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Assessment of the Integration of a He-cooled Divertor System in the Power Conversion System for the Dual-coolant Blanket Concept (TW2-TRP-PPCS12D8)

P. Norajitra, R. Kruessmann, S. Malang, G. Reimann

Institut für Materialforschung
Institut für Kern- und Energietechnik

Programm Kernfusion

Forschungszentrum Karlsruhe GmbH, Karlsruhe

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Untersuchung zur Integration eines Helium-gekühlten Divertorsystems in das Energieumwandlungssystem für das Dual-coolant Blanket Konzept

Zusammenfassung:

Der Einsatz eines heliumgekühlten Divertors zusammen mit dem Dual-coolant Blanket ist aufgrund seiner relativ hohen Kühlmittelaustrittstemperatur für das Erzielen eines hohen thermischen Wirkungsgrades des Kraftwerks von Vorteil.

Ein neues FZK He-gekühltes modulares Divertorkonzept mit integriertem Pin-Array (HEMP) wird vorgestellt. Seine Haupteigenschaften und Arbeitsweise werden ausführlich beschrieben. Das Ergebnis der thermohydraulischen Analyse zeigt, dass das HEMP-Divertorkonzept in der Lage ist, eine Wärmestromdichte von mindestens 10-15 MW/m^2 bei einem erreichbaren Wärmeübergangskoeffizienten von ca. 60 kW/m^2K und einer vernünftigen Pumpleistung abzuführen.

Die Integration dieses Divertorkonzeptes in das Energieumwandlungssystem - unter Verwendung eines geschlossenen Brayton-Gasturbinensystems mit dreistufiger Kompression - führt zu einem Nettowirkungsgrad des Blanket-/Divertor-Kreises von ca. 43 %.

Abstract:

Application of a helium-cooled divertor together with the dual-coolant blanket concept is considered favourable for achieving a high thermal efficiency of the power plant due to its relatively high coolant outlet temperature.

A new FZK He-cooled modular divertor concept with integrated pin arrays (HEMP) is introduced. Its main features and function are described in detail. The result of the thermalhydraulic analysis shows that the HEMP divertor concept has the potential of resisting a heat flow density of at least 10-15 MW/m^2 at a reachable heat transfer coefficient of approx. 60 kW/m^2K and a reasonable pumping power.

Integration of this divertor concept into the power conversion system using a closed Brayton gas turbine system with three-stage compression leads to a net efficiency of the blanket/divertor cycle of about 43 %.

Contents

1 Introduction

The dual-coolant (DC) blanket [1] (Figures 1, 2) is one of the EU advanced blanket concepts to be investigated in the frame of the long-term power plant conceptual study (PPCS). It is based on the use of a helium-cooled first wall and steel structure, a self-cooled Pb-17Li breeding zone, and SiC/SiC flow channel inserts serving as electrical and thermal insulators. The blanket is divided into large modular segments which, together with the divertor, build up an overall torus coverage and shielding for the magnets behind them. A considerable fraction of the heat energy of up to 15 % is released in the divertor. As already shown in the PPA 99 [2], integration of the divertor heating power in the power diversion system would help to significantly increase the thermal efficiency, leading to a cost reduction for electric power production in a commercial power plant. In principle, divertors cooled by helium, water, or liquid metal are conceivable together with the DC blanket. However, it is reasonable to use He-cooled divertors because of their relatively high outlet temperature of at least 700 °C which is suitable for combination with a gas turbine system.

2 Recent development of He-cooled divertor concepts

The development of He-cooled divertors for DEMO and commercial reactors started late in comparison to the blanket developments. It was launched within the framework of the preparation of the power plant conceptual study/plant availability 1999 (PPA 99) and turned out to be a major project with little knowledge available. Ever since, the development has been enforced and by now especially FZK and ENEA have set up work programs to precede the project.

A first study on He-cooled divertors was performed by Kleefeldt and Gordeev within the framework of PPA 99 [3]. They proposed a porous body under the divertor target plates to enhance the surface for heat transfer (so-called “unconventional design”). This concept could reach a heat transfer coefficient h.t.c. of about 20 kW/m²K and a power ratio (i.e. pumping power to thermal power) of 1.6 % (total thermal power of the divertor: 670 MW). The heat flux was limited to about 5 MW/m², which of course is not sufficient to cope with the foreseen peak heat flux of 15 to 20 MW/m². An advantage of this concept is the large heat transfer surface, which goes at the expense of a high pressure loss.

The next step of development was the slot concept by Hermsmeyer and Kleefeldt [4] within the framework of the PPCS II Study, 2001. Here, the porous body is replaced by a narrow slot, mainly to reduce the pressure loss. A h.t.c. of 14 kW/m²K seems to be possible at a pumping power ratio of 1.7 %. But again, this concept is limited to a heat flux of about 5 MW/m². Like the porous body concept, this structure would lead to hot spots on the target surface, because the distance over which the heat needs to be transported is not evenly distributed over the heated surface.

Therefore, Hermsmeyer and Malang [5] combined some ideas of the two existing concepts in the modified slot concept. Instead of a circular slot, an almost rectangular shaped channel is used now. The most loaded heat exchanger surface is additionally enhanced with a pin array. This concept is assumed to withstand also heat loads of 10 MW/m² at a h.t.c. of about 60 kW/m²K and a pumping power ratio of about 4 %.

Soon after that, it was generally decided to move to modular concepts, that means to split up the target plate into smaller finger-like units that could be cooled in parallel.

This way, thermal stresses within the target plate as well as within the structure should be reduced. Also, the pressure loss of the cooling system could be minimized by parallel flows.
Since 1995 already has ENEA [6] proposed a modular water-cooled concept for ITER: This is the so-called high-efficiency thermal shield (HETS) concept. Here, the body consists of small hemispherical cavities, of which seven are coupled in parallel into a group and four groups are connected in series. This idea was further developed in [7] for a helium-cooled concept. The system is able to withstand heat fluxes of 10 MW/m² at a h.t.c. of 17 to 31 kW/m²K along the radial path of flow. The pressure loss (occurring predominantly in the nozzle system) amounts to 0.8 MPa, leading to a pumping power of about 25 %.
In Table 1, a comparison of the recently developed concepts for cooling the divertor target plate is presented.
In parallel, FZK decided to develop another modular He-cooled divertor concept [8] on this basis with the goal to reach a heat flux limit of at least 15 MW/m^2. The conceptual design and thermohydraulic layout will be described in detail below.

3 The HEMP divertor concept

3.1 General

Designing high-performance divertors for a power plant needs quite a different approach than known from experimental reactors as ITER. Whereas the design of the ITER divertor is based on water at low temperature as coolant and copper as heat sink material, a power plant divertor has to be operated at much higher temperatures to keep the structural temperature above the embrittlement temperature (DBTT) of the refractory alloys and ensure suitability for high-efficiency power conversion systems. Cooling divertor plates with water at temperatures below 200 °C would waste some 10-20 % of the total power. Hence, a gas-cooled concept is required, allowing for high heat fluxes and coolant temperatures suitable for efficient use in the power conversion system.
A modular design instead of large plate structures is favourable to reduce the thermal stresses which limit the performance with respect to allowable peak heat fluxes and fatigue. Design issues for high-heat-flux (HHF) components also include a minimisation of temperature and temperature gradients and thermal stresses by cooling the high-heat-flux area with a coolant close to the inlet temperature and short heat conduction paths from the plasma-facing side to the cooled surface in order to keep the maximum structure temperature below the re-crystallisation limit.
Gas-cooled divertor concepts are favourable for reasons of safety and compatible with any blanket concept. Inlet cooling temperature is assumed to be about 600 °C to keep the structural temperature above the DBTT of refractory alloys and to achieve high power efficiency. The coolant temperature is limited at the lower boundary by the DBTT of the refractory materials under irradiation and at the upper boundary by the strength of the structure material (e.g. advanced reduced-activation ferritic-martensitic (RAFM) steels like ODS EUROFER or Ni-based alloys). Other restrictions result from the currently available materials for plasma-near structural application. In principle, the desired structural material is ODS EUROFER steel, but for this solution transition pieces between steel and refractory alloys have to be developed due to the different thermal expansion coefficients of the two materials. Since appropriate

solutions have not yet been made available, molybdenum alloy (TZM) is assumed to be the structural material for simplicity reason. The operating temperature window of refractory alloys is limited from below by irradiation embrittlement. Currently, 700 °C and 800 °C are estimates for TZM and tungsten, respectively, with a potential improvement (i.e. decrease) of up to 100 K. Upper bounds are set by the re-crystallisation temperature of the alloys, if they are used as structural material. As the latter is strongly dependent on the time of exposure, the limits are not strict. For TZM and tungsten, 1150 °C and 1200 °C, respectively, are reasonable figures, with a potential increase of 100 K. Hence, using a combination of W and TZM broadens the potential design window.

3.2 The conceptual design

The divertor is toroidally divided into 48 cartridges of 7.5° each. The principal design of the divertor cartridge with the sub-division of the divertor plate into smaller divertor modules is illustrated schematically in Fig. 3. A concept of a He-cooled modular divertor with an integrated pin array (HEMP) has been proposed [8]. Fig. 4 (left) shows the radial-toroidal cross section of HEMP divertor modules with all dimensions of interest. The below numbers given in brackets refer to this Figure. Details of the thimble are shown on the right hand side. The HEMP concept employs small tiles made of tungsten (1) and brazed to a finger-like (2) (or thimble-like) structure which in this study is assumed to be made of Mo alloy (TZM). These fingers have a width of 16 mm and a wall thickness of 1 mm and are inserted into a front plate (6a) made from TZM. This plate is connected to a back plate (6b) by parallel walls (in this study with TZM as material). Helium with a pressure of 10 MPa and an inlet temperature of 700 °C flows upwards to the pins (3) at the outer wall and via an inner tube wall (4) is then passed downwards to the He manifolds (5). The tiles are of quadratic shape with a mean area of about 16 x 16 mm^2 and 5 mm thick. In order to improve the heat convection at the top of the finger, a plate is inserted (by brazing) with a pin fin array (3). Fig. 5 shows an example of the pin arrangement which could be further optimised with respect to size, shape, and distance.

4 Energy balance and global thermohydraulic layout

The energy balance of the DC blanket concept was determined on the basis of neutronic calculations [9] and system code analyses [10] which are summarised in Table 2 and illustrated as an energy flow chart in Fig. 6. On the basis of an electric output of the power plant of 1500 MW, the fusion power was determined to be 3410 MW assuming a net efficiency for the blanket cycle of 0.43 and an energy multiplication factor of 1.17. The total blanket power of 3408 MW is divided into fractions of 1432 MW for He cooling and 1976 MW for the Pb-17Li circuit. The total divertor power amounts to 583 MW. It consists of power fractions of 335 MW for the divertor bulk and 248 MW as surface heat power (alpha and heating power) for the divertor target, respectively. A power distribution between inboard and outboard targets of 1:4 was assumed, thus leading to a surface heat power of 49.6 MW for the inboard and 198.4 MW for the outboard target, respectively. For a 7.5° divertor cartridge the size of an outboard target plate is about 810 mm x 1000 mm (toroidal x poloidal), leading to an average surface heat load of about 3.5 MW/m^2 for the overall target, and 5.1 MW/m² for the outboard. Taking into account the size of a divertor finger tile of about 16 x 16 mm^2, the number of rows in toroidal direction will be about 51 per cartridge and the number of finger units will amount to about 63 per row in poloidal direction.

Helium inlet and outlet temperatures at the target of 700 °C and 800 °C (with a temperature rise of 100 K), respectively, are assumed. The necessary helium mass flow rate to remove the divertor target heat amounts to 0.156 kg/s per outboard row. Since the peak surface heat load (in this study: 10 MW/m^2) is expected in a lower region of the target plate of about 1/3 of the poloidal plate height, cooling of divertor finger units within this region has to be increased in accordance with peak overheating (by about a factor of 2). This leads to a maximum helium mass flow rate of about 0.005 kg/s per finger unit.

5 Thermohydraulic setup and analyses for the divertor target

The thermohydraulic assessment of the new divertor concept included an estimation of the heat transfer coefficient h.t.c. and the pressure loss Δp. The goal was to reach a high h.t.c. and, at the same time, to keep the total pumping power lower than 10 % of the removed heating power. These contradictory requirements led to an optimisation problem.

A first assessment of the heat transfer coefficient was made with standard correlations taken from [11]. By lack of any better model that would describe the pin fin array, it was modelled by means of a tube bundle heat exchanger. This way, only the pins were taken into account, not the porting surfaces. Therefore, this model was believed to be rather conservative. The Nusselt number is then given by

$$Nu = \frac{1+(n-1)*f_A}{n}(0.3+\sqrt{Nu_{lam}^2+Nu_{turb}^2}), \quad (1)$$

$$Nu_{lam} = 0.664*\sqrt{\mathrm{Re}}\sqrt[3]{\mathrm{Pr}}, \qquad Nu_{turb} = \frac{0.037\,\mathrm{Re}^{0.8}\,\mathrm{Pr}}{1+2.443\,\mathrm{Re}^{-0.1}(\mathrm{Pr}^{2/3}-1)}. \quad (2, 3)$$

The Reynolds number Re is defined by

$$\mathrm{Re} = \frac{wl}{(1-\pi/4a)\nu} \quad (4)$$

with n being the number of pins, f_A a correction term for staggered arrays (≈ 1.2), Prandtl number Pr = 0.7, w inlet velocity of the gas, l “wetted” perimeter of the tube, and ν kinematic viscosity. The parameter a describes the normalised spacing between the pins: a = distance from centre to centre divided by diameter d (within one row). Since in our geometry, spacing and diameter d vary from row to row, the arithmetic mean value was used for the calculations. For the inner pins in blade form, an equivalent diameter was calculated as if the same top surface would belong to a cylindrical pin.

The pressure loss within the pin array is calculated by the correlations

$$\Delta p = N\xi \frac{s}{d'}\frac{\rho}{2}w_m^2, \qquad \xi = 2\left(\frac{64}{\mathrm{Re}} + \frac{2}{\mathrm{Re}^{0.18}}\right), \qquad \mathrm{Re} = \frac{w_m d'}{\nu} \qquad (5, 6, 7)$$

with N denoting the number of contractions between the pins in flow direction, s being the length of a contraction, d' the equivalent diameter ($= d(4a/\pi - 1)$), ρ the density of the fluid, and w_m the mean velocity

$$w_m = \frac{w}{1-(\pi/4a)}. \qquad (8)$$

The pressure loss in the structure is calculated from

$$\Delta p = \rho \sum \left[\left(\psi \frac{l}{d} + \sum \varsigma_n\right)\frac{w^2}{2}\right], \qquad (9)$$

with the necessary factors for friction ψ and form ς_n taken from literature [12].

Calculations were performed for the above-stated conditions: inlet pressure of helium 10 MPa, inlet temperature 700 °C, temperature rise 100 K, maximum heat flux to the surface 10 MW/m². For calculating the heat transfer coefficient h.t.c. and the pressure loss, the worst case was assumed. So, instead of basing on the mean gas flow through the finger, the maximum gas flow was used as if all fingers would encounter the maximum heat flux. Then, for the outboard target fingers, 0.005 kg/s of helium gas are necessary.

First, the inlet velocity and the velocity in the smallest gap between the pins were calculated by dividing the mass flow $\dot{m}$ through the density ρ and the inlet surface A (either the total inlet surface to obtain the inlet velocity or the reduced surface without the space the pins take to obtain the velocity in the narrowest gap):

$$w = \frac{\dot{m}}{\rho A} = 30.8\, m/s\ . \qquad (10)$$

In this example, the density was calculated for an inlet pressure of 10 MPa and a mean gas temperature of 1023 K using correlation [13]

$$\rho = 48.091\, \frac{p[bar]}{T[K]} \quad [kg/m^3] \qquad (11)$$

For the correlations, the kinematic viscosity of helium is needed as well. For dynamic viscosity, a correlation exists:

$$\eta = 0.4646 * T^{0.66} \quad [kg/ms, T\ in\ K] \qquad (12)$$

from which the kinematic viscosity can be deducted.

From these data and the geometric values, the h.t.c. and the pressure loss could be calculated using the above-mentioned correlations. The total pressure loss (finger unit and He manifolds) then leads to the necessary pumping power which was set in relation to the target heat power.

Table 3 shows the results for the geometrical arrangement of the pin array shown in Figure 5. It contains two rows of pins of decreasing diameter and a third row of pins alternating in circular and in blade form. The gas flow was assumed to be directed from outwards to inwards. The contraction because of the decreasing diameter was not taken into account. The h.t.c. and the pressure loss were calculated. The h.t.c. was then converted into a flat plate case with an area factor of 2.8 (= area with pins/area without pins) for comparison sake.

A value of about 61,000 W/m²K was achieved for this geometry. The pressure loss and, therefore, a pumping power of 5.5 % result for this case.

For divertor cooling, a h.t.c. of about 60,000 W/m²K is considered widely sufficient, so that Δp can be kept at a low level. The envisaged maximum pumping power of 10 % leaves enough margin to improve the heat transfer without violating this limitation.

An optimisation of the pin geometry (e.g. slight changes in arrangement and diameter of the pins) would contribute to a significant increase of the h.t.c. and decrease of Δp, which plays an important role for the pumping power. Therefore, other geometrical arrangements have to be investigated. Further, a deeper study of the microscopic flow field for the determination of h.t.c. and Δp should be undertaken by means of a commercial CFD Programme. Finally, experimental investigations are planned to confirm the theoretical findings.

Based on these results and first assessments of thermal stresses, the surface temperature at the tile can be estimated to be 1715 °C and the temperature on the pin ground to be 915 °C, which is far below the re-crystallisation limit.

6 The power conversion system and net efficiency

The reference power conversion system for the DC blanket concept is based on the use of a closed 3-compression-stage Brayton gas turbine cycle (Fig. 7). This solution offers an important advantage in avoiding the contact of liquid metal with water and tritium permeation losses to the environment. For the secondary He loop, a high He pressure of 15 MPa was chosen, which does not explicitly affect the thermal efficiency of the power conversion system, but is required to simultaneously achieve a high efficiency of the intermediate heat exchangers (IHX) and a low pressure loss ratio. To adequately adapt the thermal powers and the coolant temperatures between the primary and secondary loops, a system of 4 heat exchangers is chosen (Fig. 8). The detailed data of the four-stage IHX (1: Blanket FW, He/He, 1432 MW; 2: Divertor bulk, He/He, 335 MW; 3: Blanket interior, Pb-17Li/He, 1976 MW; 4: Divertor target, He/He, 248 MW) are summarised in Table 4. The total heat power to be transferred amounts to 3991 MW. The examination [14] for this case resulted in a maximum equivalent bundle size (for reactor system with single IHX loop) OD x H of e.g. about 6 m x 7 m for the intermediate heat exchanger IHX-1 (He/He, helical tube type) with a heat transfer surface of 18,520 m² and pressure losses in the primary and secondary loop of 0.06 and 0.07 MPa, respectively.

The calculation of the thermal efficiency of the power conversion system [14] was carried out with a FORTRAN program based on the basic rules given in [15]. In these calculations a turbine efficiency of 0.94 and a compressor efficiency of 0.92 were assumed, which are recommended in [16]. The recuperator efficiency was determined by (T10-T3)/(T2-T3) to amount to 0.94. The overall compression ratio amounts to 4.3 (1.63 for each compressor stage). This leads to a thermal efficiency of 0.44. Taking into account the total pumping power of 38.4 MW (Pb-17Li: 5 MW, blanket He: 30 MW, divertor He: 3.4 MW), the net efficiency of the blanket/divertor cycle becomes approximately (0.44*4074 MW – 38.4 MW)/4074 MW = 0.43.

7 Conclusions and outlook

A new concept for a divertor based on helium cooling has been developed. The target plate is split up into smaller modules, which are favourable to reduce thermal stresses. The cooling surface under each module is enhanced by a pin array. Tungsten is envisaged as a material for the tiles and the pins as well, while the structure will preferably be made of TZM. The operational window is limited by the re-crystallisation temperatures at the upper limit and by the DBTT at the lower limit.

A first calculation of the heat transfer coefficient gave a value around 60,000 W/m²K and a pumping power ratio (necessary pumping power to gain in heat power) of 5.5 %. This is considered to be sufficient to remove the envisaged heat load of 10 MW/m² on the divertor target with a reasonable effort.

Integration of the divertor in the power conversion system would raise the efficiency of the fusion reactor to 44 %.

In the near future, the geometrical arrangement of the pins will be further investigated by the help of a commercial CFD Program. The results will be compared to the experiments for pressure loss and heat transfer measurements, which are already under way. Manufacturing techniques for the pin arrays are currently discussed.

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Table 1: Comparison of recent developments of the divertor target plate

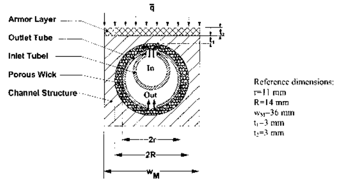
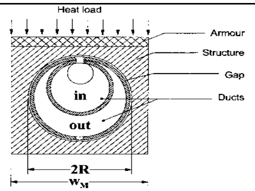
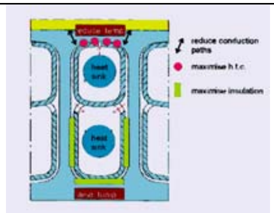
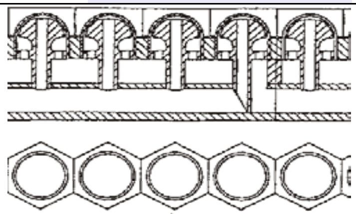
Concept		Author	Feature	HTC (W/m²K)	ΔT (K)	T_{max} wall (°C)	Pressure loss (MPa)	Power ratio (%)	Heat flux limit (MW/m²)
Porous body		FZK, IRS (1999/2000)	Porous medium	3000 .. 20,000	168	1230	0.2	1.6	about 5
Slot concept		FZK, IKET (2001)	Slot	14,000	200	1090	0.14	1.7	about 5
Modified slot concept		FZK, IKET (2002)	Slot with pin fin array	61,000	100	1080	about 0.25	4	10
HETS concept		ENEA (2002)	Impinging jet	32,000	100	1550	0.8	25	10

Table 2: Main data of the DC blanket concept.

Overall plant	
Electrical output [MW]	1500
Fusion power [MW]	3410
Neutron power [MW]	2728
Alpha-particle power [MW]	682
Thermal power [MW]	3991

	Blanket	**Divertor**
Average neutron wall load [MW/m^2]	2.27	1.7
Max. neutron wall load [MW/m^2]	3.0	
Average surface heat load [MW/m^2]	0.45	0.67
Max. surface heat load [MW/m^2]	0.59	10
Alpha-particle surface power [MW]	546	136
Heating power [MW]		112
Neutron power [MW]	2445	283
Energy multiplication	1.17	1.17
Thermal power [MW]	3408	583
Surface area [m^2]	1077	69.3 (target)

Coolant:		
Helium:		
- Inlet temperature [° C]	300	700 (target)
- Outlet temperature [° C]	480	800 (target)
- Pressure [MPa]	8	10 (target)
- Mass flow rate [kg/s]	1528	473 (bulk) 477 (target)
- Pumping power, η = 0.8 [MW]	30	3.4
Pb-17Li:		
- Inlet temperature [° C]	480	
- Outlet temperature [° C]	700	
- Mass flow rate [kg/s]	46053	
- Pumping power, η = 0.8 [MW]	5	
Secondary helium:		
- Inlet temperature [° C]	285	
- Outlet temperature [° C]	700	
- Pressure [MPa]	15	
Thermal efficiency (power conv. system) Net efficiency (blanket/divertor cycle)	0.44 0.43	

Table 3: Results of the thermohydraulic assessment of geometry No. 5.

Inlet pressure (MPa)	10
Inlet velocity for highest heat flux (m/s)	31
Max. velocity in narrowest gap (m/s)	162
Heat transfer coefficient (W/m²K)	61228
Pressure loss in pin array (MPa)	0.033
Pressure loss in finger supply (MPa)	0.09
Pressure loss per row (MPa)	0.114
Total pumping power, reactor (MW)	13.7
Removed surface heat power (MW)	253
Percentage pumping power of heat power (-)	5.5

Table 4: Intermediate heat exchanger (IHX).

	IHX-1 Blanket He	IHX-2 Divertor bulk	IHX-3 Blanket LM	IHX-4 Div. target
Heat transfer (MW)	1432	335	1976	248
Medium: - primary loop - secondary loop	8 MPa He	10 MPa He	Pb-17Li	10 MPa He
IHX Tin/out (°C): - primary loop - secondary loop	 480 / 300 285 / 434	 480 / 615 434 / 469	 700 / 480 469 / 674	 800 / 700 674 / 700
Total heat transfer surface (m^2)	18520	2910	16300	2100
Tube dimensions OD x s (mm)	17 x 2	16 x 2	14 x 2	16 x 2
Bundle - type - equiv. size[1] OD x H (m)	 helical 6.2 x 6.8	 straight 3.4 x 2.5	 helical 6.5 x 4.0	 straight 4.1 x 1.2
He pressure losses (MPa) - primary loop - secondary loop	 0.06 0.07	 0.01 0.03	 - 0.07	 0.04 0.01

[1] for reactor system with single IHX loop

Large module segmentation of DC (model C)

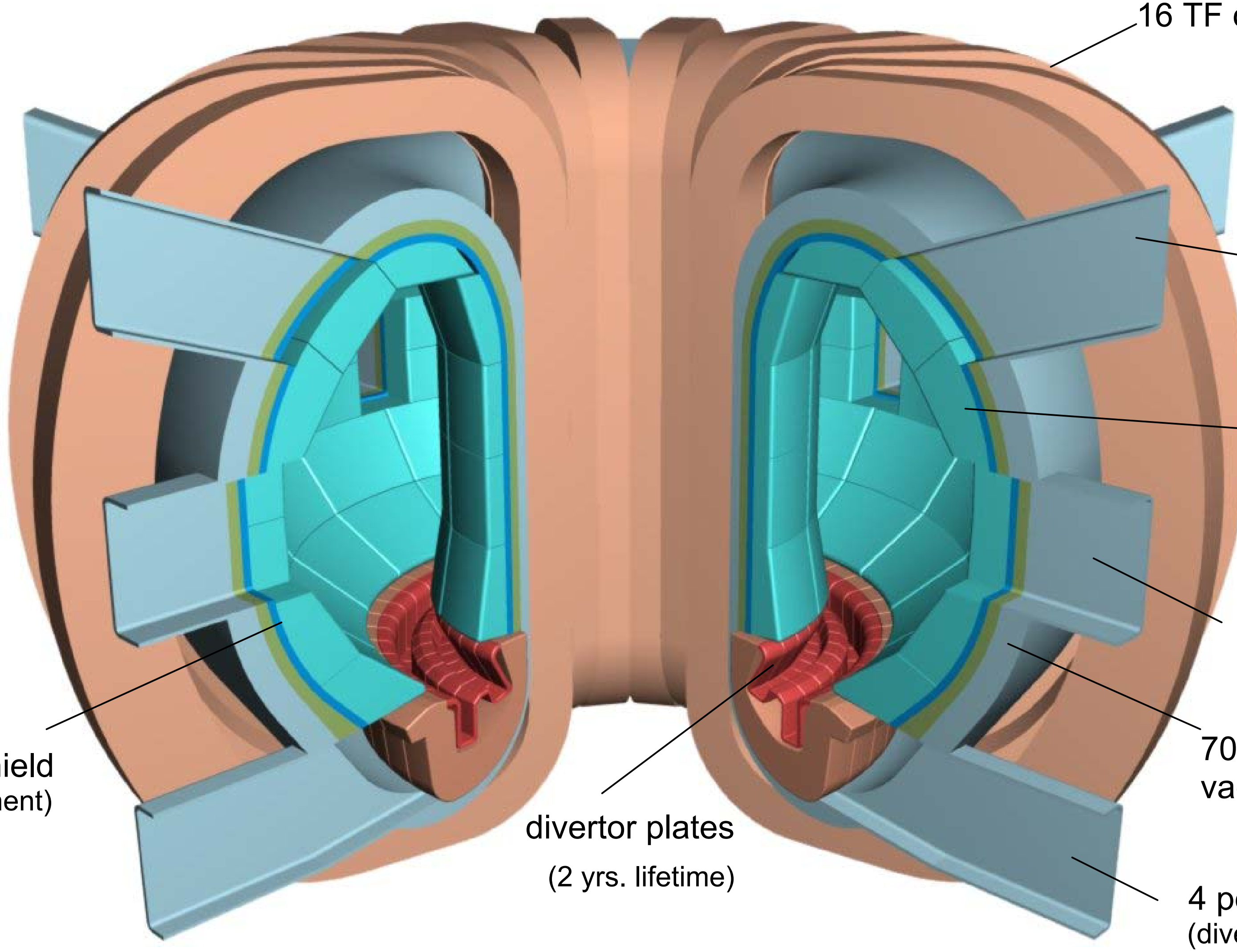


Fig. 1: Large module segmentation of the dual-coolant blankets.

Dual-coolant blanket (model C)

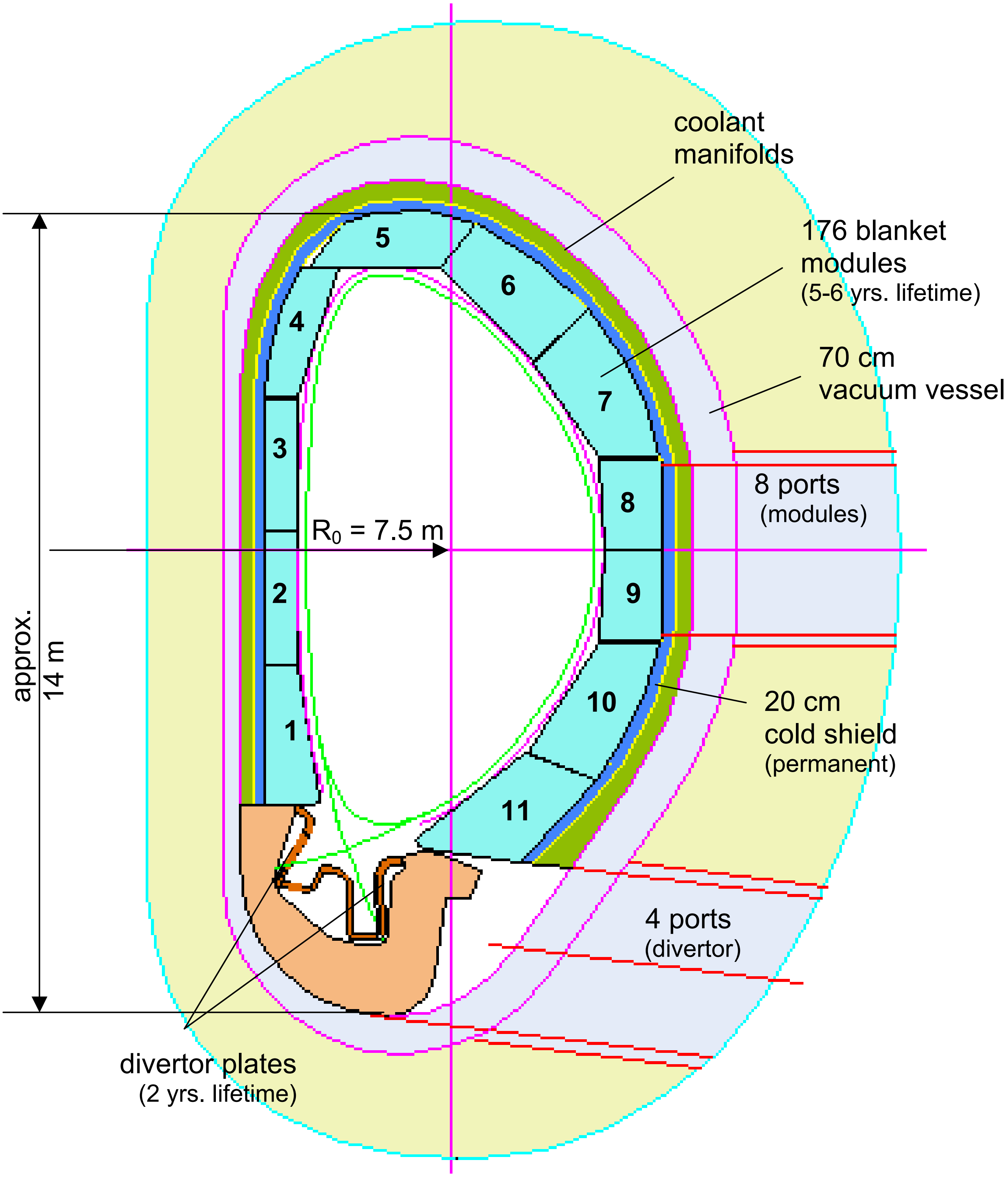


Fig. 2: Cross section of the fusion reactor torus with dual-coolant blanket modules.

He-cooled Divertor

Principle design (7.5 °)

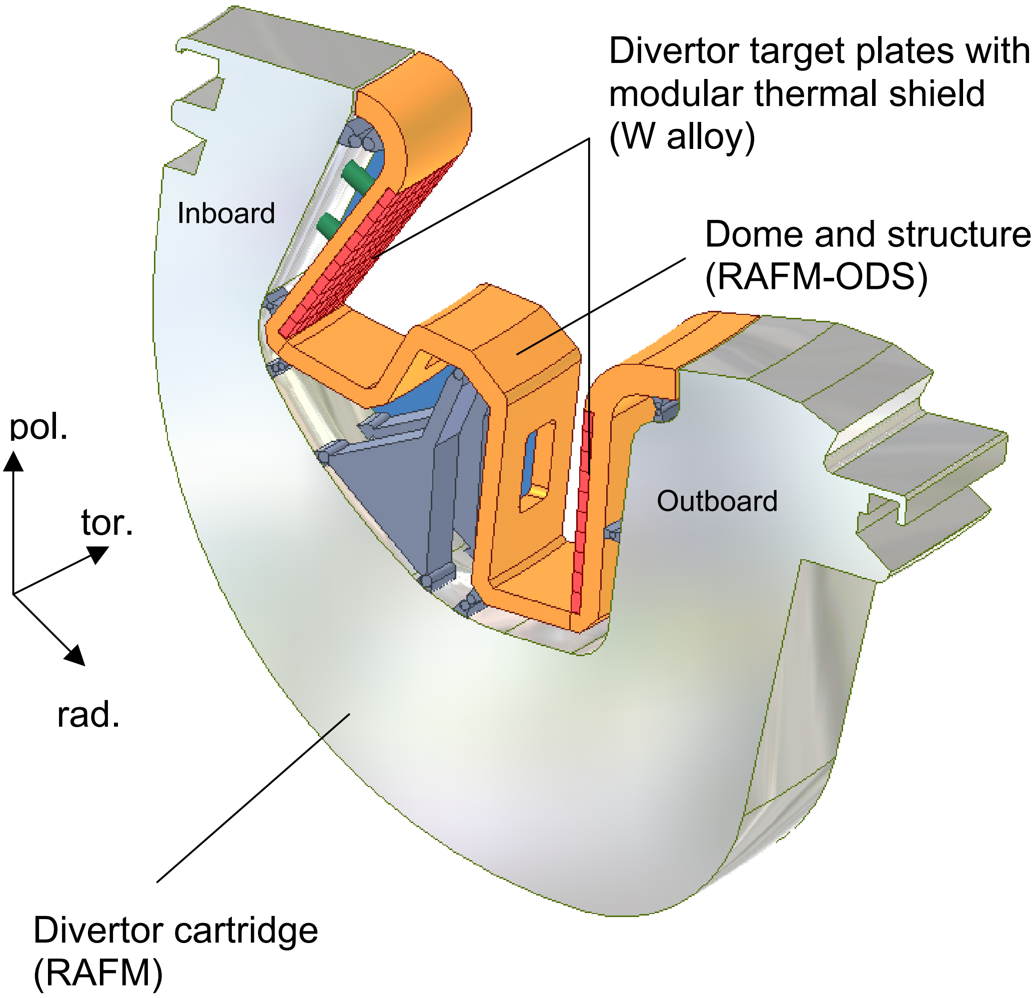
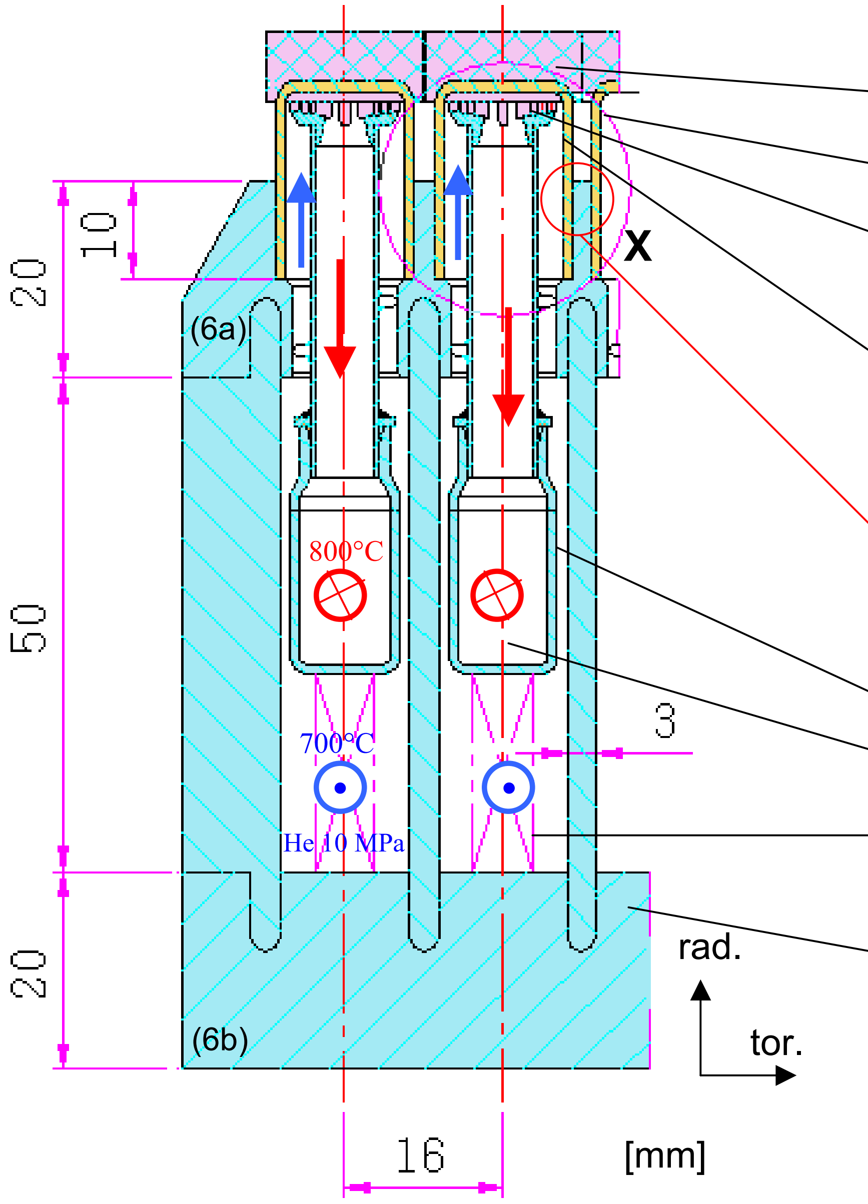


Fig. 3: Principle design of a 7.5 ° divertor cartridge.



Function	Material
(1) Tiles	W
(2) Thimble	TZM
(3) Pins	W
(4) Cooling unit: tube wall	TZM
Transition zone (T)	
(5) He manifold outlet inlet	TZM
(6) Structure (6a) front plate (6b) back plate	TZM

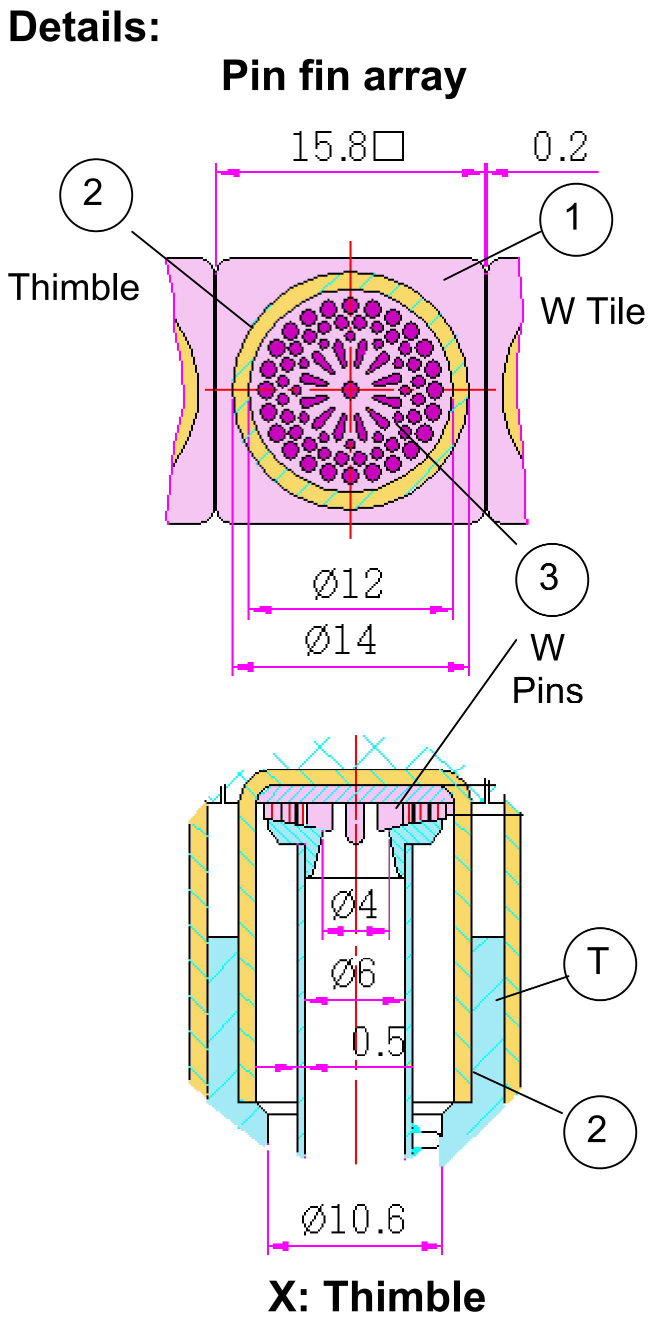


Fig. 4: The FZK modular divertor concept with integrated pin array (HEMP).

HEMP: Arrangement of W Pin Array (Ø 0.6-1.0 mm, H=1-2.5 mm)

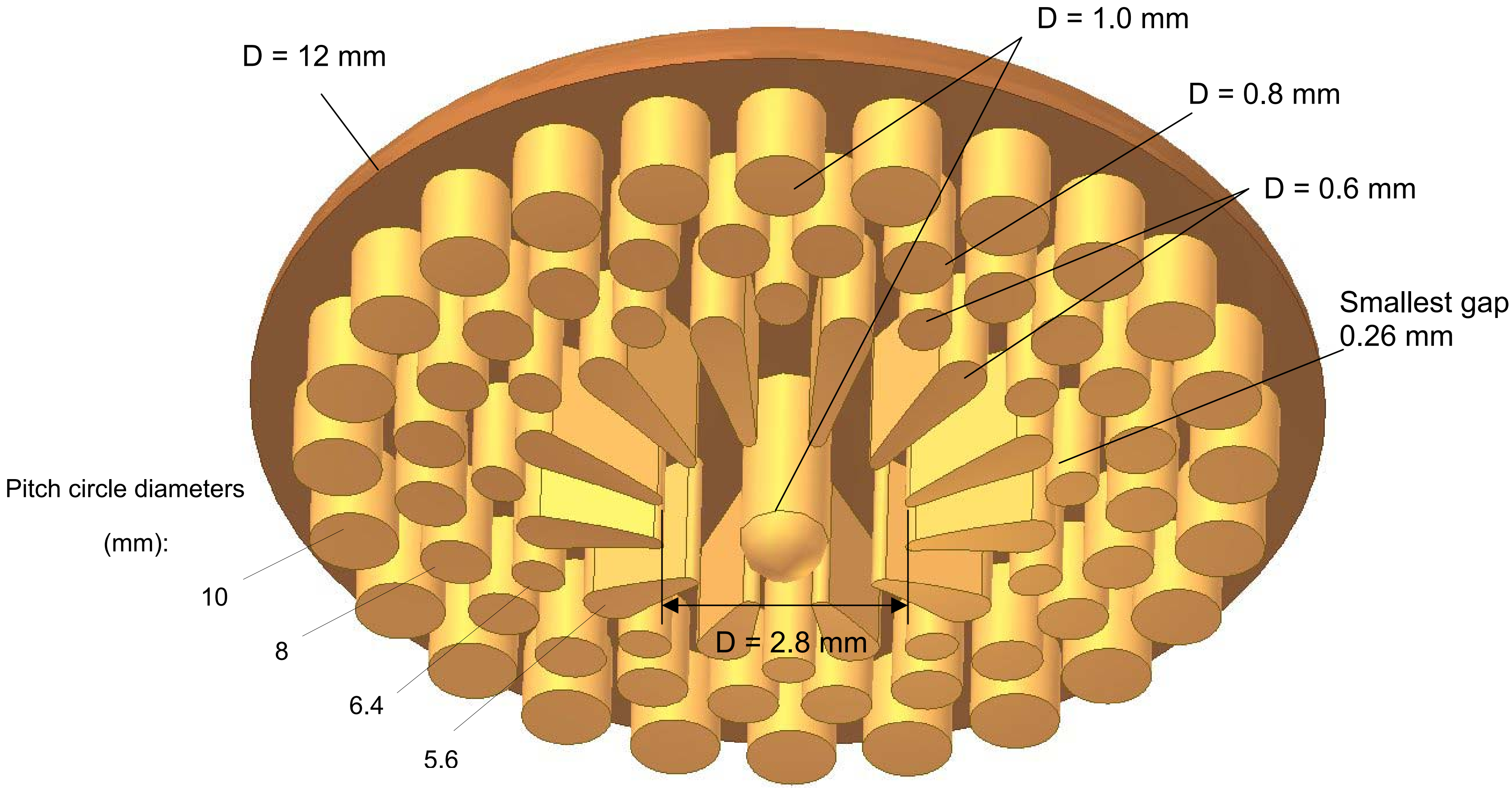
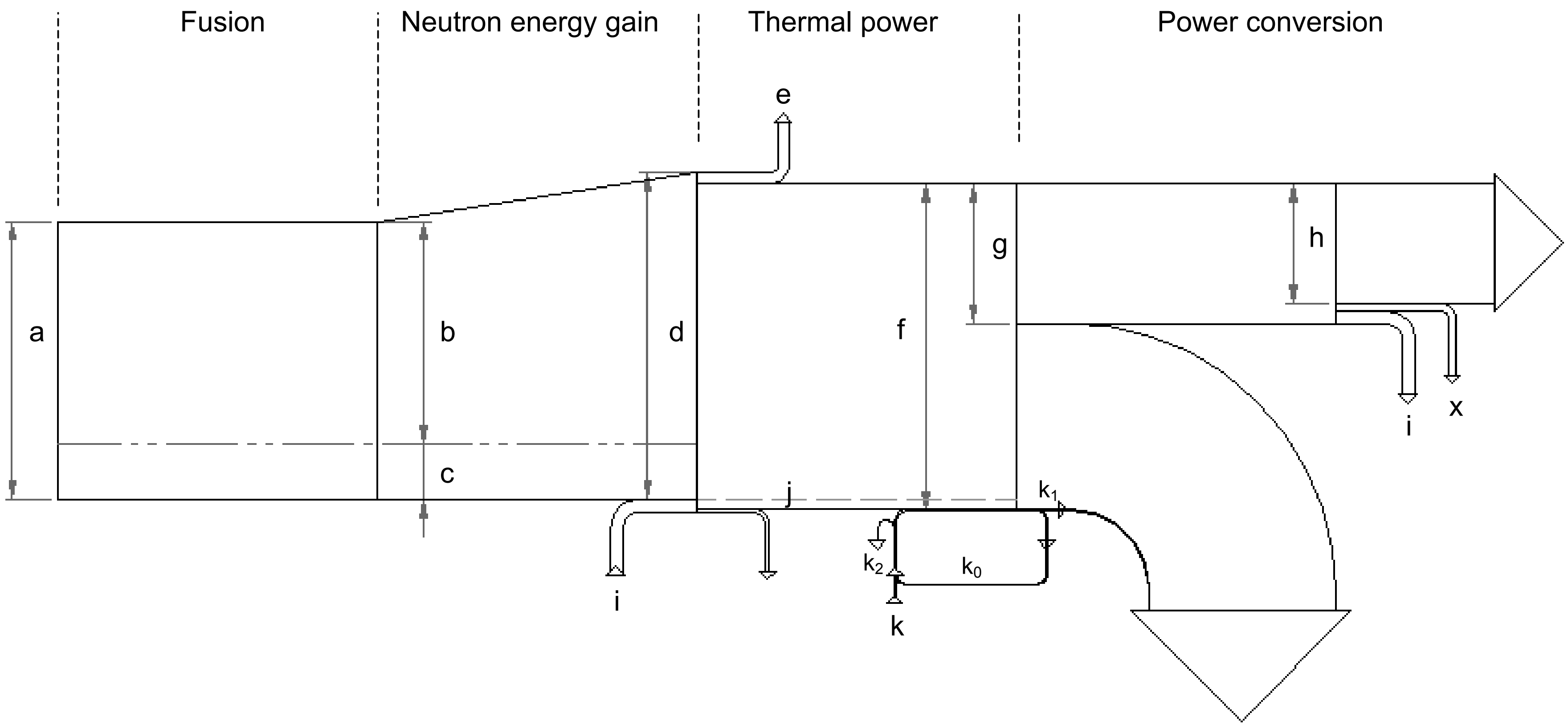


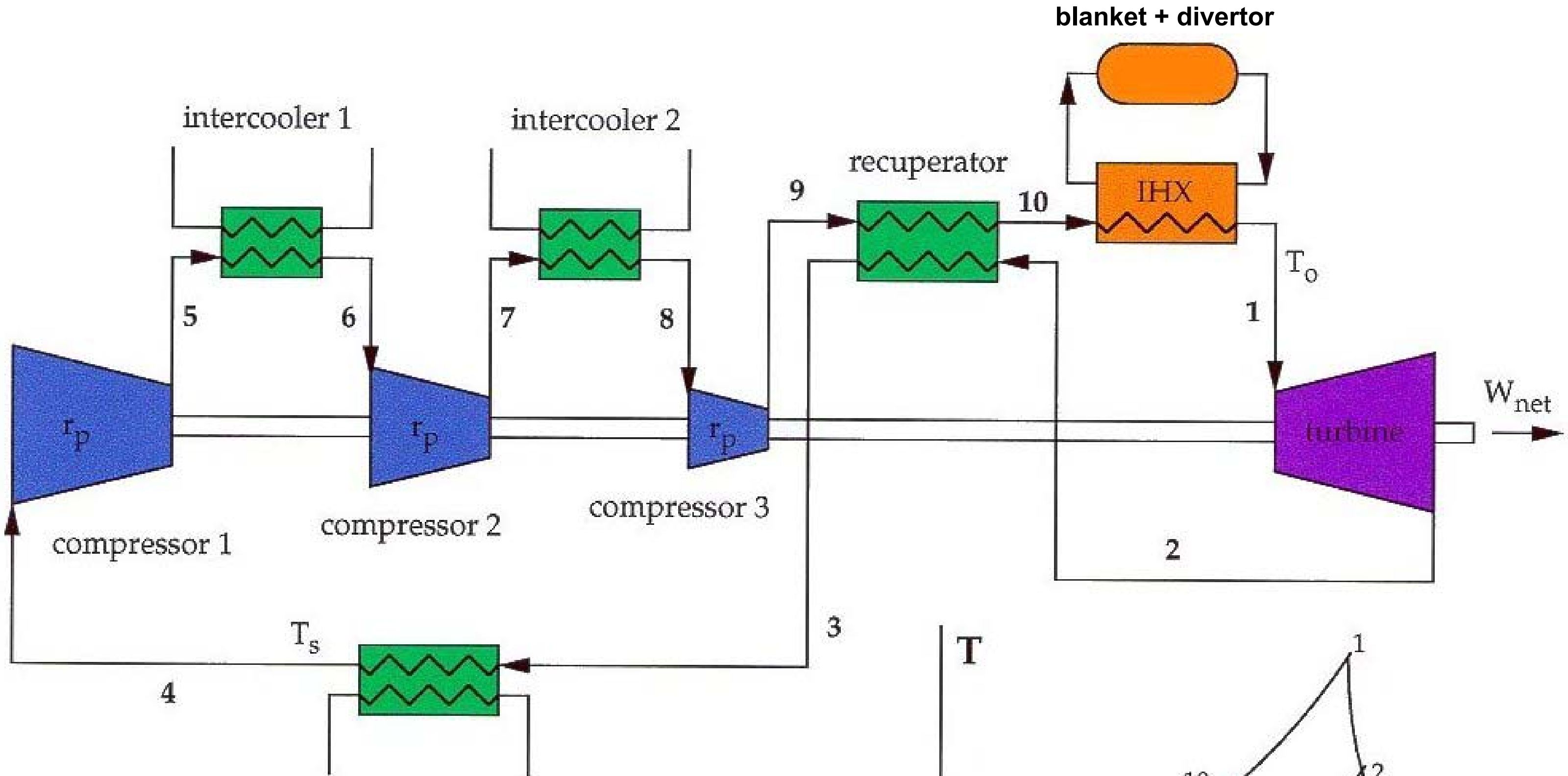
Fig. 5: Layout of the pin array.

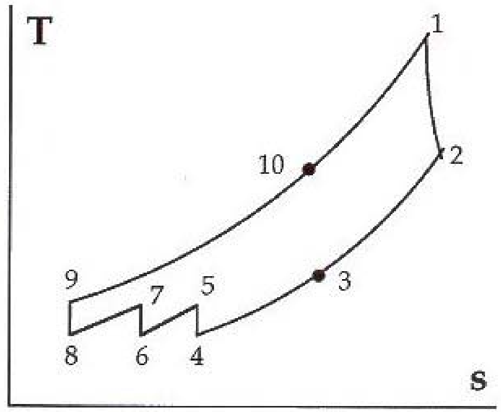


Power in (MW)

a:	Fusion power:	3410	h:	Net electric power	1500
b:	- Neutron power fraction (blanket / divertor)	2728 (2445 / 283)	i:	Heating power, electric	160
c:	- Alpha power fraction (blanket / divertor)	682 (546 / 136)	j:	Heating power, thermal	112 (η = 0.7)
d:	Thermal power (ideal: 3333+682 MW)	4015	k:	Pumping power, electric (*$k_0 \approx 13.2$, $k_1 \approx 17.5$, $k_2 \approx 7.7$ MW*)	$\approx$25.2 (η = 0.8)
e:	Power losses (cold shield + VV)	136	x:	Power consumption, auxiliaries	56
f:	Total thermal + heating power *(d-e+j)* (blanket / divertor)	3991 (3408 / 583)		Net efficiency blanket cycle: g / (f+k1) $\approx$ 0.43 Overall efficiency plant: h / a = 0.44	
g:	Total electric power produced	1716			

Fig. 6 : Energy flow diagram.





Temperatures (°C)

T1	700	T6	34
T2	295	T7	108
T3	118	T8	34
T4	34	T9	108
T5	108	T10	285

r = 4.3 (r_p = 1.63) , thermal efficiency = 0.44

Fig. 7: A closed Brayton gas turbine cycle with 3-stage compression.

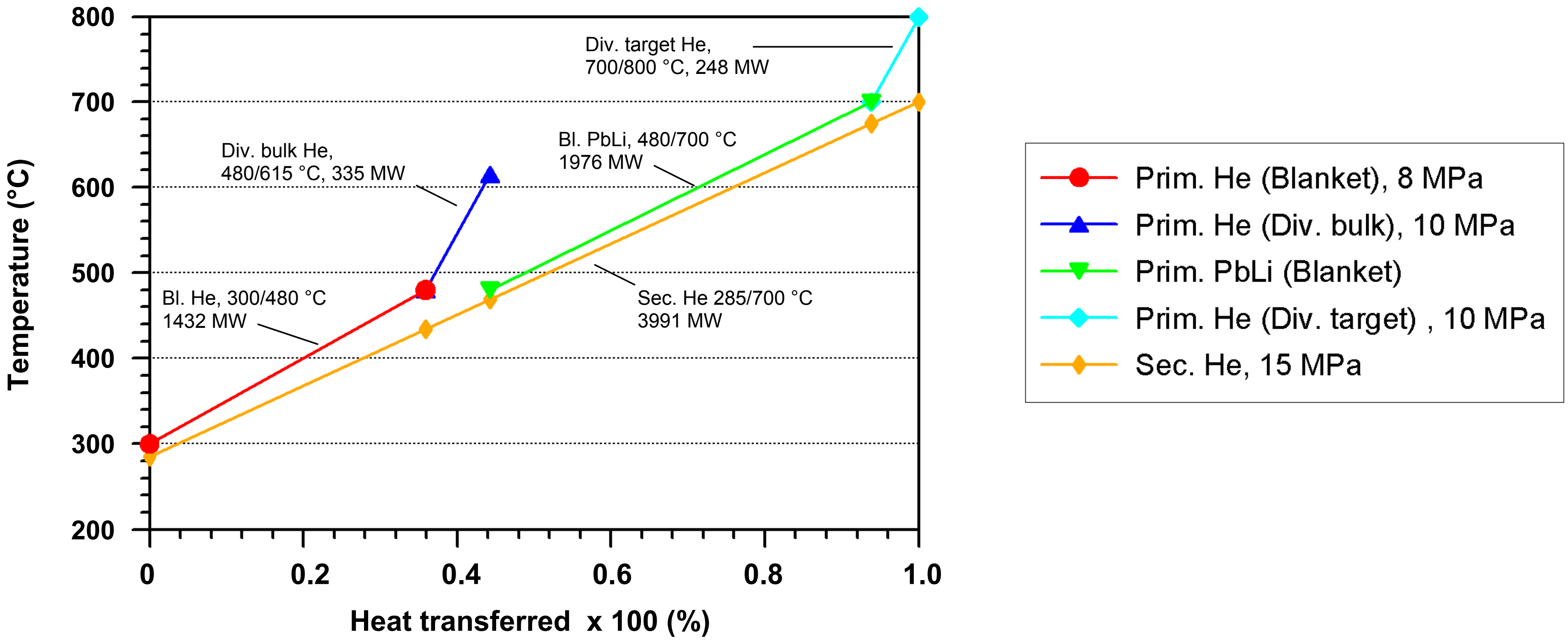


Fig. 8: Heat transfer diagram of a 3-stage gas turbine cycle with the heating power of the helium-cooled divertor integrated in the power conversion system.