COMPACT STEAM GENERATOR FOR NUCLEAR APPLICATION

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ABSTRACT

Diffusion-bonded heat exchangers, with hydraulic diameters in the range of 0.5 to 6 mm, offer the opportunity to produce high power density devices and therefore save space and cost. Although their use is now extensive in the industry for single-phase to single-phase heat transfer, steam generation has never been practiced. There are concerns regarding the small diameter of the channels, since fouling and corrosion could lead to channel blockage. There are also challenges in accurate prediction of two phase flow and heat transfer in such geometries with existing models. However, should practical steam generation be possible in compact systems, significant equipment volume reduction is likely.
A 1-D computer code is created to simulate compact steam generator (CSG) performance in an integral pressurized water reactor. A sensitivity study confirms the presence of large uncertainties arising from existing critical heat flux and pressure drop models, as shown by a previous MIT study. The reference design opts for vertical, straight channels, which enjoy more consistent predictions. The reference design allows generation of superheated steam in a once-trough flow path, which saves space and boosts efficiency above conventional U-tube steam generators and even coil type steam generators. A factor of 13-20 in volume reduction, together with a gain of 0.3% to 0.5% in cycle efficiency could be achieved.. In addition, primary-to-secondary leakage (steam generator tube failure accident) becomes less likely, if not eliminated altogether. A study of the equipment volumes of once-through compact steam generation and steam generation through expansion into a steam drum, finds that the compact steam generator provides more economy of volume. It requires less space both in and out of the vessel of the integral PWR.

KEYWORDS
compact steam generator, diffusion bonded heat exchanger, boiling, heat exchanger

1. INTRODUCTION

Compact, plate-type heat exchangers have been used successfully for several decades by the industry. Their compactness, high efficiency (close temperature approaches) and cost effectiveness make them very attractive. Progress in diffusion-bonding technique has enabled manufacturing large, robust and leak-tight devices. However their use is limited to single-phase to single-phase heat transfer, for either liquid or gas. Concerns about impurity deposition, which could lead to channel blocking, have precluded their use for phase change and their application to steam generation in power equipment. In addition, boiling phenomena in such small geometries are complex and not well represented by existing models. Experimental data are scarce. It is difficult to make local measurements without disturbing the thermal-hydraulics at play.

At MIT, after having highlighted the major benefits that compact steam generator technology could bring to a small and medium reactor (SMR) [1], it was decided to investigate via numerical simulation and small-scale experimental testing the performance of a CSG. This paper expands on assessment of the potential of compact steam generator for nuclear application. The latest developments of the technology are first presented in section 2. Then, modeling of a CSG is performed, which underlines the limits of the present knowledge. Finally a reference design is defined and compared to other types of steam generators.

2. BACKGROUND

Compact, diffusion-bonded, heat exchangers for energy applications typically present hydraulic diameters in the range of 0.5-6mm, often referred to as “mini-channels[1]”. Diffusion-bonded heat exchangers, such as the ones manufactured by Heatric™, can withstand the pressure and temperature of a steam generator (see Table I). The reliability and robustness of this technology brings confidence in the fact that it can match the standards of the nuclear industry.

Table I. Typical diffusion-bonded, compact heat exchanger operating conditions [2]

1-phase printed circuit heat exchanger (PCHE)	
Hydraulic diameter	0.5 – 6 mm
Plate thickness	0.5 – 5 mm
Operating temperatures	Up to 890ºC
Max design pressure	65 MPa
Typical Reynolds number	Gas 1,000 – 100,000
	Liquids 10 – 5,000

The high power density of compact heat exchangers is mainly due to the small size of the channels (10 to 1000 larger heat transfer area per unit volume) rather than the increase in heat transfer coefficient [3]. However, it was reported that the small hydraulic diameter could enhance boiling heat transfer in particular geometries and operating conditions [4]. The confinement effect flattens the bubbles, which reduces the thickness of the liquid film near the heated wall and boosts the evaporation process. At the same time the early coalescence of the bubbles in confined space favors dryout [4, 5, 6], therefore the overall effect of mini-channel geometries on boiling heat transfer is questionable. Literature is scarce on the subject, due to the difficulty of obtaining precise experimental data and thus building reliable and universal correlations.

Compact heat exchangers (CSGs) have never been used in practice for steam generation in nuclear steam supply systems, or other power equipment, but paper concepts have emerged during the past few years. Two major manufacturers, Westinghouse and Areva, have designed small modular power plants that take advantage of this technology to integrate steam generators inside the reactor pressure vessel in an integral layout. Westinghouse presented two concepts of integral PWRs. The first one, the Westinghouse SMR, features recirculating, once-through, straight tubes steam generators [7]. Although the heat exchanger is not made of plates but of classical shell-and-tubes, the solution adopted to shrink the internal part of the SG is uncommon. The moisture separation equipment are located outside the vessel, in an external component named steam drum, where recirculation occurs. The steam quality at the outlet of the heat exchanger and inlet of the steam drum is 60% [7]. The steam drum is massive and as high as three stories tall to achieve 800MWth of steam generation [8]. The second integral LWR design of Westinghouse is the Integral, Inherently Safe (I2S) LWR, developed together with GeorgiaTech. It also comprises a steam drum but the shell-and-tube heat exchanger has been replaced by a plate heat exchanger with mini-

[1] By opposition to “micro-channels’ found in the electronics industry.

channels that operate at higher pressure [9]. The heat exchanger has become a single-phase to single-phase heat exchanger. The fluid exiting the HX is subcooled, and steam is generated exclusively in the lower pressure steam drum.
Areva recently disclosed patents of a compact steam generator for nuclear application [10]. Unlike Westinghouse steam generators, 100% steam flow quality exits the secondary channels. Moreover, the steam is superheated. The channels are vertical, almost straight and – noteworthy – connected to each other within a plate so that cross flow between channels can occur [11]. Cross flow helps homogenize the temperature and mass flow rate among channels. Hot and cold channels have both rectangular cross-sectional shape but have different dimensions. The plates are made out of a titanium alloy (TA6V) [12] and, according to the authors, can sustain up to 10 MPa pressure difference between each other [11].
Both the Westinghouse and Areva Compact Steam Generators (CSG) feature vertical straight channels. Reliability and inspectability are major design drivers. Namely, it led Westinghouse to abandon the idea of generating steam inside the channels for fear of channel blocking by impurities. It led Areva to design straight channels of constant hydraulic diameter to use eddy current inspection techniques [13]. These operational aspects are discussed in Section 4.

3. COMPACT STEAM GENERATOR MODELING AND SIMULATION

A 1-D simulation tool is created to compare different designs and correlations, and assess uncertainties. Later on, this tool will be validated with experimental results.

3.1. Tool Description

The compact steam generator is modeled by a 1-D channel code in the form of a MATLAB script. Hot and cold channels are discretized in meshes in which thermal-hydraulic conditions are volume-averaged. Meshes exchange mass, momentum and energy at each time-step. The flow conditions at the inlet of the channels, as well as the geometry, are inputs controlled by the user. The code computes the conditions at the outlet and returns the total power rating of a representative channel.
The solution of the set of finite difference equations is obtained by matrix iteration over a fictive[2] time-scale. Temperature, pressure, void fraction in the meshes are used to calculate heat flux and pressure drops which give a new set of thermal-hydraulic conditions in the channel at the next step. Since the inlet conditions do not change, the equations converge toward a steady-state solution. Iteration parameters Δt, number of spatial meshes N and convergence criteria are set by the user. Typically, the channels are discretized into 20 to 60 nodes. It takes about 5 minutes to converge within 1% power error for 0.5-meter channels with 20 meshes.

3.2. Model Comparison

In the absence of experimental data, only verification of the code by comparison to others can be achieved. Here, the results are compared to those obtained earlier by the models of Shirvan et al [1, 14]. Shirvan et al used a similar 1-D channel code to calculate the performance of a printed-circuit heat exchanger. That heat exchanger exhibits semi-circular channels of 2mm diameter for a plate thickness of 1.6mm. Hot and cold channels are counter-current. They are of zigzag geometry: the ratio of total length vs effective (heated) length is 0.84. The plate is made of stainless steel whose conductivity is 21 W/m-K. Last but not least, the channels are horizontal[3]. The same correlations as Shirvan's for heat transfer and pressure drop are used here. The results for heat transfer are very close to those obtained by Shirvan et al.

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[2] The code is not rigorously speaking time-dependent because the mass flux is kept constant rather than pressure-dependent and because the initial conditions are unrealistic.
[3] Classical layout for single-phase PCHE by Heatric

– less than 1% difference of total power rating. The single-phase pressure loss on the primary side is similar as well, with less than 1% difference. However, the two-phase pressure drop calculation on the secondary side leads to a very different value: 8 kPa vs 40 kPa calculated by Shirvan et al. It seems that an error in implementation of the friction pressure drop calculation is at the origin of the 40kPa results. Measurements will be needed in the future to validate the pressure drop calculation. Fortunately, the system pressure at the outlet is not sufficiently affected (40 kPa within 5.8MPa) to significantly impact the temperature and heat transfer. To assess the robustness of the calculation, a sensitivity study is performed on the PCHE performance.

3.3. Sensitivity Study

Variation of the conduction length (mean distance between hot and cold channels), single-phase heat transfer as well as post-CHF heat transfer correlations do not change the results significantly: less than 1% difference on power rating. A much larger impact is observed on the pressure drop when the correlations are modified. Up to 60% difference for single-phase and 33% for two-phase pressure drops occurred. The peculiar zigzag geometry seems to be the main contributor to the uncertainty because few correlations exist for such case. Moreover the Reynolds number at the inlet is out of the range of the reference Ishizuka correlation (Ishizuka given for 3500<Re<22000 in zigzag channels) [15]. On the two-phase models, a change in the void fraction model has no effect on total pressure drop. Changing the two-phase flow multiplier expression causes minor effects. For future designs, the straight-channel geometry is recommended in order to reduce the uncertainty of pressure-drop predictions.
Finally, the influence of the critical heat flux prediction is studied. The location of dryout is of major importance for the performance of the heat exchanger. Due to the drop in heat transfer coefficient after dryout, a larger surface area is needed to get a given power rating of the device. Shirvan et al. selected a value of Xcrit = 0.90, and augmented the heat transfer surface area of the final design by 10% to accommodate uncertainty in the critical heat flux calculation.
No critical heat flux model exists for the narrow channel dimension of interest. A possible approach is to use the values of Groeneveld in his look-up table [18] and apply correction factors due to the channel configuration: 1.2mm hydraulic diameter instead of 8mm, and horizontal layout instead of vertical, so that[4]

$$q''_{crit}(D_e, P, G, x) = k_{hor} \left(\frac{0.008}{D_e}\right)^n q''_{Groeneveld}(P, G, x) \tag{1}$$

For the horizontal correction factor k_{hor}, two approaches are found in the literature: one by Duckler, which considers stratification in the channel based on mass flux, and one by Wong, which considers transit time of bubbles toward the upper part of the channel [19]. This latter correlation requires calculation of the void fraction. Therefore two void fraction models were tested: the CISE model and the Zuber (drift flux) model [20]. The correction for horizontal layout is significant, with CHF occurring at a critical quality between 0.5 and 0.8 instead of nearly 0.9 without the correction. Moreover, the different correlations give very different correction factors, which illustrate the uncertainty about CHF values in horizontal channels.
The correction factor due to the narrow channels (1.2mm hydraulic diameter here) instead of the reference 8mm diameter channels adds additional uncertainty (see Figure 1). Groeneveld suggests a correction form of $\left(\frac{0.008}{D_e}\right)^n$, with $n = 0.5$. However his data cover experiments down to 3mm diameter channels only. A reference by Tanase et al [Tanase, 2009] gives different correction factors at low mass flux and high steam quality. In our case (G<250kg/m^2-s, x>0.5), Tanase recommends $n = -0.3$. The sign of n is in this

[4] Note that no correction factor for non-circular flow areas has been found. It is a first caveat of the CHF calculation for semi-circular or rectangular channels.

case more in accordance with observation and physical intuition, for which CHF occurs earlier for narrow channels than for large channels of the same mass flux, pressure and quality, due to bubble coalescence. Tanase's article is based on CHF data covering channels down to about 1mm in hydraulic diameter.

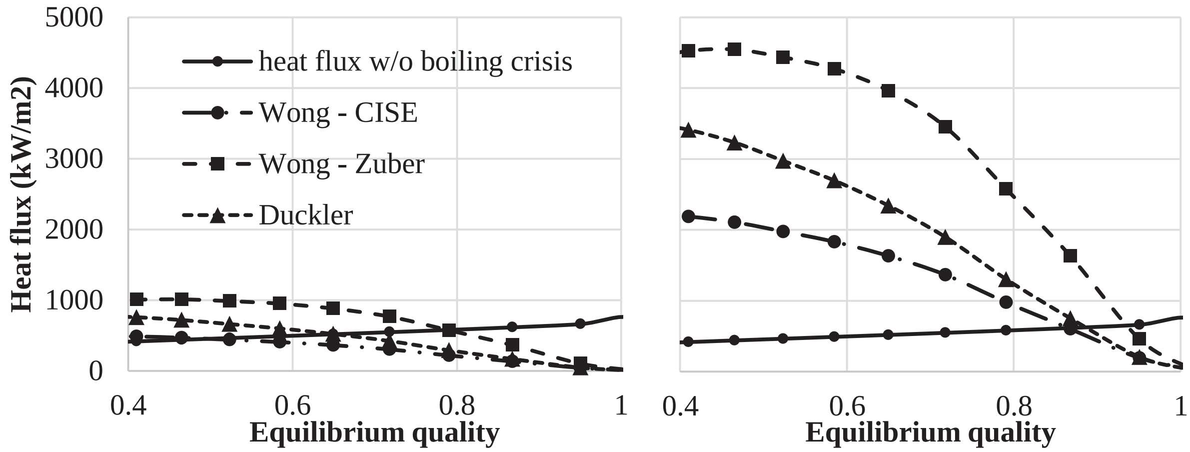

Figure 1. Critical heat flux with two different diameter correction factors: Tanase correction factor n=-0.3 (left) and Groeneveld correction factor n=0.5 (right). The scale is the same for both plots.

As shown in Figure 2, the critical heat flux in narrow channels suffers unacceptable uncertainty. The CHF correlations are out of their domain of validity and give results spanning from Xcrit = 0.9 to Xcrit = 0.5. However, adopting a vertical layout yields more consistent results, since more CHF data exists – although for larger diameter channels – and because the horizontal correction factor is eliminated.

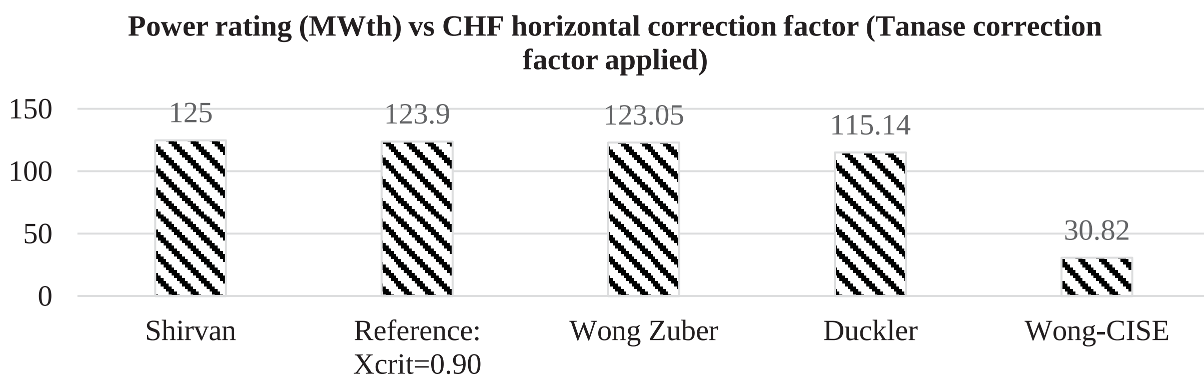

Figure 2. The critical heat flux model strongly affects CSG performance. In the case of Wong-CISE model, power drops by 75%. An upflow vapor blanketing phenomenon occurs. Further investigation is needed to determine if the cause is physical or numerical.

4. CSG REFERENCE DESIGN

4.1. Operation, Inspection and Maintenance Considerations

Compact steam generators raise operational challenges because of boiling in mini-channels. In the boiling process, impurities not carried by the steam flow are left in the channels. Deposited on the channel wall, they lead to reduction of the flow area and increased pressure drop. In the case of mini-channels, this fouling effect is dramatic. It is therefore vital to control water chemistry closely. The entire condensate will need to be filtered and cleaned through polishers and demineralizers to ensure high water quality at the SG inlet. Experience exists with full-flow water processing, because it is already in place in PWRs equipped with once-through steam generators (B&W design). More recently, significant progress has been made in water chemistry control to support the development of combined cycle fossil power plants [21]. This advancement can ensure reliability of future compact steam generators. Strainers can also be installed to prevent channel blockage by large impurities, at the expense of a higher pressure drop. Then, if prevention of fouling failed, mitigation is possible in the form of compact heat exchanger cleanup. Heatric, for instance, offers cleaning services for PCHE using high pressure water jets. The process seems effective [2]. Ultimately, if plant shutdown time needs to be minimized at all cost, spare compact steam generators can be connected and operated in lieu of the fouled ones. In a PWR, these spare CSG could be dedicated to safety functions (decay heat removal) at the beginning of a cycle. At the end of the cycle, they would participate in normal steam generation while the fouled CSG would be dedicated to safety functions, which require less power rating. CSG are likely more affordable and moveable than classical steam generators, so spare CSG could make an economical sense.

During transients, CSGs behave as once-through steam generators. They suffer from a small secondary water inventory, therefore the transients are sharper than with traditional boilers. They require a complex regulation system. However CSGs have some favorable safety features: the primary-to-secondary boundary – barrier to radioactive species – is much more robust. Indeed, the plates are diffusion bonded and resist high loads as well as corrosive environments [12]. There is no vibration, and the boundary is made of a unique material with very continuous properties. Failure within the steam generator is so unlikely that the steam generator tube failure and steam generator tube rupture accident (SGTR) could be eliminated *in practice*[5]. This would be a remarkable achievement.

Finally channel inspection is challenging but should remain doable in a compact steam generator by using eddy current inspection technique [13]. Eddy currents require a constant material thickness around the channels, associated with low radii of curvature. The use of this technique eliminates zigzag channels or complex flow paths. The reference design thus uses straight channels (see Section 4.4)

4.3. Vertical Layout

A major design choice for compact steam generator design is the orientation layout: vertical or horizontal. This choice has multiple consequences:

- *Margin to boiling crisis*: gravity degrades boiling crisis margin in the horizontal layout. As seen in Section 3, the horizontal arrangement induces a drop in the critical heat flux. In addition, there is more CHF experimental data for vertical channels. Albeit these data cover mostly conventional-size channels, there is still more knowledge with vertical channels than horizontal ones.
- *Pressure drop and natural circulation*: a vertical layout offers better opportunity for natural circulation flow in the channels for small size CSGs. If the hot fluid flows downwards, and the boiling secondary fluid flows upwards. The total pressure drop is reduced and, if designed properly, natural circulation can provide sufficient driving head to remove the decay heat of the reactor without the need for pumps.
- *Manufacturability and operational experience*: existing PCHE, such as those manufactured by Heatric, can accommodate both vertical and horizontal layout. The plates are nevertheless limited to 600 x 1500 mm by the manufacturing process [Heatric, 2014]. In the Heatric design, the plates are etched (or "printed") which produces semi-circular channels. Areva claims that a rectangular cross-sectional shape is also possible by machining and assembling twice as many plates [12]. The diffusion bonding process in the Areva case aims to produce and assemble plates of more than 2,500 mm height. Adopting vertical

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[5] Strictly speaking, SGTR is a *postulated* accident. Eliminating it from the list of possible accidents requires modification of the current regulation. In the absence of tubes, failure in the SGTR channels would have to be postulated. That being said, all the possible failure modes of the SG would need to be analyzed thoroughly for licensing purpose.

channels between 1.5 and 2.5m long is therefore doable for a CSG. On the operational side, vertical channels are easier to drain, which should accelerate cleaning by high-pressure water jetting.

All aspects drive the designer to select a vertical layout. It is the orientation adopted here. For the ease of calculation and promotion of natural circulation, a simple counter-current flow pattern is chosen. The channels of diffusion-bonded heat exchangers can be semi-circular (Heatric design) or rectangular (Areva design). Both configurations are studied here.

4.4. Inlet Conditions and Geometry

Compact steam generators are once-through steam generators (OTSG) and are fundamentally different from U-tube boilers in their mode of operation. In a U-tube boiler, steam quality is typically 20–25% when entering the separators region, and recirculation occurs in the SG shell. In a OTSG, secondary water boils off completely in its channel and exits in superheated state. Therefore, the outlet steam pressure does not set up the steam temperature as in a U-tube boiler. Steam and feedwater regulations require a more complex control system [22]. Typically, steam pressure is kept constant during operation (e.g. during load transients) and feedwater mass flow rate and temperature are adjusted to keep a minimum steam superheat of about 20 ºC. In this mode of operation, the primary-to-secondary differential temperature is relatively constant[6], which is different from a U-tube boiler-equipped PWR. SG power is instead controlled by the length of the nucleate boiling region, which is roughly proportional to the feedwater flow or load [22]. It covers 60% of the channel at 100% power. For the initial design investigation, a superheat of 20 ºC is chosen. The primary coolant is set to 15.5MPa and 325 ºC, which are classical primary coolant conditions at SG inlet. On the secondary side the pressure is set to 6 MPa (Tsat = 275.6 ºC). An inlet temperature of 215.6 ºC, corresponding to 60 ºC subcooling is selected. The mass flux is adjusted to obtain a superheat of 20 ºC at the outlet. These operating conditions are similar to those of the B&W plant designs, which use OTSGs (Table II).

Table II. Desired operating pressure and temperature of the initial CSG design at 100% power

	B&W plant [NRC, 2011]	**mPower [Generation mPower, 2014]**	**Initial CSG design**
Primary side			
Pressure	15.0 – 15.5 MPa	14.1 MPa	15.5 MPa
Inlet temperature	~ 316.8 ºC	~ 320 ºC	325 ºC
Average temperature = (Tout+Tin) /2	303.3 ºC	308.5 ºC	307.5 ºC
Outlet temperature	291.0 ºC	~ 297 ºC	290 ºC
Secondary side			
Pressure	**6.4 MPa**	**5.7 MPa**	**6.0 MPa**
Feedwater temperature	260 ºC	212 ºC	215.6 ºC
Average temperature = saturation temperature	279.8 ºC	272.3 ºC	275.6 ºC
Steam outlet temperature	298.9 ºC	300.3 ºC	295.6 ºC
Primary-to-secondary differential temperature	**23.5 ºC**	**36.2 ºC**	**31.9 ºC**
Feedwater subcooling	**19.8 ºC**	**60.3 ºC**	**60 ºC**
Outlet steam superheat	**19.1 ºC**	**28 ºC**	**20 ºC**

[6] In the range 15-100% power

The compact steam generator design has vertical, straight, channels. Two different cross-sectional shapes are considered (Figure 3). The first one is rectangular, of dimensions 4mm x 2mm and 1 mm metal thickness (Areva design). The second one is semi-circular, with 2mm diameter and 2.65mm pitch (Heatric single phase HX design). Hot and cold channels have the same dimensions, although in a future design they could have different sizes.
The CSG simulation tool is used. The correlations are referenced in Table III.

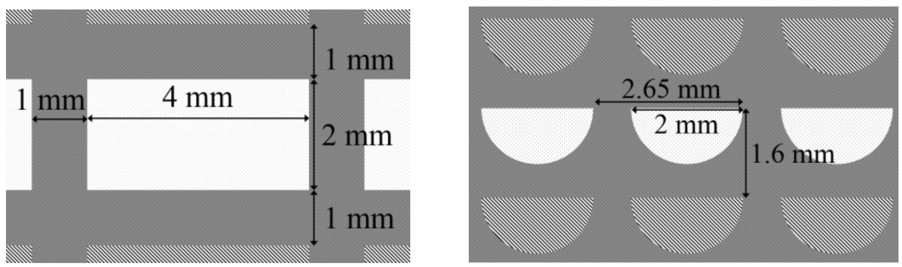

Figure 3. Two channel geometries are considered. The rectangular geometry has 2.66mm hydraulic diameter and 8 mm^2 of flow area. The semi-circular geometry has 1.2mm hydraulic diameter and 1.57 mm^2 flow area. Surface densities are 800 and 1,212 m^2/m^3 respectively.

Table III. Correlations used for the vertical straight channel CSG design. For the transition region between laminar and turbulent, a 5th order polynomial interpolation [24] **is used.**

	Correlation used		**Reference**
HEAT TRANSFER			
Single-phase	Laminar (Re<2000)	Shah & London (rectangular) or Nu = 4.089 (semi-circular)	[24]
	Turbulent (Re>5000)	Colburn	[24]
Two phase	Max(nuc. boiling htc, conv. boiling htc) (rectangular)		[25] [3]
	Nucleate boiling	Cooper	[25]
	Convective boiling	Taylor (dependence on Lockhart-Martinelli parameter)	[25]
Critical heat flux		Groeneveld CHF look-up table with Tanase diameter correction	[18] [5]
Post-CHF		Groeneveld film boiling htc look-up table	[26]
VOID FRACTION		CISE	[20]
PRESSURE DROP			
Single-phase	Laminar (Re<2000)	Shah & London (rectangular) or f = 63.3/Re (semi-circular)	[24] [27]]
	Turbulent (Re>5000)	Blasius	[24]
Two-phase	Two-phase flow multiplier with C recommended by Chisholm and Lockhart-Martinelli parameter calculated with Taylor recommendations		[25] [3]

4.5. Optimization

The goal of the optimization process is to achieve the maximum power density in order to reduce the cost of the CSG. The design is nevertheless constrained in several directions:

- The pressure drop must be reasonable, in the range of 50-100 kPa.
- The flow velocity should not be too low to avoid deposition of impurities in the channels.
- The channel length should not exceed 2 meters, in order for the CSG to fit in an integral PWR.

- The specified outlet temperatures must be met. In particular, steam has to exit in a superheated state, namely with 20ºC superheat (see Table II).

The operating pressure on the secondary side is maintained at 6MPa. Going to 7MPa would bring a marginal increase in cycle efficiency (0.4%), while it significantly increases the surface area needed to achieve the same power rating (40-50% increase) because of the reduction of the primary-to-secondary temperature difference.

Increasing the mass flow rates for a given geometry and inlet conditions results in larger power output, because it creates more turbulence and extends the subcooled boiling region to a longer portion of the boiling length (on the secondary side). However, it reduces the superheat and creates larger pressure losses.

The optimization procedure is as follows:

1) Define the channel geometry (cross sectional areas and metal thickness)
2) State the desired inlet and outlet thermodynamic conditions (temperatures and pressures)
3) Derive the associated enthalpy change and the mass flux relationship

$$\frac{G_{prim}}{G_{sec}} = \frac{\dot{Q}/(\Delta h_{prim} A_{prim})}{\dot{Q}/(\Delta h_{sec} A_{sec})} = \frac{\Delta h_{sec}}{\Delta h_{prim}} \quad (2)$$

4) Guess the channel length
5) Take the maximum mass flux that does not cause excessive pressure drop. Usually the primary mass flux is more limiting.
6) Compute the outlet conditions with the CSG simulation tool. Keep good accuracy (1%).
7) If the outlet conditions do not match the desired ones, return to step 4 and change the channel length
8) Adjust the number of channels to match the power requirement of the heat exchanger. In the absence of extensive experience in CSG,, take a margin for channel blockage and uncertainties.

4.6. Result

The optimized designs for both the rectangular and semi-circular channel geometries are defined in Table IV[7]. The enthalpy change is the same for both designs, and the pressure drops are nearly identical[8]. The number of channels was adjusted to give the same power rating of 81.25 MW. Both designs fit in the RPV of an integral SMR. The semi-circular design results in 30% less volume, and is about twice shorter than the rectangular design. However, during the calculation it is noticed that the shorter the heated length, the more sensitive is the outlet temperature to the inlet mass flux. For instance in design #2, a 10% increase in mass flux results in 13 ºC drop of superheat. This suggests that a short CSG would react rapidly to disturbances in feedwater flow, and would consequently be more complicated to operate.

The heat flux in both designs is comparable, therefore the difference in power density comes from the different densities of heat transfer area.

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[7] The volume does not include the headers, nor margin for design uncertainties. The total CSG volume including margins could be up to 30-50% larger than the volume reported in the table, e.g. the volume of the heat transfer region.

[8] Keep in mind that the uncertainty on the pressure drop calculation is large (cf. section 2.2)

Table IV. Design parameters of two Compact Steam Generators: The semi-circular channel design is more compact than the rectangular one but may be complicated to operate due to its small thermal inertia.

	Design #1: rectangular channels	**Design #2: semi-circular channels**
Number of channel pairs	**26,016**	**147,165**
Primary side		
Mass flux	2,000 kg/ m^2-s	1,750 kg/ m^2-s
Reynolds number at inlet	68,047	27,285
Inlet temperature	325.0 ºC	325.0 ºC
Outlet temperature	290.9 ºC	289.8 ºC
Average heat transfer coefficient	1,446 W/ m^2-K	3,621 W/ m^2-K
Inlet pressure	15.5 MPa	15.5 MPa
Pressure drop	55 kPa	39 kPa
Secondary side		
Mass flux	200 kg/ m^2-s	180 kg/ m^2-s
Reynolds number at inlet	4,263	1,758
Total mass flow rate	41.62 kg/s	41.62 kg/s
Inlet temperature	215.6 ºC	215.6 ºC
Outlet temperature	297.5 ºC	297.7 ºC
Superheat	22.0 ºC	22.2 ºC
Average heat transfer coefficient	668 W/ m^2-K	1,256 W/ m^2-K
Inlet pressure	6.0 MPa	6.0 MPa
Pressure drop	8 kPa	6 kPa
Power	**81.25 MWth**	**81.25 MWth**
Channel heated length	**1.15 m**	**0.5 m**
Volume L*Pitch*2Th*$N_{channel}$	**0.897 m^3 (0.6 x 1.3 x 1.15 m)**	**0.624 m^3 (0.6 x 2.08 x 0.5 m)**
Average heat flux in channels	226.3 kW/ m^2	214.8 kW/ m^2
Power density	**90.53 MW/ m^3**	**130.2 MW/ m^3**

5. COMPARISON WITH OTHER STEAM GENERATOR TYPES

5.1. Recirculation Steam Generator

The computed power density of the CSGs is 90 to 130 MW/m^3. By comparison, in a 900-MWe French PWR (3 boiler-type SGs), the 3361 steam generator tubes represent a heat exchange surface area of 4751 m^2 for 928.3 MWth. The tube plate has 3.454 m diameter and the mean tube length is 10.25 m. The SG volume taken by the shell and tubes would be 141 m^3 . This gives a power density of 6.58 MW/m^3. Therefore, by assuming comparable thermodynamic conditions, design margins, and avoiding the separators and dryers, the reference compact steam generator is 13 (rectangular channel) to 20 times (semi-circular) more compact. It generates superheated instead of saturated steam, which, depending on the temperature, yields 0.3 to 0.5% increase in theoretical efficiency. Together with reducing the materials needed for manufacturing, the compact steam generators seem to be economically advantageous.

5.2. Helical-Coil Once-Through Steam Generator

The helical-coil SG of the IRIS reactor has been successfully tested in the past [28]. It is a once-trough steam generator. The reference design has a rated power of 125 MW and fits in a volume of for 55 m^3 (headers). This gives a power density of 2.2 MW/ m^3, which is more than 40 times lower than the compact steam generator designed in this study.

5.3. Steam Drum

It was seen in section I that Westinghouse in its integral PWR designs uses steam drums to generate steam in lieu of direct steam generation in the compact heat exchanger. To evaluate this design choice, let's consider two steam conversion cycles coupled to a given pressurized water reactor. Cycle 1 produces superheated steam from a compact steam generator, whereas cycle 2 produces saturated steam from a steam drum fed by a single-phase HX (see 4). The pressure at the turbine inlet (8.4 MPa) and at the turbine outlet (condenser, 8.6 kPa) are assumed the same in both cases.

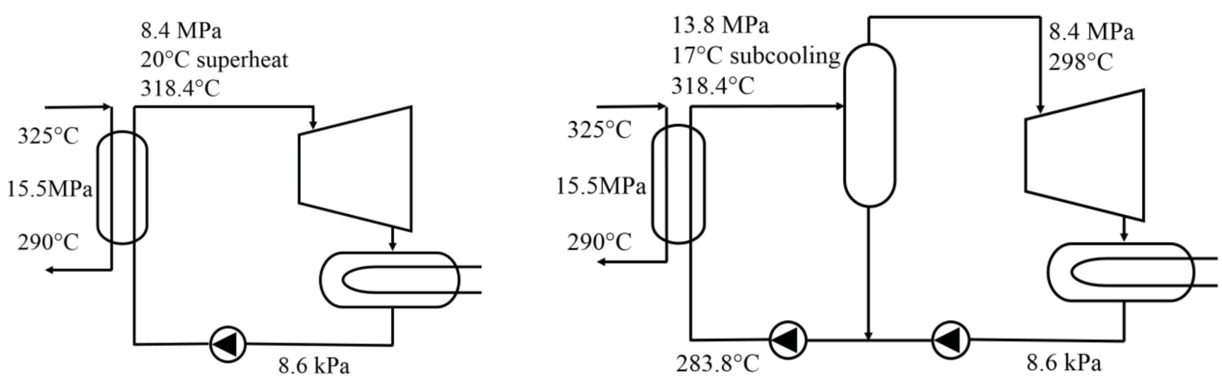

Figure 4. Steam conversion cycle 1 (left) with compact steam generator, and cycle 2 (right) with steam drum. Turbines and condensers are nearly identical.

For the first cycle a conservative superheat of 20ºC, corresponding to an absolute temperature of 318.4ºC is chosen. In the second cycle, the liquid flow exiting the heat exchanger and entering the steam drum is set to 13.8 MPa and 318.8ºC, corresponding to 17ºC subcooling. This value was taken from the Westinghouse compact steam generator design [9] and maximizes the enthalpy of the fluid entering the steam drum. The efficiencies of both cycles are calculated assuming 85% and 91% pump and turbine efficiency respectively. Steam is assumed to flash adiabatically in the drum [9]. The results are shown in Table V. The compact steam generation case yields a thermodynamic efficiency 2.1% higher in absolute value. This can be explained by the fact that the steam quality of the mixture generated in the steam drum is low: only 8.0%. Consequently the feedwater mass flow rate is 14 times higher to achieve the same turbine work. The larger pumping power harms dramatically the efficiency of cycle 2.
The diameter of the flashing drum is governed by the maximum velocity of the vapor [29, 7]:

$$u_{g,max} = K\sqrt{\frac{\rho_f-\rho_g}{\rho_g}} \quad , \text{ with } \quad K = 0.35 - 0.01\frac{P_{drum}-100psi}{100psi} \qquad \text{(3) and (4)}$$

The height of the flashing drum is given by empirical formulas documented by Wankat [29]. The steam drum is specified to drain in one minute in case feedwater was lost. For a 650 MWth reactor equipped with 8 steam drums, the calculation gives steam drums of 1.83 m diameter and 10.10 m height. This is a volume of 26.56 m^3. The HX volume is calculated using the newly-created CSG design code to match the required power rating: 650/8= 81.25 MWth. Inlet conditions on the primary side are set to 15.5 MPa, G= 1250 kg/m^2-s, T= 330 ºC. The same correlations as the reference case are used. The vertical, straight, semi-circular channels were assumed for the HX.
The results are shown in Table V. The single-phase heat exchanger is nearly twice as big as the two-phase one. This is mainly explained by the fact that the average temperatures are very different. In the single-phase case, the secondary side average temperature is 300.7 ºC whereas in the two-phase case it lies somewhere between 181.5 ºC and the saturation temperature 298.4 ºC.

Table V. The in-vessel single-phase heat exchanger requires twice as much volume as the single step steam-generator.

	Cycle 1: 2-phase HX	**Cycle 2: 1-phase HX**
Number of channels	**148,894**	**189,385**
Primary side		
Mass flux	1,750 kg/m^2-s	1,400 kg/m^2-s
Inlet temperature	325.0 ºC	330.0 ºC
Outlet temperature	290.1 ºC	290.9 ºC
Inlet pressure	15.5 MPa	15.5 MPa
Pressure drop	89 kPa	87 kPa
Secondary side	(2-phase)	(1-phase)
Mass flux	130 kg/m^2-s	1,450 kg/m^2 -s
Total mass flow rate	30.4 kg/s	431.4 kg/s
Inlet temperature	43.2 ºC (no preheater)	283.8 ºC
Outlet temperature	319.8 ºC	317.6 ºC
Inlet pressure	8.4 MPa	13.8 MPa
Pressure drop	4 kPa	62 kPa
Power	**81.25 MWth**	**81.25 MWth**
Channel heated length	**1.0 m**	**1.5 m**
Volume L*Pitch*2Th*Nchannel	**1.263 m^3**	**2.409 m^3**
Power density	64.35 MW/ m^3	33.73 MW/ m^3

As a result, the simplified evaluation of the steam drum option shows that there is a significant loss in efficiency (2.1 % absolute) and also a loss in total compactness compared to the compact steam generator option. The volume of a steam drum is comparable to the size of an integral reactor pressure vessel, and the heat exchanger itself is twice larger (Table VI). In other words, there is a strong incentive to pursue a two-phase compact heat exchanger design for nuclear application. The expected benefits in size and efficiency justify further development to overcome the operational risk raised by boiling in mini-channels, and the reduction in thermal inertia of the steam generator.

Table VI. The single-phase heat exchanger requires more volume plus a bulky steam drum for steam generation.

	Cycle 1: compact steam generation / 2-phase HX	**Cycle 2: steam drum / 1-phase HX**
Thermal power	650 MWth	650 MWth
Steam generated	Superheated (8.6MPa, 20 ºC superheat)	Saturated (8.6 MPa)
Total feedwater flow rate	0.37 kg/s per kWth	5.3 kg/s per kWth
Cycle efficiency	**33.9 %**	**30.8 %**
Net electrical power	220 MWe	200 MWe
# of HXs	8 (2-phase flow)	8 (1-phase flow)
HX unit volume	**1.263 m^3 (0.6 x 2.105 x 1.0 m)**	**2.409 m^3 (0.6 x 2.677 x 1.5 m)**
# of steam drums	0	8
Steam drum unit volume	**-**	**26.56 m^3**

6. CONCLUSIONS

The viability of compact steam generators was investigated with numerical simulation. Existing diffusion-bonded designs – from Areva and Heatric – were taken as basis to assess the optimal geometry and configurations of a compact heat exchanger for nuclear steam generation. The calculations confirm the low reliability of the CHF and pressure drop models for two-phase flow in mini-channel. Experimentation is needed to predict the performance of a commercial compact steam generator with acceptable accuracy. The uncertainty was reduced by avoiding the horizontal and zigzag channel configuration in favor of straight, vertical channels for the reference case.

The optimization of the designs yields steam generators with power densities of 90 and 130 MW/m^3 with rectangular and smaller semi-circular channel geometries respectively. If operational challenges (such as channel blocking by impurities) are successfully addressed and thermal performance confirmed, such CSG would show major benefits with respect to classical boiler steam generators, such as a reduction in size by a factor of 13 to 20, provision of superheated steam, and elimination of SG tube rupture accidents. The study also shows that the external steam drum option coupled to single-phase HX for an integral LWR steam conversion cycle yields poorer efficiency and larger volume requirements.

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