusselt number is defined by $\mathrm{Nu}_{\mathrm{D_H}} = \frac{h D_{\mathrm{H}}}{\lambda}$ and the Prandtl number $\mathrm{Pr} = \frac{\mu\, C_P}{\lambda}$ with C_P the specific heat capacity (J.kg^{-1}.°C^{-1}), λ thermal conductivity (W.m^{-1}.°C^{-1}).

The correlations of Manglik & Bergles (1995) [9] have been developed from the data of Kays & London, 1984 [10], in the following range:

$0.646 \leq D_H \leq 3.414$ (mm)
$0.135 \leq w/h_r \leq 1.034$
$0.012 \leq e/l_s \leq 0.060$
$0.038 \leq e/w \leq 0.132$

The hydraulic diameter is defined as follow:

$$D_H = \frac{4wh_r l_s}{2(wl_s + h_r l_s + eh_r) + ew}$$

According to authors, the range of validity of these correlations is : $300 < \mathrm{Re}_{D_H} < 10000$:

$$f = 9.6243 \mathrm{Re}_{D_H}^{-0.7422} \left(\frac{w}{h_r}\right)^{-0.1856} \left(\frac{e}{ls}\right)^{0.3053} \left(\frac{e}{w}\right)^{-0.2659} \times \left[1 + 7.669 \times 10^{-8} \mathrm{Re}_{D_H}^{4.429} \left(\frac{w}{h_r}\right)^{0.920} \left(\frac{e}{l_s}\right)^{3.767} \left(\frac{e}{w}\right)^{0.236}\right]^{0.1}$$

$$j = 0.6522 \mathrm{Re}_{D_H}^{-0.5403} \left(\frac{w}{h_r}\right)^{-0.1541} \left(\frac{e}{l_s}\right)^{0.1499} \left(\frac{e}{w}\right)^{-0.0678} \times \left[1 + 5.269 \times 10^{-5} \mathrm{Re}_{D_H}^{1.340} \left(\frac{w}{h_r}\right)^{0.504} \left(\frac{e}{l_s}\right)^{0.456} \left(\frac{e}{w}\right)^{-1.055}\right]^{0.1}$$

Using these correlations, all the experimental results are predicted with an error less than 20 %.
Based on these correlations, a sizing of the IHX is carried out.

2.5 SIZING OF THE IHX

Using nominal working conditions (see paragraph 1) of the IHX and considering a plate fin technology, the sizing is done using a method developed by R.K.SHAH (1981) [11].

To size the IHX, further information about plate fins heat exchanger are necessary:
The plate is made of 3 different areas: 2 distribution areas to feed and to collect the fluid and a thermal active area in which the main heat transfer occurs. To optimise thermal performances and to limit thermal gradient along the width of the heat exchanger, it is wished to avoid fluid mal distribution within the active thermal area. To limit mal

distribution, pressure drop and thermal exchange have to be limited in the distribution area; it can be achieved by using different type of fins as illustrated in figure 14.

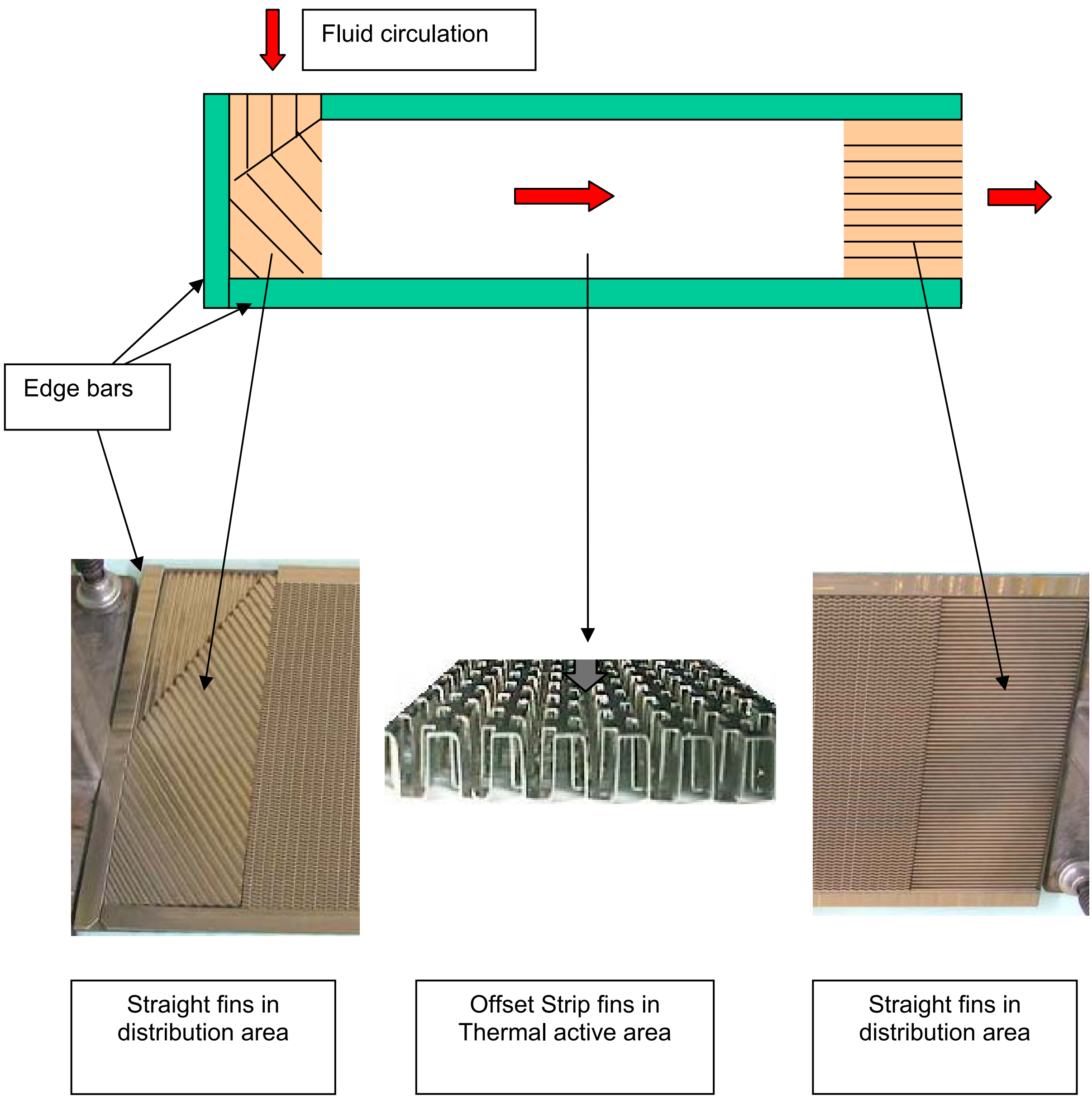


Fig.

Fig.14 Typical internal plate with different brazed fins

Since it is not possible to cancel thermal exchange in distribution area, a symmetrical feed of fluids in the heat exchanger should limit distortion linked to cross flow (figure 15).

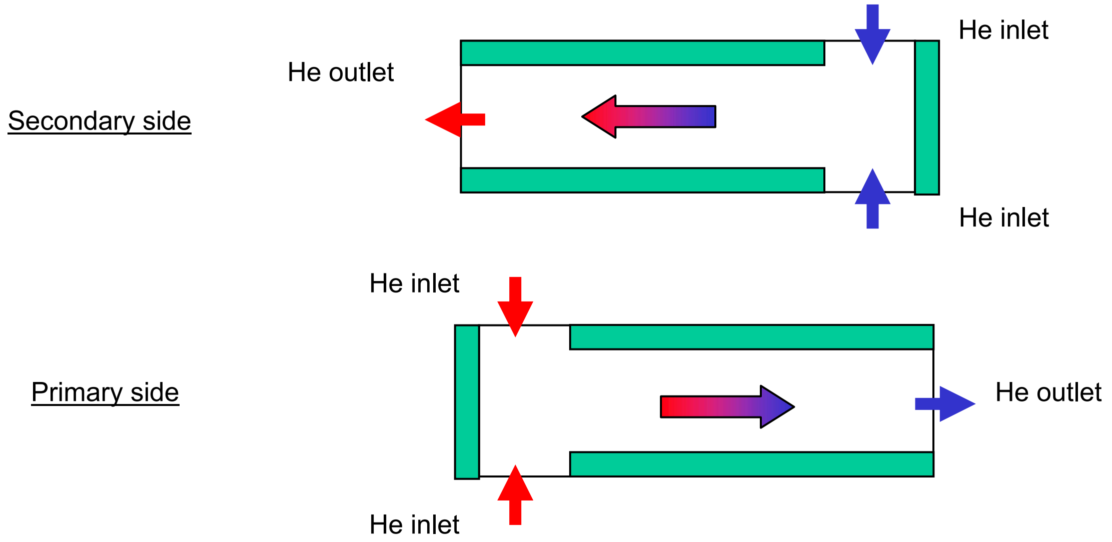


Fig. 15 Flow configuration

Generally, fins in distribution areas are sized to insure mechanical resistance only and the path between 2 fins is quite large compared to fins path located in thermal active area. Their low contribution to thermal exchange is usually not taken into account for the sizing and it may be considered as a safety margin in term of thermal performances.

2.6 ASSUMPTIONS TO SIZE THE IHX

1. The whole IHX is made of several modules in parallel. The number of modules is equal to 48. This important number of modules allows limiting the height of the stack and has a benefit effect on the generated stress.
2. The width of the plate is equal to 700 mm which is typically the size of a vacuum furnace in which brazing operation is done.
3. The width of edge bars is equal to 40 mm.

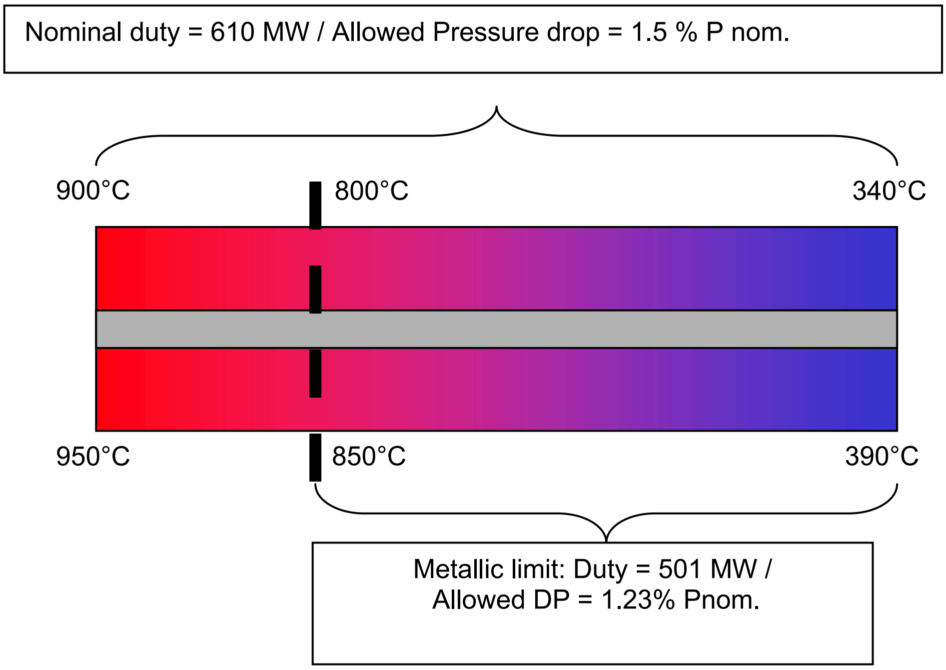


Fig. 16 Working conditions considerations

From materials point of view, it seems reasonable to split the IHX into 2 stages of heat exchangers in series to reach temperatures around 950°C; the first one made of ceramic or ODS material to achieve a part of the duty at highest temperature and the second one made of metallic material to work in lower range of temperature typically below 850°C.

Therefore, the IHX sizing given in table 2 is based on the metallic stage and the expected duty is then limited. In the same way, allowed pressure drops are decreased regarding to initial specifications.

2.7 RESULTS OF THE SIZING

	Hot side		Cold side
Fluid	Helium		Helium
Inlet data			
Inlet Temperature (°C)	850		340
Inlet Pressure (MPa)	5		5
Mass flow rate (kg/s)	210		210
Geometrical data			
Number of modules		48	
Total width (mm)		700	
Active width (mm)		620	
Active thermal length (mm)		575	
Total length (mm)		1075	
Number of channels / fluid / module		99	
Height of a module (mm)		455	
Performances			
Calculated outlet temperature (°C)	390		800
Duty (MW)		501	
Pressure drop (Mpa) (% Pnom)	0.055 (1.11% Pnom)		0.052 (1.04% Pnom)

Table 2 : IHX sizing

2.8 CHARACTERISTICS OF THE FINS

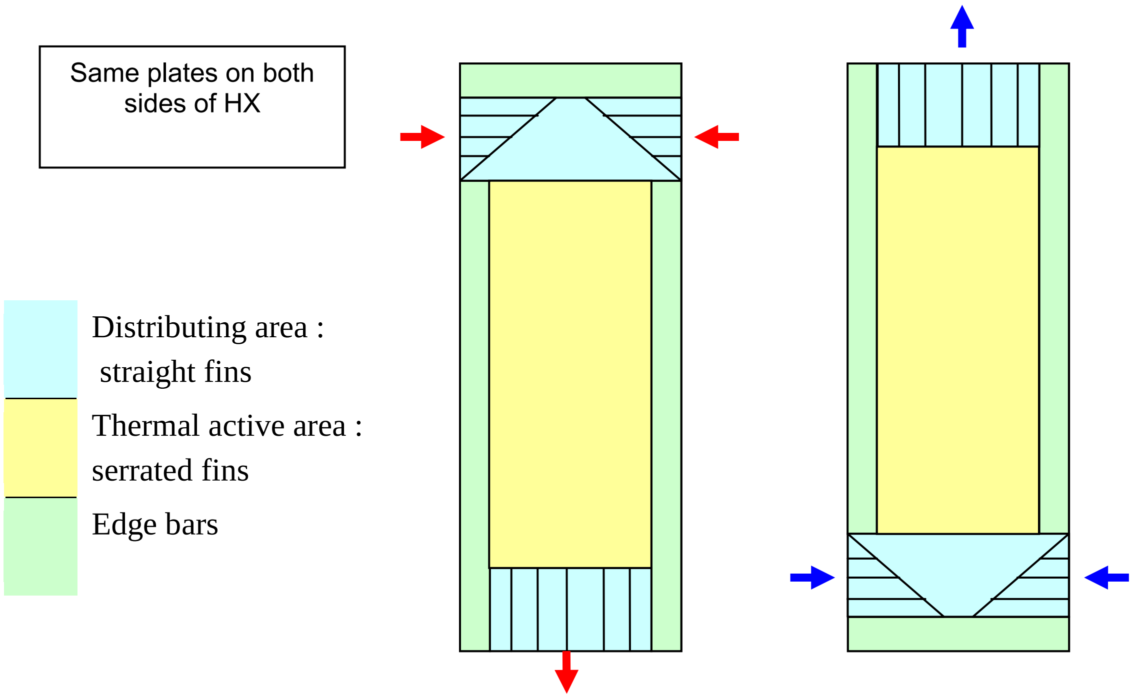


Fig. 17Internal pattern of the plates

	Hot side	Cold side
Distributing area (blue part)		
Type of fins	Straight	Straight
Thickness (mm)	0.2	0.2
Height (mm)	1.5	1.5
Density (Fin per Inch)	6	6
Hydraulic diameter (mm)	1.97	1.97
Thermal active area (yellow part)		
Type of fins	OSF	OSF
Thickness (mm)	0.2	0.2
Height (mm)	1.5	1.5
Density (Fin per Inch)	20.5	20.5
Hydraulic diameter (mm)	1.12	1.12
Length of fins (mm)	5	5
Thickness of separating sheets (mm)	0.8	

Table 3 : Fins characteristics

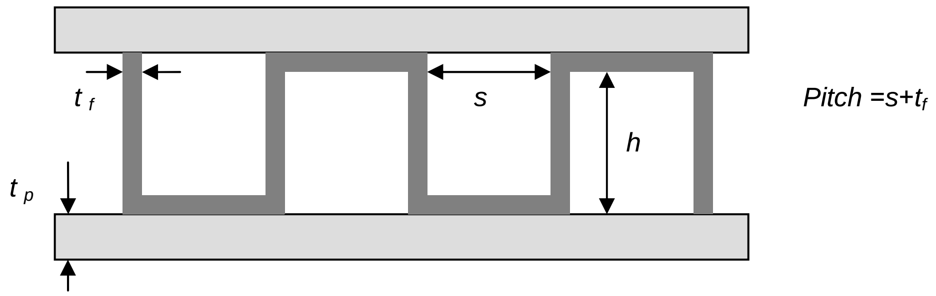
2.9 SIMPLIFIED MECHANICAL CALCULATION BASED ON FINS SIZE

A simplified approach is used to check the effect of fins parameters on generated stresses by pressure only. Of course, to go further, FEM calculations and tests would be required.

Considering channel under pressure (nominal working conditions and accidental ones), generated tensile stress may be estimated as a function of space (*s*) between 2 adjacent fins, thickness of fin (t_f) and pressure (*P*):

$$\sigma = \frac{P \times s}{t_f}$$

To equilibrate this stress generated inside the channel, the plate, on which fins are brazed, can be submitted to tensile, bending or shear stress.



Therefore each potential stress term is estimated with characteristics of the fins and separating plate:

- Tensile stress:

  - $$\sigma = \frac{P \times h}{t_p}$$

- Bending stress:

  - $$\sigma = \left(\frac{pitch}{t_p}\right)^2 \frac{P}{2}$$

- Shear stress:

$$\tau = \frac{P \times pitch}{2\,t_p}$$

Considering nominal working conditions means almost equilibrated pressure between both sides of the heat exchanger: at hot end of the heat exchanger, the considered pressure is equal to pressure drop of cold side.

In accidental conditions, it is necessary to consider P = 5 MPa (total depressurisation on side of the IHX).

	Nominal working conditions	Accidental conditions

	(P = 0.05 MPa)	(P=5 MPa)
Tensile stress within the fins	0.27 MPa	26 MPa
Tensile stress within plate	0.08	8.1
Bending stress within plate	0.06	6
Shear stress within plate	0.04	3.9

This simplistic approach shows that pressure effects are quite limited under nominal working conditions. In accidental conditions, generated stresses become more significant especially in the fins. The level of stress is the same order of magnitude than creep rupture strength of the material. As it was said previously, further investigations would be necessary from a mechanical point of view, the aim of this table is just to check if the considered fin sizes are acceptable or not.

Of course, thermal gradient effects should be added and no consideration about corrosion is taken into account.

2.10 COOLERS

For direct cycles, the use of precooler and intercooler in front of 2 compression stages allowed generation of hot water for different applications such as desalinisation of sea water or district heating.
For indirect cycles, coolers are also necessary but the application is different since they are used to cool some components. Typically, the main blower has to be cooled and the use of a compact technology may be useful to save space inside the vessel.
For this application different technologies are available, such as plate fin exchanger or eventually printed circuit heat exchanger. Nevertheless, it was decided to focus the study on a other type of technology. It is a Compabloc Heat exchanger which is developed by Alfa Laval. Usually, this type of plate heat exchanger is used when fouling fluids are used since it is a wholly dismountable apparatus and is fabricated also, under a different series name, as non dismountable apparatus without gaskets as standard fabrication, when stringent gas leak tightness is requested. The description of this Compabloc is made in the AREVA NP – France part of this deliverable and therefore it is not entirely repeated in this CEA contribution.
A cooler is necessary on helium test loop at ENEA to test a recuperator. This recuperator has already been tested in air and it is wished to check the influence of helium on its performances. Consequently, a Compabloc cooler has been bought for the helium loop and it is the opportunity to test it later at CEA.
Since this technology is a good candidate for main blower cooling, a sizing was done using correlations of literature and tests at CEA will allow checking these data.

A Compabloc Heat Exchanger is made of a stack of corrugated plates which are fed in cross flow. Depending on the duty and allowed pressure drops, Alfa Laval provides different standard sizes of plates and the internal number of plates can be adjusted.

2.11 MAIN BLOWER COOLING SPECIFICATION

	Helium		Water
Inlet temperature (°C)	96		30
Mass flow rate (kg/s)	14.35		8.6
Inlet pressure (MPa)		5.5	
Duty (MW)		1.2	

Table 4: Main blower cooling specification

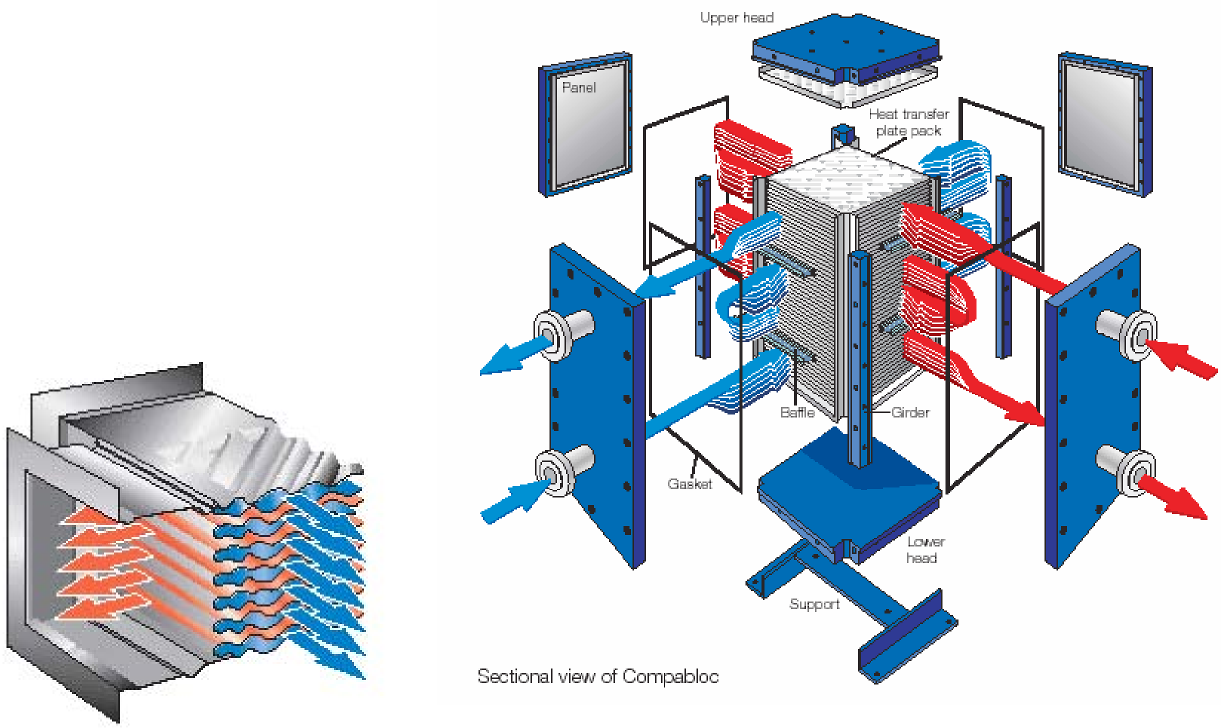


Fig. 18 Compabloc technology

Since ENEA required duty for recuperator tests is quite low, the number of plates and the plate size are limited (smallest Compabloc manufactured by Alfa Laval: 30 plates and typical size: 150 * 150 mm). Such prototype can not fit to cooler specification given in table 4.

2.12 HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF CORRUGATED CHANNELS

2.12.1 Plate description

The classical corrugated channel is characterised by (figure 19):

- the heat transfer length (L_{ht})
- the port to port length (L_{port})
- the plate width (b)

- the port diameter ($\varnothing_{port}$)
- <u>the corrugation angle (α, relative to the main flow direction)</u>
- <u>the corrugation pitch (p)</u>
- <u>the corrugation height (e)</u>

Only underlined items are necessary for Compabloc technology.

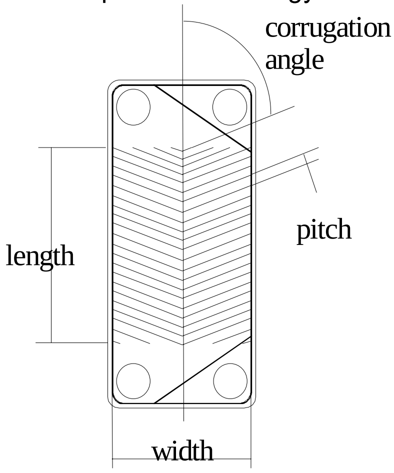


Fig. 19 Plate geometry

Generally, the manufacturers of plate heat exchangers have for a given size two types of plates. Plates with a low corrugation angle ($\alpha \approx 30°$) and plates with a higher corrugation angle ($\alpha \approx 60°$). Plates with a low corrugation angle are named 'soft plates' and those with a high corrugation angle 'hard plates'. For Compabloc Heat Exchanger, the angle is unchanged and equal to 45°.

2.12.2 Geometric characteristics

<u>Cross sectional area</u>

The cross sectional area of a corrugated channel is given by:

$$Ac = b\ e \qquad (1)$$

The cross sectional area is identical for all the channels, and the total cross sectional area is obtained multiplying the individual cross sectional area by the number of channel per fluid in parallel (*Nc*).

The fluid velocity is given by:

$$u = \frac{\dot{M}_i}{\rho\ Ac} \qquad (2)$$

where $\dot{M}_i$ is the channel mass flow rate.

$$\dot{M}_i = \frac{\dot{M}}{Nc} \qquad (3)$$

<u>Heat transfer surface</u>

The heat transfer surface is required to estimate the heat flux from the local heat transfer coefficient (α). For a plate heat exchanger either the projected surface or the developed surface is used.

The projected heat transfer surface is given by:

$$S_{pr} = L_{ht}\ b \tag{4}$$

The developed heat transfer surface is given:

$$S_{dev} = L\ B\ \Phi \tag{5}$$

Where Φ is the surface extension coefficient. For industrial plates this coefficient ranges between 1.1 and 1.5. The exact value of this coefficient depends on the corrugation pitch and height, but also on the corrugation profile and angle.
The total heat transfer surface of the plate heat exchanger is:

$$S_{tot} = (2\ Nc - 1)\ S_i \tag{6}$$

if there are the same number of channels for the hot and cold fluid, and

$$S_{tot} = 2\ Nc\ S_i \tag{7}$$

if there is one more channel for one of the two fluids

<u>*Hydraulic diameter*</u>
The general definition of the hydraulic diameter is:

$$d_h = \frac{4 \times Channel\ volume}{Heat\ transfer\ surface} \tag{8}$$

for a corrugated channel it gives:

$$d_h = \frac{2\ e}{\Phi} \tag{8}$$

As Φ is not always known, a simplified definition of the hydraulic diameter may be used:

$$d_e = 2\ e \tag{9}$$

The channel Reynolds number is then given by:

$$Re = \frac{2\ \dot{M}}{\mu\ b} = \frac{\rho\ u\ d}{\mu} \tag{10}$$

This definition of the channel Reynolds number is based on the equivalent hydraulic diameter (d_e).
These two definitions of the hydraulic diameters are used in the open literature for plate heat exchanger, and when carrying out comparison of correlations great care should be taken on the definition of the heat transfer surface and hydraulic diameter. As there is no theoretical justification to use the first definition rather than the second one, it could be recommended to adopt the most simple (d_e and S_{pr}).

Different sets of tests have been performed at CEA for different angles of corrugation and pitch to height ratios. These data were used to determine thermal and hydraulic correlations as function of Reynolds, Prandtl numbers and geometrical parameters.

2.12.3 Pressure drop in corrugated channels

The effect of the corrugation angle is clearly outlined. Compared to a smooth rectangular channel, at Reynolds number of 1000, the friction factors is respectively increased by a factor 5 and 50 for 30° and 60°corrugation angles. For a Reynolds number of 10000, the increase is respectively by a factor 10 and 100.

2.12.4 Heat transfer in corrugated channels

The experimental apparatus consist in a single corrugated channel that is heated by Joule effect. The fluid temperature is measured at the inlet and outlet as well as on several location of the heat transfer surface (20 thermocouples). The tests were performed using water and oil and have covered Reynolds number between, 30 and 15000 and Prandtl numbers between 5 and 140.
The flow seems to be turbulent even for Reynolds numbers down to 50. This characteristic gives a very high heat transfer enhancement compared to a smooth of the heat transfer performances for low and intermediate Reynolds numbers: at a Reynolds of 1000 the ratio $Nu_{30°}/Nu_{smooth}$= 4.4 and $Nu_{60°}/Nu_{smooth}$= 9.4 and at a Reynolds of 5000 the ratio $Nu_{30°}/Nu_{smooth}$= 2.8 and $Nu_{60°}/Nu_{smooth}$= 6.7. It should be noticed that the heat transfer enhancement is much lower than the increase in pressure drop.

Using our available correlation for a corrugation angle of 45° and a pitch to height ratio of 3.33, the cooler of the main blower is sized. Its main characteristics are given in table 5.

	Helium		Water
Inlet temperature (°C)	96		30
Mass flow rate (kg/s)	14.35		8.6
Type of Compabloc		CP 30	
Number of plates (*)		240	
Internal volume (l) (**)		170	
Duty margin (%)		39%	
Pressure drop (MPa)	0.0142		0.0001
Outlet temperature (°C)	77.3		69.9

Table 5: General characteristics of the main blower cooler

(*): Maximum number of plates in a CP 30 heat exchanger. It would be possible to decrease the number of plates since the duty margin is quite important. Nevertheless, keeping this maximum number of plates gives an order of magnitude of the generated volume.

(**): The given volume is the volume of the plates pack to which volume of external plates has to be added.

This sizing aims to show it is possible to get small heat exchanger to fit to specification. The proposed solution can be still improved by modifying eventually the number of plates.

Moreover, this technology is certainly not the most suitable one and more compact solutions can be used.

To check the previous design, it will be possible to validate correlations during helium tests at ENEA on the recuperator and during air tests which are foreseen in air at CEA. Even if the mock up is smaller, the internal geometry remaining the same, the performances in term of non dimensional parameters must be identical.

2.13 CONCLUSION

This document is the CEA contribution for deliverable D-CT1.7 - Design study for IHX, isolation valves and cooler done within the RAPHAEL FP6 Project.
This document is linked to IHX and cooler components for HTR and VHTR application. Based on thermal and hydraulic correlations, these 2 components are sized after a technical presentation of different types of heat exchangers.
The major technical challenge is the development of a compact IHX to transfer heat from primary side to secondary one taking into account high level of temperature and pressure. Within this project, 2 technologies are studied in parallel: tubular heat exchanger already experimented by the past on German and English gas cooled reactor and plate heat exchanger.
The advantage of a plate heat exchanger is linked to its compactness but the main drawback is linked to its higher sensitivity to thermal gradients.
Sized at nominal working conditions (850°C), the generated volume of a plate fin heat exchanger is reduced (16.5 m^3 including distributing areas).
Looking only at pressure effect, generated stresses are quite acceptable since such heat exchanger technology is able bear high pressure.
The main trouble may be due to stresses generated by thermal gradients which can be quite important compared to creep resistance data of Inconel 617 or Haynes 230. To go further in the analysis, it is absolutely necessary to use FEM studies, tests and mock up manufacturing. So the next stage will concern FEM carried out by NRG on this concept.

For cooler and looking at specification, the challenge is easier to reach since several compact technologies could be used (temperature quite low with high pressure on both sides). A sizing is done with a Compabloc technology which will be tested at ENEA with
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For cooler and looking at specification, the challenge is easier to reach since several compact technologies could be used (temperature quite low with high pressure on both sides). A sizing is done with a Compabloc technology which will be tested at ENEA with helium and at CEA with air.

2.15 BIBLIOGRAPHY

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3. THERMO-MECHANICAL ASPECTS OF HEAT EXCHANGERS (BY AMEC)

3.1 INTRODUCTION

The Very High Temperature Reactor (VHTR) is the next generation of fission reactors offering sustainable electric energy and heat for hydrogen production or other high temperature processes [1]. Such Generation IV type reactors require high gas outlet temperatures close to 1000°C to achieve high efficiencies of 70% [2].

The intermediate heat exchanger (IHX) is perhaps the most critical part of the VHTR and HTR. Its development relies on selecting specific materials that can deal with high operating temperatures and pressure drops. The thermal, environmental and service life of the reactor makes the design and selection of high-temperature materials for this component a significant challenge.

As part of the investigations within RAPHAEL SP CT Work Package 1, this report provides a contribution to the thermo-mechanical aspects of the IHX. The work is based on past and current experiences within the UK on gas cooled reactor systems.

3.2 OBJECTIVE

The purpose of this report is to review the thermal and mechanical aspects of the VHTR IHX. Aspects of the design and development are considered to be beyond existing industrial experience, particularly at very high temperatures (>950^{o}C). The most promising candidate materials are listed and the thermal and mechanical aspects of the reactor discussed. Additionally, the knowledge attained from UK experience with the commercially developed Advanced Gas-cooled Reactor (AGR) and other gas cooled systems are given to provide a basis for development. The review focuses on the tube and shell designs used in the UK.

3.3 GENERAL REQUIREMENTS OF MATERIALS FOR THE INTERMEDIATE HEAT EXCHANGER

The heat exchanger component is vital in any nuclear power plant. Its compactness and efficiency has been highlighted to be of particular importance in the VHTR. The purpose of this section is to review the available materials for the IHX, as outlined within Raphael-IP SP-ML WP2.

The thermo-mechanical conditions of the IHX that affect material selection are as follows [1]:

1. The normal operating temperatures range from 350°C to 1000°C (maximum)
2. The expected mechanical load is from the pressure i.e. 1 to 10MPa
3. Impurities in the helium are a major concern for metals, since they can lead to environmentally induced degradation
4. Aging effects and embrittlement are important since they can affect performance, especially during thermal transients
5. Welding and fabrication issues in relation to design

3.4 OPTIONS FOR MATERIAL SELECTION OF THE VHTR

Nickel based alloys have been accepted as the leading material for the IHX. This is due to their relatively high strength at high temperature and good corrosion resistance. Two nickel based alloys have been used for past heat exchanger applications (gas-cooled). The candidate materials are as follows [1-2]:

1. Alloy 617 from past German HTR systems
2. Alloy 230, alternative French HTR selection (Figure 1)
3. Alloy X and Alloy XR from Japanese HTTR (Figure 2)
4. ODS Alloys, attractive for increased temperatures up to 1000°C (or higher) but so far has little related industrial development

A recent report issued by SP ML on material selection [2] reviews the temperature and operational limits for these materials. The below table provides a summary of the SP ML findings and recommendations for future material selection actions:

Table 1: Heat Exchanger material development

Material	Temp Limit ^{o}C	Limiting Factor	Further Development	Issues for VHTR/ RAPHAEL
Mod 9Cr 1Mo steel	Operating: 550^{o}C	Long term Creep strength	ODS version capable of 100^{o}C Higher temperatures - Fusion development –	Established material for sodium and Gas cooled systems— moderate temperatures
	Max: 650^{o}C			
Austenitic Stainless Steel	Operating: 550^{o}C	Long term Creep strength		Established material for sodium and Gas cooled systems— moderate temperatures
	Max: 750^{o}C			
Alloy 800 **Alloy 800H**	Operating: 650-750^{o}C	Long term Creep strength		Used in gas cooled and Sodium cooled systems.
	Max: 920-980^{o}C	Corrosion		
Alloy 617	Operating: 750-850^{o}C	Long term Creep strength <10MPa @ 100,000h	ODS version may be possible for Higher temperatures Effect of cold work (forming) on creep strength - limit of 5%	Established in German HTR Programme - KTA Rules developed. For ODS version Micro-structural development required as a first step -
	Max: 920-980^{o}C	Corrosion; Cr_2O_3 evaporation		
Alloy 230	Operating: 750-850^{o}C	Long term Creep strength <10MPa @ 100,000h	ODS version may be possible for Higher temperatures Effect of cold work (forming) on creep strength absent from National Programmes	Selected as industrially developed alternative to IN 617 – for corrosion resistance For ODS version Micro-structural development required – Investigation of cold work on creep strength & aging Developments to be included in next stage of RAPHAEL programme
	Max: 920-980^{o}C	Corrosion; Cr_2O_3 evaporation		
Alloy XR	Operating: 750-850^{o}C	Long term Creep strength <10Mpa @ 100,000h	Effect of cold work (manufacturing) on creep strength - limit of up to 10% Tests @800-1000^{o}C	Established for HTTR Operation at 750^{o}C
	Max: 920-980^{o}C	Corrosion; Cr_2O_3 evaporation		
ODS - Ferritic	Operating: 550^{o}C		Manufacturing development	Not developed for heat exchanger application –

	Max: 650°C		required. Material being Investigated in EXTREMAT	significant programme required on joining/forming involving manufacturer
ODS - Austenitic	Operating: >550°C		Manufacturing development required. Material being Investigated in EXTREMAT	Not developed for heat exchanger application – significant programme required on joining/forming involving manufacturer
	Max: >650°C			

3.5 GENERAL REQUIREMENTS WITH RESPECT TO DESIGN FOR THE INTERMEDIATE HEAT EXCHANGER

The IHX is identified as a key component, which provides the interface between the primary and secondary or other application circuits. There are stringent requirements to be faced since it aims to transfer the full reactor power:

1. High heat transfer performance

2. Due to varying operating temperatures from the core inlet to the core outlet in different parts of the component, it has to withstand high thermal loads generated by high temperature gradients in permanent and transient conditions

3. It has to also withstand the corrosive environment of the helium coolant, which although chemically inert in nature, is not actually inert due to unavoidable impurities

4. It must be as compact as possible both from an economic and a size aspect.

In order to get a compact design of the IHX, two main types of design arrangement are currently being considered, i.e. the Plate type, which includes the Plate Machined Heat Exchanger (PMHE), the Plate Fin Heat Exchanger (PFHE) and the Plate Stamped Heat Exchanger (PSHE) (Figure 3) and the conventional tube and shell arrangement (Figure 4). The compactness for both tube and/or plate concepts and their integrity under specific loads requires significant development for the HTR/VHTR system.

For the RAPHAEL Project Sub-project System Integration (SP SI) are providing the necessary system input information to enable the research and development activities to proceed in an integrated manner. The industrial projects on which the SP SI information is based are the French ANTARES reactor system and the South African PBMR. The main characteristics of the French ANTARES reactor system are summarised below in Table 2 [3]. For both designs the plate concepts are the first choice arrangements.

Table 2: Main characteristics of the ANTARES HTR

Thermal power (MW)	600
Core outlet temperature (°C)	850
Core inlet temperature (°C)	400
Primary fluid	He
Pressure (bar)	60
Fuel enrichment	~15%
Burn-up (GWd/tHM)	150
Secondary IHX outlet temperature (°C)	800
Secondary fluid (for electricity production)	80% N_2, 20% O_2

For the NNC contribution to VHTR IHX thermo-mechanical investigations, experiences from the shell and tube design arrangement are provided. These are provided from the AGR and gas cooled fast reactor (GCFR) system.

3.6 UK EXPERIENCE WITH GAS-COOLED REACTORS

The AGR is a development of the Magnox system in which the maximum temperature of the coolant gas was raised by 250–300°C to 650°C, thereby giving rise to greater thermal efficiency and enhanced economic performance of the power station. An AGR power station consists of two reactor units and a turbine house combined into a single complex. Each reactor unit, including the core, heat exchangers and gas circulators, is enclosed within a prestressed concrete pressure vessel. A schematic representation of the reactor is shown in Figure 5.

3.7 AGR HEAT EXCHANGER DESIGN

In the original design of the AGR system, the boilers are located in the annulus between the gas baffle and the liner wall of the prestressed concrete pressure vessel. There are four boilers per reactor, each consisting of three separately encased main units and three reheater units. Each boiler occupies a quadrant. The general arrangement of a boiler unit is shown in Figure 6. The detailed construction of the unit is shown in Figure 7. A schematic representation of the AGR boiler system is shown in Figure 8.

There are essentially two parts to the boiler:

(a) the high pressure boiler

(b) the reheater

The high pressure boiler is the main source of heat removal from the reactor. It consists of the following sections:

(i) primary economiser

(ii) secondary economiser and primary evaporator

(iii) secondary evaporator and primary superheater

(iv) secondary superheater

Different materials are used for the tubing in the various sections of the boiler. The primary economiser tubing is made from mild steel and the secondary superheater tubing is made from AISI Type 316H stainless steel. The tubing in-between these two sections is made from 9% chromium, 1% molybdenum ferritic steel. Transition joints are therefore provided between the primary and secondary economiser and the primary and secondary superheater banks. The secondary economiser and primary evaporator sections of the boiler are contained within a single set of platens (the lower 9% chromium bank). The secondary evaporator and primary superheater sections are also contained within a single set of platens (the upper 9% chromium bank). The tube elements are supported in the banks by welded spacers of the same material as the tubing. These spacers are supported in the banks from cross-members attached to the main unit casing.

The three reheater units of each boiler are located immediately above the high pressure units. Each reheater unit is suspended from the liner roof by four slings. The reheater casing is connected to the high pressure boiler casing below it by links at the centre of the short sides. These links maintain vertical alignment between the two units but permit vertical movement. A seal is provided between the reheater and the high pressure boiler casings which ensures continuity of the gas flow between them. Both units are fitted with a continuous vertical baffle which detunes the casing cavities to avoid acoustic coupling. The baffles extend over the full height and breadth of their respective casings and divide the space within them into two equal cavities. The baffle effectively divides each boiler unit into two half-units on the gas side: the inner and outer half-units.

Feedwater enters the high pressure boiler via two economiser penetrations at the bottom of each unit. Superheated steam leaves the boiler via three penetrations at the top of the unit. After passing through the high pressure cylinder of the turbine, cold reheat steam enters the reheater units via the reheater inlet penetrations. The steam tube within each of these penetrations extends into the vault forming an inlet header. The cold reheat steam is conveyed from the inlet header to the reheater platens via a series of tailpipes. There are 36 platens within each unit, each of which consists of four separate flow paths. The four inlets and four outlets to each platen are connected in pairs using bifurcations. The hot reheat steam leaves the unit via an outlet header which again is an extension of the reheater penetration steam tube.

The full power operating parameters for the standard design of AGR boiler were originally as follows.

Boiler gas inlet temperature	634°C
Boiler gas outlet temperature	288°C
Gas flow rate per boiler unit	314 kg s-1
Feed temperature	156°C
Superheater outlet pressure	166 bar
Superheater outlet temperature	541°C
HP steam flow per boiler unit	39.9 kg s-1
Reheater inlet pressure	42.1 bar
Reheater inlet temperature	343°C
Reheater outlet pressure	40.7 bar
Reheater outlet temperature	541°C
Total heat transferred to the boilers	1517 MW

The design of most of the later AGRs followed that of the original system, but four reactor systems (two stations) incorporated helically coiled once-through pod boilers. Each such reactor incorporates four boilers, one boiler consisting of two pod boiler units which are connected on the feedwater and steam sides externally to the pressure vessel. The boilers are of the single pressure once-through type, with the reheater arranged above the high pressure boiler surface. Both the boiler and reheater surfaces are in counter flow with the downward flow of reactor coolant gas. The general arrangement of the pod boiler units within the reactor is shown in Figure 9. Each unit is suspended in its pod from a top closure head. A gas circulator is located at the base of each pod.

In the pod boiler, the reheater and high pressure surfaces are arranged in the form of concentric contra-rotating multi-start helices (see Figure 10). The number of starts per row, as well as the constant helix angle, is chosen such that all tubes in the high pressure boiler, as well as in the reheater, are of the same length. The high pressure surface is composed of 285 tubes in parallel, each 81 m long, arranged in 19 rows with six starts in the inner row and 24 starts in the outer row. The reheater surface is comprised of 312 tubes in parallel, each 6.5 m long, arranged in 13 rows with 12 starts in the inner row and 36 starts in the outer row. Each of the 285 water circuits in the high pressure boiler is fed separately by its associated restrictor tube. Flexibility spirals at the base of the unit connect the restrictor tubes to the heating surface. The heating surface then extends upwards in one continuous length economiser, evaporator and superheater. The surface is comprised of low fin tubing, formed in mild steel (Economiser I), 9% Cr, 1% Mo steel (Economiser II, evaporator and Stage I superheater) and Type 316 stainless steel (Stage II superheater). The reheater surface is comprised of low fin Type 316 stainless steel tubing. A representation of the pod boiler, showing dimensions and the tube materials, is presented in Figure 11.

At the outlet from the high pressure surface, adjacent tube circuits are bifurcated and the steam is led through Type 316 stainless steel tubes to the tube sheets in the superheater outlet manifolds positioned in the top closure head penetrations. These tail tubes are led upwards past the reheater surface through an annulus which is formed between the pod boiler unit outer casing and the smaller diameter reheater unit casing. The superheater outlet tails are arranged in a single row in this annulus, in spiral form to absorb differential thermal expansion, and they pass through a gas baffle in the annulus designed to prevent hot gas from bypassing the reheater surface. The reheater surface is trifurcated at inlet and outlet. The reheater inlet tails are of Type 316 stainless steel and are arranged in spiral form, passing down the annulus in two rows to connect to the trifurcation at the inlet to the heating surface.

The full power operating parameters for the AGR pod boiler are as follows:

Number of pod units per reactor	8
Gas inlet temperature	644°C
Gas outlet temperature	280°C
Superheater steam outlet pressure	171 bar
Superheater steam outlet temperature	543°C
Feed temperature	157°C
Steam generation per reactor	481 kg s-1

Reheater inlet pressure	41.5 bar
Reheater inlet temperature	337°C
Reheater outlet pressure	39 bar
Reheater outlet temperature	539°C
Reheater steam flow per reactor	439 kg s-1

3.7.1 Review of Thermo-Mechanical Problems in the AGR Heat Exchanger

The following provides a summary of the main thermo-mechanical problem areas experienced in the operation of the AGR heat exchangers:

(1) 9% Chromium steel oxidation - When the evaporator and primary superheater sections were designed it was expected (from 20 kh laboratory data) that protective oxidation kinetics would apply at 520°C for 200 kh. Maximum metal temperatures were therefore set at 485°C for tubes and 518°C for the support structure. It was concluded that even if the risk of tube failure was low, accumulation of oxide debris within the gas circuit could significantly impair reactor operations and maintenance. Maximum tube temperatures were therefore reduced from 485°C to 446°C.

(2) Stress-corrosion - Damage from stress-corrosion is difficult to predict quantitatively since it arises from the complex interaction of stress, material, temperature, pressure and environment. The original design concept was to avoid wetting the austenitic superheater section. Research results showed that austenitic superheater tubes could be wetted for short periods. Implementation of these recommendations gave rise to a dramatic decrease in the number of reactor trips and the eventual return to full-load capability for the steam generators.

(3) Creep of transition joints - When small defects were found in the root run of transition welds between 9% chromium steel and Sanicro transition pieces, concern was expressed for the long-term integrity of such welds. An extensive testing programme was mounted to determine materials composition and thermal expansion coefficients of the components forming the transition joint, together with tensile, creep, corrosion and fatigue properties of the weldments on both uniaxial and tubular specimens. It was concluded that failure due to stress-corrosion and fatigue (high and low cycle) was unlikely provided the weld was operated in a dry environment.

3.8 DRAGON HEAT EXCHANGER DESIGN

The primary heat removal circuit of the Dragon reactor was contained entirely within the carbon steel pressure vessel with its six heat exchanger branches. The coolant path of the primary heat removal circuit was arranged in such a way that the vessel walls were in contact with the cold (330-370°C) reactor inlet stream only. The hot leg of the circuit consisted of a large plenum chamber above the core and six duct liners, which fed the reactor outlet coolant to the primary heat exchangers. A general arrangement of the heat exchanger, and its associated circulator, is shown in Figure 12.

A heat exchanger unit consisted of 63 mild steel tubes in a one pass parallel flow arrangement. Each tube left one of three inlet headers, passed downwards between the heat exchanger shell and a skirt, then went through a 90° bend to traverse the gas flow horizontally and enter the helical bundle. The helical section was arranged in seven concentric cylinders of evaporator tubes, the number of starts increasing from six to 12 with increasing helix diameter. As a result of this arrangement, the angles of inclination as well as the heated lengths were equal despite the large diameter difference of the helices. Tubes from any one inlet header could go to any one of the three outlet headers, the exact route and the height of exit from, or entry to, the headers being chosen to preserve equal overall tube lengths. There was no systematic correlation between the tube arrangement at headers and the tube arrangement in the bundle. The dimensions of the unit are given below.

Outer diameter of tubes	18 mm
Inner diameter of tubes	14 mm
Water side flow area	0.0097 m^2
Helix tube length (approx.)	4.3 m
Horizontal tube length at inlet to helix (mean)	0.2 m
Tube length in cross flow	4.5 m
Tube length downcomer (mean)	2.1 m
Tube length riser (mean)	0.8 m
Total tube length (mean)	7.4 m
Water content of primary heat exchanger (cold)	95.5 kg
Water content of secondary circuit (approx.)	300 kg

Helix diameter (Coil No. 1)	245 mm
Helix diameter (Coil No. 7)	545 mm
Helix diameter (mean)	395 mm
Heat transfer area, gas side (cross flow)	16.0 m^2
Heat transfer area, water side (total effective)	12.7 m^2

All components associated with the water circuits were fabricated from carbon or low alloy steels

3.8.1 Review of Thermo-Mechanical Problems in the Dragon Heat Exchanger

The following provides a summary of the main thermo-mechanical problem areas experienced in the operation of the Dragon heat exchangers:

(1) Tube failures - A number of tube failures occurred in the primary heat exchangers, partly attributed to local overheating of tubes at the gas inlet due to poor velocity distribution. The flow distribution was measured in a test rig and it was found that the local gas velocity around the inner tubes was twice as high as the mean gas velocity. By inserting a perforated diffuser between the duct outlet and the bundle inlet, a much more homogeneous flow distribution was achieved. With this design modification and the insertion of orifices at the tube inlets (water-side) a new set of primary heat exchangers operated without further incident.

Two other explanations were: corrosion of the mild steel tubes due to the water chemistry, and flow reversal and dry-out as a result of unstable flow. The immediate difficulties of reactor operation were overcome by a combination of orifices on the tube inlets, changing the water treatment to include lithium hydroxide, and improving the velocity profile of the high temperature gas by a gas baffle. A theoretical analysis of the heat exchangers showed that: without inlet orifices the flow in the heat exchanger tubes was unstable in the parallel channel mode.

(2) Corrosion damage - Extensive analysis carried out on the damaged heat exchanger clearly indicated that the corrosion was confined to the limited region of nucleate and slug boiling in those tubes where the highest heat transfer occurred. Every leak occurred in this region and on the tube wall subjected to high heat transfer. It was concluded that the corrosion resulted from violent boiling conditions in the horizontal section giving rise to conditions which affect the stability of the magnetite film.

(3) Flow fluctuations - Parallel channel instability occurs in a bank of tubes when the flow in individual tubes fluctuates in an oscillatory manner although the total flow

and pressure drop between headers remains constant. This is made possible by different tubes oscillating out of phase with each other. In the Dragon primary heat exchanger, the headers are assumed to be the six water and steam manifolds, each having 21 tube connections. To determine the stability of the flow, the pressure difference across a tube is monitored following a disturbance introduced in the flow. The response to a disturbance in the flow rate was computed at power levels of 5, 10, 15 and 20 MW. It was found that the disturbance died out quickly at low power levels (5 & 10 MW), but persisted at higher power (15 & 20 MW). From this it was concluded that the system was inadequately damped at the higher powers rather than unstable.

3.9 EXPERIENCES FROM GAS-COOLED FAST REACTOR DEVELOPMENTS

The Gas-cooled Fast reactor (GCFR) is also a generation IV advanced reactor system. The below experiences relate to the IHX unit based on a shell and tube concept. There are again important requirements on compactness but recognising that the overall size cannot be reduced to the same proportions as for the plate type design. The shell and tube arrangement is however considered the most stable configuration as the tubes are internally pressurised under normal operation.

The GCFR operates at slightly lower temperatures (~850°C) than the VHTR but similar temperatures to the ANTARES project with the primary circuit helium flowing through the shell and the secondary supercritical carbon dioxide (S-CO_2) flowing through the tubes [4].

The IHX component is housed within a pressure vessel connected to the reactor vessel by a coaxial cross-duct (Figure 4(a)). Its components are: the tube bundle; secondary-side headers; the primary gas circulator with its electric motor and cooler; and the primary-side gas baffles. The concept of a heat exchanger with an integrated circulator offers the greatest protection to the primary circuit components and provides for all the pressure boundary to be cooled by the primary coolant.

An important design principle of the IHX Is that the tube bundle has to be free to expand within the shell to minimise stresses within the tubes and the tube plates. A further design principle Is that neither hot primary nor hot secondary gas should come into contact with the primary pressure boundary. To comply with the first principle, the tube bundle and its headers have been designed to be fully floating with respect to the IHX shell. The second principle has been fulfilled by using a system of internal baffles and engineered leak paths to ensure that the entire internal surface of the shell is cooled by primary coolant at the reactor inlet temperature.

A cross-sectional view of the heat exchanger is shown in Figure 4(b). This type of arrangement could provide a good starting point for the IHX design of the VHTR. The tube bundle contains numerous straight tubes spanning between two secondary headers. The S-CO_2 enters the tube bundle from the bottom hemispherical header and exits by the top toroidal header. The bottom header is supported from a central spine-tube that is, itself, suspended from the top shell of the IHX pressure vessel. As well as supporting the bottom of the tube bundle, the central spine serves as the duct that supplies the bottom header with cold secondary coolant. The top secondary header floats on the end of the tube bundle, such that the bulk expansion of the tube bundle is accommodated without inducing any significant stresses.

The primary coolant enters the unit through the central duct of the coaxial cross duct and enters the shell-side of the tube bundle at the top. Baffles constrain the primary flow to remain within the tube bundle until it exits at the bottom end. Further baffles duct the primary flow around the bottom secondary header to the eye of the primary circulator impeller. The discharge flow from the circulator flows via a diffuser up the inside of the pressure vessel shell wall until it reaches the outer annular duct of the cross-duct, along which it is returned to the reactor. Orifices in a diaphragm above the cross duct allow a small leakage flow to pass up into, and cool, the top shell of the pressure vessel. This leakage flow rejoins the main primary flow at the bottom of the tube bundle after flowing down the outside of the central spine via a clearance gap between the top toroidal header and the spine. The design parameters of the GCFR IHX are summarised below in Table 2 [4]:

Table 3: Design parameters of the IHX

Parameter	Value
Reactor Power	600 MW
Gas circulator hydraulic power	43 MW
IHX thermal power	643 MW
Type	Shell & tube with straight tubes and counterflow
Primary (shell-side) fluid	Helium
Primary pressure	70 bar (nominal)
Primary mass flow rate	576 kg/s
Primary Inlet Temperature	680°C
Primary Outlet Temperature	465°C
Secondary (tube-side) fluid	Supercritical CO_2
Secondary pressure	200 bar (nominal)
Secondary mass flow rate	3115 kg/s
Secondary inlet temperature	445°C
Secondary outlet temperature	610°C
Log mean temperature difference	40.4°C
Vessel material	9Cr1Mo
Vessel external diameter	4.7 m
Vessel wall thickness	150 mm
Overall length of unit	37 m

The IHX tube bundle consists of 12276 straight tubes arranged in 44 rings on an approximately triangular pitch. A summary of the tube bundle parameters is presented in Table 3 [4]. The tube bundle parameters were determined with the assumption that the mean temperature difference could be represented reliably by a log-mean temperature difference. This is strictly valid for S-CO_2 because of the large fluid property changes. Provisional checks have shown that this approximation is not suitable for recuperators but is reasonably good for the IHX. The IHX bundle design parameters will be refined and consolidated in future work.

Table 4: Design parameters of IHX tube bundle

Parameter	Value
Tube outside diameter	20 mm
Tube bore	17 mm
Tube material	Inconel 625
Number of tubes	12276
Annular tube bundle arrangement	44 Rings
Inner tube row pitch circle diameter	1294 mm
Outer tube row pitch circle diameter	3519 mm
Tube circular pitch	27.1 mm
Tube radial pitch	25.87 mm
Active tube length (for heat transfer)	19.11 m
Active heat exchange surface area	14740 m^2
Tube length (between tube-plates)	23.00 m
Tube-plate diameter	3.88 m
Overall heat transfer coefficient	1080 $W/m^2/K$
Helium velocity	30 m/s.
S-CO_2 velocity	8.9 m/s.
Primary (Helium) pressure drop	0.85 bar
Secondary (S-CO_2) pressure drop	1.95 bar

The primary circulator is a single stage, radial flow device whose impeller is driven by a force-cooled electric motor. The motor will have to be custom built since the circulator power of 44 MW is beyond the power of 'off-the-shelf' electric motors. A summary of the design parameters of the circulator are presented in Table 4 [4].

Table 5: Design parameters of the primary gas circulator

Parameter	Value
Rotational speed – variable	1200 - 2970 RPM
Inlet pressure	70 bar
Outlet pressure	73 bar
Pressure ratio	1.043
Assumed polytropic efficiency	85%
Circulator outlet temperature	480°C
Circulator inlet temperature	465°C.
Specific enthalpy change	75 kJ/kg
Hydraulic power	43.2 MW
Electrical power (98% efficiency)	44.1 MW
Blade outlet angle	Radial
Number of blades	31
Impeller tip velocity	288.9 m/s
Outlet flow whirl velocity	259.6 m/s
Impeller outlet diameter	1858 mm
Impeller outlet passage width	150 mm
Outlet flow radial velocity	156.4 m/s
Outlet flow / diffuser inlet velocity	303.1 m/s
Impeller eye diameter	1100 mm
Impeller hub diameter	460 mm
Inlet flow axial velocity	156.1 m/s

In summary, although the GCFR requirements do not strictly simulate the VHTR requirements, the current IHX developments for the GCFR system provides a useful input for the thermo-mechanical development aspects for the VHTR IHX shell and tube design concept.

3.10 CONCLUSIONS AND SUMMARY

In conclusion, key points have been highlighted in this report to aid in the development of the IHX component of the VHTR. In relation to its thermal and mechanical aspects, these are:

1. It will need to withstand high temperatures up to 1000°C, and pressure drops resulting in mechanical loads of 1-10MPa

2. Encompass high heat transfer performances

3. Deal with the corrosive environment of the coolant from its impurities

4. Nickel based alloys are deemed the leading candidate materials since they retain their strength at high temperatures and have good corrosion resistance, however currently they are considered to have an outlet gas temperature limit of around 850^{o}C.

5. Compactness and efficiency are key issues

6. Specific plate concepts have been proposed for the French HTR programme to tackle issues of compactness

7. IHX design for the GCFR is a good starting point for VHTR developments for tube and shell design

8. The shell and tube concept is the more stable configuration

9. Shell and tube concept was found to be good for the IHX component of the GCFR, but not its recuperator

10. Problems with compactness in relation to this shell and tube concept for the VHTR

11. Information attained from past problems in the heat exchanger of the AGR and DRAGON.

These results emphasise the fact that although the IHX component of the VHTR is still at its early design stage, there are existing heat exchangers presented in this report that provide invaluable knowledge in its development.

3.11 LIST OF TABLES

3.12 LIST OF FIGURES

3.13 REFERENCES

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[5] B. F. Riley and S. Panchoo: 'UK operational experience with heat exchangers and valves in gas-cooled reactors' AMEC NNC, June 2006.

Figure 1: IHX and reactor vessel of the HTR connected using a co-axial cross duct (ANTARES) [3]

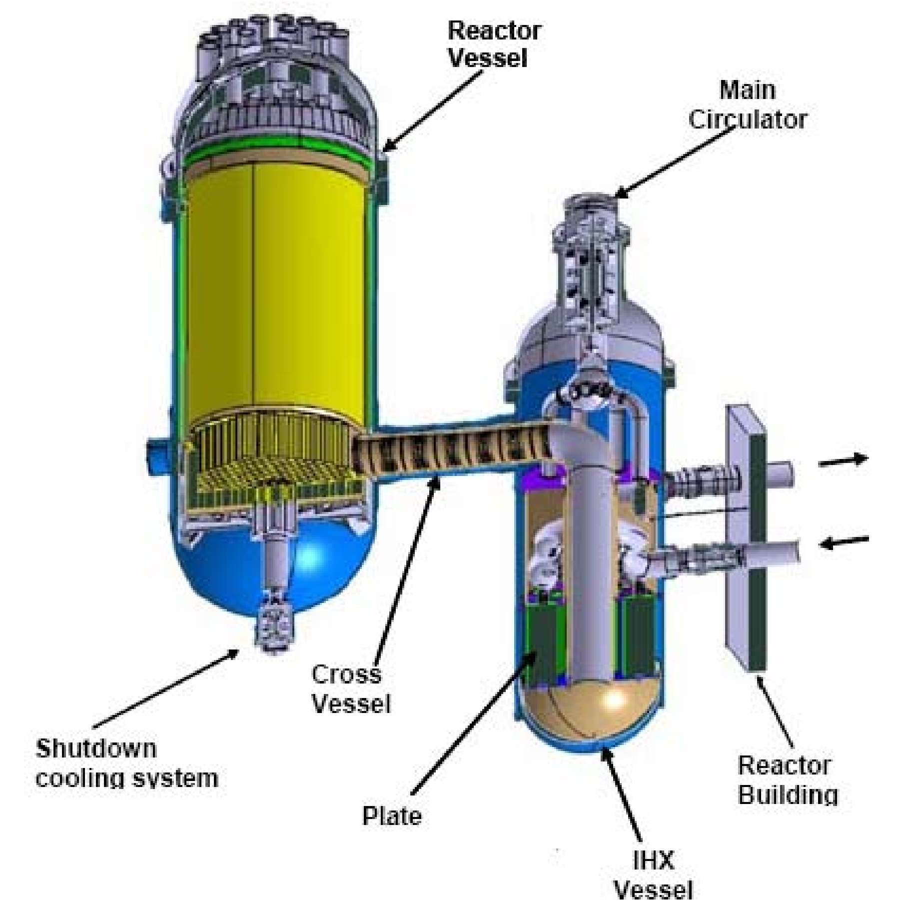
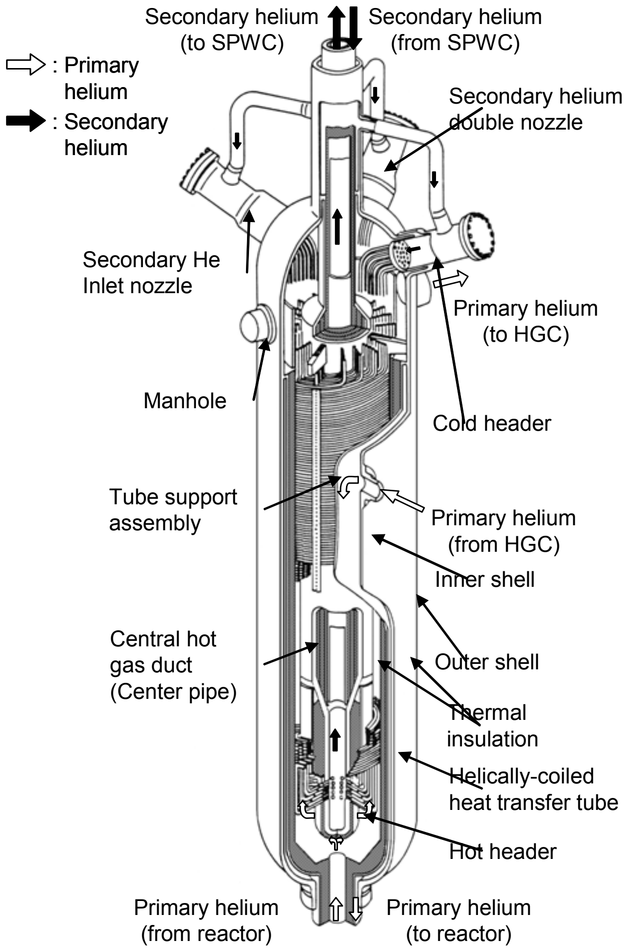


Figure 2: IHX component of the Japanese HTTR



Major specification of the IHX

Type		Vertical helically coiled counter flow type
Primary / Secondary coolant		Helium / Helium
Heat capacity		10MW
Coolant flow mass rate	Primary	15t/h／12t/h*
	Secondary	14t/h／12t/h*
IHX coolant temperature	Primary	(Inlet) 850／950°C* (Outlet) 390°C
	Secondary	(Inlet) 300°C (Outlet) 775／860°C *
Heat transfer tube	Number	96
	O. D.	31.8mm
	Thickness	3.5mm
	Material	Hastelloy XR
Shell outer diameter		2.0m
Total height		11.0m
Material	Shell	2 1/4Cr-1Mo steel
	Hot header	Hastelloy XR
	Heat transfer tube	Hastelloy XR

* Rated operation ／High temperature test operation

Figure 3: Potential plate concepts for the VHTR [3]

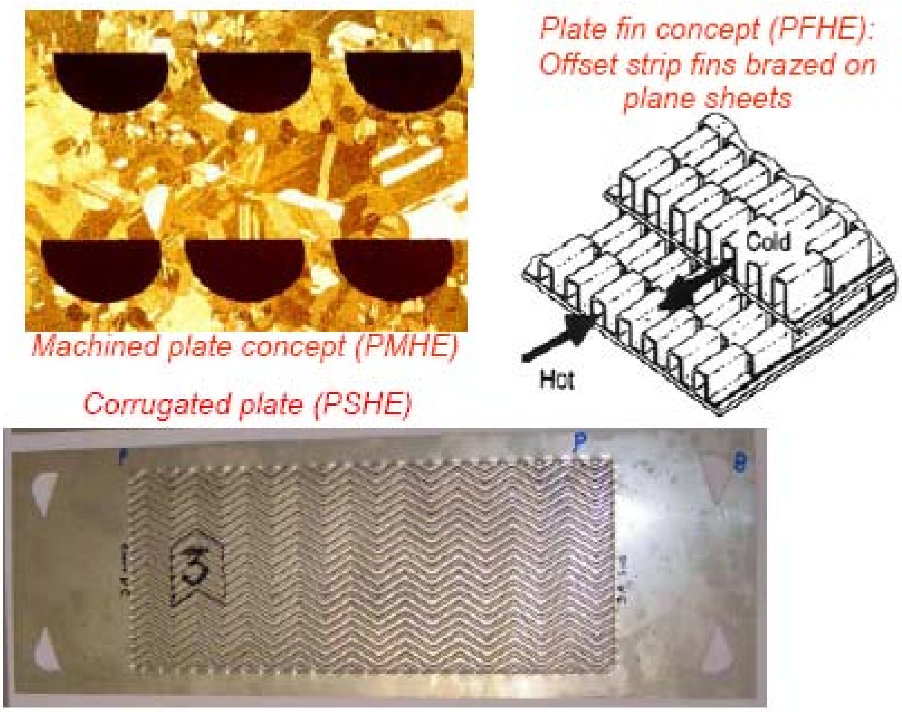
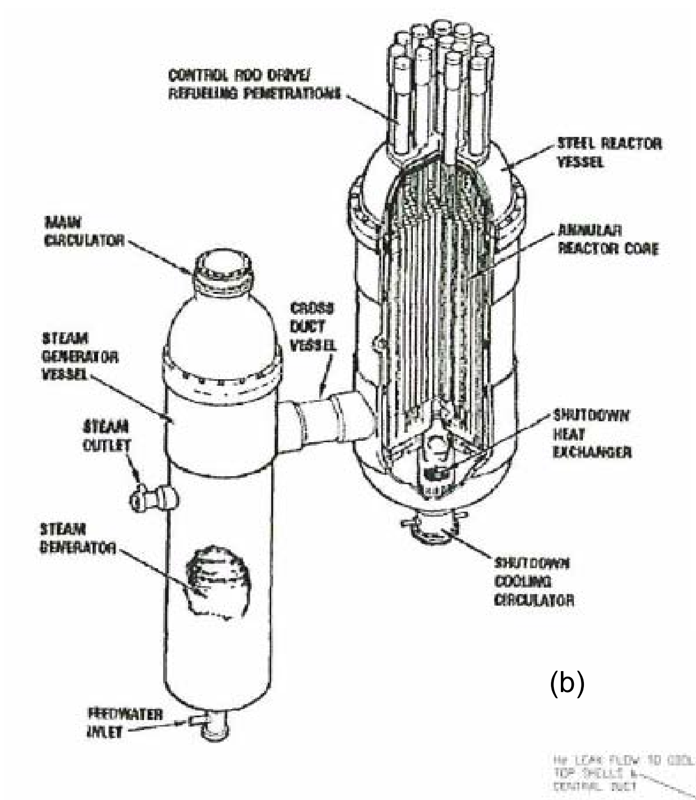


Figure 4: (a) Intermediate Heat exchanger of the GCFR (b) Cross-sectional view (starting point)

(a)



(b)

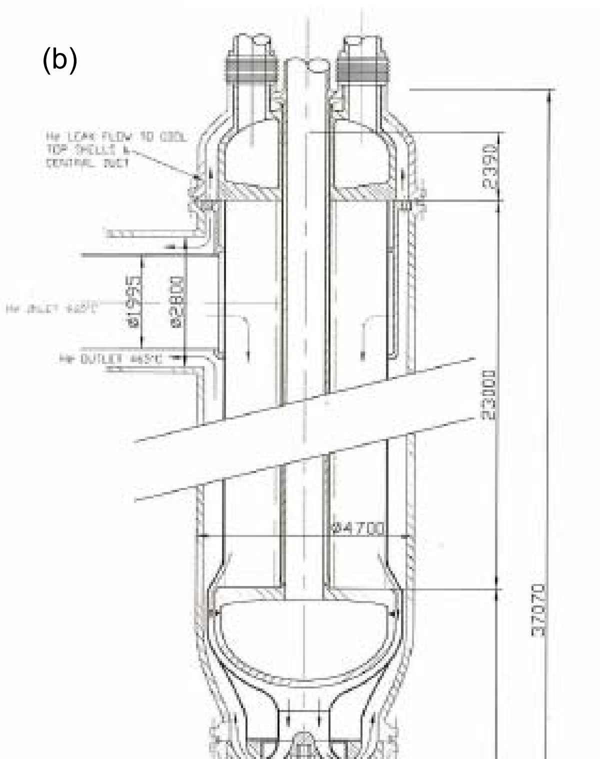


Figure 5: Schematic diagram of the AGR reactor system [5]

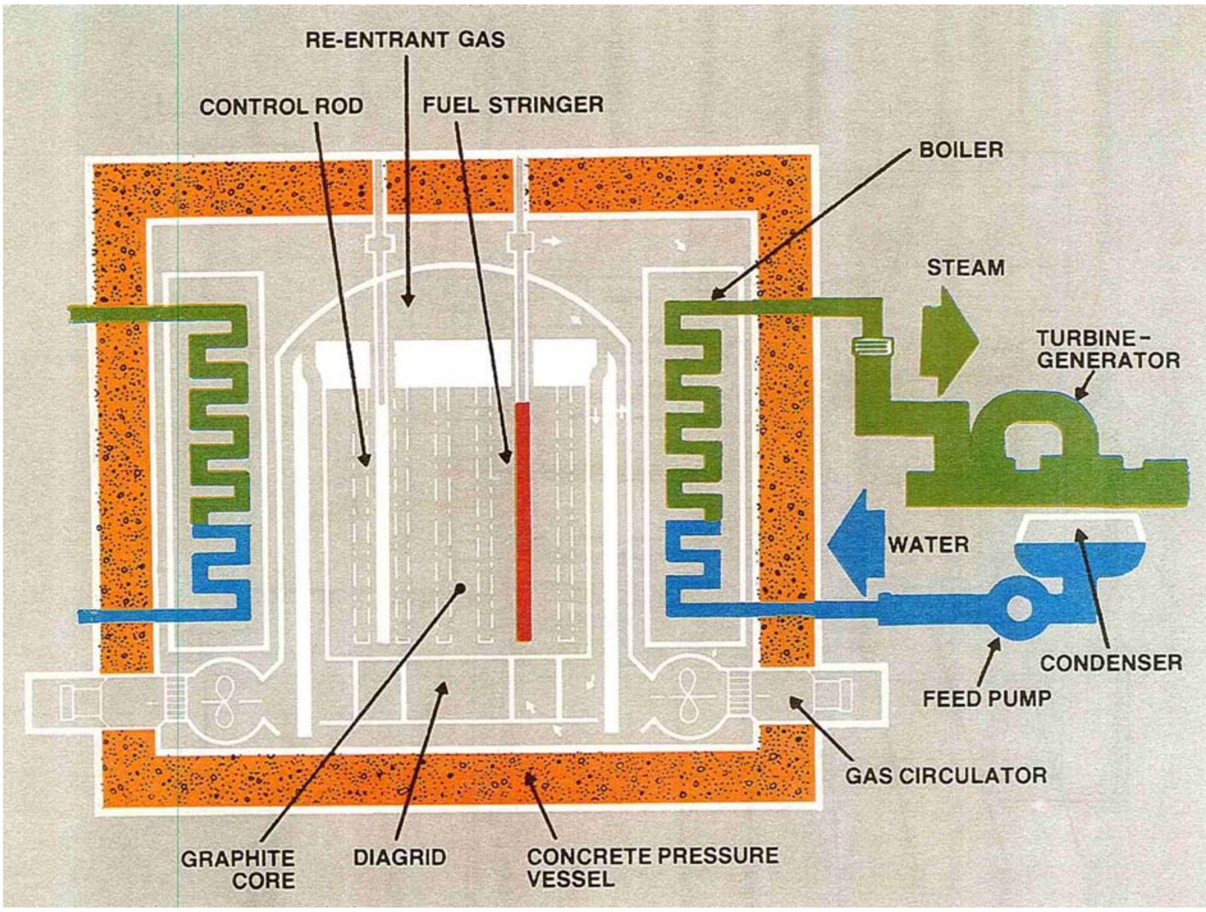


Figure 6: The AGR heat exchanger [5]

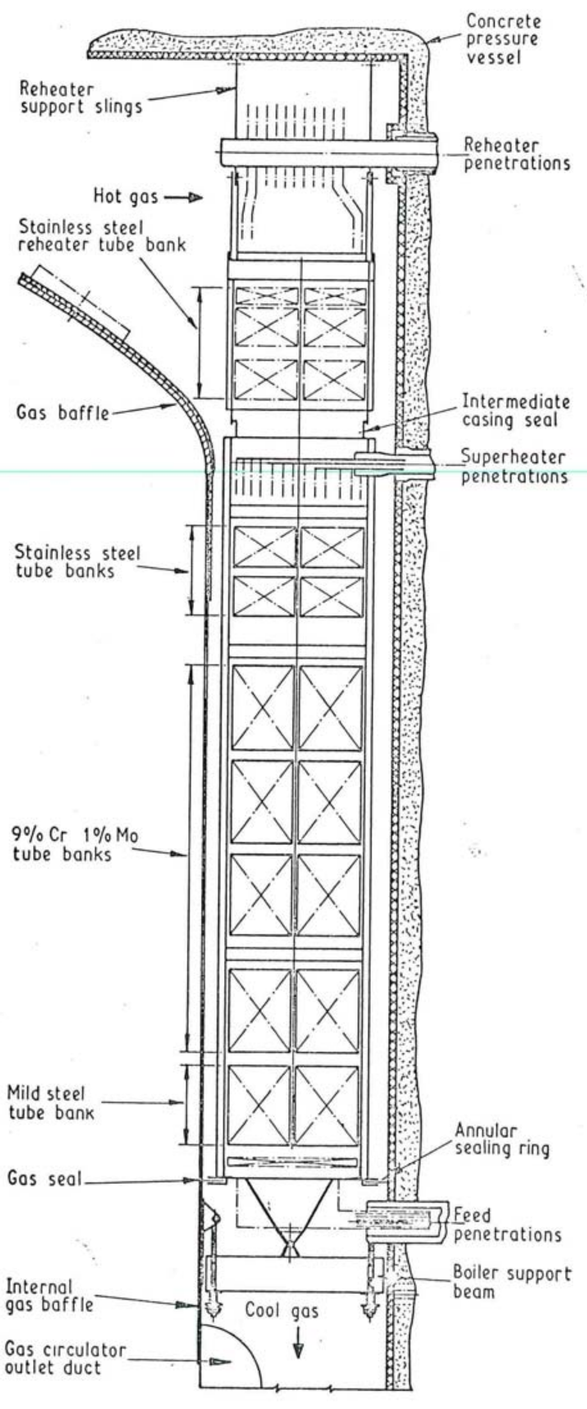

Concrete pressure vessel
Reheater support slings
Reheater penetrations
Hot gas
Stainless steel reheater tube bank
Gas baffle
Intermediate casing seal
Superheater penetrations
Stainless steel tube banks
9% Cr 1% Mo tube banks
Mild steel tube bank
Annular sealing ring
Gas seal
Feed penetrations
Boiler support beam
Internal gas baffle
Cool gas
Gas circulator outlet duct

Figure 7: Detail of the AGR heat exchanger [5]

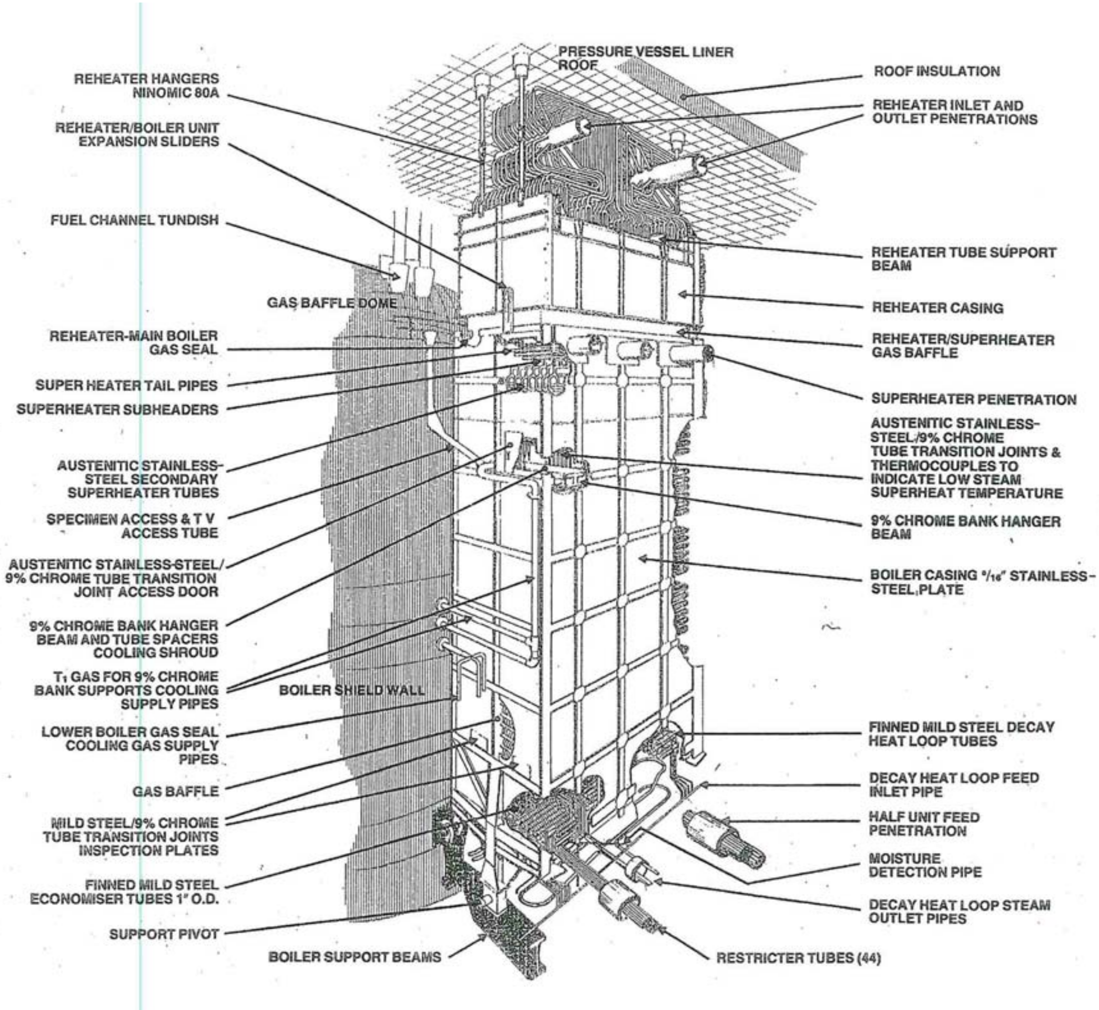


Figure 8: The AGR boiler system [5]

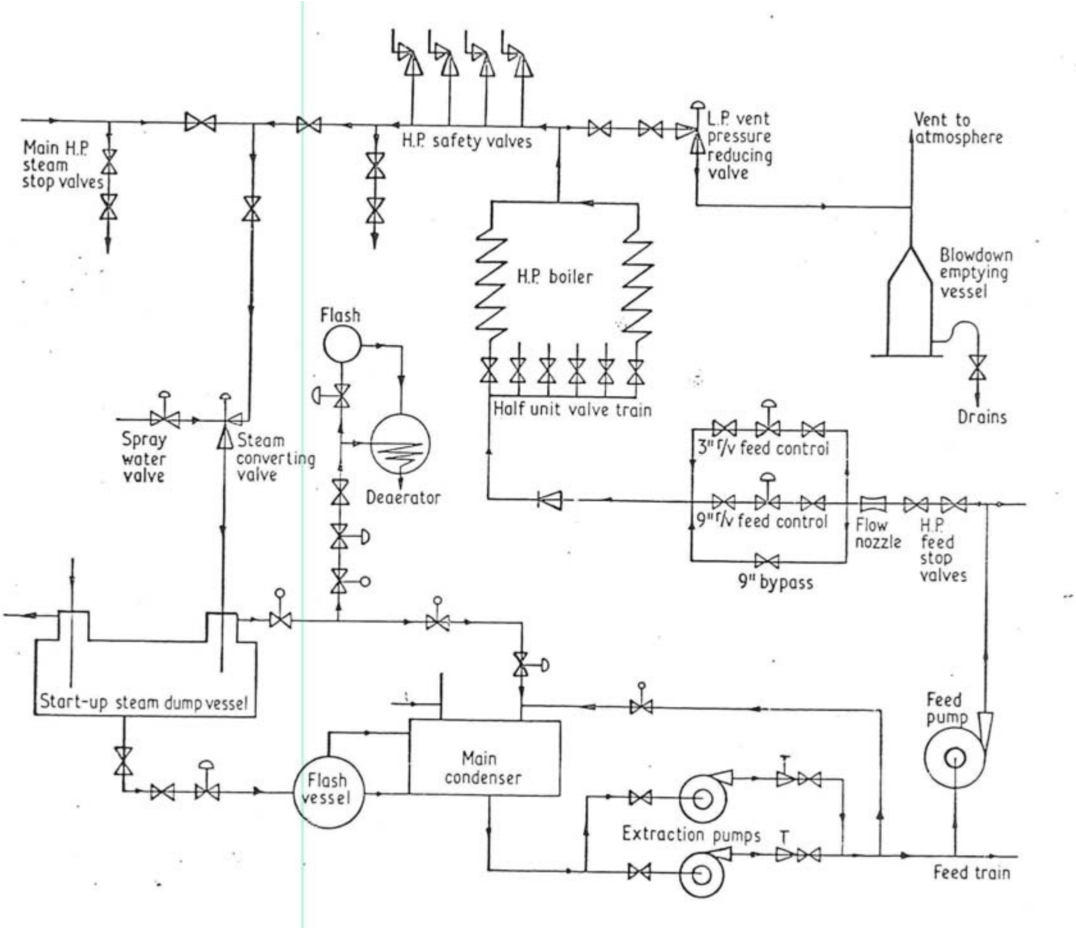


Figure 9: The AGR reactor system with pod boilers

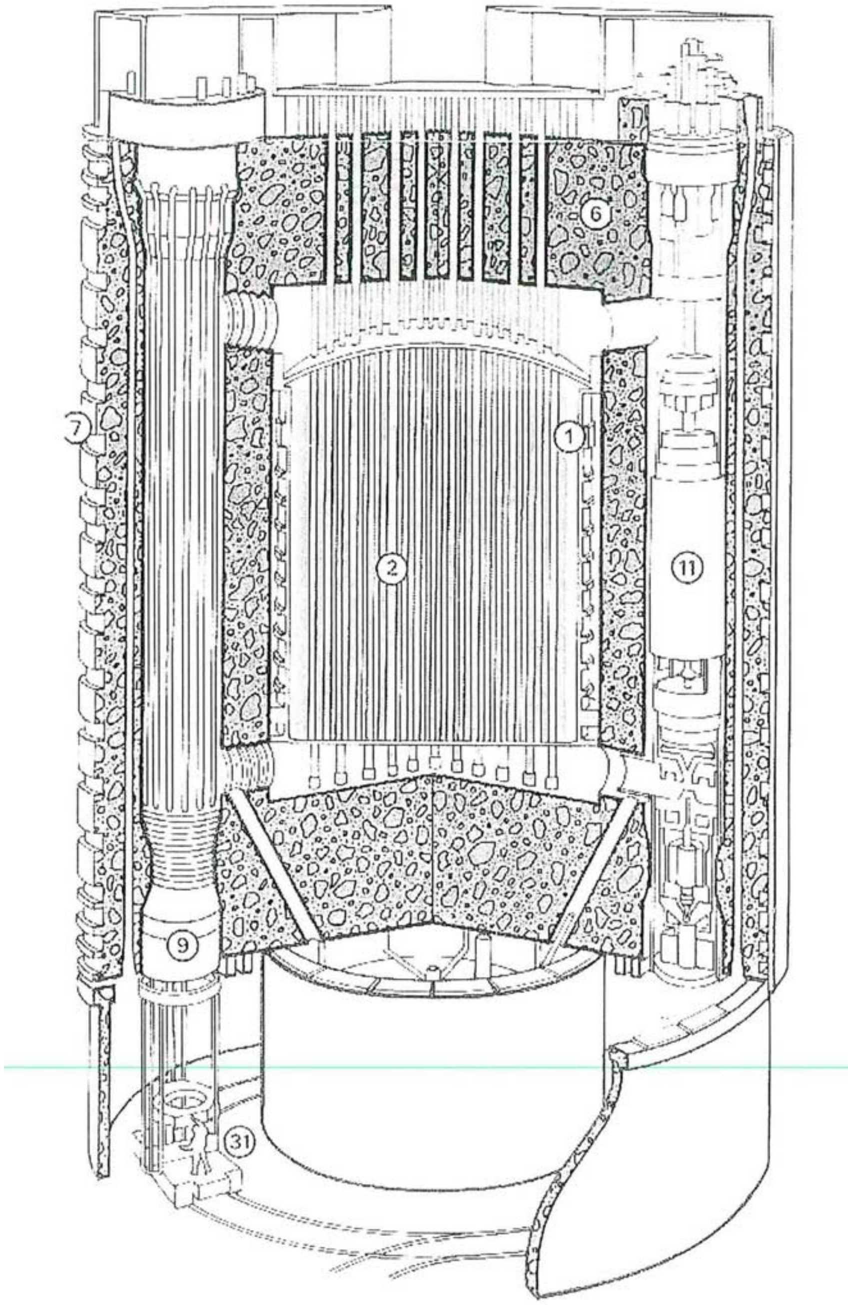


Figure 10: The AGR pod boiler – tubing arrangement

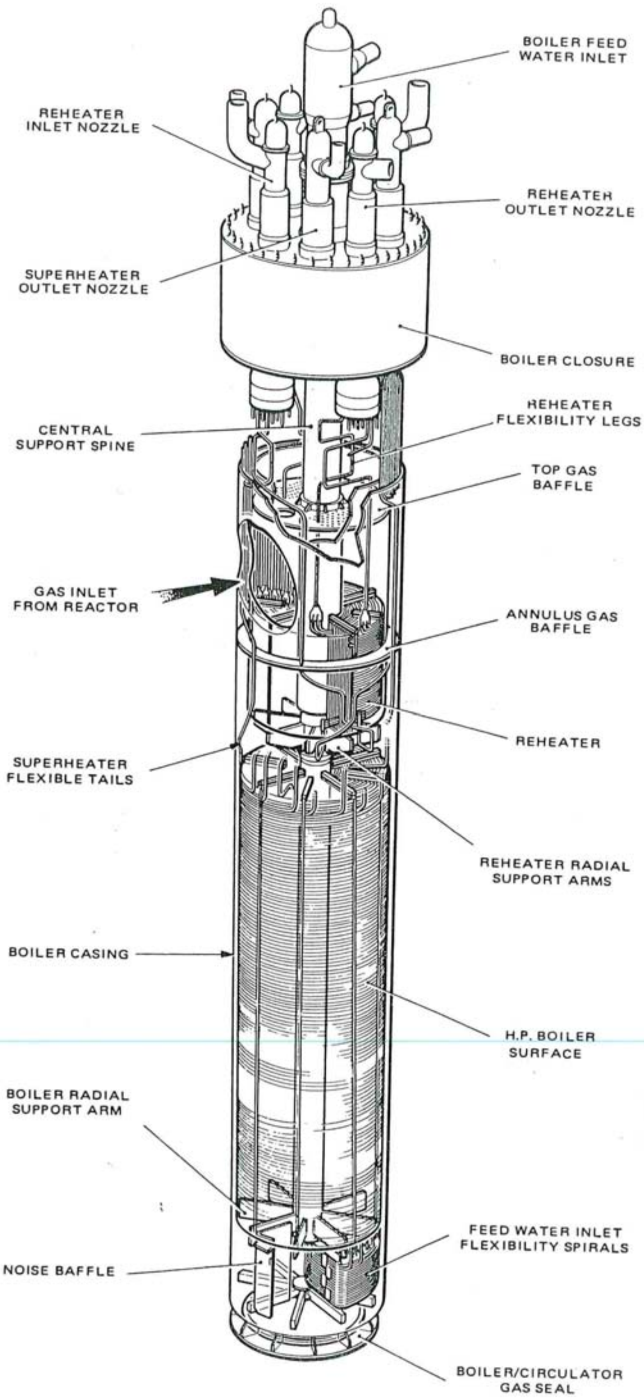


Figure 11: The AGR pod boiler – dimensions and materials

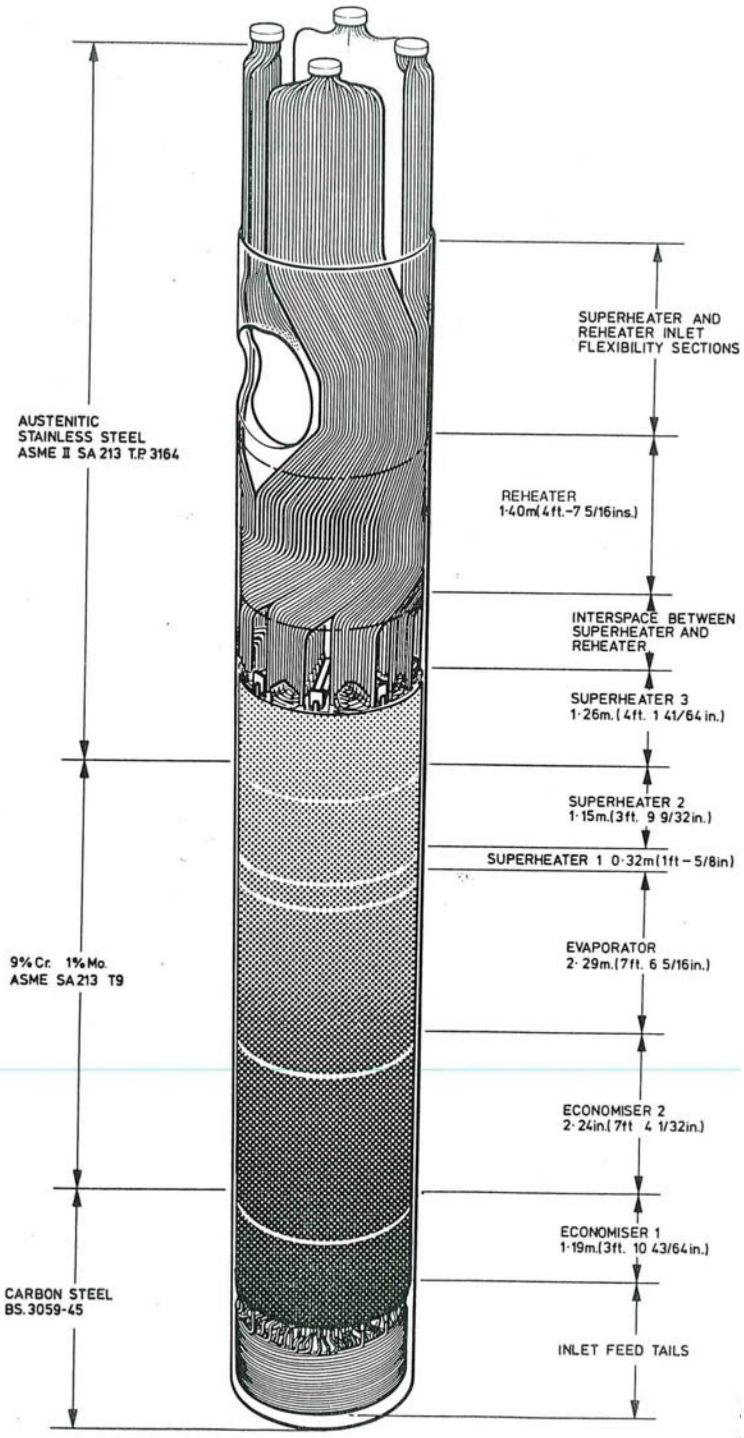


Figure 12: The Dragon heat exchanger [5]

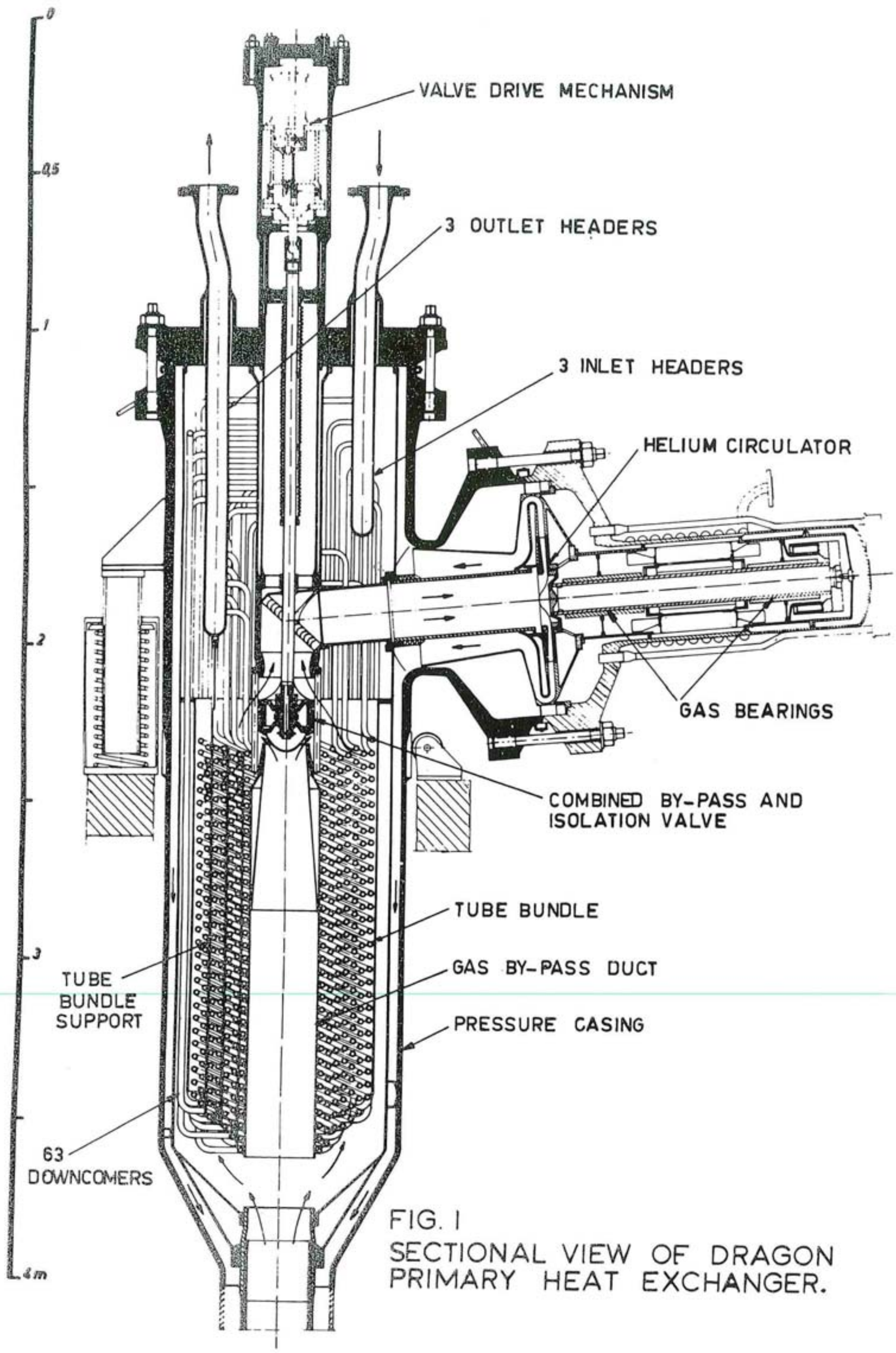


FIG. I
SECTIONAL VIEW OF DRAGON PRIMARY HEAT EXCHANGER.

4. IHX AND COOLER THERMAL HYDRAULIC STUDY (BY ANP-F)

4.1 SUMMARY

Four modules of 152.5MWth each with 2411 tubes helically wound on 28 rows, active length of 28.2m are proposed for RAPHAEL Reactor.

A helium/water cooler based on COMPABLOC ALFALAVAL technology is proposed for RAPHAEL cooler and will be tested on HE-FUS loop and CEA CLAIRE loop for validation.

4.2 INTRODUCTION

The objective of this study is to propose a pre-design of a tubular IHX based on VHTR ANTARES concept and a compact cooler based on COMPABLOC ALFA LAVAL VICARB concept. As mentioned in reference 1, the key issue for these two components is its compactness and its integrity under the applied loads.

The specification issued for RAPHAEL (Deliverable DC-T1.1, reference2) will be used as input data.

The organization of this document is as follows:

1. Pre-design of a tubular IHX, layout inside a plant design and Cost unit assessment,
2. Evaluation of a Compabloc ALFA LAVAL VICARB exchanger as a compact helium/water cooler.

This study will be defined in accordance with the recommendations issued from Design Safety, Materials, System Integration, Hydrogen Production and High Performance Helium turbine sub-projects.

4.3 TUBULAR IHX

4.3.1 Pre-design

In steady state, the tubular IHX nominal conditions are provided in the table1 for Process Heat Supply.

It is postulated that the IHX has to remove a conservative value of 610MWth power including the core power and the primary gas circulator power.

From studies performed on ANTARES, it could be state that a reasonable number of tubes is about 2200, the height and the tube bundle dimension are practicable from the manufacturing point of view.

For this study, 3 modules options have been investigated:

a) 2 modules of 305MWth,
b) 3 modules of 203MWth,
c) and 4 modules of 152.5MWth.

The number of tubes is respectively 4730 tubes (first option) and 3200 tubes (second option) with a tube diameter: **21/16.6mm, thickness: 2.2mm in INCONEL 617 material**. For these two first options, the technological feasibility needs to be checked.

Consequently, it is reasonable to select the third option (**4 modules of 152.5MWth**) with the main features as follows:

1. **2411 Tubes, 28 rows**
2. Active tube length: 28.2m,
3. Helix diameter first row: 1248mm
4. Helix diameter: last row: 3052mm

The main correlations (for information) used for the pre-design are:

d) Primary convective heat transfer correlation: $Nu_1 = 0.3 + \sqrt{(Nu_{lam})^2 + (Nu_{turb})^2}$

e) Secondary convective heat transfer correlation: $Nu_2 = \dfrac{\left(f_2/8\right)\cdot \mathrm{Re}_2 \cdot \mathrm{Pr}_2}{1 + 12{,}7 \cdot \sqrt{f_2/8}\cdot\left(\mathrm{Pr}_2^{2/3} - 1\right)}$

f) Primary pressure drop coefficients: $f_{a1} = \xi_l + \xi_t \cdot \left[1 - \exp\left(-\dfrac{\mathrm{Re}_{a1} + 1000}{2000}\right)\right]$ with laminar

$\xi_l = \dfrac{f_l}{\mathrm{Re}_{a1}}$ and turbulent $\xi_t = \dfrac{f_t}{\mathrm{Re}_{a1}^{\,0,1\cdot\left(P_{ax}/P_{rad}\right)}}$

g) Secondary pressure drop coefficients (laminar and turbulent): $f_2 = 64/\mathrm{Re}_2$

$$\xi = \frac{0{,}3164}{\mathrm{Re}_2^{0.25}}\left[1+0.095\left(\frac{d_i}{D}\right)^{1/2}\cdot \mathrm{Re}_2^{1/4}\right]$$

The design data and preliminary tube bundle geometry issued by **a computer code under development (EXCEL sheet configuration)** are presented into the tables here-after.

Design data:

Denomination	Symbol	Unit	Process heat Supply Plant
Thermal Power	P	MWth	**152.5**
Primary/Secondary fluid	-	-	**He/He**
Primary side			
Inlet Temperature	Te1	°C	**950**
Outlet Temperature	To1	°C	**390**
Primary pressure	p1	MPa	**5**
Primary mass flow	$\dot{m}$ 1	kg/s	**209.8**
Pressure loss	Δp	MPa	**0.007**
Secondary side			
Outlet Temperature	To2	°C	**900**
Inlet Temperature	Te2	°C	**340**
Secondary pressure	p2	MPa	**4.5**
Secondary mass flow	$\dot{m}$ 2	kg/s	**209.8**
Pressure loss	Δp	MPa	**0.089**
Geometry			
Outer tube diameter	OD	mm	**21.0**
Inner tube diameter	ID	mm	**16.6**
Number of tubes	N		**2411**

Tube wall thickness	Thk	mm	**2.2**
Distance between tubes	S1	mm	**33.4**
Helix angle	α	°	**25.38**
Helix inner diameter	D inside	m	**1.181** (first row - radial pitch)
Helix outer diameter	D outside	m	**3.12** (last row + radial pitch)
IHX inner surface area	A2	m²	**3544**
IHX outer surface area	A1	m²	**4483**
Length of tubes	L	m	**28.18**
Tube bundle height	H	m	**12.08**

Tube bundle geometry:

Row	Diameter (m)	Circumference (m)	Radial pitch (mm)	Tubes per row	Accumulated number of tubes
1	1.248	3.921		50	50
2	1.315	4.131	67	53	103
3	1.382	4.340	67	55	158
4	1.448	4.550	67	58	216
5	1.515	4.760	67	61	277
6	1.582	4.970	67	63	340
7	1.649	5.180	67	66	406
8	1.716	5.390	67	69	475
9	1.782	5.600	67	71	546
10	1.849	5.809	67	74	620
11	1.916	6.019	67	77	697

12	1.983	6.229	67	79	776
13	2.050	6.439	67	82	859
14	2.116	6.649	67	85	943
15	2.183	6.859	67	87	1031
16	2.250	7.069	67	90	1121
17	2.317	7.278	67	93	1214
18	2.384	7.488	67	96	1309
19	2.450	7.698	67	98	1407
20	2.517	7.908	67	101	1508
21	2.584	8.118	67	104	1612
22	2.651	8.328	67	106	1718
23	2.718	8.538	67	109	1827
24	2.784	8.747	67	112	1938
25	2.851	8.957	67	114	2052
26	2.918	9.167	67	117	2169
27	2.985	9.377	67	120	2289
28	3.052	9.587	67	122	**2411**

This design with tubular IHX is specified with four 152.5MWth components because the secondary pressure loss is limited by a requirement (**less than 2% of the total pressure**). For information, 3 Modules of 203MWth each with 2411 tubes (active length: 31.3m) could be practicable with **secondary pressure loss of 3.65%.**

4.4 LAYOUT OF THE IHX

An overview of the helix IHX is presented in the figure 1.

The tubular IHX (German design concept), due to higher heat transfer surface, need more than one component vessel to transfer the thermal power 610MWth, the general arrangement with 4 tubular IHX is presented in the figure 2.

4.5 COST ASSESSMENT

From ANTARES information (reference 3), the cost assessment is 75M€ for one tubular IHX unit in the Plant configuration including four tubular IHX and four primary blowers.
This estimated cost has to be validated by component manufacturers

4.6 HELIUM/WATER COOLER

4.6.1 General presentation

The helium/water heat exchangers have been specified for VHTR ANTARES, for the Reactor SCS (Shutdown Cooling System) and for the primary blower cooler.
But due to the operating conditions (reference 2), the water side outlet temperature (**39°C**) of the SCS seems too low for an efficient used of the heat, and this application is not retained.
The helium/water exchangers are also used to extract the losses generated by the motor of the helium main blower, the heat flux to evacuate is approximately 1.2MWth, the water outlet temperature reaches **64°C** (reference2).
This last value remains compatible with potential heat applications like desalinization or district heating using the energy extracted by the water circuit. We propose to focus the study on this kind of cooler
The design based on a compact heat exchanger (COMPABLOC ALFALAVAL concept) is suitable (to see next chapter §3.2) to fulfill such an application, see here after the specification of CP15 Product.

4.6.2 ALFALAVAL concept

The COMPABLOC ALFALAVAL core is a stack of corrugated heat transfer plates in stainless steel, welded alternatively to form channels. Each model is modularized with a standard number of plates.
The concept is presented in the figure3.
A CP15 model has been ordered (to see design specification, table 2) by AREVA NP to ALFALAVAL GENOBLE Factory, for HE-FUS loop tests at ENEA BRASIMONE Centre.

A manufacturing data report has been provided by the manufacturer (Document N° RFF 9242.01/A - Customer reference N°: 40A/1007005947).

This component (figure 5) will to be set up and tested in the HE-FUS loop as cooler during He tests of the HEATRIC mock-up combination, in agreement with the COMPABLOC Instruction Manual (COMPABLOC IM-001 Rev. B), included in the manufacturing data report.

The tests results will confirm or not the potentiality of this compact concept for our application, proposal made during the last meeting (reference 4).

4.7 CONCLUSION

The objective of this study is mainly to propose a pre-design of a tubular IHX and a compact cooler based on “COMPABLOC” ALFALAVAL concept.

Four modules of 152.5MWth each with 2411 tubes helically wound on 28 rows, active length of 28.2m are proposed for RAPHAEL Reactor.

A helium/water cooler based on COMPABLOC ALFALAVAL technology is proposed and will be tested on HE-FUS loop and CEA CLAIRE loop, for validation.

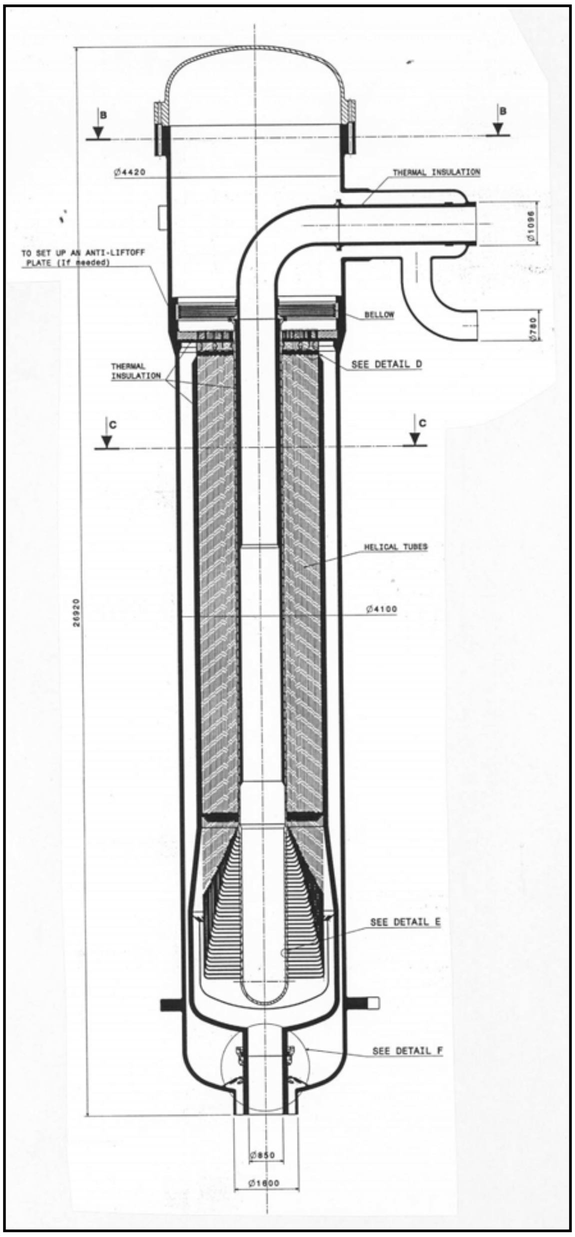
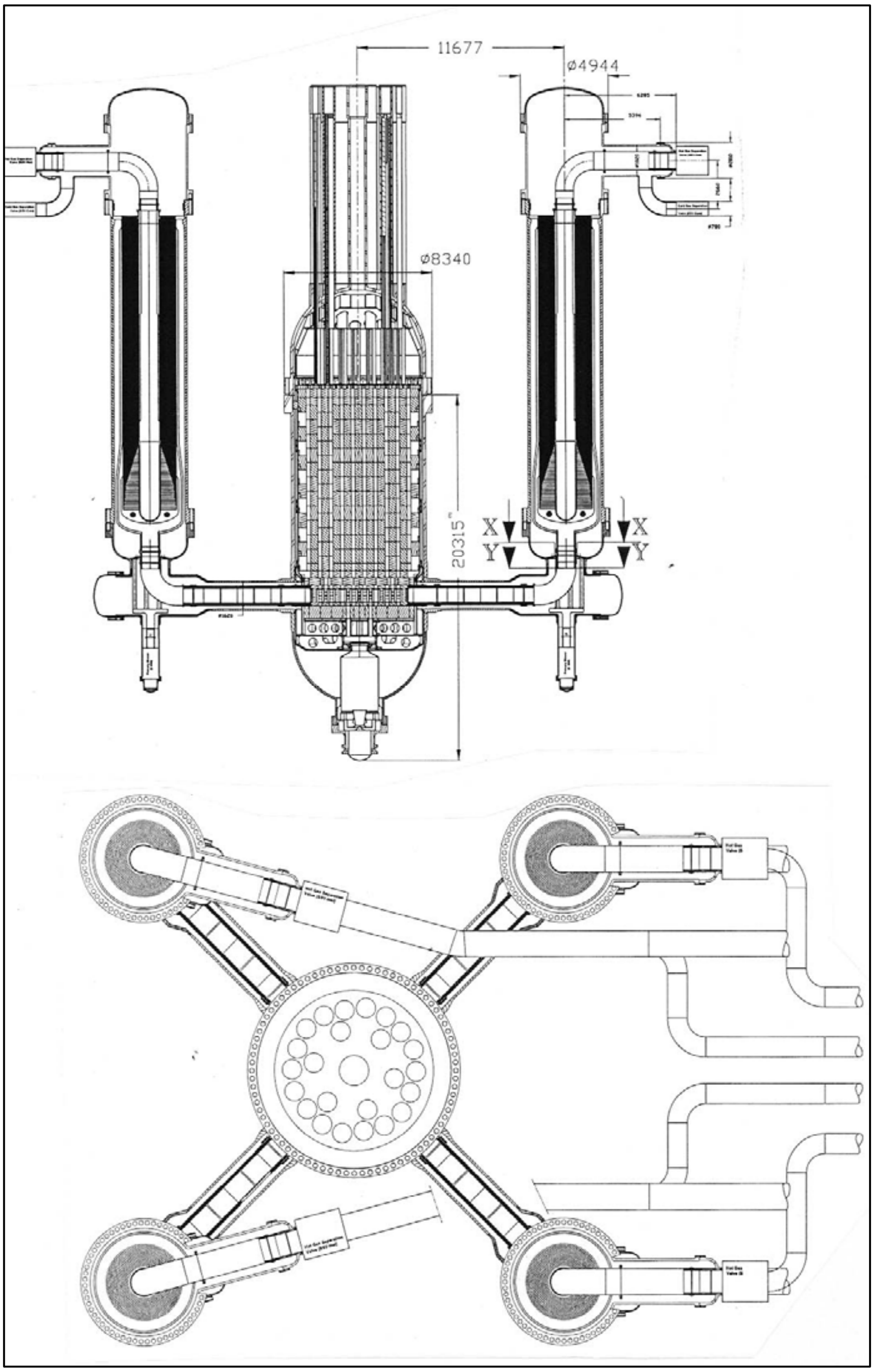


Figure 1: ihx tubular concept

- **Design options:**
- 8 radial tubes support plates welded vertically to the central tube
- Tube bundle helically « wound » alternatively on right and on left angle direction

Figure 2: ihx arrangement



OPTION: Four Component Vessels with 1 tubular IHX, 1 blower each

Figure 3: COMPABLOC ALFALAVAL CONCEPT

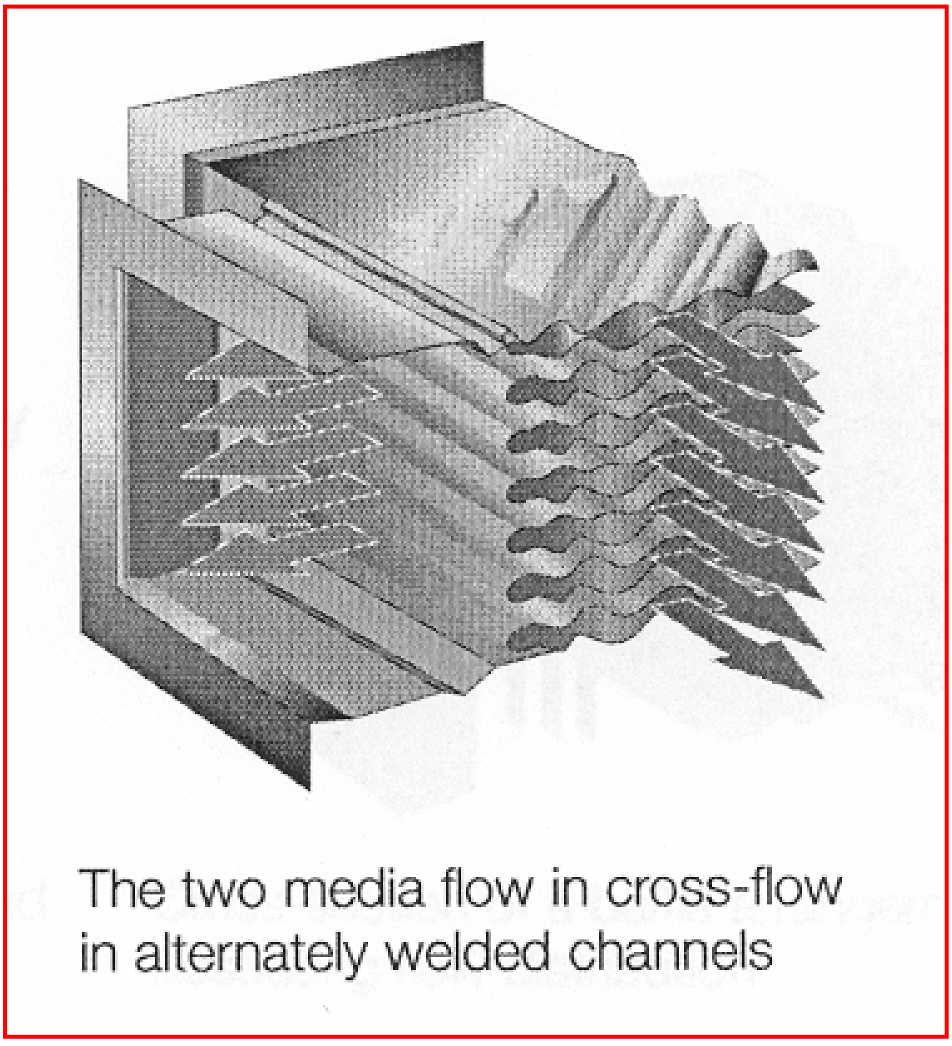


Figure 4 Alfa Laval name plate (cp15-v-30 /2007)

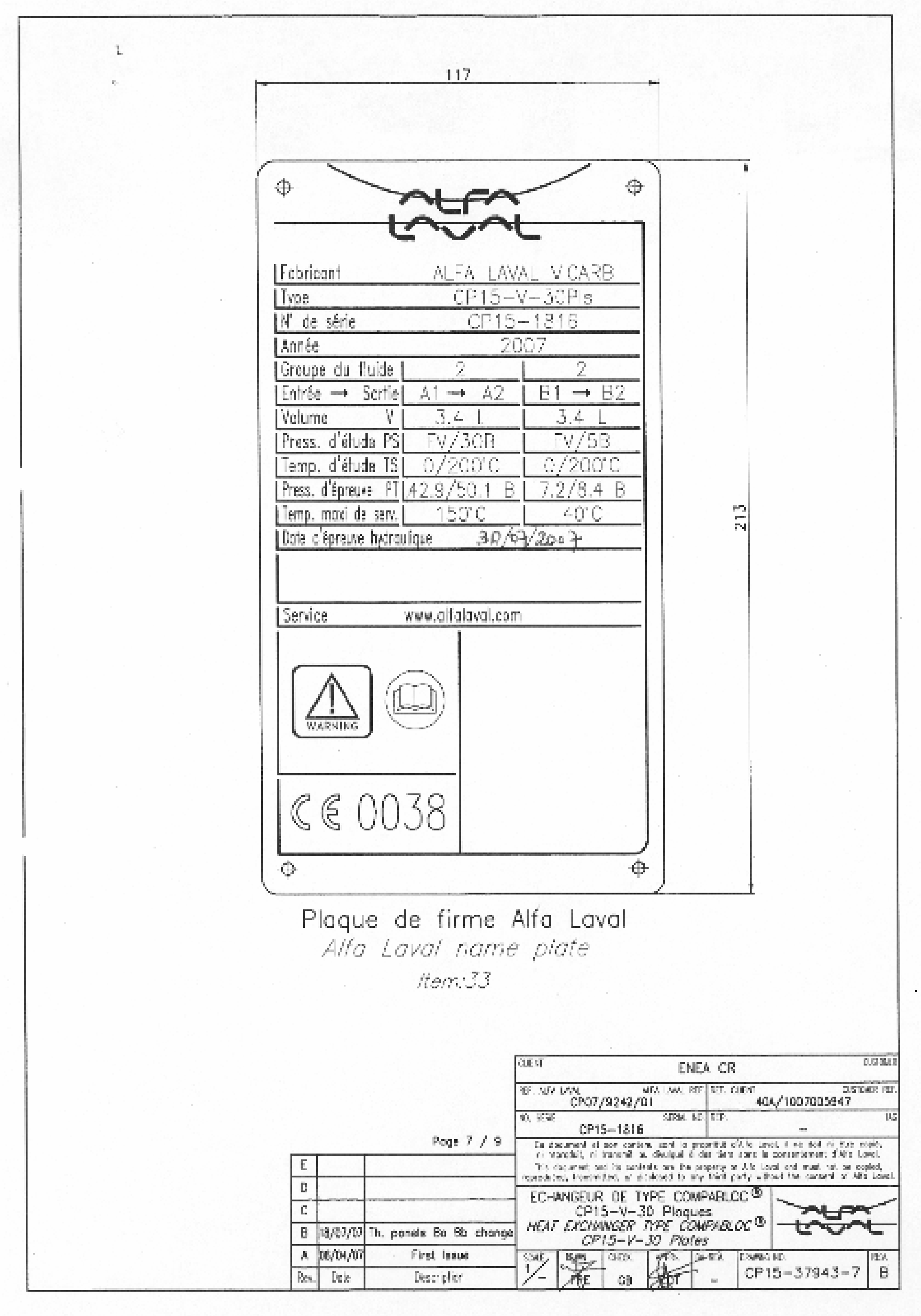


Figure 5: cp 15 alfa LAval main geometrical features (drawing cp15 37842/a)

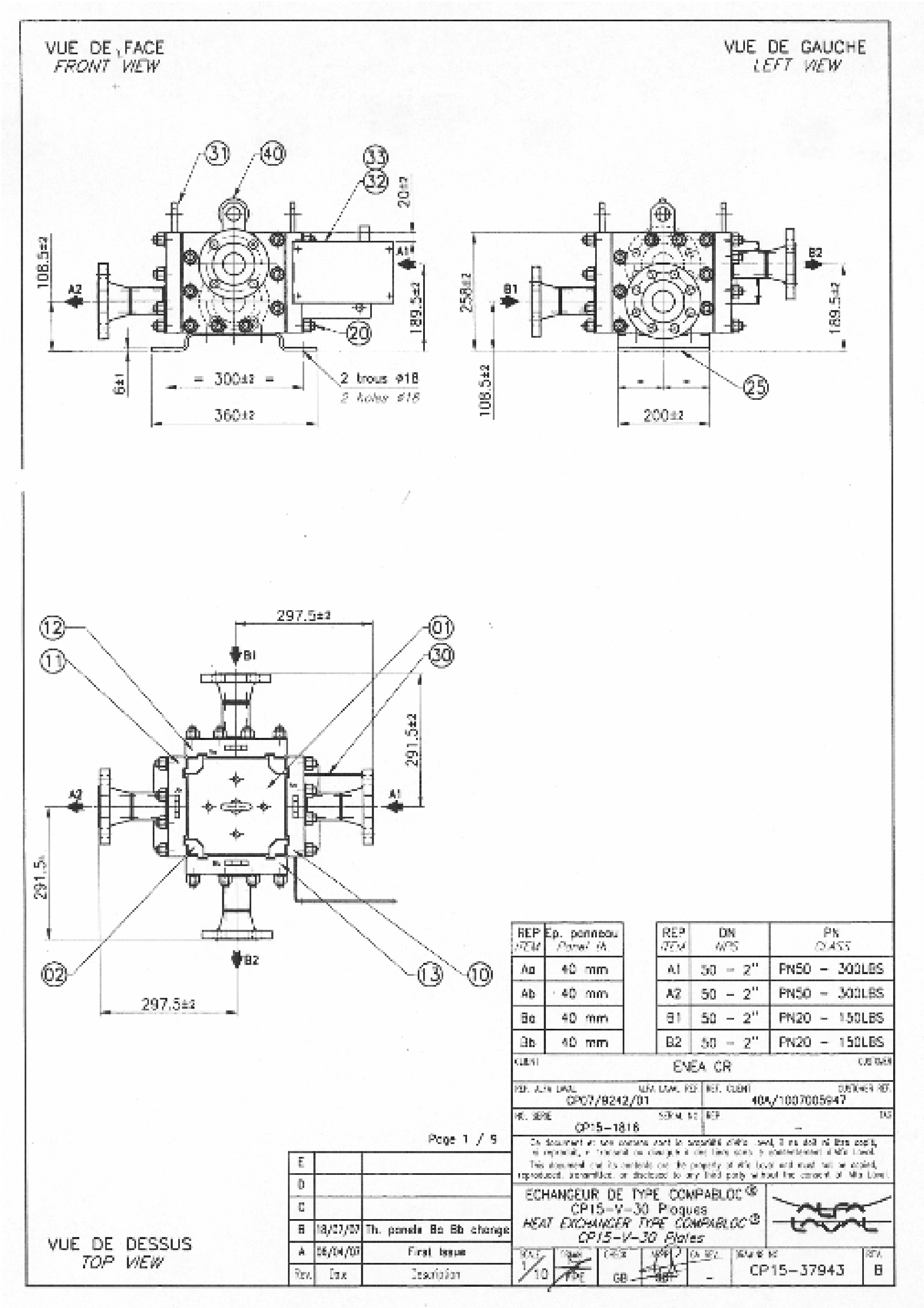


Table 1: general data (vhtr for process heat application, Ref.1)

For IHX tubular concept

IHX specifications at condition of core outlet Temperature 950°C		
	Primary (hot side)	Secondary (cold side)
Fluid	**Helium**	**Helium**
Total Flow rate (kg/s)	**210**	**210**
Inlet Temperature (°C)	**950**	**340**
Outlet Temperature (°C)	**390**	**900**
Inlet Pressure (MPa)	**5.0**	**4.5**

Effectiveness	**90 %**
Maximum Primary Pressure Loss	**2 %**
Maximum Secondary Pressure loss	**2 %**

Table 2: COMPABLOC HEAT EXCHANGER SPECIFICATION (he-fus tests)

Spécification Echangeur Compabloc Alfa Laval

Société : Areva
Echangeur Type : CP15
Projet : 07/PEE/FRGRFMI-426(a)
N°/Ref : 1 Date : 09/03/2007

		Coté Chaud :	Coté Froid :
Fluide		Helium	Water/Steam
Débit massique	kg/h	360.0	1119
Fluide Condensé/Vaporisé	kg/h	0,000	0,000
Température d'Entrée	°C	150,0	20,0
Température de Sortie (Vapeur/Liquide)	°C	100.0	40.0
Pression d'Entrée / Sortie	bar	18.0/18.0	
Pertes de charge (Perm/Calc)	kPa	360/1.35	100/0.0260
Vitesse dans les connexion (Entrée/Sortie)	m/s	24.9/21.9	0.159/0.160
Puissance échangée	kW	26.00	
ΔTm	K		
Coefficient d'échange (propre)	W/(m²*K)	766.8	
Coefficient d'échange (encrassé)	W/(m²*K)	408.3	
Surface d'échange	m²	0.7	
Excès de surface	%	88	
Ecoulement		Contre-Courant	
Orientation de l'appareil		Horizontal	
Nombre de plaques		30	
Nombre de passes		1	1
Groupage		1*15 M	1*16 M
Matériau des plaques / Epaisseur		AISI 316 L / 0.80 mm	
Joints		Graphite Reinforced	Graphite Reinforced
Matériau des connexions		AISI 316 L	AISI 316 L
Diamètre des connexions (A1/A2 or B1/B2)	mm	50/50	50/50
Position des raccordements		A1 -> A2	B1 -> B2
Code de fabrication		PED, Catégorie 1	
Type de raccordements		ANSI	
Pression de calcul	barg	FV/30.0	FV/5.0
Pression de test	barg	42.9	7.15
Température de calcul (min/max)	°C	100/200	20/200
Volume liquide	dm³	3,650	3.820
Poids net, vide / plein	kg	127/135	

Performances sous réserve du respect des débits et températures d'entrées, ainsi que des propriétés physiques.

4.8 LIST OF TABLES

4.9 LIST OF FIGURES

4.10 REFERENCES

1. DTS/DR/2006/14 RAPHAEL IP FP6 ST-CT-Component Development: Heat Exchangers , Valves and Vessel Work Plan of the work package WP-CT-1 Rev. B
2. NFPVE DC 1075 Deliverable D-CT1.1 RAPHAEL IP Components specifications for IHX and Coolers
3. Paper on “Development of the Intermediate Heat Exchangers (IHX) for ANTARES” ICAPP 2006 International Congress on Advances in Nuclear Power Plants June 4-8, 2006, RENO NEVADA
4. Minutes Fifth Meeting SP-CT Component Development meeting WP1 June 13 and 14th 2007, KNUTSFORD