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ENGINEERING SERVICES FOR THE NEXT GENERATION NUCLEAR PLANT (NGNP) WITH HYDROGEN PRODUCTION

Power Conversion System Alternatives and Selection Study

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EXECUTIVE SUMMARY

Pre-conceptual design studies [PCDSR 2007] for the Next Generation Nuclear Plant (NGNP) carried out by an industrial team led by General Atomics (GA team) recommended a reference Power Conversion System (PCS) for the NGNP and an alternative PCS. The reference PCS was a recuperated and intercooled closed Brayton cycle system located in the primary coolant system (direct Brayton cycle PCS). The alternative PCS was a combined-cycle system (gas turbine plus Rankine steam cycle) with the gas turbine and steam generator (SG) located in the primary coolant system (direct combined-cycle PCS).

Both of the PCS alternatives recommended by the GA team were concluded by the NGNP Project to entail significant risks associated with the lack of operational and maintenance experience for gas turbines in a gas reactor primary coolant loop, and [INL 2008] specifies that the NGNP must have an indirect PCS. Further, the NGNP Preconceptual Design Report [INL 2007] includes a requirement that the NGNP PCS be capable of producing steam for potential process heat applications. In response to these requirements, the GA team performed two follow-on conceptual design studies as part of the FY08-1 conceptual design studies. In one study [GA 2008a], the GA team evaluated various heat transport system alternatives in which the thermal energy produced in the reactor is transferred to the PCS in a secondary loop via an intermediate heat exchanger (IHX). In the second study, the GA team evaluated the pros and cons of locating the SG in the primary loop vs. a secondary loop [GA 2008b]. The result of these studies was that GA recommended an NGNP configuration having the SG in the primary loop as the reference concept to be developed during NGNP conceptual design.

Although the NGNP Project considers a gas turbine located in the primary circuit to be too risky for the NGNP, the Project recognizes that the capability of a direct combined PCS to co-generate steam for process heat applications and electricity at an efficiency approaching 50% is attractive for a commercial cogeneration plant. The Project also recognizes that an indirect combined cycle could be attractive for the same reasons, and could be a candidate for the NGNP PCS. Consequently, a comparison of direct and indirect combined cycle options for a commercial plant has been included in the scope of a PCS alternatives and selection study that the GA team has been tasked to perform as part of the FY08-2 NGNP conceptual design studies. This study is the subject of this report. The objectives of this study are to:

- Provide a recommendation for the configuration of the power conversion system (PCS) for the NGNP and the justification for this recommendation
- Provide estimates of the performance, cost, and technology readiness of the PCS configuration recommended for NGNP

- Perform a comparison of direct and indirect combined-cycle PCS options for a commercial cogeneration plant and evaluate the feasibility of using a combined-cycle PCS in an indirect heat transport configuration in the NGNP.
- Identify configurations of the PCS that should be considered for commercial applications including electric power production, cogeneration, and hydrogen production.

Power Conversion System Alternatives for NGNP

In [GA 2008b], the conclusion was reached that the PCS configuration selected for the NGNP to produce steam for potential process heat applications should have the steam generator (SG) in the primary loop. Accordingly, at the onset of the present PCS study, the configuration with the SG in a primary loop shown in Figure E-1 was selected for more detailed evaluation. An alternative PCS configuration shown in Figure E-2 having an IHX in the primary loop with the SG in a secondary loop was also included in the PCS study based on the results of the FY08-1 heat transport loop alternatives study [GA 2008a]. Based on cost evaluations in [GA 2008b], dual-loop variants of both alternatives were included in the current PCS study because dual loops were evaluated not to entail large cost penalties.

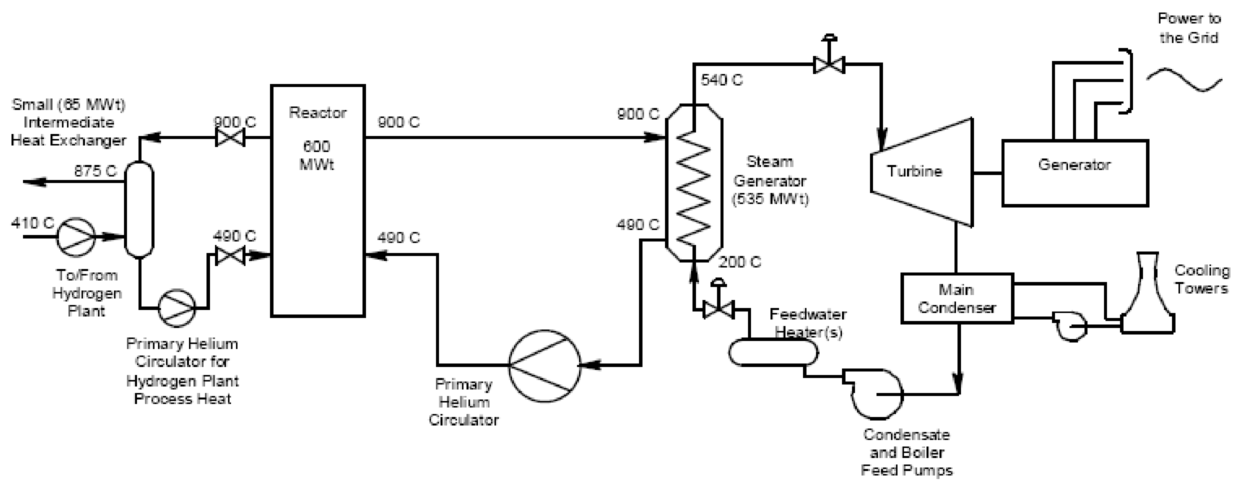

Figure E-1. Steam generator in primary loop

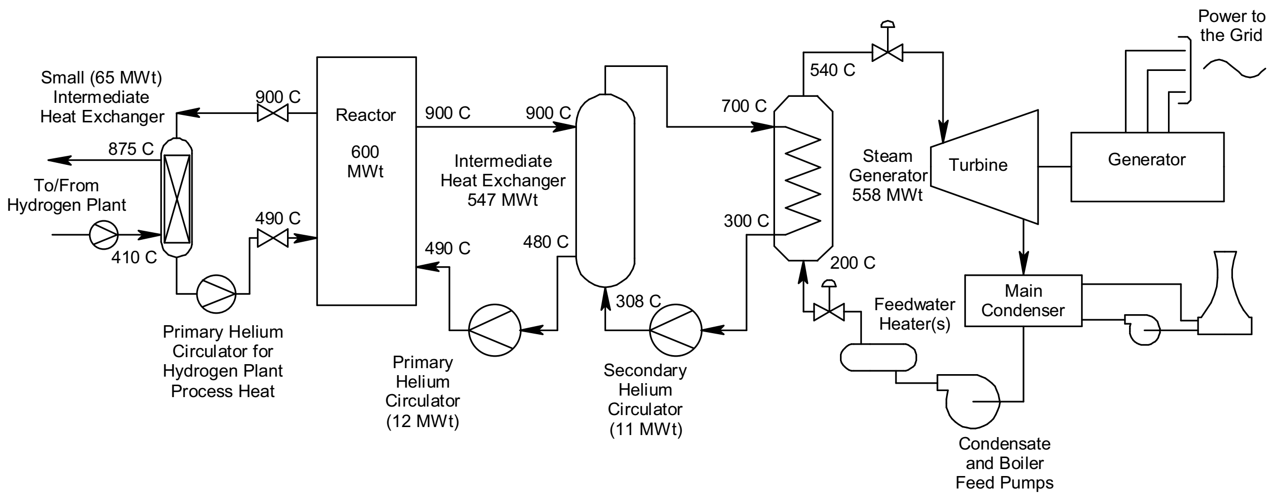

Figure E-2. Steam generator in secondary loop

Following initiation of the current PCS alternatives study, an agreement was reached in a meeting of NGNP project participants in October 2008 [DOE 2008] to reduce the reactor core outlet gas temperature objective for NGNP into the range of 750°C to 800°C. On the basis of this agreement, a core outlet gas temperature of 750°C was chosen for the current PCS study to enable use of much of the existing high temperature gas reactor (HTGR) design and technology from previous U.S. HTGR programs (e.g., MHTGR, NPR, Large HTGR, Fort Saint Vrain, Peach Bottom). Also, the parallel primary loop shown in Figures E-1 and E-2 to supply high-temperature process heat to an engineering-scale hydrogen production plant (or plants) was deleted from GA's reference NGNP plant configuration because neither of the advanced hydrogen production processes under consideration (e.g., high-temperature electrolysis or the S-I thermochemical water splitting process) are considered economical if heat is provided to the process at less than 750°C.

The resultant reference PCS configuration is as shown in Figure E-3 and the alternative PCS configuration is shown in Figure E-4. In these figures, a partitioning is shown that divides the PCS into a NHSS side and a BOP side. For both of these configurations, dual-loop variants on the NHSS side were considered in the evaluation of PCS alternatives.

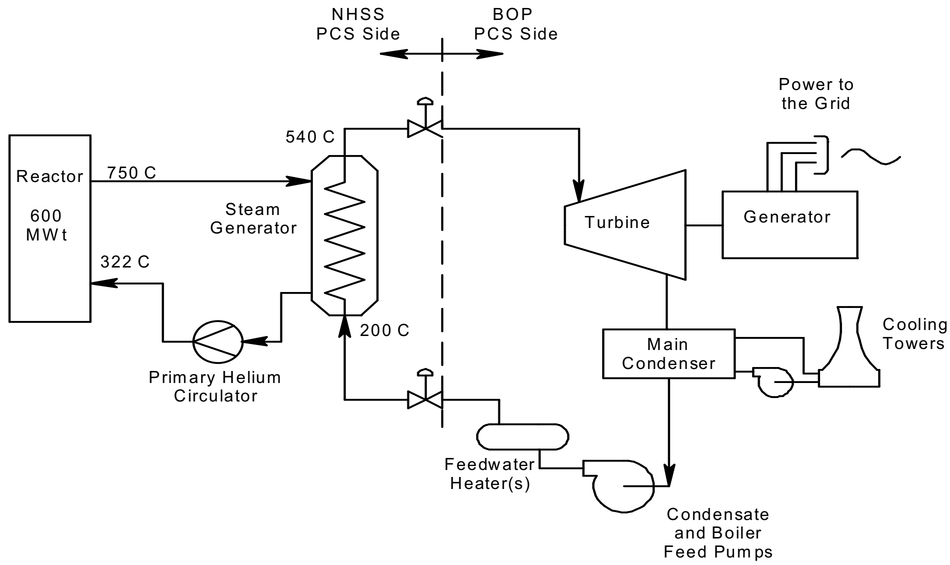

Figure E-3. Reference PCS configuration with steam generator in primary loop

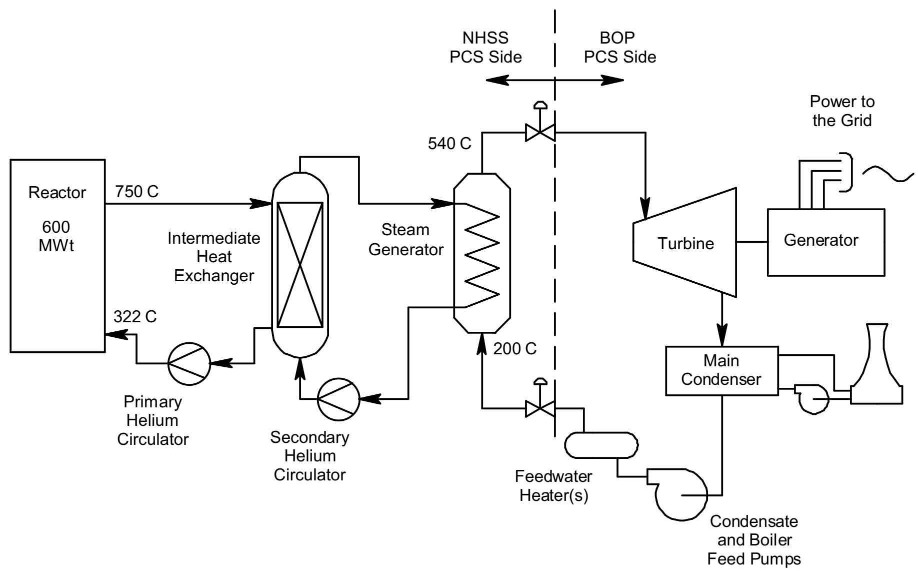

Figure E-4. Alternative PCS configuration with steam generator in secondary loop

For assessing performance, design, and cost characteristics of the PCS alternatives, SG concepts, IHX concepts, and heat balance diagrams were developed for each of the alternatives. Capital cost estimates were also prepared for plants employing each of the PCS alternatives.

Steam Generator Designs

The design of the SGs for each of the PCS alternatives was based on the SG design concept developed for the MHTGR [CDSR 1987]. This design concept was used for the NPR conceptual design and benefits from technology and operating experience obtained from similar SGs used in the Fort Saint Vrain plant. The MHTGR design arrangement is based on housing the steam generator in a steam generator vessel, with its thermal center located below that of the reactor core as shown in Figure E-5.

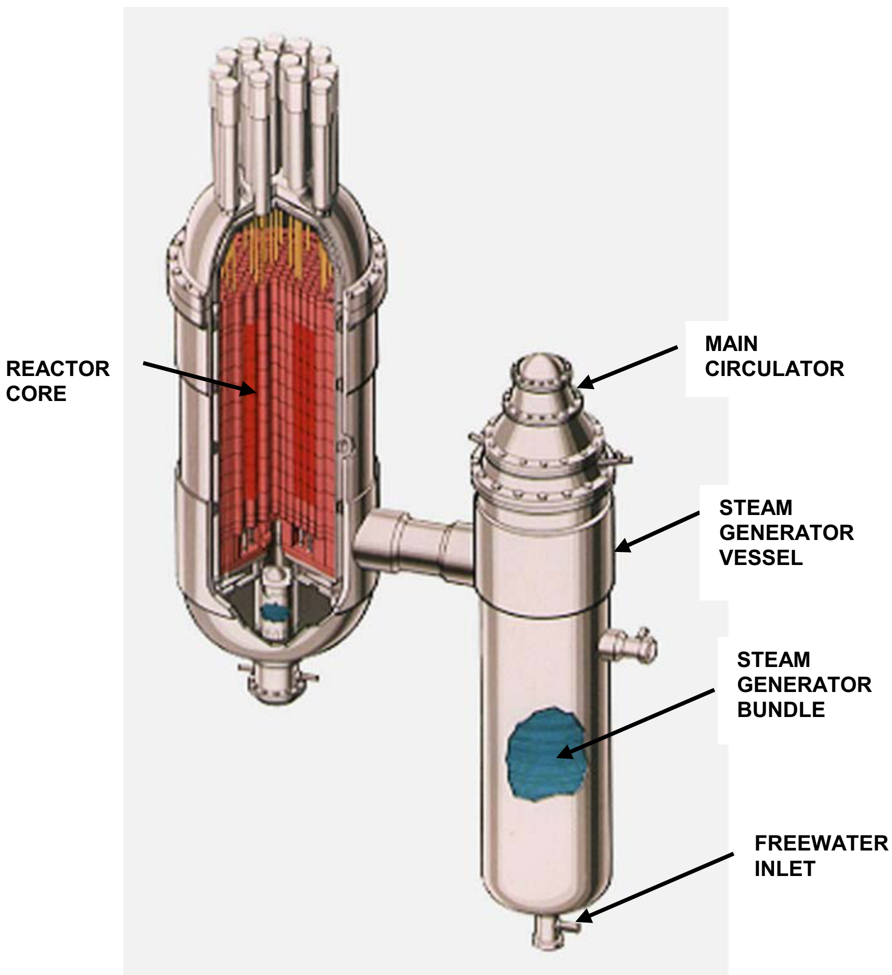

Figure E-5. MHTGR steam generator conceptual design arrangement

The MHTGR design arrangement is directly applicable for the PCS alternatives with the steam generator in the primary loop. For the indirect cycle alternatives, the IHX should also be located below the thermal center of the reactor core analogous to the MHTGR arrangement, but alternative steam generator designs and arrangements could be used. However, for sizing the steam generators for the current evaluation of alternatives, the same steam generator design concept was assumed for both the direct and indirect cycle alternatives.

The following conclusions resulted from the SG design analyzes performed for each of the configuration alternatives:

- With regard to overall size and weight considerations, all of the steam generators for all of the configuration alternatives were found to have sizes and weights that fell within acceptable ranges used in past gas reactor programs for manufacturing, handling, transportation and installation
- The direct single loop requires the least steam generator space and construction material. Dual direct loops would require ~33% more steam generator space and require ~32% more construction material by weight.
- The alternative that requires the most steam generator space and construction material is the indirect dual-loop configuration. The indirect dual loop requires ~100% more space and ~73% more construction material than the direct single loop configuration for only the steam generator portion of the PCS.

Intermediate Heat Exchanger Designs

Sizing calculations carried out for the indirect loop IHXs resulted in the conclusion that the size required (9 m diameter pressure vessel) for an IHX for a single indirect loop PCS with the selected core outlet and steam generation conditions exceeds current nuclear grade pressure vessel manufacturing capabilities. Multiple IHXs could be used in the single indirect loop but the simplest approach would be to have multiple (e.g., dual) indirect loops.

Heat Balance Diagrams

Heat balance diagrams were developed for each of the PCS alternatives using a simplified systems analysis model for coupling the components in the primary loop(s) (the reactor and heat exchangers – either SG or IHX) and secondary loop(s) (IHX and SG), as applicable. The circulator power requirements from the heat balance diagrams are summarized in Table E-1.

Table E-1. Circulator Power Requirements

NHSS PCS Alternative	Primary Circulator Power MW(t)*	Secondary Circulator Power MW(t)*
Single Direct Loop	6.5	---
Dual Direct Loop	3.1	---
Single Indirect Loop	7.5	2.0
Dual Indirect Loop	4.0	1.0
* As shown in this table, MW(t) is the thermal energy added to the heat transport system by the helium circulator. The MW(e) rating of the helium circulator is 10% to 20% higher.		

An assessment of the current state of helium circulator technology contained in [GA 2008a] indicates that:

- The technology required to produce high-temperature helium circulators is well understood and relatively available for circulators of up to about 5 MW(e)
- A credible vendor (Howden) confirmed that circulators of about 6 MW(e) are currently considered to be viable. This includes circulators featuring the preferred bearing option, Active Magnetic Bearings (AMBs).
- Higher powered circulators are feasible but will require more development. Development costs are expected to increase rapidly as the machine size approaches 10 MW(e). The largest practical size for the helium circulator is around 15 MW(e).

Based on this assessment, all of the circulators in Table E-1 are potentially viable, but the most practical approach would be to limit the circulator power to ≤ 6 MW(e). In this approach, the dual-loop options should be considered, or two circulators operating in parallel should be used for the single loop alternatives.

Plant Capital Cost Estimates

To evaluate the cost significance of the different types, quantities, and sizes of components required by the NHSS PCS alternatives, full-scope plant capital cost estimates were prepared for each of the PCS alternatives for both an electric-only generation plant and a cogeneration plant. "Full scope," as used here means all of the direct and indirect capital costs per the Energy Economic Data Base (EEDB) code of accounts [ORNL 1993] for all Systems, Structures and Equipment for complete plants (both NHSS and BOP plant sides).

Table E-2 contains a comparison of the relative total plant capital costs. The relevant conclusions are:

- The plant having the least plant capital cost is the direct NHSS PCS single loop cogeneration plant
- The dual-loop PCS configuration increases plant capital cost by ~6%
- Indirect PCS configurations increase plant capital costs 16% to 20%

Table E-2. Comparison of Relative Plant Capital Costs

NHSS PCS Alternative	One Loop Electric Plant	Two Loop Electric Plant	One Loop Co-Gen Plant	Two Loop Co-Gen Plant
Direct	1.09	1.14	1.00	1.06
Indirect	1.25	1.34	1.16	1.25

Prior economic analyses [GCRA 1993] indicate that a NOAK 4x600 MW(t) high temperature gas reactor electric generation plant with the direct NHSS PCS alternative should be economically competitive, although the economic advantage may not be great without credit for passivity or carbon emissions. Presuming the direct NHSS PCS electric generation plant is competitive, a cogeneration plant variant should be equally competitive. However, if the plant capital cost is 25% higher, as in the case of the plant based on use of the indirect PCS configuration, the economic viability would be questionable.

Recommended NHSS PCS Configuration

The direct NHSS PCS configuration shown in Figure E-3, either in a single loop configuration (the preferred configuration) or in a dual-loop configuration, is the recommended configuration based on the performance, design, and cost evaluations performed. For the preferred single loop configuration, a parallel design and development path is recommended for providing the required circulator capacity. The dual path would include development of a single circulator of the required power capacity in parallel with development of a design that uses two circulators in parallel.

BOP PCS Configuration

There is a great variation in character of the energy mix required by cogeneration users. A large number of BOP configurations have been developed over the years for industrial, commercial, and district heating applications. Consequentially, the BOP PCS design is impacted significantly more than the design of the NHSS by the process conditions needed for a specific application.

Because of the large variation in BOP PCS configurations applicable to cogeneration, the range of configurations can be organized into four general categories. Three of the alternatives, identified as Alternates A, B, & C are conceptualized to provide process steam. Alternative A, the reference case is illustrated in Figure E-6. In the fourth alternative (Alternate D), the nuclear steam generator provides direct process heat to a particular liquid/gas process stream and steam for the production of electricity to be used in the process or sold it to the grid.

The specific configuration chosen for the generation of steam and electricity is dependent on both the required steam conditions and the amount of electricity to be generated. Therefore, the first three alternatives are based on a generalized separation of possible steam conditions into three groups. These are:

- Medium-pressure process steam - Alternative A
- High-pressure process steam - Alternative B
- Low pressure process steam - Alternative C

In Alterative A, medium-pressure process steam is extracted from [a] intermediate stages of the turbine-generator or [b] between different turbine generators. As shown in Figure E-6, the high-pressure steam is used to make electricity, as well as medium-pressure process steam. In this case, the NHSS could include an IHX followed by the high-pressure steam generator; or any configuration in which there is no IHX, but a secondary reboiler downstream of the turbine generator extraction point. The electricity produced would be used by the plant and extra power could be sold to the grid.

In Alternative B, the high-pressure process steam flow may be extracted from the high-pressure steam turbine discharge as in the reference case or from the steam line between the steam generator and the inlet to the high-pressure stage of the turbine generator. The electricity produced could be used by the plant and extra power could be sold to the grid.

In Alternative C, low-pressure process steam is extracted either from the low pressure stages of the turbine generator or at the turbine generator outlet. In this configuration a steam turbine generator would be included to produce electric power using the high-pressure steam before the low pressure stream was extracted, sent to a reboiler to generate steam for sending to the process plant. The electricity produced would be used by the plant and extra power could be sold to the grid.

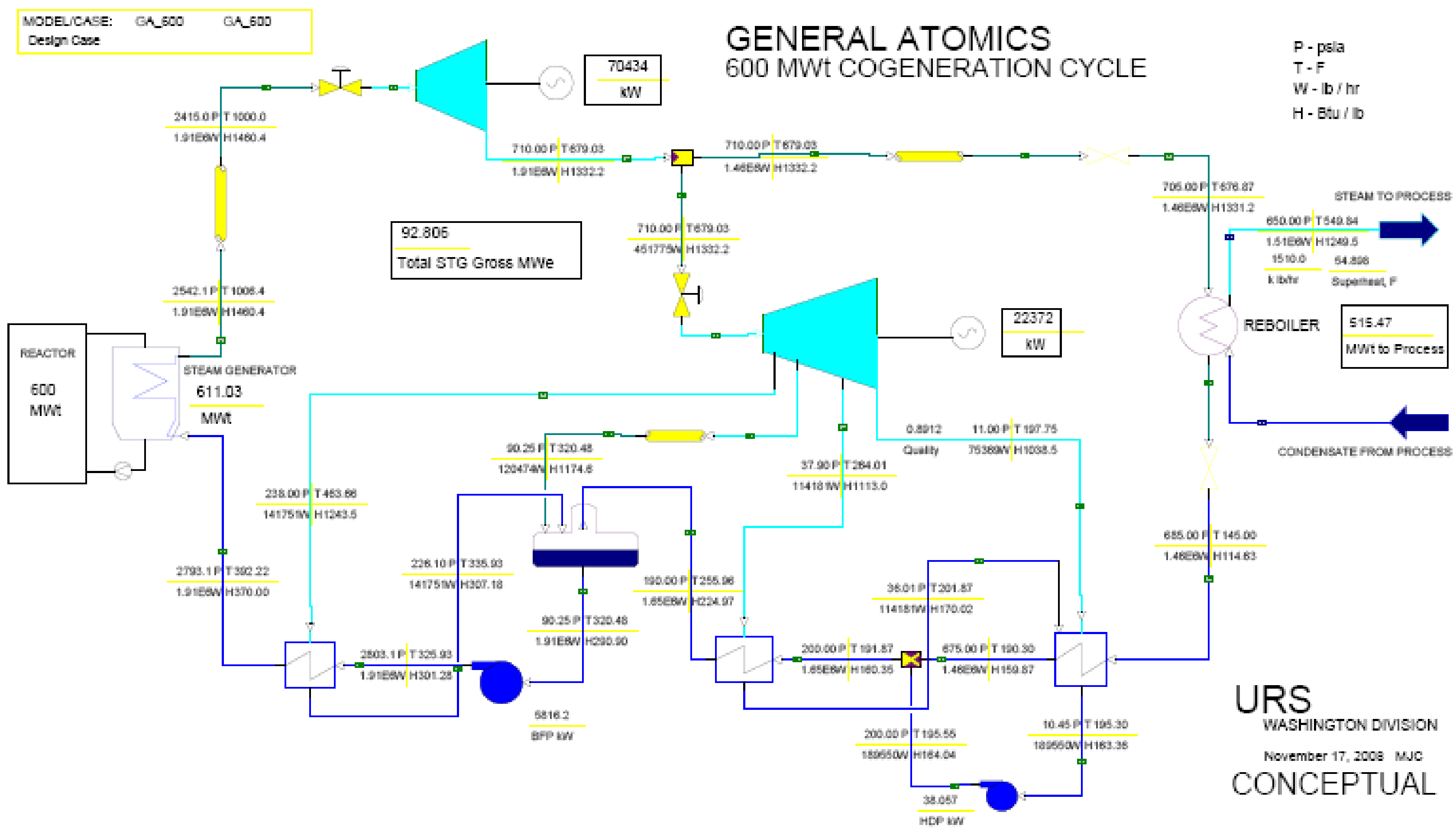

Figure E-6. COG-MHR process diagram

In Alternative D, the high temperature heat generated by the reactor is used to both directly heat a process flow stream from the process plant; and to generate high temperature steam to be used in making electricity and medium or low pressure process steam that will be used elsewhere in the process plant. In this case, the NHSS could include an IXH followed by a custom heat exchanger. This heat exchanger could have two sections. The first section would heat the process fluid/gas stream before it returns to the process plant. This second section would produce high-pressure steam for use by the turbine generator and subsequently as low pressure process steam.

Performance and Capital Cost Variations

A qualitative comparison of the capital cost variations for the alternatives is provided in Table E-3.

Table E-3. Qualitative Comparison of Overnight Capital Costs for Various BOP PCS Alternatives

Alternative	Installed Cost Change Relative to Reference Plant	Comments
Alternative A [Reference]	Baseline	
Alternative B	Small Increase	Smaller turbine generator but larger reboiler plus high-pressure steam transmission pipeline
Alternative C	Moderate Decrease	Larger turbine generator bur smaller reboiler
Alternative D	Small Decrease	Smaller turbine generator but much more complex reboiler
NOTES: Analysis assumes no change in the NHSS and for reboiler located after steam turbine island		

BOP PCS Configuration Recommendation

The PCS BOP cogeneration plant configuration selection can be made to accommodate the process mix need of various end users. An important objective in this regard for the NGNP program is to show that the HTGR NHSS modules can be sited and licensed at the end user facility.

Indirect Combined Cycle PCS

As noted above, the NGNP Project placed two important constraints on the NGNP PCS following the preconceptual design phase. One was that the NGNP must have an indirect cycle to reduce risk. The second was that the NGNP PCS must be capable of producing steam. These constraints rule out a pure gas turbine cycle, but a combined cycle (which Rolls-Royce proposed during the preconceptual design studies) is still an option. In the current PCS study, Rolls-Royce was tasked with further investigating combined cycle PCS options. Rolls-Royce's work was broken down into the following three main elements:

- Evaluate an indirect combined cycle PCS for a commercial plant
- Compare this with a direct cycle PCS, and make a recommendation on the best option for a commercial plant
- Determine how the technology for the recommended commercial plant option would best be demonstrated in the NGNP

Rolls-Royce study of an indirect combined cycle concluded that there are very significant reductions in risk/cost for this arrangement compared to a direct combined cycle:

- Conventional bearings can be used – no Active Electromagnetic Bearing risk.
- Working fluid in gas turbine can be more "air-like" – reduced risks of the unknowns associated with Helium aerodynamics and leaks.
- Compressor looks like a conventional aeroengine compressor – only 6 stages compared with 18 for Helium direct combined cycle. Turbine is only 2 stage compared with 5 for Helium direct combined cycle. This would allow large cost savings in the turbomachinery compared with a direct combined cycle.
- Gearbox can be used instead of power electronics for gas turbine generator providing a significant cost saving.
- Turbine blade cooling is not likely to be required, even with a 950°C reactor outlet temperature.
- Turbomachinery maintenance work will be much easier and cheaper with indirect cycle. Turbomachinery does not get contaminated with radioactivity. There is an opportunity to replace turbine blades that are creep life – expired when the reactor is refueled every 18 months.

Balanced against these significant benefits in cost and risk for an indirect combined cycle, a large IHX would be needed, which brings with it large risks and costs. Rolls-Royce compared the costs and risks of direct and indirect combined cycles and concluded that, although the direct cycle is predicted to be more efficient (50.2% versus 48.6% for the indirect at 850°C reactor outlet temperature), an indirect combined cycle would be a less-expensive and lower-risk option for a commercial electricity plant.

Furthermore, in comparing the indirect combined cycle plant with a pure steam cycle, the addition of an IHX and a gas turbine provides excellent value by increasing the cycle efficiency from 42.6% to 48.6%, which is a large benefit in terms of both plant capacity and fuel/waste processing costs. However, the combined-cycle efficiency of 48.6% is obtained at a much higher reactor outlet temperature than is the steam-cycle efficiency of 42.6%, and this has significant implications on the cost and risks associated with the reactor itself. It is therefore important to understand the relationship between reactor outlet temperature and cycle efficiency so that the point at which it becomes economically worthwhile to add a gas turbine can be judged. The relationship between net electrical efficiency and reactor outlet temperature is given in Figure E-7. In all cases the steam inlet temperature is 580°C.

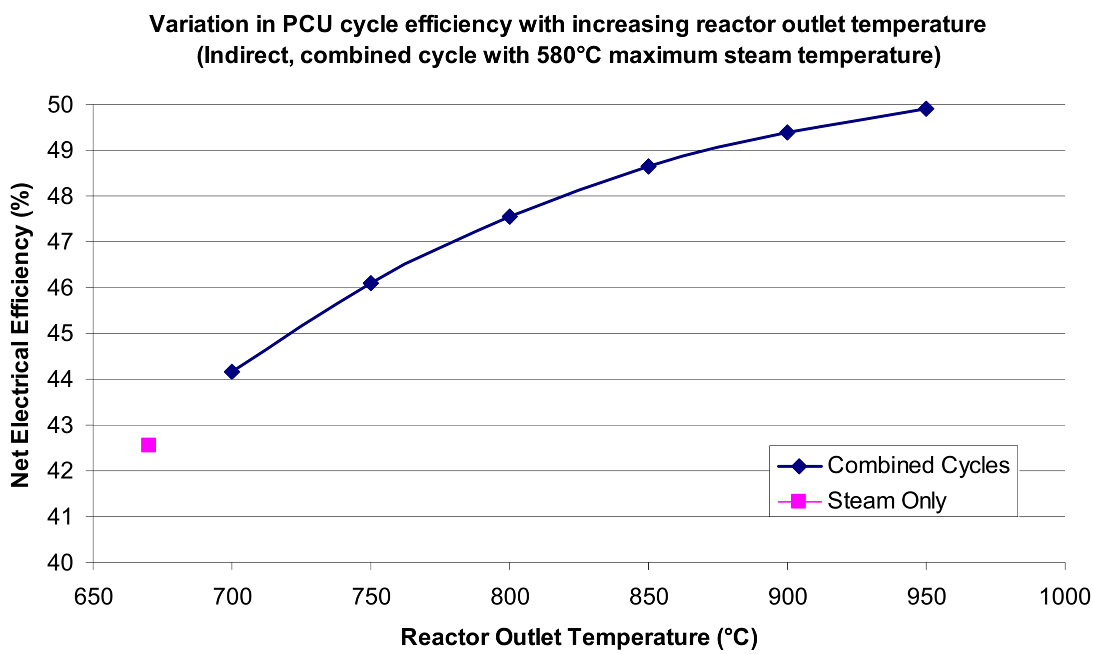

Figure E-7. Cycle efficiency vs. outlet temperature for steam only and combined cycles

It is clear that the combined-cycle points and the steam-only cycle point form a continuous curve. Including a gas turbine in the cycle can be seen to be worth around 3.5%pts of efficiency at a reactor outlet temperature of 750°C, 5%pts at 800°C, and 6%pts at 850°C. It is clear that the combined-cycle greatly benefits from increasing reactor outlet temperature, but as noted above higher reactor outlet temperatures involve increased reactor costs and risk.

Consideration of the reactor costs and risks associated with increasing reactor outlet temperature was outside the scope of the current Rolls-Royce combined-cycle PCS study.

Based on PCS considerations, Rolls-Royce recommends an indirect combined cycle as the best choice for a medium-term commercial plant for electricity-production. In the longer term, a direct combined cycle may merit reconsideration because it is slightly more efficient, and this effectively translates as more electrical capacity for a given reactor. For the benefits of this higher efficiency to be realized, the turbomachinery (running on pure helium as the working fluid and utilizing advanced active magnetic bearing technology) would have to be absolutely, dependably reliable. In a direct cycle, the pure gas turbine option, while not quite as efficient as a combined cycle, has the lowest footprint and is the most elegant. However, this option has a challenging active magnetic bearing and also requires a large robust recuperator with a delicate internal structure.

Evaluation of Compact IHX Design Issues

Because of the importance of the IHX to any indirect PCS option, more detailed evaluations of compact IHX design issues remaining from the FY08-1 IHX and heat transport alternatives study [GA 2008a] were included as part of the current PCS alternatives study. The evaluations were performed by Toshiba Corporation and included calculations to determine the effect of heat transfer assumptions on IHX size and a structural analysis to estimate the effect of thermal stresses on IHX lifetime. Toshiba was also tasked to evaluate the impact of using an 80 wt% nitrogen/20 wt% helium mixture in the secondary loop on the size and cost of a PCHE-type IHX.

The results of the IHX evaluations performed as part of the current PCS alternatives study are as follows:

- The zigzag method of calculating PCHE module heat transfer and pressure drop gives estimates that are in best agreement with Heatric Corp. PCHE module specifications.
- The results of the refined PCHE module stress analyses performed in the current study suggest that the service-lifetime of all of the IHX designs considered could potentially be 60 years. However, these stress calculations were for normal operation and cold shutdown conditions only.
- The effects of thermal and environmental aging on the IHX will need to be accounted for in more rigorous analyses and could potentially reduce the service lifetime of the IHX. Accumulation of graphite dust in the PCHE modules could also potentially reduce the IHX service lifetime.
- The estimated costs for all of the IHXs considered in this study increased relative to the costs estimated in the FY08-1 IHX study because use of the zigzag method to re-size the IHXs resulted in increased IHX pressure drops that were offset by increasing the

PCHE module length and the number of PCHE modules for each IHX. (The IHXs were re-sized to maintain the same pressure drop).

- The impact of using an 80 wt% nitrogen/20 wt% helium mixture as the working fluid in the secondary on the IHX would be to further increase the size and cost of the IHX due to the decrease in heat transfer performance.

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ACRONYMS AND ABBREVIATIONS

AMBs	active magnetic bearings
BOP	balance of plant
bpd	barrels [40 gallon] per day
CCGT	combined cycle [gas turbine]
CF	creep factor
COE	cost of electricity
COG-MHR	Cogeneration MHR
CPR	compressor pressure ratio
DOE	Department of Energy
DRL	design readiness level
DS	direction solidification [turbine blade]
EEDB	Energy Economic Data Base
EPC	engineering, procurement and construction [costs]
EPRI	Electric Power Research Institute
FE	finite element
FOAK	first-of-a-kind
GA	General Atomics
GT-MHR	Gas Turbine-MHR
HP	high pressure
HRSG	[steam generator]
HTGR	High Temperature Gas-Cooled Reactor
HTS	Heat Transport System
IHX	intermediate heat exchanger
INL	Idaho National Laboratory
LMTD	log mean temperature difference
LP	low pressure
MHR	Modular Helium Reactor
NGNP	Next Generation Nuclear Plant
NHSS	nuclear heat supply system
NOAK	N^{th}-of-a-kind
NRC	Nuclear Regulatory Commission
O&M	operations & maintenance
PCHE	printed circuit heat exchanger

PCS	power conversion system
PCDSR	Pre-conceptual Design Studies Report
psia	pound per square inch absolute
RF	reserve factors [safety factors]
RPM	revolutions per minute
RRAP	Rolls-Royce Aero-engine Performance [synthesis program]
SC-MHR	Steam-cycle MHR
SG	steam generator
SSCs	systems, structures and components
SSS	synchronous self shifting [clutch]
TRL	technology readiness level
Uhub	[hub line blade speed]

1 INTRODUCTION AND BACKGROUND

Pre-conceptual design studies [PCDSR 2007] for the Next Generation Nuclear Plant (NGNP) carried out by an industrial team led by General Atomics (GA team) recommended a reference Power Conversion System (PCS) for the NGNP and an alternative PCS. The reference PCS was a recuperated and intercooled closed Brayton cycle system located in the primary coolant system (direct Brayton cycle PCS). The alternative PCS was a combined-cycle system (gas turbine plus Rankine steam cycle) with the gas turbine and steam generator (SG) located in the primary coolant system (direct combined-cycle PCS).

Both of the PCS alternatives recommended by the GA team were concluded by the NGNP Project to entail significant risks associated with the lack of operational and maintenance experience for gas turbines in a gas reactor primary coolant loop, and [INL 2008] specifies that the NGNP must have an indirect PCS. Further, the NGNP Preconceptual Design Report [INL 2007] includes a requirement that the NGNP PCS be capable of producing steam for potential process heat applications. In response to these requirements, the GA team performed two follow-on conceptual design studies as part of the FY08-1 conceptual design studies. In one study [GA 2008a], the GA team evaluated various heat transport system alternatives in which the thermal energy produced in the reactor is transferred to the PCS in a secondary loop via an intermediate heat exchanger (IHX). In the second study, the GA team evaluated the pros and cons of locating the steam generator in the primary loop vs. a secondary loop [GA 2008b]. The result of these studies was that GA recommended an NGNP configuration having the steam generator in the primary loop as the reference concept to be developed during NGNP conceptual design.

Although the NGNP Project considers a gas turbine located in the primary circuit to be too risky for the NGNP, the Project recognizes that the capability of a direct combined PCS to co-generate steam for process heat applications and electricity at an efficiency approaching 50% is attractive for a commercial cogeneration plant. The Project also recognizes that an indirect combined cycle could be attractive for the same reasons, and could be a candidate for the NGNP PCS. Consequently, a comparison of direct and indirect combined cycle options for a commercial plant has been included in the scope of a PCS alternatives and selection study that the GA team has been tasked to perform as part of the FY08-2 NGNP conceptual design studies. This study is the subject of this report.

The current PCS alternatives and selection study is organized into four tasks:

<u>Task 1</u>. Recommended Configuration for NGNP. Provide a recommendation with respect to GA's preferred indirect cycle PCS configuration and a justification for the recommendation.

<u>Task 2</u>. Cost and Performance for the Recommended NGNP Configuration. Provide cost and

performance estimates for the PCS configuration recommended in Task 1. As originally defined, the study was to take the following criteria into account:

- The objective of initiating plant operation in 2018
- The potential that the plant will be initially operated at reactor outlet temperatures in the 750 °C to 850 °C range
- The principle that the design of the plant should not preclude operating with a reactor outlet temperature of up to 950 °C

However, the above criteria have changed as a result of two major programmatic decisions that were made since inception of the study. First, the date for initial plant operation was slipped from 2018 to 2021. Second and more importantly, the reactor outlet gas temperature was reduced to 750°C to 800°C and the requirement to design the plant not to preclude operating with a reactor outlet gas temperature up to 950°C was eliminated. As discussed below, these programmatic decisions necessitated some changes in the planned approach to the study.

<u>Task 3</u>. Evaluate the Combined Cycle PCS Alternative (for the NGNP). Evaluate the feasibility of using a combined cycle PCS configuration in an indirect heat transport configuration in NGNP. This task was assigned to Rolls-Royce. Because the purpose of the NGNP is to demonstrate technologies that are attractive for deployment in commercial reactors, it was decided that the logical approach for this task would be to first determine the most attractive combined cycle option for a commercial plant and then assess how the technology could best be demonstrated in NGNP. Thus, the scope of the task evolved to include the following elements.

- Evaluate indirect combined cycle options for a commercial plant and select a preferred option
- Compare cost and performance estimates for the preferred indirect combined cycle option with a direct combined cycle option, and make a recommendation on the best option for a commercial plant
- Determine how the technology for the recommended commercial plant option could best be demonstrated in the NGNP
- Perform an assessment of the technology readiness level (TRL) of the direct and indirect combined cycle options

Because of the importance of the IHX to any indirect PCS option, more detailed evaluations of compact IHX design issues remaining from the FY08-1 IHX and heat transport alternatives study [GA 2008a] were included as part of the current PCS alternatives study. The evaluations were performed by Toshiba Corporation and included calculations to determine the effect of heat transfer assumptions on IHX size and a structural analysis to estimate the effect of thermal stresses on IHX lifetime. Toshiba was also tasked to evaluate the impact of using an 80% nitrogen/20% helium mixture in the secondary loop on the size and cost of a PCHE-type compact IHX.

Task 4. PCS Alternatives for the Commercial Applications. Identify configurations of the PCS that should be considered for commercial applications including, as a minimum, electric power production, cogeneration and support of hydrogen production.

Since inception of the PCS alternatives and selection study, the NGNP Project has reduced the reactor outlet gas temperature to 750°C to 800°C and eliminated the requirement to design the plant not to preclude operating with a reactor outlet gas temperature up to 950°C [DOE 2008]. As a result, the work under Tasks 1, 2 and 4 was refocused on the 750°C to 800°C reactor outlet gas temperature range. Also, as a result of the reduction in reactor outlet gas temperature, the plant configurations considered for commercial applications have been limited to electric power production and cogeneration of process steam and electricity.

For the purposes of the present PCS study, the PCS has been partitioned into two parts. The first part is the Nuclear Heat Supply System (NHSS) side, and the second part is the Balance of Plant (BOP) side. The NHSS side contains all of the systems, structures and components (SSCs) associated with the production of steam using heat energy from the reactor and the BOP side of the PCS consists of the SSCs for production of process steam and/or utilization of the steam for generation of electricity.

The Task 1 and 2 results for the NGNP NHSS side of the PCS are presented in Section 2. Key system and component characteristics for the NHSS PCS alternatives that were considered are provided, and the recommended NHSS alternative is identified along with the basis for its selection. Section 3 provides the results for Tasks 1, 2 and 4. Options are identified for the BOP side of the PCS, and performance, design, and cost data are presented for a reference PCS configuration for cogeneration of process steam and electricity. Section 4 presents the results of the combined cycle PCS alternatives study performed as described above by Rolls-Royce under Task 3. Section 5 presents the results of the IHX analyses performed by Toshiba under Task 3. The results and conclusions of the various tasks are summarized at the end of each section and in the Executive Summary.

2 NHSS-SIDE PCS CONFIGURATION EVALUATIONS

2.1 Alternatives Considered

In a previous study [GA 2008b], alternative PCS configurations for production of steam using heat from the NGNP reactor were identified and evaluated. One configuration with a steam generator in the primary circuit and two configurations with a steam generator (SG) in a secondary loop were considered. The configurations with the SG in a secondary loop are shown in Figures 2-1 and 2-2, and the configuration with the SG in the primary circuit is shown in Figure 2-3.

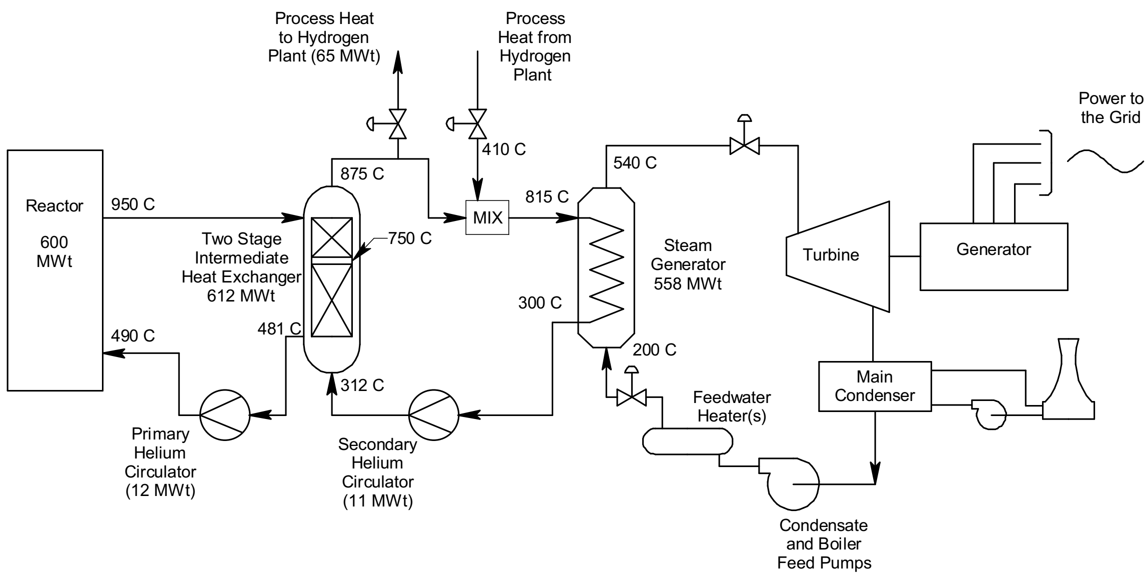


Figure 2-1. Steam generator in secondary loop – serial HTS

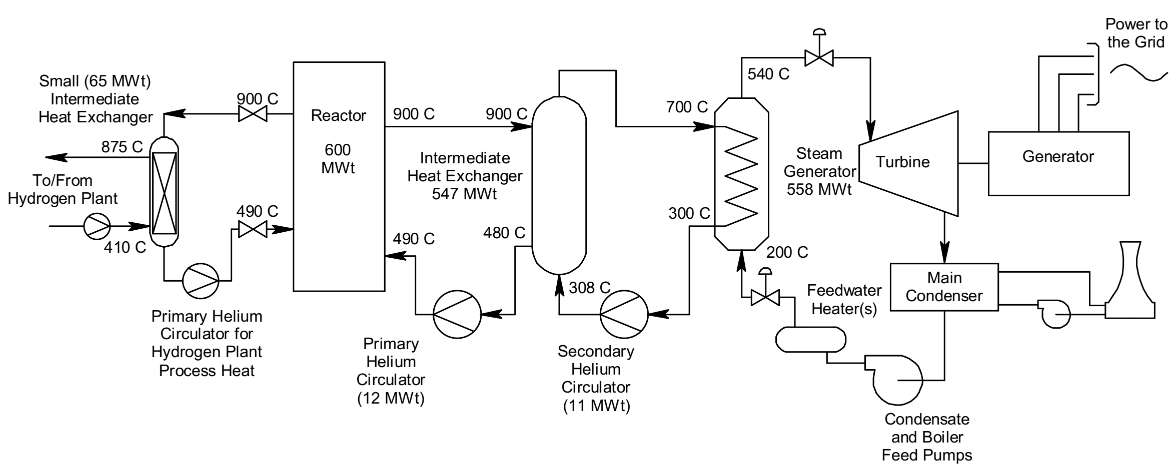


Figure 2-2. Steam generator in secondary loop – parallel primary loop configuration

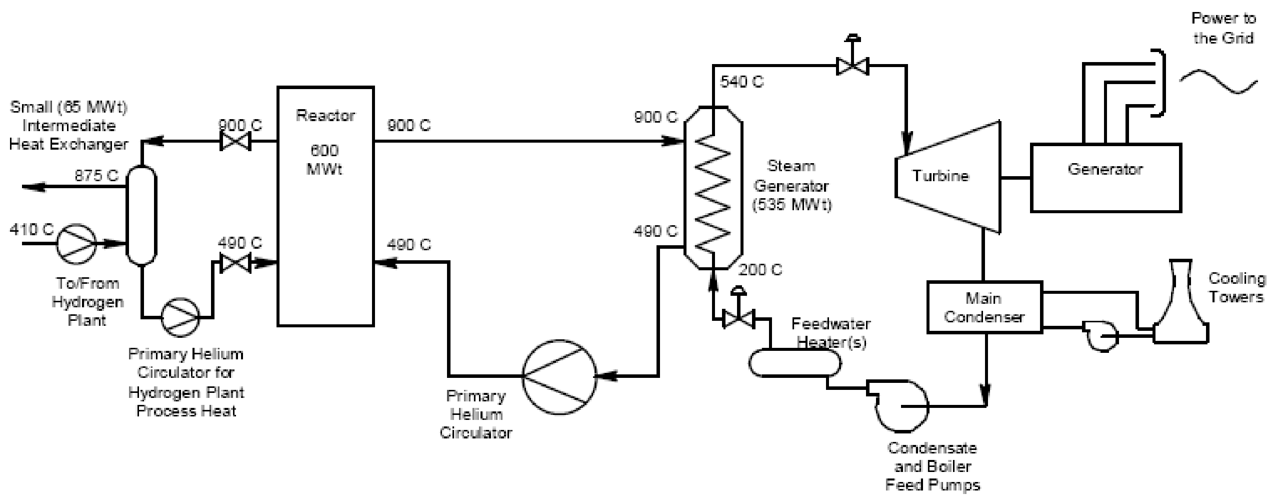


Figure 2-3. Steam generator in primary loop

It was concluded in [GA 2008b] that the configuration with the SG in the primary loop should be selected for further NGNP design development and better definition of estimated costs and performance. Accordingly, at the onset of the present PCS alternatives study, the configuration shown in Figure 2-3 was selected as the reference configuration for more detailed evaluation. The indirect PCS configuration shown in Figure 2-2 was also selected for evaluation. Based on the evaluations in [GA 2008b], it was decided to also consider dual-loop variants for both of the alternatives because dual loops were concluded not to entail large cost penalties.

However, following inception of the PCS alternatives study, an agreement was reached in a meeting of NGNP project participants in October 2008 [DOE 2008] to reduce the reactor core outlet gas temperature objective for NGNP into the range of 750°C to 800°C. On the basis of this agreement, GA chose a core outlet temperature of 750°C for the PCS study to enable use of much of the existing high temperature gas reactor (HTGR) design and technology from previous U.S. HTGR programs (e.g., MHTGR, NPR, Large HTGR, Fort Saint Vrain, Peach Bottom). Also, the parallel primary loop shown in Figures E-1 and E-2 to supply high-temperature process heat to an engineering-scale hydrogen production plant (or plants) was deleted from GA's reference NGNP plant configuration because neither of the advanced hydrogen production processes under consideration (e.g., high-temperature electrolysis or the S-I thermochemical water splitting process) are considered economical if heat is provided to the process at less than 750°C.

The resultant modified reference PCS configuration is shown in Figure 2-4, and the indirect cycle alternative is shown in Figure 2-5. In these figures, the PCS is partitioned into a NHSS side and a BOP side.

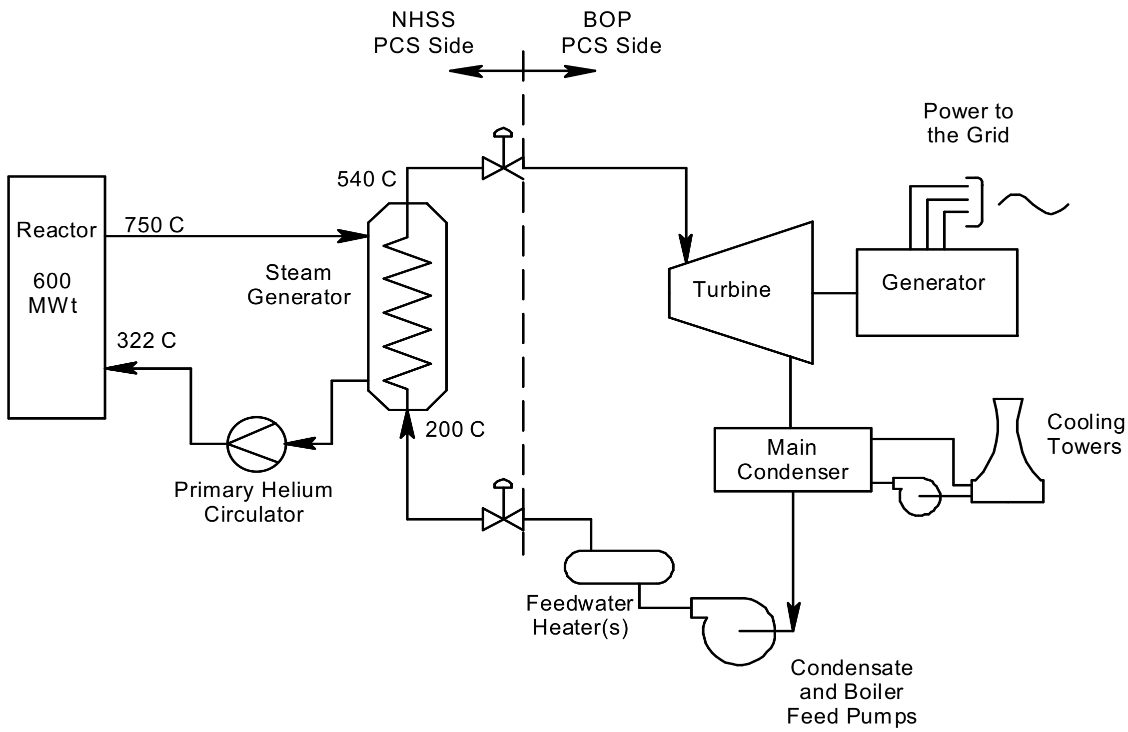


Figure 2-4. Reference PCS configuration with steam generator in primary loop

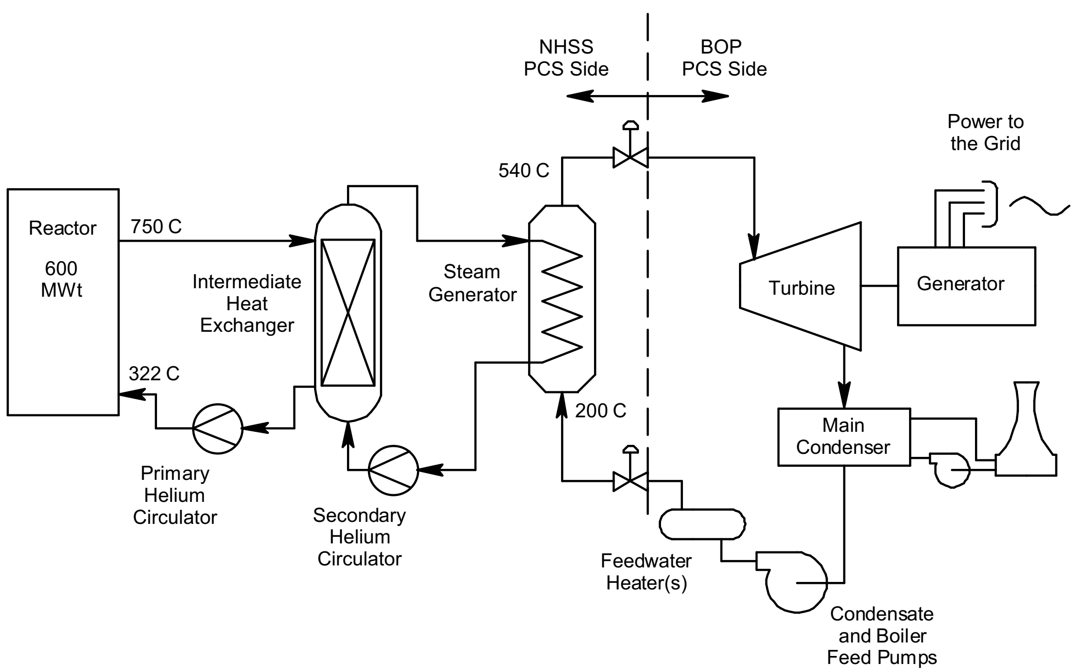


Figure 2-5. Alternative PCS configuration with steam generator in secondary loop

2.2 Performance, Design and Cost Characteristics

To assess performance, design and cost characteristics for the various PCS alternatives, SG concepts, IHX concepts, and heat balance diagrams were developed for each of the NHSS-side PCS options identified in Figure 2-6. Capital cost estimates were also prepared for commercial plant designs that include these various PCS alternatives.

NHSS-side PCS Configuration	
SG Arrangement	**No of Loops**
SG in Primary Loop (Direct Steam Cycle)	Single Loop
	Dual Loop
SG in Secondary Loop (Indirect Steam Cycle)	Single Loop
	Dual Loop

Figure 2-6. PCS alternatives included in PCS study

2.2.1 Steam Generator Designs

The design of the steam generator (SG) assumed for each of the PCS alternatives is based on the SG design concept developed for the MHTGR [CDSR 1987]. This concept was used for the NPR conceptual design and benefits from technology and operating experience obtained from similar steam generators used in the Fort Saint Vrain plant. The MHTGR design arrangement is based on housing the SG in a SG vessel, with its thermal center located below that of the reactor core as shown in Figure 2-7.

The MHTGR design arrangement is directly applicable for the PCS alternatives with the SG in the primary loop. For the indirect cycle alternatives, the IHX should be located below the thermal center of the reactor core analogous to the MHTGR arrangement, but alternative SG designs and arrangements could be used. However, in sizing the SGs for the current study, the same SG design concept was assumed for both the direct and indirect cycle alternatives.

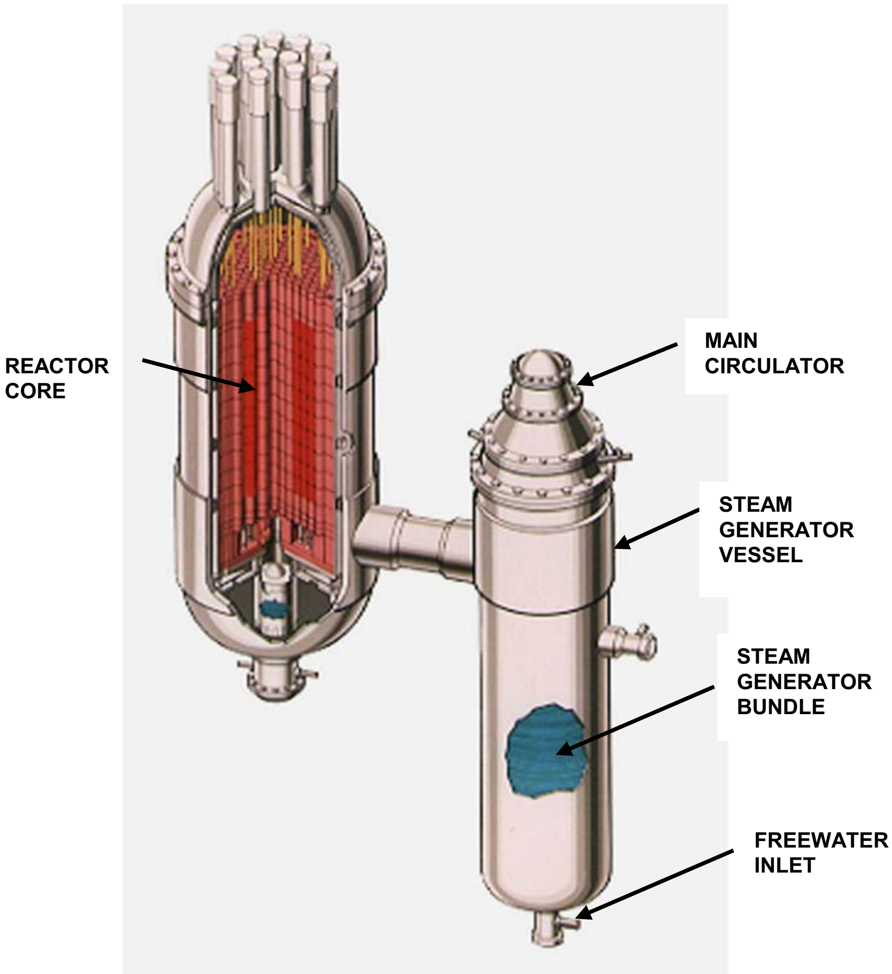


Figure 2-7. MHTGR steam generator conceptual design arrangement

In this design concept, the SG is a vertically oriented, upflow boiling, cross-counterflow, once-through shell-and-tube heat exchanger. The SG utilizes a helically-wound tube bundle. The design provides access for tube leak detection and plugging from both ends of each tube. In addition, the design makes possible the removal and replacement of the SG tube bundle through a removable upper vessel head even though the unit is designed with a service life equal to that of the plant. Support of the helical-tube bundle is by means of drilled radial support plates into which the tubes are threaded. The plates are supported vertically at the lower end of the tube bundle and horizontally by lateral restraints of inner and outer shrouds.

Feedwater is introduced through a nozzle assembly in the bottom of the SG vessel (see Figure 2-7). The superheated steam is discharged through a nozzle assembly in the upper side wall of the SG vessel. A typical SG tube is shown schematically in Figure 2-8. From the feedwater tubesheet, the tube leads into the economizer/evaporator/initial superheat section of the helical bundle, where it threads through the radially oriented, drilled- plate support structure. A transition lead-out section contains a material change from 2¼Cr - 1 Mo steel to Alloy 800H; which entails a bimetallic weld. The Alloy 800H section of the tube leads into the finishing superheater bundle and then out to an expansion zone. Some differential expansion is accounted for in the lead-in and transition lead sections of the tube, but the net axial expansion difference between the SG and the SG vessel is accommodated by the expansion loops.

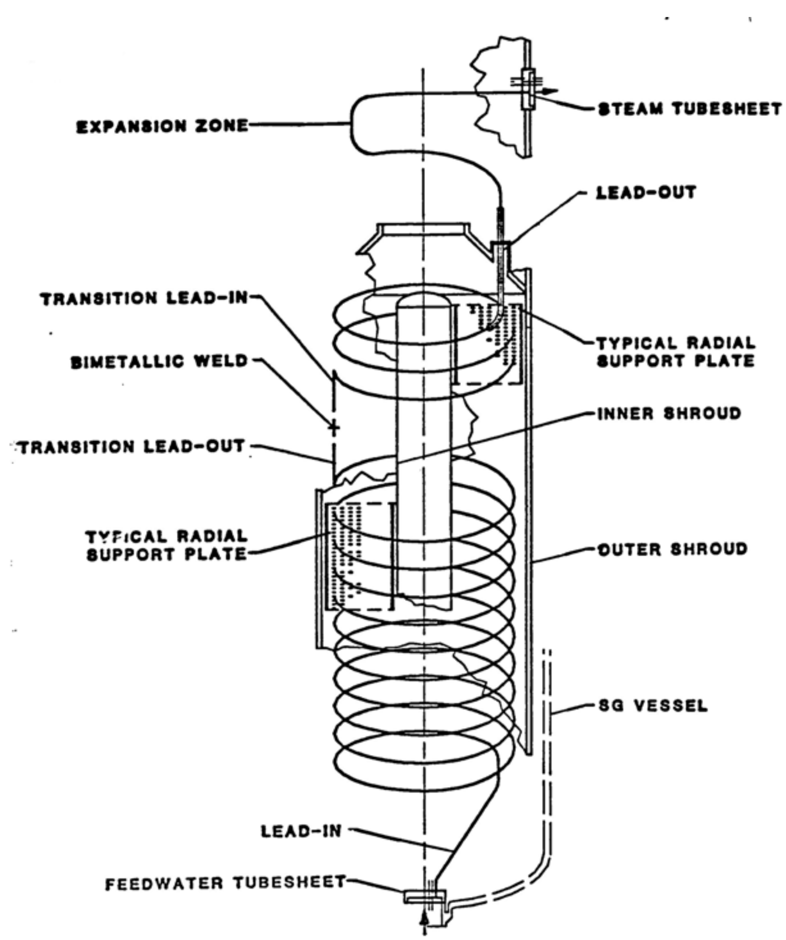


Figure 2-8. Schematic of typical SG tube

Hot helium from the reactor enters a plenum above the SG bundle, the gas flows downward across the heat transfer surface between the inner and outer SG shrouds, and exits into a lower outlet plenum. In this plenum, the helium is turned 180° and directed into a return annulus formed between the outermost shroud and the SG vessel wall. After flowing upward in this annulus, the gas enters a circulator inlet plenum.

The SG design parameters determined for each of the PCS alternatives are listed in Table 2-1. Key SG size parameters are highlighted in bold text. Table 2-2 provides a comparison of the SG volume and weight ratios.

Table 2-1. Steam Generator Design Parameters

SG Design Parameter	**SG in Primary Single Loop**	**SG in Primary Dual Loop**	**SG in Secondary Single Loop**	**SG in Secondary Dual Loop**
SG Heat Duty, MW	611	305	614	307
SG Bundle OD , m	**4.65**	**3.96**	**4.65**	**4.65**
Bundle ID, m	2.29	1.22	1.22	1.22
Coil Height, m	4.7	3.46	6.23	3.99
Total SG Height, m	**12.00**	**11.00**	**14**	**12**
Total Bundle Weight*, tonne	**125**	**75.4**	**185**	**120**
Vessel Height	**16.50**	**15.3**	**18**	**15.8**
Vessel ID, m	**5.26**	**4.57**	**5.26**	**5.26**
Vessel Weight, tonne	**351**	**268**	**377**	**341**
Tube Sheet Weight, tonne	91	30.1	90.3	30.2
Number of Tubes	604	289	600	289
Tube OD, mm	22.23	22.23	22.23	22.23
Tube Wall, mm	3.30	3.302	3.302	3.302
He Inlet Temperature, °C	750	750	700	700
He Outlet Temperature, °C	314.3	315	272	274
He Flow Rate, kg/sec	269.6	134.8	275.9	138.5
He Inlet Pressure, MPa	7.00	7.00	7.00	7.00
He Pressure Drop, kPa	10.84	3.1	9.72	1.94
Water Inlet Temperature, °C	200	200	200	200
Steam Outlet Temperature	540.6	540.6	540.6	540.6
Water Flow Rate, kg/sec	241.4	120.2	241.9	120.9
Feedwater Inlet Pressure, MPa	18.97	19.46	19.6	20.34
Steam Outlet Pressure, MPa	17.24	17.24	17.24	17.24

* Bundle weight includes total tubes (dry), shrouds and support plates weights. It does not include vessel and tube sheet weight.

Table 2-2. Comparison of SG volume and weight ratios

Single versus dual loops	**Direct**	**Indirect**
Total SG volume, dual/single	1.33	1.71
Total SG weight, dual/single	1.32	1.51
Direct versus Indirect	Single Loop	Dual Loop
Total SG volume, Indirect/direct	1.17	1.50
Total SG weight, Indirect/direct	1.15	1.32

The following conclusions are made based on consideration of only the SG sizes (diameter & length), volumes and weights:

- With regard to overall size and weight considerations, all of the SGs have sizes and weights that fall within ranges that have been found to be acceptable in past gas reactor programs for manufacturing, handling, transportation and installation.
- The direct single loop requires the least SG space and construction material. Dual direct loops would require ~33% more SG space and require ~32% more weight of construction material.
- Indirect dual loops would require ~71% more SG space and ~51% more weight of construction material than single indirect loops.
- Indirect single loops require ~17% more SG space and ~15% more construction material than direct single loops.
- Indirect dual loops require ~50% more SG space and ~32% more construction material than direct dual loops.
- The alternative that requires the most SG space and construction material is the indirect dual-loop configuration. The indirect dual loop requires ~100% more space (1.33 x 1.50) and ~73% more construction material than the direct single loop configuration for just the SG portion of the PCS.

2.2.2 IHX Designs

Design concepts were developed for the IHX required for the indirect loop PCS alternatives based on the core inlet and outlet gas temperatures shown on Figure 2-5. This was done for the purpose of obtaining a rough estimate of the size of the IHX, which was necessary to support development of the plant capital cost estimates. The printed circuit heat exchanger (PCHE) design was assumed for the IHXs. For sizing the IHXs, the simplifying assumption was made that the designs could be based on the use of a small number of heat transfer modules since the operating temperatures permit long lifetimes. Although this assumption may not be valid for the current Heatric PCHE design, the use of few heat transfer modules reduces overall cost and simplifies the design. Table 2-3 summarizes the results of the sizing analysis for two IHXs, one for the single indirect loop alternative and one for the dual indirect loop alternative.

Schematics of the IHX design arrangements are provided in Figures 2-9 and 2-10. In these schematics, a cylindrical pressure vessel is provided that encloses the heat transfer modules. The schematic of the IHX for the single indirect loop (Figure 2-9) indicates a very large pressure vessel is required (~9 m diameter). While considerable optimization of the size is potentially possible, the size of the IHX for a single indirect loop PCS appears to be impractical given that the current maximum size for manufacture of a nuclear grade vessel is on the order of 7 to 8 m. Although multiple IHXs could be used in the single indirect loop, the simplest approach would be to use multiple (e.g., dual) indirect loops.

Table 2-3. IHX Design Parameters

Design Parameter	Single-Loop Indirect PCS	Dual-Loop Indirect PCS
GEOMETRIC PARAMETERS		
Material	Alloy 617	Alloy 617
Number of Modules	4	2
Installed Heat Transfer Area, m^2	14,000	6,763
Module Height, m	5.2	5.2
Total Module Width (Includes Edges), m	2.026	2.026
Edge Distance, mm	13	13
Total Module Length, m	0.761	0.761
Radius of Helium Channels, mm	1.5	1.5
Channel Center to Center Spacing, mm	3.9	3.9
Channel Offset Pitch, mm	12.7	12.7
Height of Offset, mm	2.286	2.286
Layer Thickness, mm	2.4	2.4
Number of Layers per Module per fluid	1077	1077
Total Metal Volume Before Etching Channels, m^3	32.07	16.03
Total Weight, tonne	247	124
HEAT TRANSFER/FLUID FLOW PARAMETERS		
Required Heat Duty, MW	612	306
Calculated Heat Duty, MW	612.5	306.3
Helium Inlet Temperature, °C	750	750
Helium Outlet Temperature, °C	313	313
Secondary He Inlet Temperature, °C	275	275
Secondary He Outlet Temperature, °C	700	700
Metal Temperature at the Primary Side Inlet, °C	725.8	725.8
Metal Temperature @ Midspan, °C	495.2	495.2
Metal Temperature at the Primary Side Outlet, °C	294.6	294.6
Helium Pressure, MPa	7	7
Helium Flow Rate, kg/sec	269.32	134.66
Secondary He Flow Rate, kg/sec	276.92	138.46
Primary He Pressure Drop in Hex, kPa	28.5	28.5
Secondary He Pressure Drop in Hex, kPa	28.2	28.2
LMTD, °K	43.73	43.73

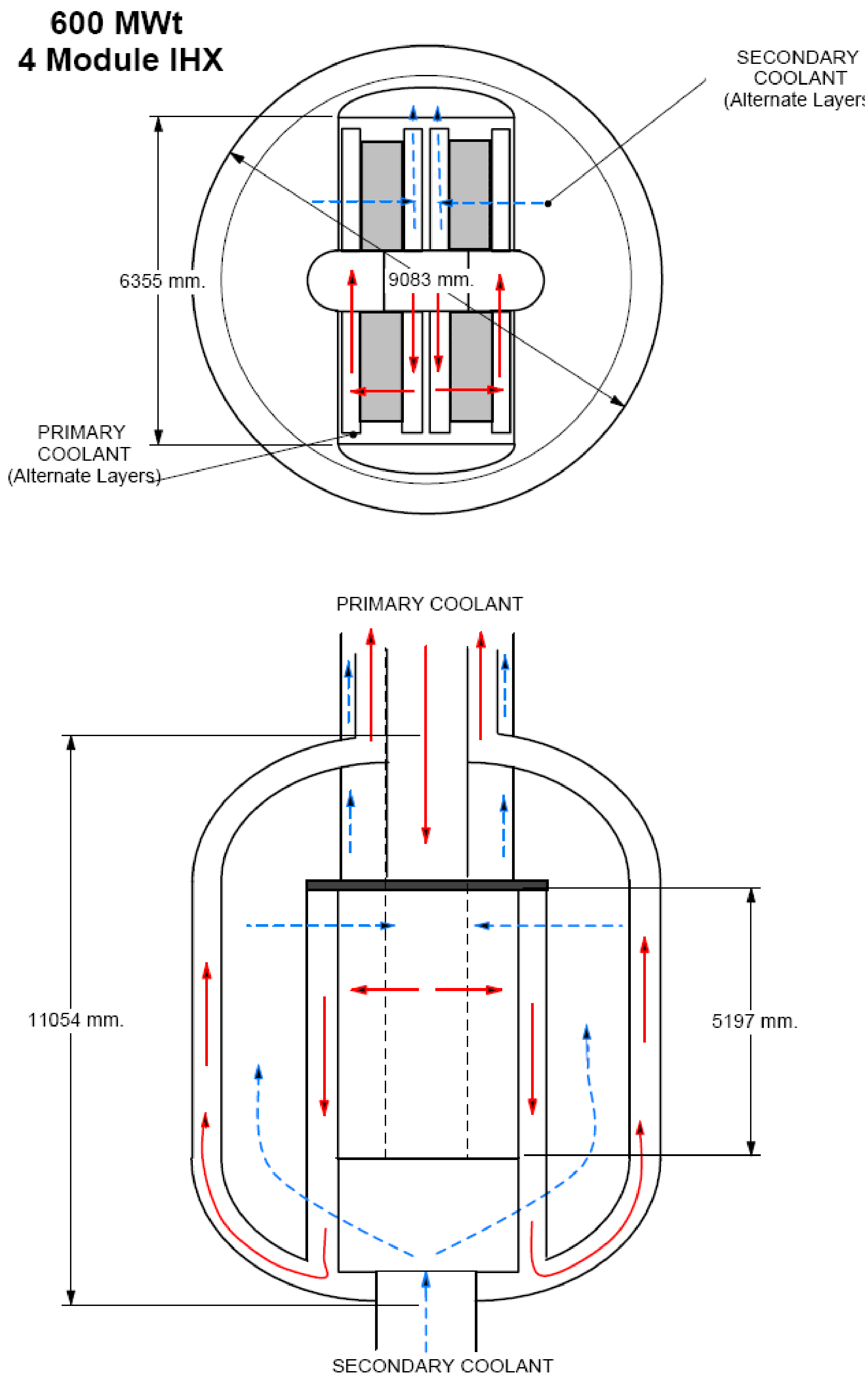
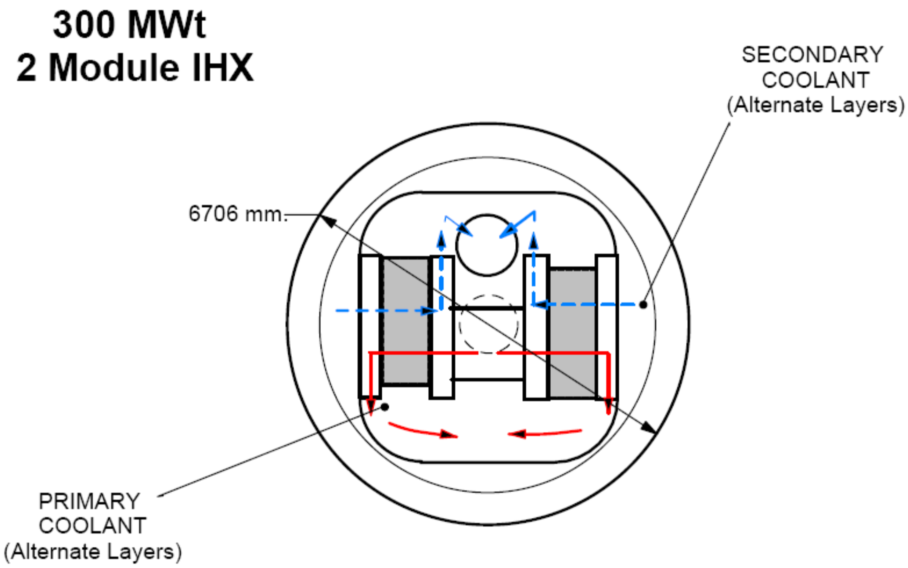


Figure 2-9. Schematic of IHX for single indirect loop PCS alternative



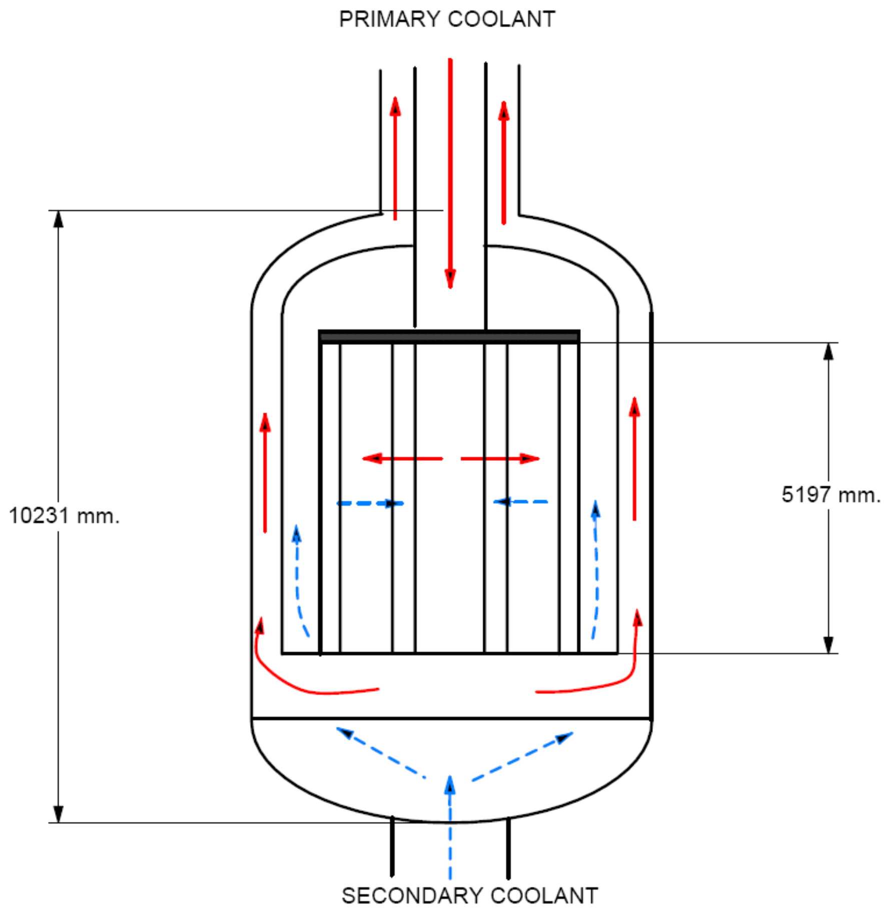


Figure 2-10. Schematic of IHX for dual indirect loop PCS alternative

2.2.3 Heat Balances

Heat balances were developed for each of the NHSS-side PCS alternatives identified in Figure 2-6. The heat balances were developed using a simplified systems analysis model for coupling the components in the primary loop(s) (the reactor and heat exchangers, SG or IHX) and secondary loop(s) (IHX and SG), as applicable. Heat balance diagrams for each of the NHSS-side PCS alternatives are provided in Figures 2-11 through 2-14.

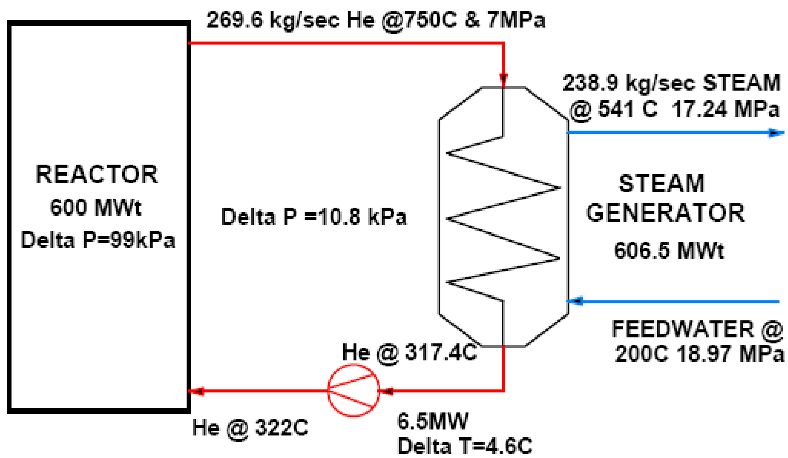


Figure 2-11. Single loop PCS configuration with SG in primary loop

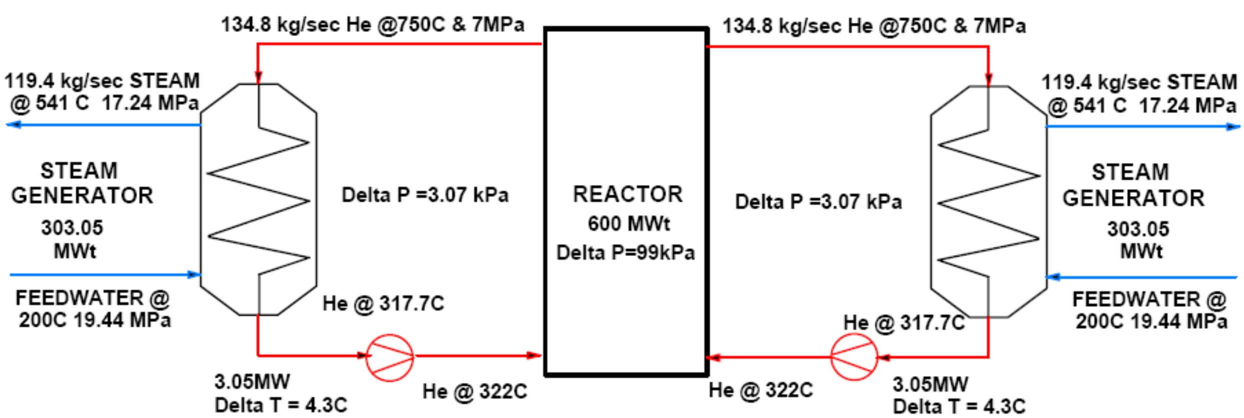


Figure 2-12. Dual-loop PCS configuration with SG in each of dual primary loops

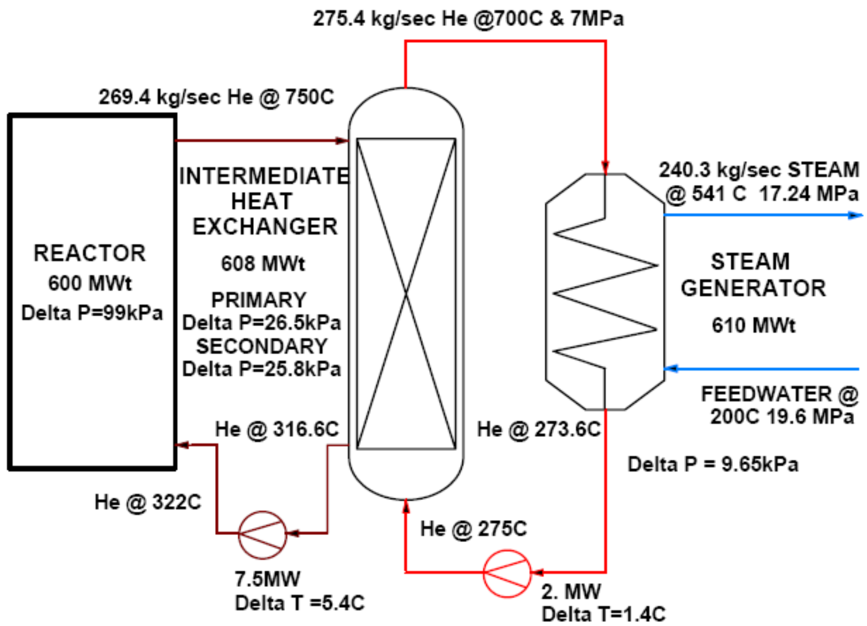


Figure 2-13. Single-loop PCS configuration with SG in secondary loop

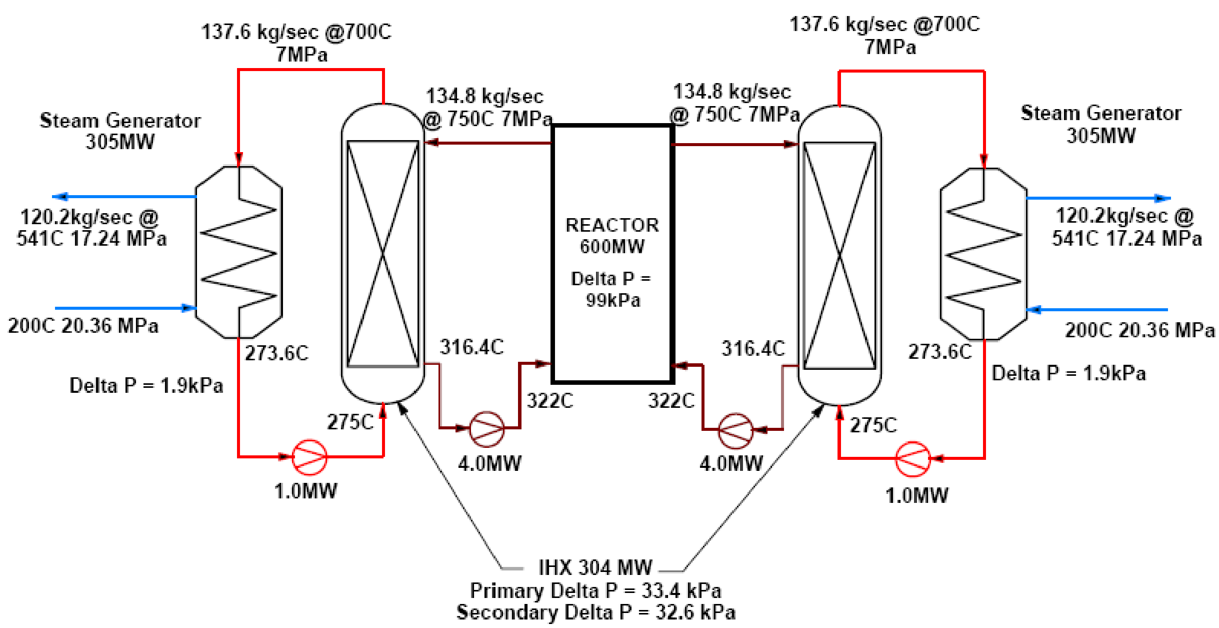


Figure 2-14. Dual-loop PCS configuration with SG in secondary loop

The primary purpose of developing the heat balances was to determine the circulator power requirements and the commercial availability of circulators having the required capacity. The circulator power requirements from the heat balance diagrams are summarized in Table 2-4[1].

Table 2-4. Circulator Power Requirements

NHSS PCS Alternative	Primary Circulator Power MW(t)*	Secondary Circulator Power MW(t)*
Single Direct Loop	6.5	---
Dual Direct Loop	3.1	---
Single Indirect Loop	7.5	2.0
Dual Indirect Loop	4.0	1.0
* As shown in this table, MW(t) is the thermal energy added to the heat transport system by the helium circulator. The MW(e) rating of the helium circulator is 10% to 20% higher.		

[GA 2008a] provides the following assessment of the current state of helium circulator technology [GA 2008a].

- The technology required to produce high-temperature helium circulators is well understood and relatively available for circulators of up to about 5 MW(e)
- A credible vendor (Howden) confirmed that circulators of about 6 MW(e) are currently considered to be viable. This includes circulators featuring the preferred bearing option, Active Magnetic Bearings (AMBs).
- Higher powered circulators are feasible but will require more development. Development costs are expected to increase rapidly as the machine size approaches 10 MW(e). The largest practical size for the helium circulator is around 15 MW(e).

Based on this assessment, all of the circulators in Table 2-4 are potentially viable, but the most practical approach would be to limit the circulator power to ≤ 6 MW(e). In this approach, the dual-loop options should be considered, or two circulators operating in parallel should be used for the single loop alternatives.

[1] The secondary loop helium circulator power (2.0 MWt) shown for the single indirect loop in Table 2-4 is much lower than that shown in Figure 2-2 (11 MWt). This difference resulted primarily from resizing of the conceptual steam generator and IHX designs in the current study, which resulted in large reductions in the pressure drops across the heat exchangers.

2.2.4 Plant Relative Capital Costs

To evaluate the cost significance of the different types, quantities, and sizes of components required by the NHSS-side PCS alternatives, full scope plant capital cost estimates were prepared for each of the PCS alternatives shown in Figure 2-6 for an electric-only generation plant and a cogeneration plant producing electricity and process steam. “Full scope” means all of the direct and indirect capital costs per the Energy Economic Data Base (EEDB) code of accounts [ORNL 1993] for all Systems, Structures and Equipment for complete plants (both NHSS and BOP plant sides) are included in the estimate.

To prepare the plant capital cost estimates, an MHTGR cost data base from [GCRA 1993] was used. The cost data base was escalated to 2007$ and scaled as required for the required plant powers, component quantities, and component sizes using appropriate cost estimation methodologies. Key assumptions upon which the cost estimates are based include:

- A generic (Middletown, U.S.) plant construction site was used as bases for the estimate.
- A project contingency of 20% was applied to the capital costs to allow for estimate uncertainties.
- The reactor core design used for all cases was the MHR 600-MW(t) core.
- The commercial plant was assumed to be the 6th four-module plant constructed, representing the nth-of-a-kind (NOAK) plant.
- Each module in the plant was assumed to have a one-on-one NHSS and BOP configuration (one turbine per reactor for the electric generation plant).
- One-half of the SG output was assumed to go to produce process steam and one-half used for electricity generation in the cogeneration plant.
- Two steam turbine designs were used in the cost estimates. A turbine having a rating of 240 MW(e) was used for the electric-only generation plant and a turbine rated at 124 MW(e) was used for the cogeneration plant.
- The Feedwater and Condensate system was based on a conditioning and full return of the condensate from the turbine.
- For the indirect cycles, the IHX was assumed to be positioned within the Reactor Building with the SG being located within a Steam Generator Building attached to the exterior of the Reactor Building, protected by isolation valves.
- The indirect cycle plant cost estimates include an additional helium circulator in the secondary loop between the IHX and SG.

Table 2-5 and 2-6 summarize the plant capital cost estimates at the EEDB 2-digit account level for the direct NHSS-side PCS alternatives and the indirect NHSS-side PCS alternatives, respectively. The estimates provided in Tables 2-5 and 2-6 are ROMs having an unknown level of uncertainty because the data base used for development of the estimates is quite old;

however, the estimates are considered useful for comparison purposes because they were all made on a consistent basis. All costs are in 2007$.

Table 2-5. Summary of Plant Capital Costs for the Direct NHSS-side PCS Alternatives

MHR DIRECT CYCLE PLANTS SUMMARY		**Cost in 2007$**			
		One Loop All Electric Direct Cycle	**Two Loop All Electric Direct Cycle**	**One Loop Co-Gen Direct Cycle**	**Two Loop Co-Gen Direct Cycle**
ACC T	ACCOUNT DESCRIPTION				
	DIRECT COSTS				
20	LAND AND LAND RIGHTS				
21	STRUCTURES AND IMPROVEMENTS	227.7	245.0	216.5	244.7
22	REACTOR PLANT EQUIPMENT	581.1	610.8	581.0	610.8
23	TURBINE PLANT EQUIPMENT	302.6	319.2	218.4	226.4
24	ELECTRIC PLANT EQUIPMENT	80.0	80.0	48.7	48.7
25	MISCELLANEOUS PLANT EQUIP	29.2	29.2	29.2	29.2
26	HEAT REJECTION SYSTEM	51.6	51.8	48.4	48.4
	TOTAL DIRECT COST	1272.1	1335.9	1142.3	1208.2
	INDIRECT COSTS				
91	CONSTRUCTION SERVICES	147.4	154.8	132.4	140.0
92	HOME OFFICE ENG AND SERV	95.4	100.2	85.7	90.6
93	FIELD OFFICE ENG AND SERV	76.1	80.0	76.1	79.9
94	OWNER'S COST	280.1	294.4	280.0	294.0
	TOTAL INDIRECT COSTS	599.0	629.5	574.2	604.6
	ENGR. PROCUR. CONSTR. COST (EPC)	1871.1	1965.4	1716.5	1812.8
	CONTINGENCY (BASED ON 20%)	374.2	393.1	343.3	362.6
	TOTAL EPC COST	2245.3	2358.4	2059.8	2175.4

Table 2-6. Summary of Plant Capital Costs for the Indirect NHSS-side PCS Alternatives

MHR INDIRECT CYCLE PLANTS SUMMARY		Cost in 2007$			
		One Loop All Electric Indirect Cycle	Two Loop All Electric Indirect Cycle	One Loop Co-Gen Indirect Cycle	Two Loop Co-Gen Indirect Cycle
ACCT	ACCOUNT DESCRIPTION				
	DIRECT COSTS				
20	LAND AND LAND RIGHTS				
21	STRUCTURES AND IMPROVEMENTS	244.2	265.4	232.9	265.0
22	REACTOR PLANT EQUIPMENT	719.9	779.3	719.9	779.3
23	TURBINE PLANT EQUIPMENT	302.6	319.2	218.4	226.4
24	ELECTRIC PLANT EQUIPMENT	80.0	80.0	48.7	48.7
25	MISCELLANEOUS PLANT EQUIP	29.2	29.2	29.2	29.2
26	HEAT REJECTION SYSTEM	51.6	51.8	48.4	48.4
	TOTAL DIRECT COST	1427.5	1524.7	1297.5	1397.0
	INDIRECT COSTS				
91	CONSTRUCTION SERVICES	165.4	176.7	150.4	161.9
92	HOME OFFICE ENG AND SERV	107.1	114.4	97.3	104.8
93	FIELD OFFICE ENG AND SERV	94.3	102.1	94.3	102.1
94	OWNER'S COST	347.0	375.6	347.0	375.6
	TOTAL INDIRECT COSTS	713.8	768.8	689.0	744.4
	ENGR. PROCUR. CONSTR. COST (EPC)	2141.3	2293.5	1986.5	2141.4
	CONTINGENCY (BASED ON 20%)	428.3	458.7	397.3	428.3
	TOTAL EPC COST	2569.6	2752.2	2383.8	2569.7

Table 2-7 contains a comparison of the relative total plant capital costs (Total "EPC" cost[2]).

Table 2-7. Comparison of Relative Plant Capital Costs

NHSS PCS Alternative	One Loop Electric Plant	Two Loop Electric Plant	One Loop Co-Gen Plant	Two Loop Co-Gen Plant
Direct	1.09	1.14	1.00	1.06
Indirect	1.25	1.34	1.16	1.25

The relevant conclusions from the numbers in Table 2-7 are as follows:

- The plant having the lowest plant capital cost is the direct NHSS PCS single loop cogeneration plant
- The dual-loop PCS configuration increases plant capital cost by ~6%
- Indirect PCS configurations increase plant capital costs 16% to 20%

These results are consistent with the conclusions reached in [GA 2008b], which were derived using a more macroscopic cost analysis model.

2.3 Summary and Recommendation

2.3.1 Summary

In [GA 2008b], alternative NGNP PCS configurations capable of producing steam for potential process heat applications were evaluated. Some of the configurations that were considered included an IHX to transfer reactor heat from the primary loop to a secondary loop containing a SG. Based on the economic, control and protection, safety, maintainability, tritium transport, and commercial prototype evaluations performed in [GA 2008b], it was recommended that an NGNP configuration that includes the SG in the primary loop be selected for further design development and better definition of estimated costs and safety performance. In the current study, further evaluations of alternative PCS configurations capable of producing steam for potential process heat applications have been performed. The two basic NHSS-side PCS configurations illustrated in Figures 2-4 and 2-5 were evaluated. Figure 2-4 shows the direct PCS configuration with the SG in the primary loop. Figure 2-5 shows the indirect PCS configuration with the SG in a secondary loop interconnected to the primary loop by means of an IHX. Dual loops as well as single loops for both the direct and indirect configurations were evaluated.

[2] EPC = Engineering, Procurement and Construction cost.

Rough sizing calculations were performed for the heat exchangers in both the single and dual-loop variants of the configurations shown in Figures 2-4 and 2-5. It was concluded that the SG sizes are acceptable for all of the alternatives and variants. The estimated IHX size is, however, considered excessive (~9 m diameter) for the single loop variant of the indirect alternative. So, for the indirect configuration, only the dual-loop version is considered viable.

Heat balances were developed for the single and dual-loop variants of both the direct and indirect PCS configurations. The heat balances indicate that the single-loop variants of both the direct and indirect PCS configurations require circulators having power capacities in excess of currently available circulator technology. However, evaluations in [GA 2008a] indicate that circulators having the required power capacity are feasible and could be developed and deployed to support NGNP startup in 2021. An alternative to developing the higher powered circulators would be to use two circulators in parallel for the single loop configurations; however, this would entail a somewhat more complicated design and control system.

A more detailed evaluation of the plant capital costs for the alternative PCS configurations was performed in the current study than was performed for the earlier SG alternatives study [GA 2008b]. Complete plant capital cost estimates were prepared for both electricity generation and cogeneration plants for single and dual-loop variants of reference direct and indirect PCS configurations. The plant having the lowest estimated capital cost is the cogeneration plant having the single-loop direct PCS configuration. The estimated plant capital cost for the dual-loop variant of this configuration is about 6% higher than the cost estimated for the single-loop variant. The capital cost of a cogeneration NOAK plant based upon use of the dual-loop indirect PCS configuration (the only viable indirect configuration) is estimated to be about 28% higher than the cogeneration plant with the single-loop direct PCS configuration.

Prior economic analyses [GCRA 1993] indicate that a NOAK 4x600 MW(t) high-temperature gas reactor electric generation plant with the direct NHSS PCS alternative should be economically competitive, although the economic advantage may not be great without credit for passivity or carbon emissions. Presuming the direct NHSS PCS electric generation plant is competitive, a cogeneration plant variant should be equally competitive. However, if the plant capital cost is 25% higher, as in the case of the plant based on use of the indirect PCS configuration, the economic viability would be questionable.

There are two issues associated with the direct PCS configuration that are the primary drivers for consideration of the indirect PCS configuration. These are:

- The potential for moisture ingress into the primary system form SG leakage. (The IHX provides an additional barrier against moisture ingress into the primary system.)

- The potential for migration of tritium from the primary system through the SG into the secondary system (no other radionuclide is capable of migrating through SG tubes at these operating temperatures). The IHX provides an additional barrier against tritium contamination of the process steam

Moisture ingress into the primary system is an issue that requires careful attention because moisture in the primary system can cause (1) corrosion of the core graphite, and (2) hydrolysis of the fuel kernels in fuel particles having failed coating layers, thereby increasing the release of fission gases, including radioiodines, from the fuel. In [GA 2008b], the safety analyses contained in [PSID 1992], which was generated for the reference MHTGR plant [CDSR 1987] were reviewed to gain insights into the relative safety hazards of locating the SG in the primary circuit. The conclusions from this review were as follows:

- Moisture ingress into the primary coolant system from SG leakage is not expected to result in unacceptable average or localized corrosion of either the bulk core moderator graphite or the graphite core support components.

- Moisture ingress into the primary coolant system from SG leakage is not expected to result in radionuclide releases in excess of regulatory limits.

The applicability of the safety analyses performed for the MHTGR operating with a core outlet gas temperature of about 700°C to an NGNP operating with a core outlet gas temperature of 950°C is questionable. However, with the core outlet gas temperature now reduced to 750°C for the NGNP, the above conclusions from the MHTGR safety analysis are considered to be valid for the NGNP.

With regard to the second issue, there is the potential for diffusion of tritium from the primary system through the SG into the secondary system steam. No direct leakage of radionuclides from the primary to the secondary loops is feasible because the secondary system operating pressure is significantly higher than the primary (~17 MPa vs. 7 MPa). Thus, migration of tritium from the primary system into the secondary loop is limited to diffusion mechanisms driven by the temperature gradient (i.e., from higher temperatures in the primary loop to lower temperatures in the secondary loop). If tritium contamination of the steam is determined to be a problem, one potential solution is to include a steam-to-steam heat exchanger (i.e., a reboiler) to transfer heat from the secondary loop to a tertiary loop, which will transport the heat to the end user. This heat exchanger would provide a further barrier against tritium transport into the process steam supply system. A heat exchanger in the steam supply system would be a more effective and economical means of preventing tritium contamination of the process steam than an IHX operating at much higher temperatures in primary coolant loop.

2.3.2 Recommended NGNP NHSS-side PCS Configuration

The direct NHSS-side PCS configuration shown in Figure 2-4, either in a single-loop configuration (the preferred configuration) or in a dual-loop configuration, is the recommended configuration based on the performance, design, and cost evaluations contained in Section 2.2 and summarized above. For the preferred single loop configuration, a parallel design and development path is recommended for providing the required circulator capacity. The dual path would include development of a single circulator of the required power capacity in parallel with development of a design that uses two circulators in parallel. For the latter case, each of the circulators would have about one-half the power capacity required for the single circulator case.

3 BOP-SIDE PCS ALTERNATIVES EVALUATION

The assessment presented herein evaluates BOP-side PCS options based on the NHSS conditions recommended in Section 2 as they pertain to the selection of a BOP for a commercial version of the NGNP. Capital cost estimates for PCS alternatives are developed for a commercial, Nth-of-a-kind (NOAK) plant consisting of four, 600 MW(t) reactor modules. The assumed NHSS configuration uses a single steam SG for each reactor module, and results in the secondary fluid conditions presented in Table 3-1.

Table 3-1. Assumed NHSS Configuration

Parameter	Value
Steam flow rate	269.6 kg/s (122.3 lbm/s)
Steam outlet temperature (from SG)	541ºC (1006ºF)
Steam outlet pressure	17.24 MPa (2500.5 psi)
Feed water inlet temperature (to SG)	200ºC (392ºF)
Feed water pressure	18.97 MPa (2751.4 psi)

GA has been developing the high-temperature, gas-cooled reactor technology since the middle 1960's for electricity production, cogeneration, and a variety of process-heat applications, including the production of hydrogen. In more recent years, GA has been developing a passively safe, modular design referred to as the Modular Helium Reactor (MHR). Consequently, GA has compiled a very large data base of technical, performance and cost information; as well as a large number of technical and cost models describing the use of the MHR for various electric and industrial applications. In addition, there is a significant amount of literature on Gas Cooled Reactor applications available as a result of projects sponsored by DOE, EPRI, the electric utility industry and other U.S. and international organizations. The information contained in these resources and in particular within several specific references was utilized and supplemented as required in the development of the assessment. Key references used for this study included, but were not limited to, [MPR 2008], [NP2010 2005], [GA 1981], [UE&C 1980], [GCRA 1987], [CDSR 1987]. The information extracted from these references was supplemented as required to complete the analysis. This included an updating of the capital cost estimates for the following two typical commercial applications.

SC-MHR (Electricity) Options: A facility whose purpose is electric power production, using Steam Cycle Modular Helium Reactor (SC-MHR) modules.

COG-MHR (Co-Generation) Options: A cogeneration facility using the same reactor and NHSS design as the electric power production facility. Given the vast range of possible cogeneration applications, an overview of such applications is provided in Section 3.2, with a reference cogeneration configuration selected for more detailed consideration in Section 3.3.

3.1 Cogeneration User Requirements

Co-generation is used in several sectors of the economy and has a wide range of applications. It is predominantly used in the industrial sector, and is also used for heating facilities in the Commercial and Residential Sectors as the basis for district heating[3]. Accordingly, this assessment is focused on the use of cogeneration in the Industrial Sector. References [MPR 2008], [NP2010 2005], [GA 1981], and to a lesser extent [UE&C 1980] address the use of co-generation in the industrial sector.

There is a large variation in the requirements for electricity, steam and process heat among Co Generation user. The references noted above all identify refineries and petrochemical industries as the most significant user. These are followed recovery of oil from tar sands or shale. There is also a large variation in the character and mix of the steam, process heat, and electricity requirements of the various users.

3.1.1 Temperature Range

Figure 3-1 from [MPR 2008] segregates the various industries and processes by temperature range.

3.1.2 Steam Pressure Range

[NP2010 2005] lists the range of conditions for the steam used by cogenerators surveyed as part of the study. On page 1-18, it states "Those that use cogeneration have chemical processes that require process steam at specific temperatures and pressures. Some typical ranges are:

Very high pressure: 1500 psi

High pressure:400 to 600 psi

Medium pressure: 100 to 300 psi

Low-pressure: 30 to 45 psi

3 District heating use is much more prevalent in Europe than the United States.

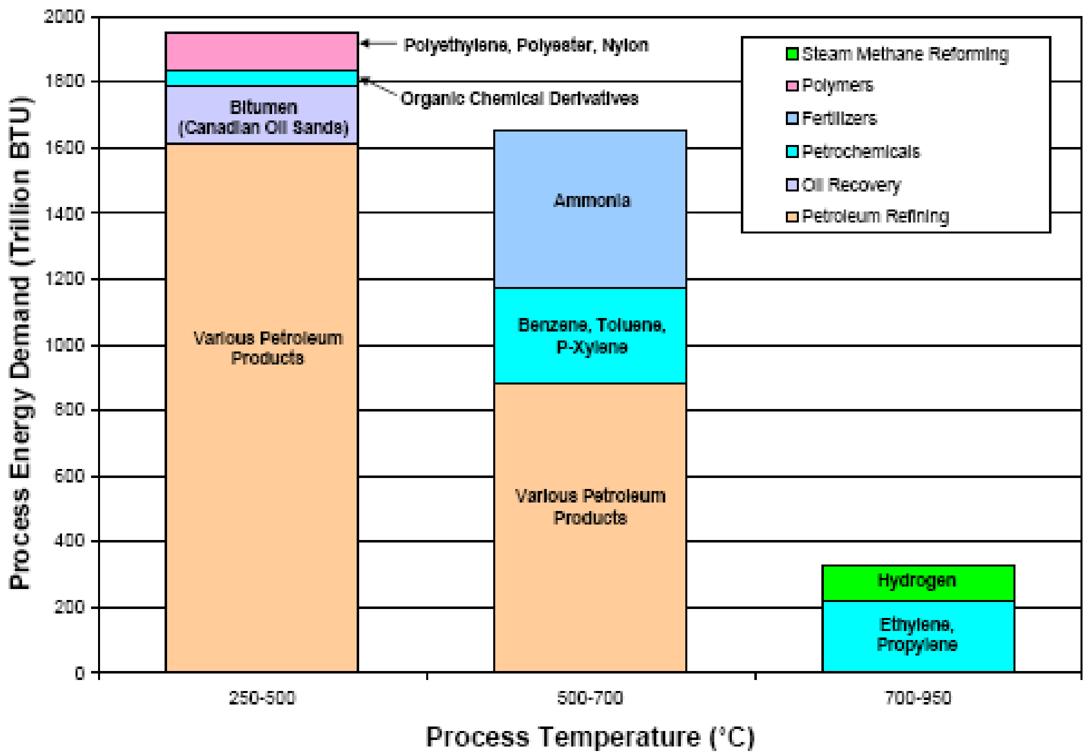


Figure 3-1. Near-term HTGR application annual process energy demand (2000-2007)

3.1.3 Temperature, Pressure and Quantity

The data in Table 3-2 below is extracted from [GA 1981]

Table 3-2. Data on Energy Mix of Various Cogeneration Users

Criterion	Units	Heavy Oil	Chemical Complex	Refinery
Steam required pressure (1)	psia	650	970	1380
Steam required temperature	Degrees F	495	767	845
Steam required quantity	MW(t)	993	2745	1929
Electric power required	MW(e)	1	582	111
Electric power produced	MW(e)	158	582	327
Notes: Highest pressure/temperature condition required for the process shown.				

3.1.4 Other Characteristics and Requirements

It is observed on pages 1-19 and 1-20 in [NP2010 2005] that:

- "… Cogeneration units, by definition are located on or adjacent to the user's site. This eliminates the need for transmission services, which cost about $5 per MWh, or 10% of the total cost of delivered electricity. They also provide protection against future [transmission] rate increases."
- "… an important advantage of cogeneration is reliability, particularly with respect to steam used in continuous manufacturing processes. To ensure this reliability, and users build an extra unit that can be called upon quickly should another unit experience a forced outage. Thus, cogeneration capacity is modular in nature, meaning it is added in the smaller increments."
- "…there is a market for [nuclear cogeneration] but that market will have the following characteristics…
  - outlet temperature is sufficient to generate steam at pressures above 400 PSI
  - ratings of between 200 and 600 MW
  - can be sited on or adjacent to chemical facility located near a major population center

[NP2010 2005] concludes that:

- "… the currently available reactor designs (those with design certifications) would have difficulty meeting…[these characteristics]"
- "There are nuclear plant designs under development that have the potential to meet these needs…"
- "The ideal nuclear cogenerator can be visualized in this way: dismantle the existing gas-fired units and replace them with a nuclear cogenerator and the customer sees no difference in the operation of its manufacturing process."

On page viii, [MPR 2008] states that: "Based on the preliminary energy, reliability and site requirements for near-term HTGR applications, the following conclusions were reached with respect to total thermal plant and module size for typical applications:

- The thermal demand for a typical 200,000 bpd complex coking refinery is approximately 1100 MW(t) ([with an energy mix of] 7% steam, 76% heat, 17% electricity). Refining reliability requirements would suggest that a minimum of three modules be provided."
- The thermal demand for 100,000 bpd of in-situ bitumen extraction is approximately 1270 MW(t) (over 90% steam). Reliability requirements would suggest that a minimum of two

modules be provided. A module size of 400-600 MW(t) could extract approximately 60 to 90 thousand bpd of bitumen."

- The thermal demand for a 100 million [standard cubic feet per day] steam methane reforming unit is approximately 130 MW(t) ([with an energy mix of] 56% steam, 37% heat, 7% electricity). Given a small module size that would be required for this application, it is likely that it would be coupled with other applications such as electricity and steam production for other processes."

Two salient conclusions drawn from the above information are that a nuclear reactor module used in cogeneration applications:

- must be designed to be very robust and flexible to accommodate significant variations in siting requirements
- must have an interface with the balance of plant that allows for a large mix of energy conversion options that will have to be custom designed to accommodate the large variations in cogeneration user energy mixes

The COG-MHR based on GA's NGNP design is well suited for cogeneration applications based on the 600 MW(t) module size, and inherently safe design features that support an Emergency Planning Zone equivalent to the Exclusion Area Boundary that is controlled by the nuclear plant licensee (i.e., the reduced need for emergency evacuation and sheltering provisions accommodates plant siting at an industrial facility even if it is in close proximity to a population center).

3.2 Design, Performance and Cost Characteristics

A large number of BOP configurations are possible in cogeneration applications, and the detailed BOP design is dictated by specific user requirements. For the purposes of this assessment, two configurations compatible with the NHSS conditions are considered. The first configuration is a cogeneration plant providing electricity and low-to-medium temperature steam. This is the "COG-MHR Reference Case" or Case 1. The second configuration is an all electric plant, which provides no process steam or process heat. This is the "SC-MHR Reference Case" or Case 2. The two configurations are summarized below:

- **The COG-MHR Commercial Plant** [Reference Case 1] consists of four NOAK nuclear cogeneration modules. Each module is composed of a single 600 MW (t) MHR and the heat cycle equipment to provide both electricity and steam.
- **The SC-MHR Commercial Plant** [Reference Case 2] consists of four NOAK electricity producing modules. Each module is composed of a single 600 MW (t) MHR and Rankine cycle equipment to provide only electricity.

3.2.1 Development of Overnight Capital Cost Projections for Commercial Plant

3.2.1.1 Approach

For each of the two configurations considered in this study, the modular HTGR is utilized as the source of steam and electric power. The reactor plant and balance-of-plant costs for design, engineering and construction are generated based on available published information adjusted for changes in the conceptual design vis-à-vis established design concepts. The effort relied heavily on previous work completed by GA. The capital cost for each of the commercial plants was developed as follows.

The Overnight Capital Cost for the SC-MHR and COG-MHR were extracted and modified from existing GA Cost Model Information to the basic commercial plant model consistent with the requirements of the current study. The Overnight Capital Cost for the COG-MHR was adapted from the SC-MHR. The adjustments included replacing the SC-MHR turbine plant and supporting system costs with new COG-MHR costs. Most significant of these was the replacement of the single shaft turbine generator in the SC model with two smaller turbine generators required by the COG heat balance; and the addition of a large steam reboiler. Indirect Costs such as Construction Services, Home and Field Office Engineering, and Owner's Cost were extrapolated as per the GA Capital Cost Estimate Directs vs. Indirect.

The resulting cost estimates are summarized below. The cost delta between the two cases is relatively small. As noted, Co-generation plants are custom designed for the needs of the individual user. Therefore, this delta is expected to vary significantly from case to case.

3.2.1.2 Ground Rules

The following ground-rules were used in the development of Overnight Capital Costs:

- Cost estimating dollars referenced to January 2007 date
- Cost estimating reference is Greenfield site located in Kenosha WI. Greenfield site cost excludes site development, temporary or permanent utilities and other services to the site
- Construction cost estimates are the most likely costs (rather than best or worst case)
- Cost estimates assume sufficient funding is available as and when needed to not impact project progress negatively
- Costs assume that on-site, nuclear grade construction is separately managed or physically separated from industrial-grade (non-nuclear) construction
- A nuclear to non-nuclear grade premium will be used
- Project organizational assumptions include:

  - Public/Private Partnership will act as the owner's agent
  - A single subcontractor is responsible for engineering and design, licensing support, manufacturing and construction management activities
- Learning curve reductions are achievable for the following:
  - Manufactured items that are not currently available commercially
  - Field labor on the same site
  - Field labor on different sites
- Engineering for the Commercial Plants is available for efficient planning and execution of construction and startup
- Estimate is based on one shift, five day, 10 hours construction work week
- Exclude costs of
  - waste disposal
  - state and local taxes
  - initial fuel load
  - interest during construction
  - escalation during construction
  - decommissioning costs
- Protective features (e.g. isolation valves, condenser dump capacity, etc.) are assumed to be equivalent among the alternatives.

3.2.1.3 Assumptions

Parametric values assumed for the development and assessment of the commercial models are summarized in Table 3-3.

Table 3-3. Scope & Parameters for COG-MHR NOAK & SC-MHR NOAK Commercial Plant

Parameters	
Project Level – (Organization)	GA Data Base Models
Overnight Construction Scope & Cost (Direct & Support [Indirect] Components) Scope Definition	
1. Production Block/Battery Scope	Included
2. On-Site to Production B/B Scope	Included
3. Annual Plant Capacity Factor	90%
4. Plant design Life	60 Years
1. Construction Material Costs	Note 1
2. Installation Unit Rates	Note 1
3. Construction Labor Unit Costs	Note 1
4. Professional Services Costs	Note 2
5. Construction Estimate Contingency	20%

NOTES:

1. GA NOAK Commercial Unit Model material costs are for Kenosha WI site.
2. For the NOAK Commercial Plants, the engineering and design of the plants is complete. Engineering will be confined to owner's site preparation effort, which will be done before construction of the NOAK Commercial Plant begins.

3.2.2 NOAK Commercial Scale Cogeneration Plant [COG-MHR]

3.2.2.1 Plant Description

This assessment focuses on the characteristics of the commercial plant as a whole as opposed to the individual subsystems and components. The COG-MHR NOAK commercial plant is adapted from the plant described in [UE&C 1980]. A new heat balance and design was developed based on the NHSS inlet and outlet conditions of the 600-MW(t) MHR module as given in Table 3-1, coupled with two steam turbine generators having steam extraction between the high- and low-pressure turbine generators. The extracted steam passes through a "reboiler" type heat exchanger at the cogeneration site boundary. The reboiler provides an additional barrier to tritium that could potentially enter the process steam. The process steam conditions generated by this configuration are the same pressure and temperature as those specified for the Gulf Oil Company's refinery in Port Arthur, Texas and used for the 1170 MW(t) facility described in [UE&C 1980].

The COG-MHR commercial plant consists of four NOAK nuclear cogeneration modules. Each module is composed of a single 600-MW(t) MHR and the heat cycle equipment to provide both electricity and steam.

Each COG-MHR module operates with a reactor thermal power level of 600 MW and supplies helium at an outlet temperature of 750°C to a SG that is coupled to a steam turbine with steam extraction after the high pressure turbine generator. The extracted steam from each module is passed through the reboiler to produce the process steam for consumption by the process plant. The steam produced by each module is collected in a common steam piping manifold and conveyed via pipeline to the user's process plant.

Waste heat is rejected from the COG-MHR modules using cooling towers in a manner similar to that for electricity-producing plants.

3.2.2.2 Nominal Plant Design Parameters

The NOAK commercial plant consists of four modules that provide steam and electricity. Since the user needs an uninterrupted steam supply, only 3 modules would be operating at one time and one module would be on "standby" as a backup. This backup module could be designed to provide either additional steam to the process when needed, or generate electricity that could be sold to the grid while in "steam standby" status.

The nominal plant design parameters are given in Table 3-4 for the condition when the COG-MHR NOAK commercial unit is operating with three modules and the fourth is on standby and NOT generating electricity. On an annual basis, the COG-MHR NOAK commercial plant at a plant capacity factor of 90% produces 17,857,260 tons of steam and 1,887,499 MWh of electricity.

Table 3-4. COG-MHR Nominal Plant Production & Energy Design Parameters

Basic MHR Information		Note	per Module	Total
[1] Number of Modules		A	1	3
[2] Electricity Produced, Net	MWh/yr	B	629,166	1,887,499
[3] Export Steam Energy	MW(t)		515.5	1,546
[4] Process Steam Amount	Tons/yr	B	5,952,420	17,857,260
Thermal Energy Balance (Steam Cycle)	MW(t)			
[1] Module Steam Generator Output	MW(t)		611.0	1,833
[2] Converted to Electricity	MW(t)	C	86.9	261
[3] Transferred to Process Steam	MW(t)		515.5	1,546
[4] Thermal Cycle Losses	MW(t)		8.6	26
[5] Rejected to Cooling System	MW(t)		0	0
[6] Recovered from Steam Process	MW(t)		Included in [3]	Included in [3]
[7] Total of [2] Thru [6]	MW(t)		611.0	1,833
[8] Gross Electricity Produced	MW(e)		92.8	278
[9] Facility Auxiliary Loads	MW(e)	D	13.0	39
[10] Net power to GSU	MW(e)		79.8	239
Notes: [A] Total quantities are shown for 3 operating modules (one on standby) [B] Assumes a 90% capacity factor [C] Net Cycle Output (Gross Output less Thermal Cycle Pump Power) [D] House electrical load is estimated and includes 5.9 MW for Thermal Cycle Pumps and 6.1 MW for NHSS and 1 MW for transformer and electrical losses and other plant use.				

3.2.2.3 Plant Process Heat Balance Diagram

Figure 3-2 shows the process heat balance for the COG-MHR Module.

3.2.2.4 Capital Costs

The overnight capital cost for the COG-MHR NOAK commercial plant is summarized in Table 3-5.

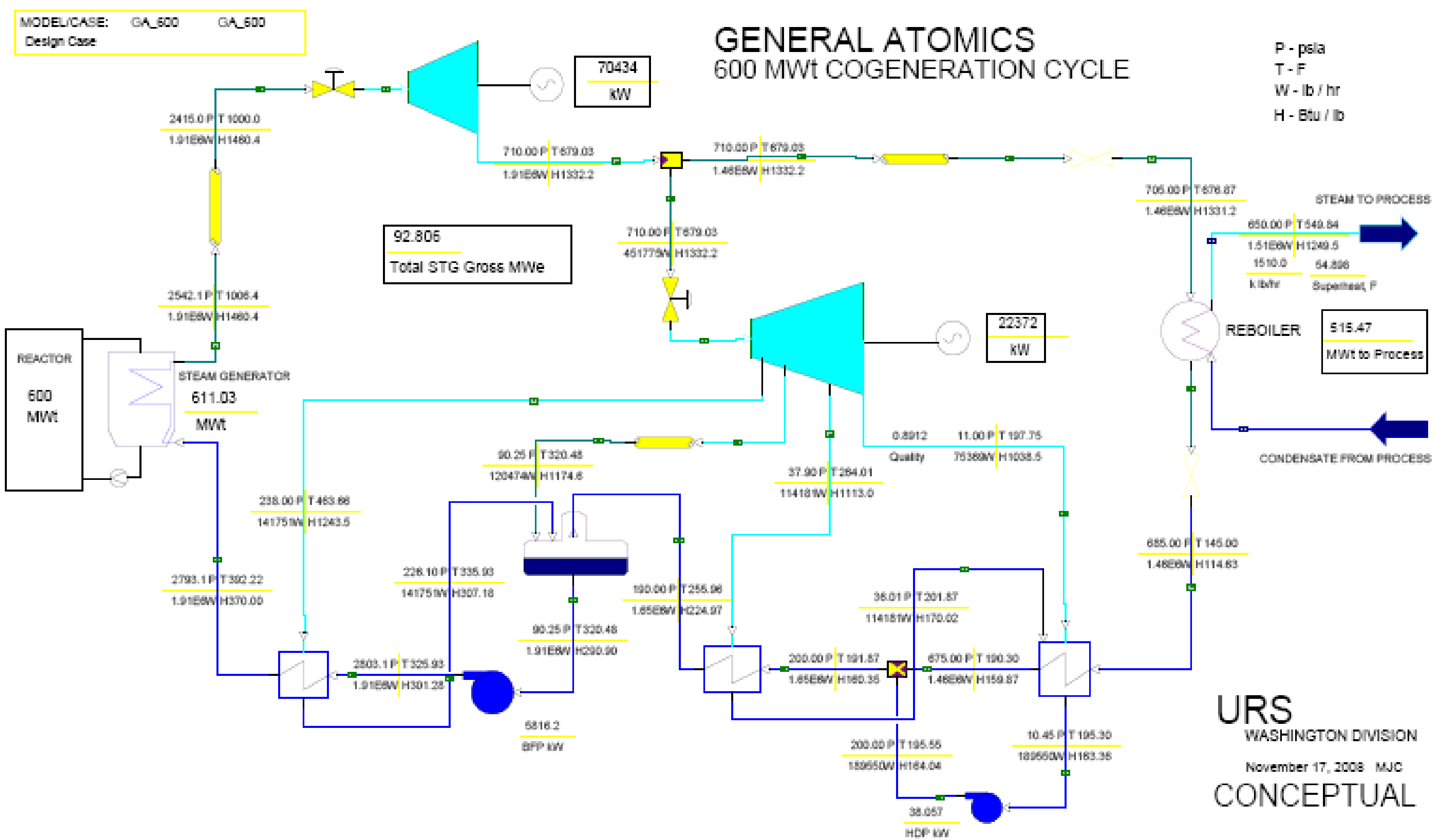


Figure 3-2. COG-MHR process diagram

Table 3-5. Overnight Capital Cost for COG-MHR NOAK Commercial Plant

ACCT	ACCOUNT DESCRIPTION	TOTAL COSTS
	DIRECT COSTS	
20	LAND AND LAND RIGHTS	Not Included
21	STRUCTURES AND IMPROVEMENTS	395,666,178
22	REACTOR PLANT EQUIPMENT	557,224,311
23	TURBINE PLANT/REBOILER EQUIPMENT	291,912,747
24	ELECTRIC PLANT EQUIPMENT	96,122,921
25	MISCELLANEOUS PLANT EQUIPMENT	34,392,646
26	HEAT REJECTION SYSTEM	42,246,358
	TOTAL DIRECT COSTS	**1,417,565,160**
91~9	INDIRECT COSTS	
	TOTAL INDIRECT COST	**668,469,134**
	BASE CONSTRUCTION COST	2,086,034,294
	CONTINGENCY @ 20%	417,206,859
	TOTAL OVERNIGHT COST	**2,503,241,153**

3.2.3 Utility Electric NOAK Commercial Plant [SC-MHR]

3.2.3.1 Plant Description

The nominal design parameters for the SC-MHR components such as the reactor, power conversion system and IHXs have been previously described in other GA reports such [CDSR 1987]. Therefore, this assessment focuses on the characteristics of the commercial plant as opposed to the individual subsystems and components. The published GA reports can be consulted for additional technical detail.

The SC-MHR commercial plant consists of four NOAK electricity-producing modules. Each module is composed of a single 600-MW(t) MHR and Rankine cycle equipment to provide only electricity.

3.2.3.2 SC-MHR Plant Arrangement

The plot plan for the SC-MHR plant is similar to that used in [GCRA 1987] for the SC-MHR Target Commercial Plant.

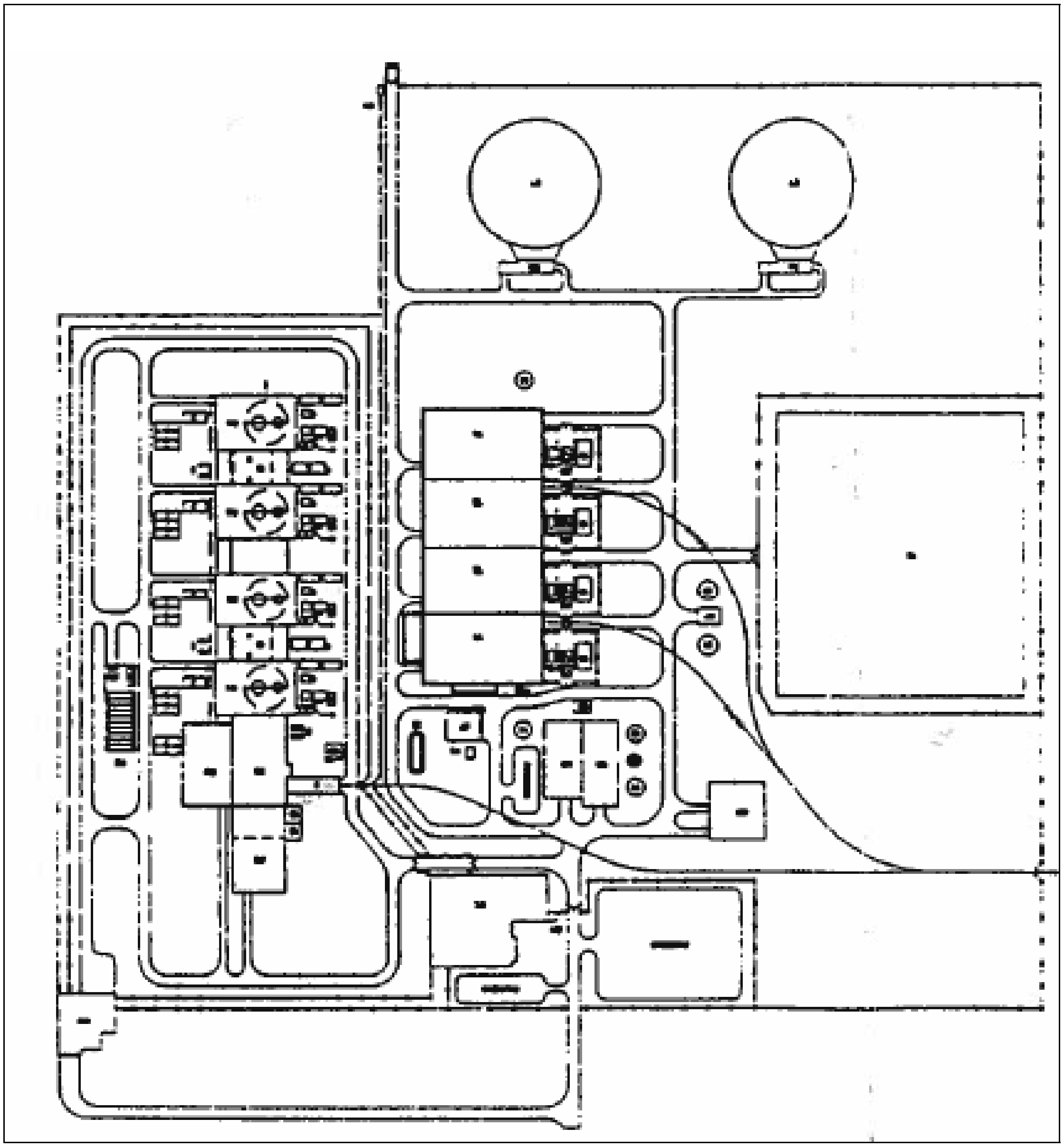


Figure 3-3. SC-MHR NOAK commercial plant arrangement

3.2.3.3 Overnight Capital Costs

The Overnight Capital Costs for the SC-MHR NOAK Commercial Plant are summarized in Table 3-6.

Table 3-6. Overnight Capital Cost for SC-MHR NOAK Commercial Plant

ACCT	ACCOUNT DESCRIPTION	TOTAL COSTS
	DIRECT COSTS	
20	LAND AND LAND RIGHTS	Not Included
21	STRUCTURES AND IMPROVEMENTS	395,666,178
22	REACTOR PLANT EQUIPMENT	557,224,311
23	TURBINE PLANT/REBOILER EQUIPMENT	300,529,575
24	ELECTRIC PLANT EQUIPMENT	99,960,228
25	MISCELLANEOUS PLANT EQUIPMENT	34,392,646
26	HEAT REJECTION SYSTEM	52,807,947
	TOTAL DIRECT COSTS	**1,440,580,884**
91~9	INDIRECT COSTS	
	TOTAL INDIRECT COST	**679,322,463**
	BASE CONSTRUCTION COST	2,119,903,347
	CONTINGENCY @ 20%	423,980,669
	TOTAL OVERNIGHT COST	**2,543,884,017**

3.3 Discussion of Cogeneration BOP Alternatives

As previously noted, there is a great variation in the characteristics of the energy mix of cogeneration users. Essentially, the BOP PCS configurations are custom designed for each installation. A large number of BOP configurations have been developed over the years for industrial, commercial and district heating applications. Consequently, the BOP PCS design is impacted significantly more than the design of the NHSS by the process conditions needed for a specific application.

Because of the large variation in BOP PCS configurations applicable to cogeneration, the following discussion is rather generic. The range of configurations is organized into four general categories. Each of the categories is represented by a single alternative. Reference Case 1, COG-MHR NOAK as described above, is one of these alternatives. It is used as a baseline against which the characteristics of the other three alternatives are discussed.

Three of the alternatives, including the reference case deal only with the production of steam and electricity. These are identified as Alternates A, B, & C.

A fourth alternative has been added because of the importance of the gas-cooled reactor as a high-temperature direct heat source for various process and industrial applications. In the fourth case (Alternate D), the SG provides direct process heat to a particular liquid/gas process stream and steam for the production of electricity to be used in the process or sold to the grid.

3.3.1 Steam and Electricity Production - Alternatives A, B, & C

The specific configuration chosen for the generation of steam and electricity is dependent on both the required steam conditions and the amount of electricity to be generated. Therefore, the first three alternatives are based on a generalized separation of possible steam conditions into three groups. These are

- Medium pressure process steam: (Alternative A, Reference Case 1)
- High pressure process steam: (Alternative B)
- Low pressure process steam:(Alternative C

In Alternative A, medium-pressure process steam is extracted from [a] intermediate stages of the turbine-generator or [b] between different turbine-generators (as shown for the reference case in Figure 3-2). As shown in Figure 3.2, the high-pressure steam is used to make electricity, as well as medium-pressure process steam.

In Alternative B, the high-pressure process steam flow may be extracted from the high-pressure steam turbine discharge as in the reference case or from the steam line between the SG and the inlet to the high-pressure stage of the turbine-generator. In either case, the high pressure steam would be passed through a reboiler to produce high-pressure steam for the user who would also use the electricity that is produced. Excess electricity could be sold to the grid.

In Alternative C, low-pressure process steam is usually extracted either from the low-pressure stages of the turbine-generator or at the turbine-generator outlet. In this configuration, a steam turbine-generator would be included to produce electric power using the high-pressure steam before the low-pressure steam was extracted. The extracted low-pressure steam would pass through a reboiler to produce steam for the process plant. The plant would use the electricity that is produced and extra power could be sold to the grid.

3.3.2 Direct Heating of Process Flow Stream and Electricity Production - Alternative D

In Alternative D, the heat generated by the reactor is used to both directly heat a process flow stream from the process plant; and to generate high-temperature steam to be used in making electricity and medium- or low-pressure process steam that will be used elsewhere in the process plant. In this case, the plant would include a custom-designed high-pressure reboiler. This reboiler would have two sections. The first section would heat the process fluid/gas stream before it returns to the process plant. This second section would produce high-pressure steam for use by the turbine generator and subsequently as low pressure steam.

3.3.3 Performance and Capital Cost Variations

Because the BOP PCS portions of a cogeneration plant are custom designed for each application, the performance is optimized for the cogenerator's specific energy mix requirements. Therefore, the general comparison given below will focus primarily on capital cost variations. Table 3-7 provides the results of the analysis.

Table 3-7. Qualitative Comparison of Overnight Capital Costs for Various Alternatives

	Installed Cost Change Relative to Reference Case	Comments
Alternative A [Reference]	Baseline	
Alternative B	Small Increase	Smaller turbine generator but larger reboiler plus high pressure steam transmission pipeline
Alternative C	Moderate Decrease	Larger turbine generator but smaller reboiler
Alternative D	Small Decrease	Smaller turbine generator but much more complex reboiler
NOTES: Analysis assumes no change in the NHSS and for reboiler located after steam turbine island.		

It should be noted, that for most cogeneration applications, the most important factor on the cost impact for the NHSS is the supplemental design enhancements that will be necessary to match the NHSS to the topographical, geotechnical, meteorological, and other natural conditions of the site in a manner that satisfies the regulators.

3.4 Summary Observations and Conclusions

A discussion of cogeneration alternatives was given in Section 3.3 above. Based on this discussion and other foregoing design, performance and cost information, the following observations and conclusions are offered.

- The essential marketing objective should be to design a nuclear fueled HTGR plant that can be used to replace both the boiler house and the gas fueled heaters located on existing refinery and chemical complexes.
- The key issue is that the NGNP energy source be robust enough so that it can be sited at the locations of current refinery and chemical complexes to facilitate the replacement of boiler houses and natural gas fueled heaters.
- Siting includes issues of population density, geotechnical features, and security requirements.

3.5 BOP PCS Configuration Recommendations

In light of the foregoing, The NGNP should:

- Be sized for 600 MW (t) or less per module
- Use an external Energy Exchanger that allows for steam and process heat applications
- Utilize an external Energy Exchanger whose final design varies with and is tailored to any BOP.
- Be characterized by its flexibility in siting:
  - Population density,
  - Geotechnical features
  - Security requirements
- Be characterized by flexibility in serving a large variation in PCS/BOP requirements:
  - Industrial process heat
  - Industrial cogeneration and
  - District heating

The PCS/BOP selection should be made to accommodate the process mix need of the Industry Financial Backers when the NGNP program can show that the HTGR NHSS modules can be sited and licensed at their existing facility.

3.6 Other Considerations Not Addressed

There are some areas where additional sensitivity analysis might be of value, but these areas were not within the scope of this effort. One of these areas is an assessment of the impact of a CO_2 penalty on the price of natural gas and its use as a cogeneration fuel and feedstock.

4 COMBINED-CYCLE PCS STUDY

This study was performed by Rolls-Royce. Because the purpose of the NGNP is to demonstrate technologies that are attractive for deployment in commercial reactors, it was decided that the logical approach for this task would be to first determine the most attractive combined cycle option for a commercial plant and then assess how the technology could best be demonstrated in NGNP. Thus, the scope of the task included the following elements.

- Evaluate indirect combined cycle options for a commercial plant and select the preferred option
- Compare cost and performance estimates for the preferred indirect combined cycle option with a direct combined cycle option, and make a recommendation on the best option for a commercial plant
- Determine how the technology for the recommended commercial plant option could best be demonstrated in the NGNP
- Perform an assessment of the technology readiness level (TRL) of the direct and indirect combined cycle options

The results of the study are presented in Rolls-Royce Report DNSD146266 [Rolls-Royce 2008]. The information presented in the following sections has been excerpted from the main body of the Rolls-Royce report with only minor editorial changes. Additional details concerning cycle modeling assumptions, net electrical efficiency calculations, and the compressor and turbine mechanical work are provided in Appendix A and B of the Rolls-Royce report.

4.1 Choice of Performance Cycle

This section describes the development of the indirect combined cycle starting with prior work on gas turbine power conversion cycles for the Modular Helium Reactor.

4.1.1 Previous Work

Two cycles have previously been developed as potential solutions for the power conversion unit of the current reactor design. Both were direct cycles, hence included a gas turbine in the primary circuit. The earlier of the two designs, the GT-MHR cycle, was an intercooled and recuperated closed Brayton cycle with helium as the working fluid. The minimum helium pressure in the cycle was 25 times atmospheric with an overall pressure ratio of 2.8. Power was generated from a gas turbine (~300MW) in the primary circuit. The cycle had a high net electrical efficiency of 48% at a reactor outlet temperature of 850°C.

In 2007, a second cycle was developed by Rolls-Royce [PCDSR 2007]. This was done in an attempt to further improve the efficiency and to mitigate some of the risks perceived in the GT-MHR design. The cycle proposed was a direct combined cycle. The reactor outlet temperature was 850°C and to reduce the temperature to a level acceptable to the steam equipment, a gas turbine was installed in the primary circuit. This gas turbine had an overall pressure ratio of 1.87 giving a steam turbine inlet temperature of 580°C. The gas turbine power was ~50MW and the steam turbine power ~250MW. The predicted net electrical efficiency was 50.2%.

4.1.2 Development of the Indirect Cycle

Following from the work described above, the current project is to develop an indirect cycle of optimum efficiency. An indirect cycle is achieved by introducing an IHX and moving the turbo-machinery into a secondary loop. The coolant in the primary loop is now driven by a motor driven circulator.

The move to an indirect cycle provides the flexibility to select a more appropriate working fluid for the gas turbine circuit. To the first order, the choice of gas in the gas turbine circuit does not affect efficiency from a cycle viewpoint. There are second order effects, such as the pressure drop in the heat exchangers, which for a heavier gas, with a lower specific heat capacity will be higher. However, the choice of the working fluid is dominated by turbo-machinery considerations and as described in Section 4.4, the most appropriate working fluid was decided to be a mix of nitrogen and helium in an 80/20 proportion by mass. The primary coolant remains helium.

In the development of an optimum cycle, several different power conversion unit designs were considered. These included an indirect Brayton cycle, an indirect pure steam cycle, and several indirect combined cycles; one of which was an indirect version of the 2007 direct combined cycle. The potential of each cycle was evaluated and the more promising designs modeled using the performance package IPSEpro. From this down-selection process, it was decided that efforts should be concentrated on the development of an indirect version of the 2007 direct combined cycle.

During the development process, the efficiency was seen to be highly sensitive to the temperature difference at the ‘pinch point’ in the SG economizer. From an assessment of the SG in 2007, 22°C was determined to be an achievable temperature difference with a reasonably compact SG. It will therefore be important for the 22°C economizer temperature difference to be achieved in order for the target efficiencies to be reached in the operational plant.

The introduction of the IHX also has an effect on the cycle efficiency. This must be considered in the IHX design since for the same type of heat exchanger, the size is directly related to the temperature and pressure drops across it. A larger unit should provide smaller temperature and pressure drops and a more efficient cycle for a fixed reactor outlet temperature. Of course, a larger unit will be more expensive, especially considering that the IHX must be located within the containment building. A 50°C temperature drop was considered a reasonable cycle assumption offering a compromise between efficiency and IHX size and cost.

The indirect combined cycle enables the primary and secondary circuit mass flow rates to be controlled independently. It was therefore possible to optimize the design points for reactor outlet temperatures of 850°C, 900°C and 950°C by varying the primary circuit mass flow rate. For each case, the gas temperature rise through the core was restrained to remain in the range 360°C - 460°C. It was found that the optimum efficiency is achieved where the reactor temperature rise temperature is close to the minimum allowable.

4.1.3 Description of the Optimized Indirect Combined Cycle

The description of the proposed indirect combined cycle presented here is based on a reactor outlet temperature of 850°C, which provides a net electrical efficiency of 48.6%. 900°C and 950°C reactor outlet temperatures were also modeled; the associated efficiencies for the direct and indirect cycles are shown in Table 4-1. The direct combined-cycle values are quoted from [Rolls-Royce 2007]

Table 4-1. Direct Combined and Indirect Combined-Cycle Efficiencies

Reactor Outlet (°C)	Direct (%)	GT Inlet (°C)	Indirect (%)	Indirect Cycle Penalty (%pts)
850	**50.2**	800	**48.6**	1.6
900	**51.4**	850	**49.3**	2.1
950	**52.4**	900	**49.8**	2.6

A diagram of the proposed indirect combined cycle with optimized operating conditions shown for 850°C reactor outlet temperature is shown in Figure 4-1.

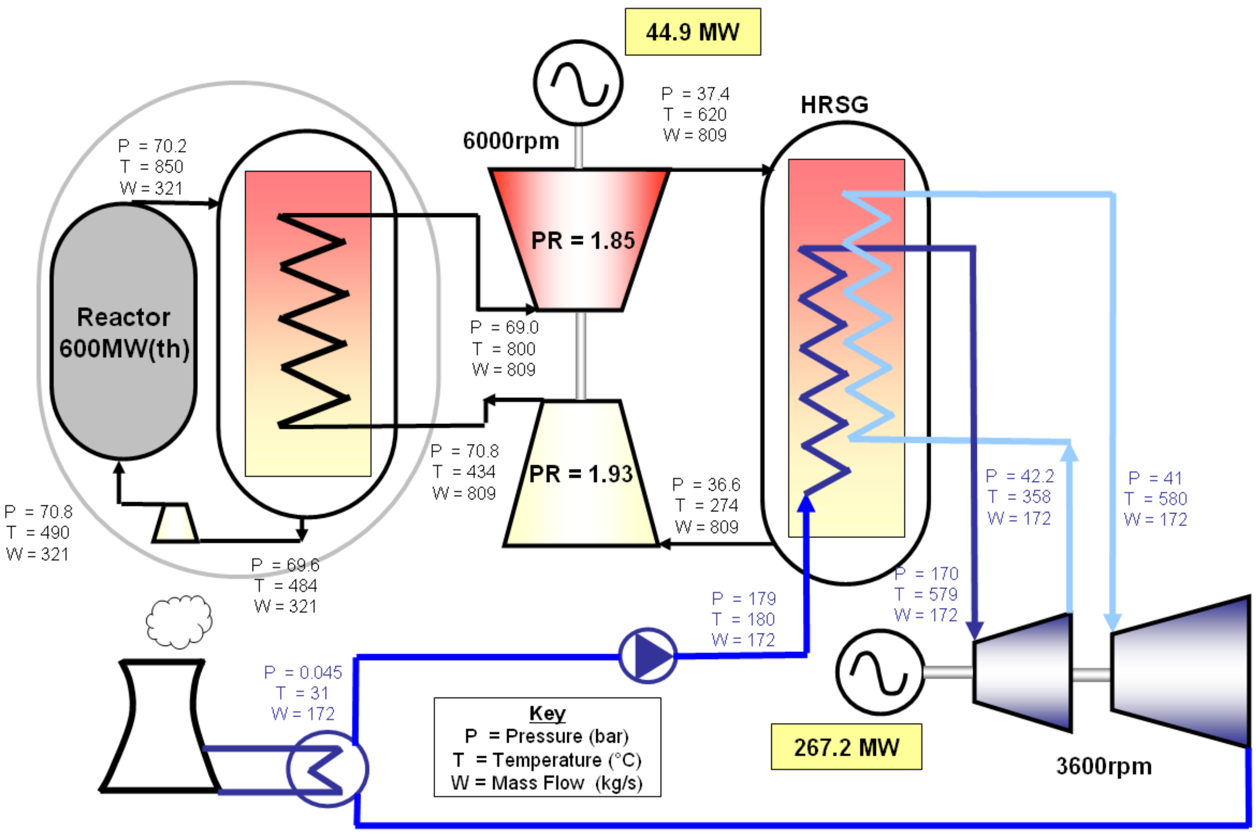


Figure 4-1. Diagram of indirect combined cycle with 850°C core outlet temperature

The cycle is similar to the direct combined cycle described in [Rolls-Royce 2007] but includes an IHX as described in Section 4.7. The cycle operates with a minimum pressure of 36.6 bar and an overall pressure ratio of 1.85. For a fixed reactor outlet temperature, the indirect cycle has a 50°C lower gas turbine inlet temperature compared to the direct cycle. Therefore, in order to maintain the steam turbine inlet temperature at 580°C, the work done and power generated in the gas turbine is less than for the direct cycle. However, the steam circuit mass flow rate is increased slightly compared to the direct equivalent because of the heat introduced to the primary circuit by the circulator. This serves to slightly increase the proportion of work done in the steam circuit, which takes more advantage of the phase change in water.

Controlling the primary and secondary circuit mass flow rates independently enables a higher reactor inlet temperature to be achieved compared to the direct cycle. For the indirect cycle an inlet temperature of 490°C can be achieved (compared to 437 °C for the direct cycle). The mass flow rate through the gas turbine is larger than for the direct cycle due to the lower specific heat capacity of the nitrogen/helium mix compared to helium. This increases the pressure drop in the heat exchangers compared to the direct cycle to 1.8 bar; up from 0.6 bar.

4.1.4 Evaluation of the Indirect Combined Cycle Compared to Direct Combined Cycle

For the 850°C cycle, the 1.6% reduction in the efficiency of the indirect cycle compared to the direct cycle can be attributed to the following factors:

- Disadvantage of a reduced gas turbine inlet temperature: -0.7%pts
- Disadvantage of Increased pressure drop in heat exchangers: -0.6%pts
- Disadvantage of the power consumption of the primary circuit circulator: -0.83%pts
- Advantage of the optimization of the primary circuit mass flow rate: +0.53%pts

Overall, the total reduction in efficiency is smaller than may have been expected. The expected losses due to the above items are offset in part by the advantage gained in being able to control the primary and secondary circuit mass flow rates independently.

4.2 Off-Design Performance

4.2.1 Introduction

The IPSEpro tool was used for modeling the design point of the cycle. This tool, however, is incapable of modeling the cycle at off-design conditions. Nevertheless, it is possible to predict the likely performance of the indirect combined cycle at off-design conditions by drawing on information from two sources. The first of these is [Rolls-Royce 2007]. The second source is the Rolls-Royce Aero-engine Performance synthesis program (RRAP) modeling work that was done in 2007 on both the GT-MHR and direct combined cycles. The indirect combined-cycle has enough in common with these other cycles that useful conclusions can be drawn and issues that require further attention can be highlighted.

4.2.2 Review of 2007 work on GT-MHR and Direct Combined Cycles

The 2007 modeling of the GT-MHR and direct combined cycles drew the following key conclusions:

- Performance of the GT-MHR cycle on a hot day was compromised as it appeared that, in addition to the normal 'Carnot' effects, the reactor power would have to be reduced. Similar effects were also observed if it were necessary to de-rate the reactor to run at reduced outlet temperature.
- In comparison, the direct combined cycle appeared to offer better hot day performance. This is because the steam mass flow can be varied to control the helium temperature at compressor entry and hence the reactor is 'unaware' of the ambient temperature change. It was speculated that this extra control variable would also provide benefits at other off-design conditions.
- The GT-MHR cycle bypass valve would need to pass around 45% of the main flow in order to reduce the net power on the gas turbine shaft (i.e. the power developed in the turbine less the power absorbed in the compressor) to zero. This action would be required during the start sequence and also in the event of the power station being dropped by the grid.
- The starting sequence proposed in the supplied GT-MHR documentation appeared to be feasible and there were no threats to compressor stability at part load.

4.2.3 Indirect Combined Cycle at Part Load

It is suggested that inventory control should be used as the main strategy for running at part load. This has the advantage of maintaining turbo-machinery non-dimensional operating points and cycle temperatures and thus keeps cycle efficiency high. To both minimize the pressure differential across the IHX and maintain the required relationship between reactor power and reactor temperature rise, the inventory would also need to be reduced in parallel in the reactor circuit.

It is believed that it will be essential to have control over the gas turbine pressure ratio. This is so both the net power on the gas turbine shaft and the gas temperature into the top of the boiler can be controlled. Since the gas turbine is to be connected to the main steam turbine/generator shaft (through a clutch) the gas turbine shaft will, when the machine is synchronized, run at constant speed. Controlling pressure ratio across a compressor running at constant speed requires the use of either variable geometry in the compressor or a bypass valve (similar to that proposed in the GT-MHR cycle). It is considered very unlikely that variable compressor geometry alone would offer the required control. It is therefore concluded that a bypass valve, between compressor outlet and turbine outlet, will be required.

By extrapolation from the 2007 RRAP work on the GT-MHR cycle it is clear that the by-pass valve would need to pass a substantial flow. It would need to be of sufficient size that the net power on the gas turbine shaft could be reduced to zero during the startup sequence and in the event of the power station being dropped by the grid. The pressure ratio of the indirect combined cycle is lower than that of GT-MHR and this means the bypass flow would probably need to be in excess of 45% of the 809kg/s secondary circuit flow. This is because, at lower pressure ratios, the ratio of compressor power to turbine power is lower. Therefore, the net power is a large fraction of the turbine power and the turbine power thus needs to be sharply reduced to reduce the net power to zero. The practicalities of the bypass valve passing such a large flow, while also being able to exert fine control over the speed of the shaft when it is not synchronized, will need careful consideration.

4.2.4 Indirect Combined Cycle on Hot Day/Cold Day

On a hot day the steam flow can be varied which means that the compressor inlet temperature can be controlled. This in turn means that the primary and secondary circuits would be unaffected by day temperature, and there would be no need to reduce reactor power on a hot day. Although the net output and efficiency of the cycle would be reduced on a hot day, due to the increased condenser back-pressure, it is expected that reductions would be in line with those of conventional fossil fired combined-cycle plant. Similarly, on a cold day, improvements in efficiency and net electrical output would be expected to be broadly in line with conventional combined-cycle plants.

4.2.5 Indirect Combined Cycle – Transients

The most onerous transient is loss of grid. In order to prevent gas turbine/steam turbine shaft over-speed the net torque on the shaft will need to be dropped to zero very rapidly. For the steam turbine this can be done in the conventional way by slamming a stop valve closed and dumping steam. For the gas turbine the by-pass valve will need to be fully opened rapidly. This places a further design requirement on the by-pass valve in addition to those noted above.

For other transients a combination of inventory control and by-pass flow can be used to vary power output whilst maintaining compressor stability. An advantage of the indirect combined cycle over the direct combined cycle is that the IHX acts as a ‘damper’ between the reactor and gas turbine circuits. This means that the reactor is somewhat isolated from gas turbine transients and also, the gas turbine is isolated from reactor transients. This should make control of transients, and especially fault transients, easier to manage.

4.2.6 Indirect Combined Cycle – Starting

A starting sequence has been proposed based on RRAP modeling of the GT-MHR starting sequence. In general, the indirect combined cycle has more variables which can be controlled than was the case for the GT-MHR cycle. For example the reactor, gas turbine and steam circuit flows can all be independently varied. It is concluded that the availability of extra control variables must make starting easier than for GT-MHR. Since no issues were identified which would make starting the GT-MHR difficult, it is further concluded that starting the indirect combined cycle is unlikely to present insurmountable problems.

4.3 Technology Demonstration in NGNP Program

The exact configuration of the NGNP demonstrator plant is yet to be fully defined but it is believed that the plant is likely to be configured to produce process heat and steam. It is clear, however, that a commercial plant for generating electricity would almost certainly have a gas turbine included to maximize plant efficiency. It would therefore be desirable if the gas turbine technology could be demonstrated in some way in the demonstrator plant.

This section seeks to answer the following questions:

1. What would be the efficiency of a pure steam turbine plant with no gas turbine?
2. What is the relationship between reactor outlet temperature and efficiency for an indirect combined-cycle plant? Down to what reactor outlet temperature is it worthwhile or feasible to include a gas turbine in the plant?
3. How could a gas turbine be included into the NGNP demonstrator plant?

4.3.1 Efficiency of a Pure Steam Turbine Plant

The current state of the art for steam turbines is for steam inlet temperatures of 600°C, the limiting factor being the properties of the materials used in the thick-walled headers, pipes, valves and rotors. The indirect combined cycle presented above assumes a slightly conservative steam inlet temperature of 580°C, with a maximum steam pressure of 180 bar. The cycle is reheated at 40 bar, also to 580°C.

When a pure steam cycle is modeled in the IPSEpro tool at the same steam conditions, a net electrical efficiency of 42.6% is predicted. The efficiency of a steam cycle is very dependent on the condenser pressure. This in turn depends on how the condenser is cooled and the temperature of the air or water used to cool it. The steam cycle model was developed during the 2007 study and the aim was to compare against the Brayton cycle concept on a basis that would not unfairly favor either. The Brayton cycle assumed that cooling water was available which could cool the helium in the precooler and intercooler to 26°C. Achieving this would probably require a coastal location in a fairly cool climate. To give a fair comparison a condenser pressure of 45 mbar, achievable in a similarly cool coastal location, was specified in the steam cycle.

It is of interest to compare this efficiency prediction with modern fossil-fired steam plants. A literature search has shown that state of the art steam plants can have net electrical efficiencies well in excess of 40%. Denmark, in particular, has a number of very high performing plants which benefit from coastal locations and low sea temperatures which allow very low condenser pressures (down to 23 mbar). A net electrical efficiency of 47% is claimed for the Nordjylland 3 plant with a double reheat steam cycle operating at 'ultra-supercritical' steam conditions of 580°C and 290 bar. There are a number of other examples around the world where net electrical efficiencies well in excess of 40% are claimed. It is therefore concluded that the steam cycle efficiency of 42.6% modeled here is reasonable.

When comparing the efficiencies of the various cycles the relative values are more important than the absolutes. It is explained above why the steam cycle is considered to be modeled on a fair basis against the original intercooled and recuperated closed Brayton cycle. When modeling steam only and combined cycles, care has been taken to ensure that the steam parts of the cycle are consistent, i.e., they produce the same power output per unit heat input. Therefore, although the efficiency of the steam plant would vary depending on climate and how the condenser is cooled, it is expected that the relative efficiencies of steam only and combined-cycle plants would stay constant.

In the IHX options study report [GA 2008a], a steam inlet temperature of 540°C was assumed. IPSEpro modeling suggests that, at a fixed steam pressure, this would penalize the net electrical efficiency by around 0.3% pts compared to a 580°C inlet temperature.

4.3.2 Reactor Outlet Temperature and Value of Gas Turbine

4.3.3 Feasibility of Including a Gas Turbine in the NGNP

The IHX options study report by GA presents two possible layouts for the NGNP demonstrator plant, the serial HTS (Heat Transport System) configuration and the parallel primary loop configuration. Both of these assume the inclusion of a hydrogen production plant and therefore have high reactor outlet temperature (900°C). Both cycles assume the inlet temperature to the steam plant is 540 °C.

A plant with a gas turbine inserted would ideally have the following characteristics:

- The gas turbine would operate at conditions (i.e. temperature, pressure, pressure ratio) consistent with those foreseen for the commercial plant.
- The gas turbine would operate with the helium/nitrogen mix proposed for commercial plant.
- If the gas turbine were not available, this should not prevent the running of the demonstrator plant in other modes.

- The plant should be designed so that a gas turbine is not required at the outset but could be fitted part way through the demonstrator program.
- The plant should be designed so the gas turbine could be inserted at minimum additional cost.

These requirements have been reviewed against a number of possible layouts for the demonstrator plant which include a gas turbine. The most promising of these is based on the parallel primary loop configuration and is described in the subsection below.

4.3.4 Proposal for Including a Gas Turbine in the NGNP

The parallel primary loop configuration is shown in Figure 4-3. There are two primary loops, one which provides 65 MW of heat to a hydrogen plant and the second which provides steam. The second of the primary loops exchanges heat through an IHX to a helium secondary circuit. The IHX drops 200°C, and a further 160°C is dropped across the SG in order to provide 540°C steam at the steam turbine inlet. Helium is circulated at 70 bar in the secondary circuit by an 11 MW circulator.

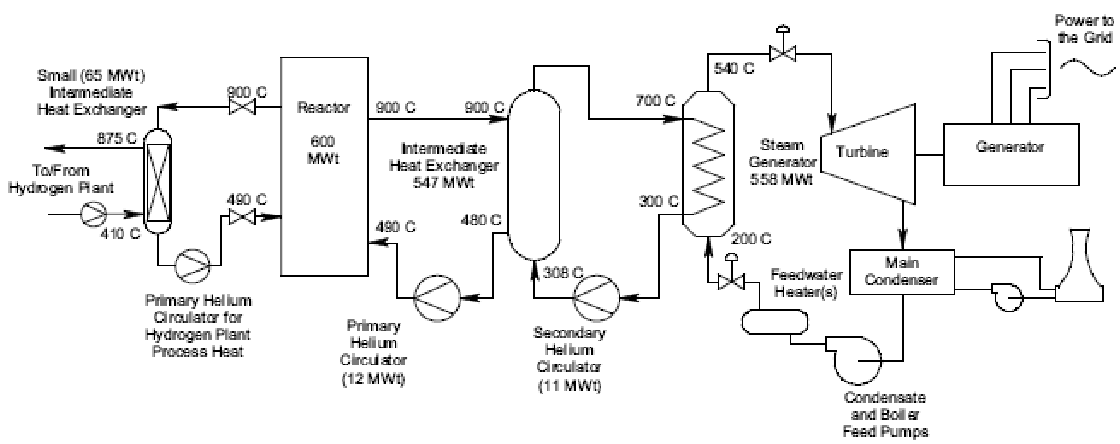


Figure 4-2. Parallel primary loop configuration from IHX alternatives study

The following changes to this configuration are proposed to allow a gas turbine to be included:

- Replace the helium in the secondary circuit with a helium/nitrogen mix, so that when the gas turbine is included, it can run on the correct working fluid.
- Insert a larger/more efficient IHX between the primary and secondary circuits to reduce the temperature drop across it. This opens up a temperature difference between which the gas turbine can operate.

- Insert a larger/more efficient SG. This again opens up the temperature difference between which the gas turbine can operate.
- Insert a cooler in the secondary circuit between the IHX and the SG. The purpose of this is to reduce the temperature at the SG inlet to a satisfactory level in the event that the gas turbine is not operational.

The gas turbine can then by installed by bypassing the cooler on the hot side with the turbine and bypassing the circulator on the cold side with the compressor. The layout is shown in 4-4. The layout and conditions when the gas turbine is operational are shown as solid lines. The dashed lines and boxes show the layout and conditions when the gas turbine is not operational.

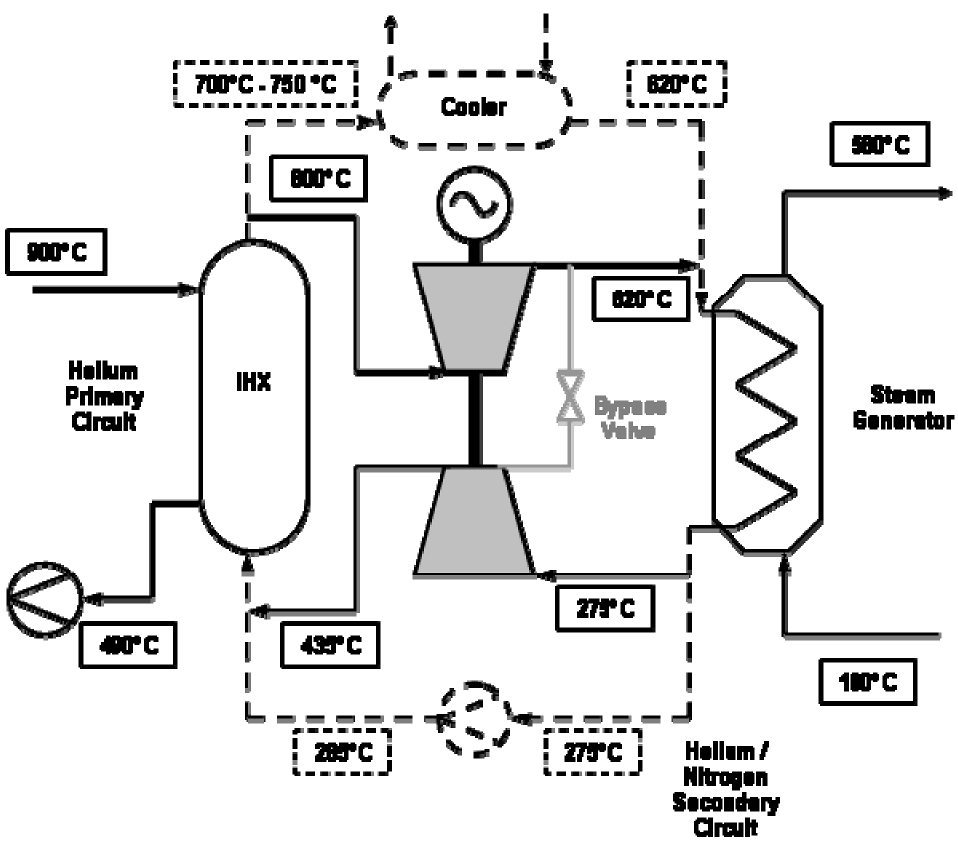


Figure 4-3. Proposed layout for inclusion of gas turbine in the NGNP

This layout would cost more than the existing parallel primary loop configuration even without the cost of the gas turbine and its ducting. This is because the heat exchanger and SG would need to be larger (to give smaller temperature differences and because the working fluid is no longer pure helium), and the cooler is an additional component. The heat exchangers would also need to be carefully designed so that they could operate satisfactorily in both operating regimes.

The pressure on the cold side of the IHX would be 70 bar when the gas turbine is in operation. This means that the pressure differential introduced by the gas turbine around the secondary circuit does not affect the IHX. When the gas turbine is not operational, the whole secondary circuit can be pressurized to 70 bar. When the gas turbine is operational, the pressure at the SG will of course be lower (around 37 bar), but it is not unusual for SGs to operate at pressure differentials such as this.

The proposed layout is very flexible because it allows the plant to operate either with or without the gas turbine in place. The addition of the cooler allows the temperature at the SG inlet to be controlled, which means that the steam inlet temperature can be controlled independently of the reactor outlet temperature. This gives extra flexibility when commissioning the plant and would allow it to be tested over a wider range of conditions.

The layout also allows the gas turbine plant to be tested at exactly the conditions envisaged in a commercial plant, with temperatures and pressures being reproduced correctly. The inclusion of the bypass valve allows that component to be properly tested and fault conditions, such as dropped load, could be simulated.

4.4 Secondary Working Fluid

A number of pure gases and gas mixtures have been assessed as options for the secondary working fluid. It can be relatively simply shown that the “ideal” gas requirements for the turbo-machinery and IHX are contradictory; compact and efficient turbo-machinery requires a higher density gas for lower volumetric flow rate while a heat exchanger requires a gas with good thermal conductivity. Issues such as corrosion, oxidation and nitriding can also have a significant impact on gas selection. Preliminary investigations into a number of pure gases suggested that most would result in complex turbomachinery or an excessively large IHX.

4.4.1 Choice of Working Fluid

A study was conducted into the choice of secondary working Fluid. This study involved looking at the size and nature of the turbine for each of the following fluids. From a turbomachinery point of view the relative merits of each are briefly summarized below.

- Helium. A helium turbine is large, has many stages and a low expansion through the turbine. The development of a helium turbine has additional costs due to the lack of experience of designing turbines for noble gases.

- Argon. Argon is similar thermodynamically to helium, but denser. It would result in a much smaller turbine with one third the number of stages. It is widely available as an industrial gas. The development of an argon turbine would be similarly expensive.

- Xenon. Xenon is also similar to helium and would result in an even smaller turbine. The development of a xenon turbine would be similarly expensive. Furthermore, xenon may have disadvantages due to the possibility of poisoning the reactor in the event of a leak

.

- Carbon Dioxide. This is widely used in current reactor designs as a primary coolant; however, there is little experience in gas turbine turbomachinery. It would result in a large turbine with a high expansion ratio and would be difficult to design.

- Air/Nitrogen/Nitrogen-Helium Mixtures. An air turbine is roughly half the size and half the number of stages of a helium turbine. There is a wealth of experience in designing air turbines and therefore it would be a much lower risk option. Adding helium to the system has not got any advantages to the turbo-machinery but has significant advantages elsewhere in the secondary system.

As a result of this study a mixture of helium and nitrogen by mass was selected as the secondary working fluid. From a turbomachinery perspective a mixture of nitrogen and helium behaves in exactly the same way thermodynamically as pure nitrogen; however, an overall system benefit was found to be achievable by introducing some helium. This benefit arises because of the contradictory nature of the dependence of the IHX and turbo-machinery components on gas properties.

Selection of an approximate 80/20 by mass nitrogen/helium gas mixture is recommended on the basis of balancing risk in turbo-machinery and the IHX and reducing cost. Introduction of a small mass inventory of helium results in turbo-machinery key parameters similar to existing Rolls-Royce aero and energy products, but significantly reduces the IHX size. Further discussion is detailed in Sections 4.5 and 4.7. It should be recognized that the exact specification of the working fluid will vary depending on the final detailed cycle and component design; however, the completed analysis indicates that a gas composed of 20-30% helium by mass offers significant benefits.

4.4.2 Gas mixture properties

Detailed consideration has been given to gas mixtures of helium and nitrogen. Some academic literature identifies an inconsistently large improvement (relative to the helium inventory) from the addition of helium into other gases. Data from selected academic papers has been analyzed to determine underlying non-dimensional heat transfer functions, and it has been demonstrated that the perceived performance improvements result from systematic error in the experimental method. Independent models of helium/nitrogen gases based on approximation methods of Wilke and Wassiljewa at various helium inventories were produced to estimate transport properties, and the effect on both turbo-machinery and IHX assessed.

An overall system benefit was found to be achievable by introducing approximately 20% helium by mass. This benefit arises because of the contradictory nature of the dependence of the IHX and turbo-machinery components. Mass and density dependant properties of the gas rise linearly with the mass inventory of helium in the gas mixture. Conversely, expansion ratio (γ) and thermal conductivity are dominated by gas kinetics and hence rise linearly with the helium molar fraction.

Mass and density effects primarily affect the turbo-machinery stage count and achievable velocity ratios; hence, a linear increase in turbo-machinery stages is observed relative to the helium mass inventory. Compressor annulus area is governed by expansion ratio hence rises linearly with helium molar fraction. This increase in area can be effectively managed to maintain blade aspect ratios by increasing compressor mean line. Figure 4-5 demonstrates the dependence of compressor stage count on gravimetric helium inventory.

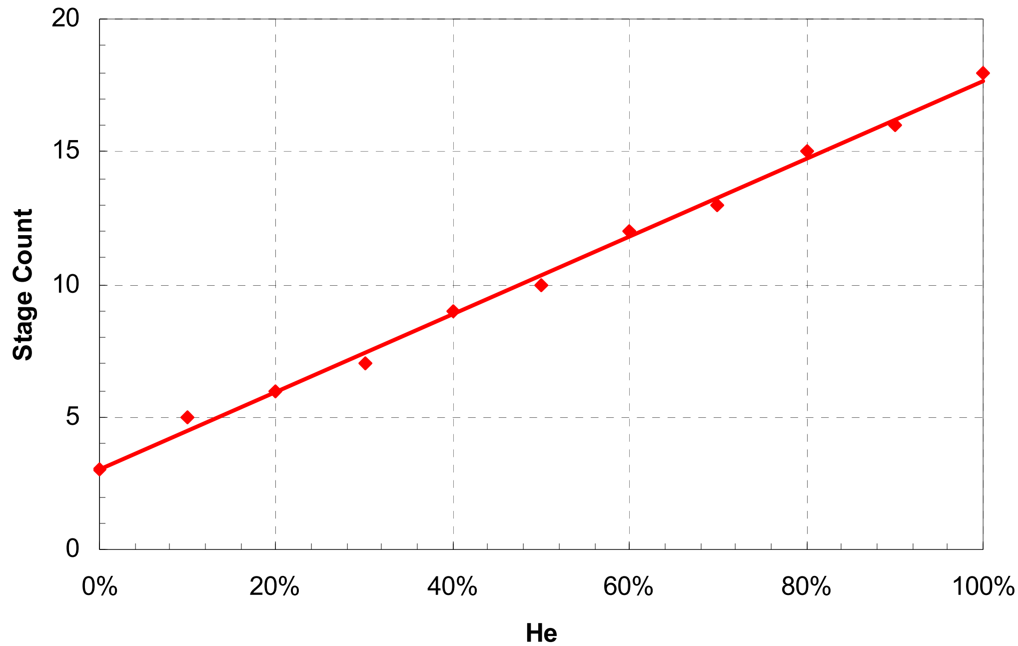


Figure 4-4. Compressor stage-count relative to gravimetric helium inventory

Heat exchanger sizing is affected by both volume flow rate and thermal conductivity; however, it is apparent that thermal conductivity dominates within the examined working range. Intermediate working fluids are exchanged on the basis of constant fluid heat capacity (mc_p). Figure 4-6 illustrates the normalized reduction in heat exchanger size from the introduction of helium to the working fluid.

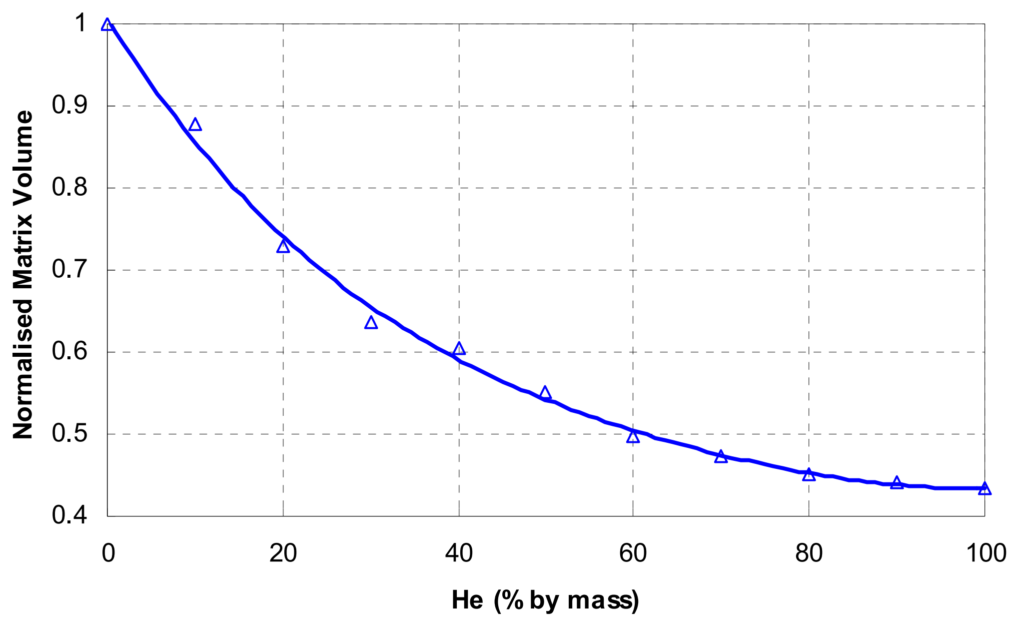


Figure 4-5. Normalized matrix volume relative to gravimetric helium inventory

4.5 Compressor and Turbine Aerodynamic design

The compressor and turbine aerodynamic design are discussed in this chapter. Section 4.5.1 describes the compressor aero design and Section 4.5.2 the turbine aero design. Following chapter will then discuss the mechanical design aspects. Choosing a turbomachinery working fluid that is much more similar to air than the helium required for a direct cycle gives turbo machinery designs that are very similar to Rolls-Royce gas turbine experience. The design task is also much less of a stretch for Rolls-Royce's well established aerodynamic design tools. The turbomachinery risks are therefore significantly reduced with the indirect combined cycle.

4.5.1 Compressor Aerodynamic design

The philosophy of the aerodynamic design of the compressor is to base the NGNP design on an existing design that has been developed for an aero engine application; for this study we have chosen the 6 stage Trent aeroengine family style High Pressure Compressor. Using an existing and well proven design will significantly reduce risks.

A conventional Trent compressor uses air as the operating fluid. The change from air to Helium/Nitrogen mixture changes the gas properties of operating fluid which will result in the compressor having less overall compressor pressure ratio (CPR) capability, this though is not a problem as the cycle requires a lower CPR so fits well with this design concept. To have a compressor that is strongly based on an existing design, the basic aerodynamic properties must be obtained. These are individual stage enthalpy rise divided by the square of stage mean blade speed ($\Delta H/U^2$) and the stage inlet axial velocity divided by the stage mean blade speed (Va/U). By maintaining these parameters the blade mean radius geometries will be very similar to the original compressor but will be operating at lower Mach numbers as a result of the speed of sound of the mixed gas being higher than in air.

Once the above design is completed the compressor is scaled such that it will operate at the required inlet non-dimensional flow parameter, massflow times the square of the temperature divided by the total pressure ($M\sqrt{T}/P$) and the revolutions per minute (RPM) is changed to maintain the original hub line blade speed (Uhub). A further correction is made to achieve the required design speed of 6000 rpm. This is achieved by moving the hub line of the compressor to a new position such that Uhub is maintained. With this new hub line the casing line is moved to maintain the flow areas through the compressor.

Table 4.2 illustrates that the above process gives a compressor that is very similar to a Trent compressor. This can be seen by the mean aerodynamic parameters and blade deflections being the same. The mean axial Mach numbers are lower for the mixed gas design, because of the gas properties being different; this will give a slight efficiency advantage. The different gas properties and operating point between the two compressors does result in a lower exit hub/tip radius ratio and high mean blade aspect ratios[4]; these again will give an efficiency advantage.

Table 4-2. Comparison of Trent Compressor and Suggested He/N_2 Mixture Compressor

Property	Trent Compressor	Mixed Gas Compressor
Mean $\Delta H/U^2$	0.443	0.443
Mean Va/U	0.6	0.596
Mean Rotor deflection	19.4	19.4
Mean Stator deflection	32.1	32.1
Mean axial Mach number	0.42	0.275
Inlet hub/tip radius	0.813	0.806
Exit hub/tip radius	0.921	0.851
Mean Aspect Ratio	0.6	0.888

4 Ratio of the height of the blade to the blade chord where the chord is the length of the blade between the leading edge and trailing edge

In summary, because this compressor is so closely related to a well developed gas turbine compressor, there would be very high confidence of a “right-the-first-time” design. Indeed, rig validation testing may well not be required to be certain of achieving design point efficiency and surge margin. This would save significant technology development costs compared with a helium direct-cycle gas turbine.

4.5.2 Turbine Aerodynamic Design

4.5.2.1 Previous Helium Turbine Design for the Direct Cycle

Rolls-Royce has previously studied turbomachinery options for a helium cycle gas turbine. This included an evaluation of the OKBM GT-MHR direct cycle concept, a simpler more efficient Rolls-Royce design for the GT-MHR and a Rolls-Royce combined-cycle concept. To understand the benefits of the current design for a Nitrogen-Helium turbine, it is important to understand the design drivers behind the previous work.

Helium has a very high specific heat capacity and a high ratio of specific heats (gamma). Together, these properties mean that the speed of sound in helium is very high. However, it has a very low molecular weight and hence is not very dense. This choice of working fluid therefore drives the design in the following ways.

- As the helium is not dense, a large flow area is required and the axial velocity of the helium is very high.
- As the velocity of the helium is high, the blade speed needs to be high. This pushes the design towards a fast shaft speed and a high diameter turbine.
- The high gamma means that a large amount of work can be extracted from the fluid with like density change. This means there is a small change in flow area through the turbine.
- The low density gas means many blades are required for each stage and the shape of each blade is unconventional.
- The large, high speed turbine means the mechanical design of the blades and disks is challenging. This forces design choices such as blade cooling, thermal barrier coating and exotic materials.

Despite these challenges, a suitable design for the direct combined cycle was found to be a five stage turbine rotating at 5000 rpm with a diameter of around 1.5 m. Although the cycle design has changed, this turbine performs roughly the same power output as the current design. It is therefore useful for comparison with the Nitrogen-Helium design described in section 4.5.2.2.

4.5.2.2 Turbine Design for Indirect Cycle

A realistic, achievable concept design for the turbine has been completed. The design has the following features:

- To provide the smallest turbomachinery and to match the requirements of the compressor design, a high shaft speed of 6000 rpm was selected.
- Two stages
- Diameter of 1.2 m
- Shroudless blading is used. This keeps costs down and simplifies the mechanical design for a very small performance penalty.
- The outer hade line is parallel, giving good sealing and good tolerance to large axial movement.
- The vortex is optimized to keep the first stage cool and to help the exit diffuser.
- The aerodynamics are chosen to keep blade stress to a reasonable level. Hence no thermal barrier coating or blade cooling is required.
- The turbine is lightly loaded and an efficiency of 90% should be possible.
- As shown in Figure 4-7, the blade design is conventional and would look very similar to an aero engine.

Hence, in comparison to the helium turbine described in 4.5.2.1 above, the nitrogen-helium turbine is one-eighth of the weight, with only two stages (compared to five), considerably simpler with no blade cooling and existing design tools are available. It is therefore a very favorable design.

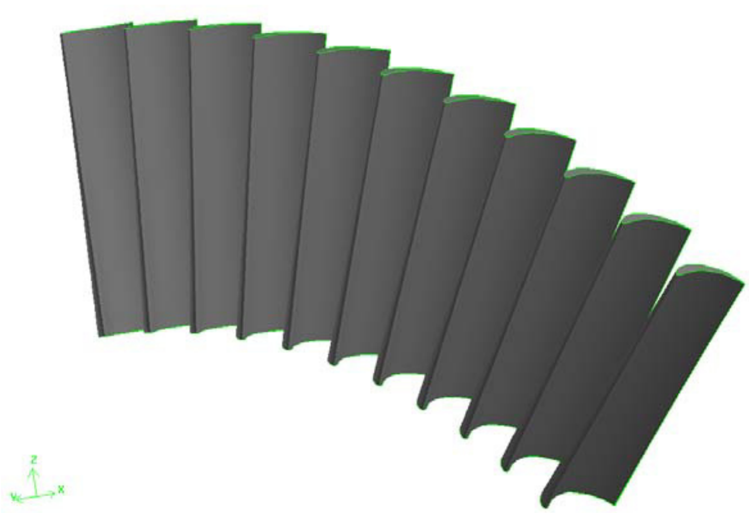


Figure 4-6. Suggested turbine blade design

In summary, similar to the compressor design the turbine design is closely related to a conventional well developed gas turbine, there would be very high confidence of a 'right first time' design. Indeed, rig validation testing may well not be required to be certain of achieving design point efficiency and surge margin and again would save significant technology development costs compared with a helium direct-cycle gas turbine.

4.6 Compressor and Turbine Mechanical Design

The compressor and turbine mechanical design are discussed in this section. Section 4.6.1 describes the compressor mechanical design, Section 4.6.2 the turbine mechanical design, and Section 4.6.3 discusses the selected compressor and turbine materials

4.6.1 Compressor Mechanical Design

For the compressor mechanical design Rolls-Royce has completed an assessment of a six-stage compressor operating at 6000 rpm (see Figure 4-8). In both cases the design and stress analyses used gas paths from Rolls-Royce compressors which apply Trent style 'architecture'. The latter is the preferred compressor solution aerodynamically operating within a N_2/He gas closed circuit.

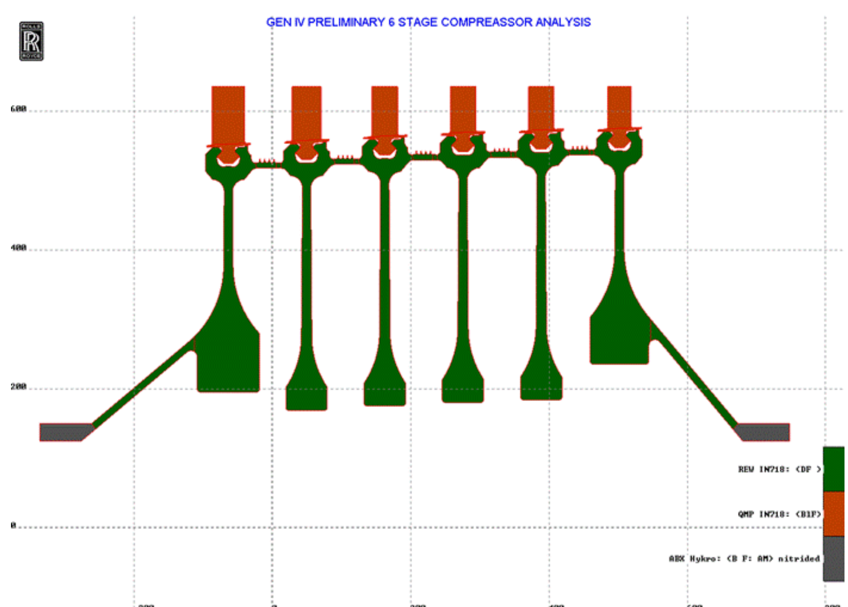


Figure 4-7. Six-stage compressor operating at 6000 rpm

The design of both the turbine and compressor is driven by the creep life requirements to achieve 18-months operation which allows repair and overhaul activity during the period when the nuclear plant is being refueled. This requires a life of 13,000 hours and an assumed minimal number of cycles. Therefore, the objective with the compressor has been a life of 13,000 hours or a multiple (e.g. 26,000 hours) to allow a modular maintenance regime.

For the compressor temperatures and Creep Factor (CF) loads, a range of materials have been considered, also noting the cost implications. For the blades INCO 718 has been considered as the most suitable material, with discs either in INCO 718 or possibly Titanium IMI834 at the forward end. Titanium is a more expensive solution and would lead to a more complex bolted rotor with the added risk of difficulties in designing out the consequential thermal fight at the joint. Hence, an all INCO solution would be preferable.

For the six-stage compressor with INCO blades and discs, an initial Finite Element (FE) analysis has been completed. This has shown that suitable RF values for burst, rim peel and creep over the defined temperature gradients are achievable within the design concept. The preliminary rotor-dynamic analysis results indicate consideration is needed in the next design phase regarding the bounce and bending mode frequencies (from Campbell diagram analysis around 100 Hz) created with the 6000RPM operation. This would have to be re-assessed during a refinement of the rotor support bearing stiffness in the design.

From the assessment of the I_P/I_D (Polar Inertia/Diametral Inertia) relationship the compressor ideally should be lengthened slightly. The addition of a balance piston for rotor thrust compensation would also help to improve the result.

Hence, in summary, the six-stage compressor concept is mechanically viable to suit this application.

4.6.2 Turbine Mechanical Design

The aerodynamic design for the turbine favors a two-stage solution at 6000 rpm (see Figure 4-9). A mechanical study has been completed with turbine inlet temperatures from 750°C to 850°C, with an expected load of 226 MN and a life of 13000 hours. The turbine blade material is expected to be 50°C lower than the turbine entry temperature.

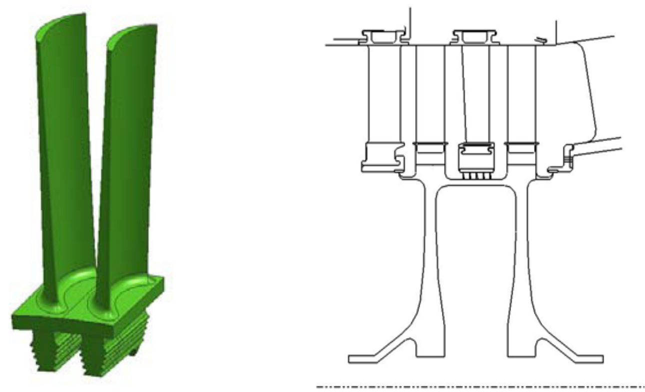


Figure 4-8. Two-stage turbine solution at 6000 rpm

In order to maximize the chances of having acceptable creep life without blade cooling, an unshrouded turbine blade has been selected. With this configuration, with the slightly cooler turbine operating temperatures than with a direct cycle, acceptable creep life looks possible without the need for blade cooling, even at 950°C reactor outlet temperatures. The proposed design has 79 blades for each stage. The blade has been supported by a Trent style root and a layout familiar to this type of design.

The blade material selection process has been completed in relation to the range of temperatures and the creep requirements. In completing this assessment creep of 0.1% and 0.2% have been considered. For energy applications 0.2% is considered to be appropriate. In all cases to achieve these requirements, aerospace type Nickel alloy material solutions have been identified as the most appropriate, using either Direction Solidification blade (DS) or single crystal solutions. The objective has been to use uncooled blades to allow a less complex design solution. At 750°C DS (M002) materials may be used; however, beyond 800°C Single Crystal solutions (CMSX4) are more suitable and at 850°C CMSX4 or higher temperature RR3000 (a Rolls-Royce developed high temperature material) may be considered.

Udimet 720 Li has been applied as the most suitable material for the Turbine disc in the expected temperatures. The Turbine blades and disc have been assessed using FE modeling against the initial layouts and material choices shown. The assumed temperature distribution (Figure 4-10) and worst X-Y principal stresses at 6000 rpm (Figure 4-11) are shown below.

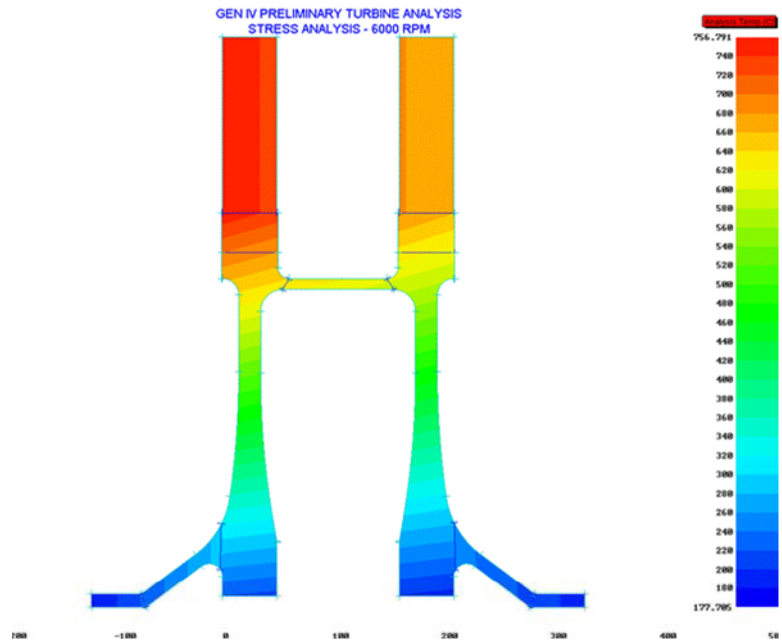


Figure 4-9. Temperature distribution

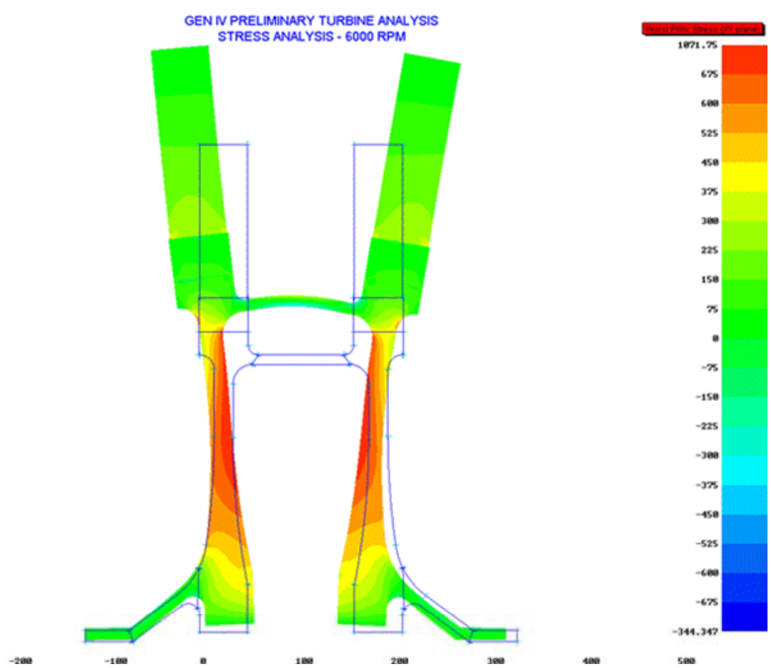


Figure 4-10. Worst X-Y principal stresses

The rotor dynamics of the turbine with the associated design layout and Campbell analysis have shown an acceptable solution with stiff bearings at this stage of the concept. The first engine mode at 6000 rpm from the analysis is below the pitch, bending and bounce modes.

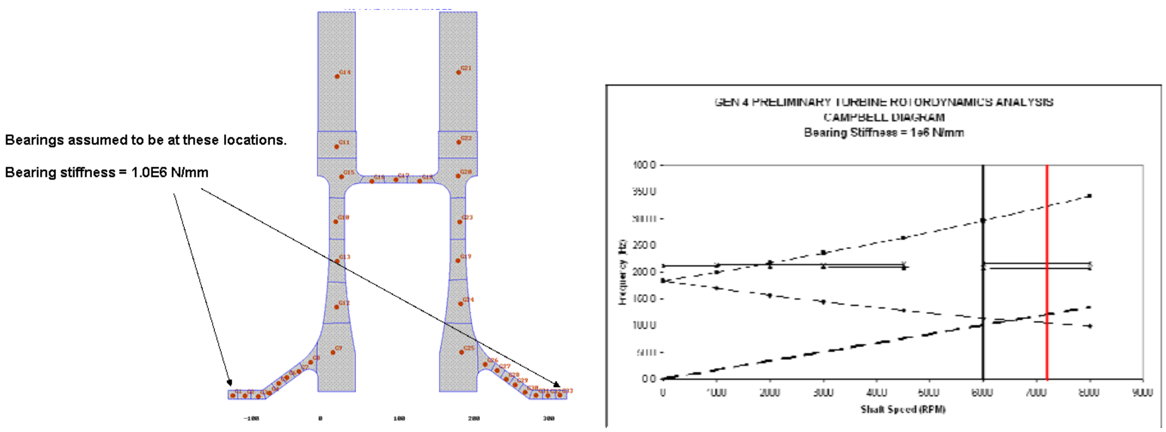


Figure 4-11. The first engine order at 6000 rpm, pitch, bending and bounce modes

4.6.3 Compressor and Turbine Materials

Consideration has been given to the materials being applied to both the Compressor and Turbine, the implications of use within a N_2/He closed cycle environment and when mixed with oil within the closed cycle. With the solutions being proposed Nickel is a significant constituent but also with major traces of Chromium, Iron and Cobalt. Other minor traces include Tantalum, Tungsten, Niobium, Titanium, Molybdenum, Aluminum, Carbon, Boron and Zirconium. The assessment completed by a literature search of academic material considered the implications with oxidation and corrosion.

It was found that the oil, helium, and nitrogen would not contribute to the corrosion of the compressor and turbine materials. Greater corrosion risks are presented by contaminants or impurities such as steam or air, with the main corrosion mechanism being carburization and decarburization. Because the nitrogen/helium mix will have fewer impurities than air, corrosion and nitriding are considered likely to have less impact in this environment than in a normal land-based or aero gas turbine.

Further consideration would be needed if a direct cycle were to be used because of the implications of turbomachinery contamination by fission products.

4.7 IHX - Choice and Sizing

Although the IHX was not part of the Rolls-Royce work scope, some preliminary investigations were performed to reach conclusions on the choice of best cycle and best working fluid for the secondary cycle.

The IHX design options are discussed in this Section. Section 4.7.1 discusses the heat exchanger type, Section 4.7.2 discusses material selections and Section 4.7.3 discusses the sizing of the IHX for various nitrogen/helium gas mixtures.

4.7.1 Heat Exchanger Type

The design operating conditions derived from the cycle performance model infer that several heat exchanger styles may be appropriate for the IHX. During design-point operation, the heat exchanger is subjected to a relatively small pressure differential at elevated temperatures, thus plate-fin and tube-and-shell designs were considered.

Cursory analysis of transient conditions and failure cases, however, suggests that more rigorous design criteria exist to handle development of a severe pressure differential across the heat exchanger boundary arising from a loss of secondary coolant. The severity of this pressure differential is likely to exceed the design pressure capability of a plate-fin heat exchanger and result in significant damage to the matrix and loss of primary coolant boundary integrity.

Nuclear Regulatory Commission (NRC) requirements for inspection during manufacture of heat exchangers providing the primary coolant boundary within high-temperature gas reactors are presently unknown. Plate-fin heat exchangers require significant welds and lack capability for full inspection of jointed sections. Conversely, modern welding techniques for tube-and-shell heat exchangers produce a welded joint that is fully inspectable.

It is therefore suggested that a tube-and-shell style heat exchanger is the most robust solution capable of meeting both operational, regulatory and safety design requirements. Accordingly, Rolls-Royce has limited its IHX evaluation below to a shell-and-tube style heat exchanger.

4.7.2 IHX Materials

At the elevated temperatures and long service duration required, progressive inclusion of nitrogen into heat exchanger materials may result in embrittlement and a reduction in structural integrity of duplex steels. The significant material volumes in both the pressure vessel and header assembly may be sufficient such that nitriding results in only a marginal and manageable reduction in overall design strength; hence, the use of duplex steels may be permitted to reduce cost. The relatively thin walls of the tube assembly may suffer a significant reduction in strength, and thus it may be necessary to utilize nickel alloy materials (such as Alloy 800H) in their construction.

4.7.3 IHX Sizing

Preliminary estimates of heat exchanger size have been completed on the basis of the indirect 850°C cycle. The estimates assume a heat exchanger composed of numerous 1” outer diameter tubes with a 5% wall thickness. The heat exchanger is based upon a counter flow arrangement using the ε-NTU method, and no specific modeling of entrance or exit losses has been conducted. The specified design fluid containing 20% helium by mass reduces the heat exchanger volume by 27% from pure nitrogen relative to a 56% reduction for a pure helium working fluid. It has also been demonstrated that heat exchanger size is strongly dependent on the allowable pressure drop, selection of which has implications for the performance of the plant cycle. The figure below illustrates variation of normalized matrix volume with pressure loss. Data from the Areva PCS study is marked in for reference, as well as the Rolls-Royce assumption. It should be noted that some differentiation occurs from the use of helical tubes in the Areva study against straight tubes in the simplistic Rolls-Royce model. Further assessment is required to strengthen the confidence in these numbers.

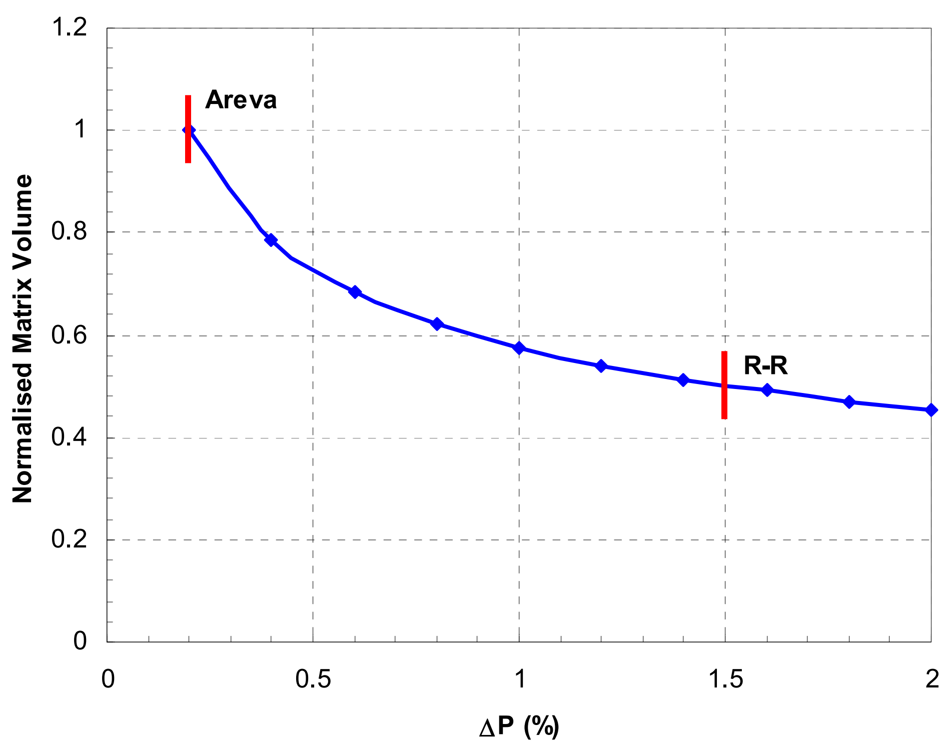


Figure 4-12. Normalized matrix volume scaling

It would also be possible to adopt a modular heat exchanger arrangement consisting of a number of smaller heat exchangers. Due to the timescale and because the IHX design is not included in the Rolls-Royce work scope for this project, this was not investigated.

4.8 Plant Layout

The suggested plant layout and arrangement is discussed in the following subsections. The overall site layout is shown in Section 4.8.1. Other subsections discuss layout of the ducting, Reactor, IHX, Containment Building, power generation equipment and water system.

4.8.1 Site Layout

An illustration of a potential plant layout is shown in Figure 4-14. Although full optimization of the layout was beyond the scope of this study, the proposed layout is considered to be sensible. Where sufficient information was available to approximate equipment dimensions quickly and easily, this has been done.

A workshop, stores, control room, office building is included for completeness, but no electrical substation has been included. Also, no refueling management and spent fuel handling buildings have been included.

4.8.2 Ducting

Concentric ducts have been assumed to carry the helium to and from the reactor although the benefits of this have not been assessed.

Pressure losses in the system are considered to be critical in achieving the required performance so the ducting has been sized to give velocities below Mach 0.1 where possible.

The ducts to and from the IHX are concentric with the lower temperature compressor exit flow on the outside. Further study is required to assess if concentric ducts is the optimum solution with stresses in the ducts being weighed against the complexity and stresses in the transition casing.

The pipes from the low pressure (LP) steam turbine have been limited to 1.5m diameter for practicality which results in a velocity of Mach 0.4. This is considered acceptable as the pipes are straight and quite short resulting in a small pressure loss.

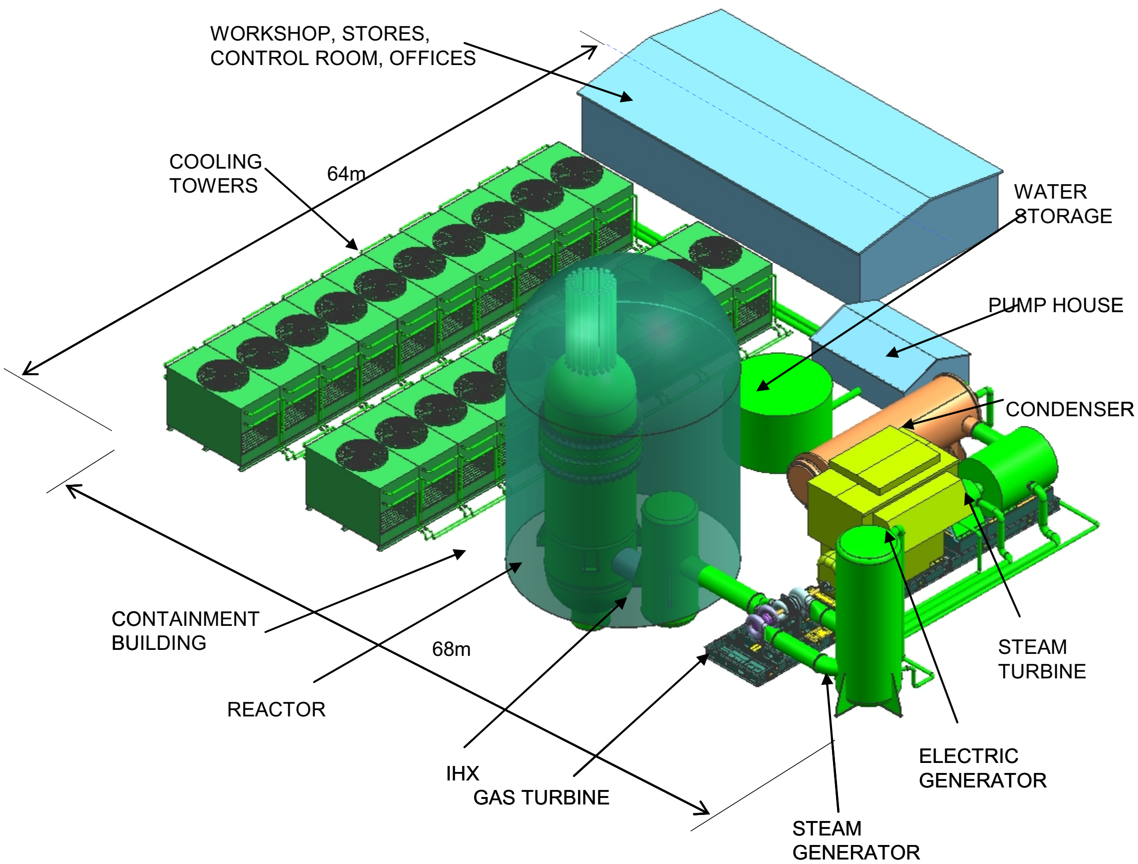


Figure 4-13. Plant layout

4.8.3 Reactor, IHX, and Containment Building

The reactor dimensions have been taken from the direct cycle design study and measures approximately 24m tall (not including the control rods) and 7-m diameter.

The IHX dimensions have been approximated to suit the matrix dimensions defined in section 4.7 and are approximately 11-m tall and 4-m diameter to accommodate the pressure vessel and headers.

A booster compressor/circulator, not shown, is required between the IHX and reactor to circulate the helium and recover the pressure from 69.6 bar at IHX outlet to 70.8 bar at reactor inlet. This may necessitate separation of the ducts. Both the reactor and IHX will be installed within a containment building. The containment building has been sized to fit over the reactor and IHX leaving some space for additional equipment if required including multiple IHXs.

4.8.4 Power Generation Equipment

The compressor and turbine dimensions approximate to initial gas path estimates but no detailed design has been undertaken.

The compressor outlet scroll will need to incorporate an effective diffuser at the front end and some further diffusion in the scroll is still probable. Careful design will be required to minimize losses. Similarly, the turbine exhaust scroll and diffuser will need careful design to ensure pressure recovery is maximized and will need to be structural to withstand the 37 atm pressure. The compressor and turbine inlet scrolls will have some contraction so design should be more straightforward.

A double ended generator is assumed where the gas turbine drive is at one end and the steam turbine drives from the opposite end via a synchronous self shifting (SSS) clutch to prevent the gas turbine from trying to drive the steam turbine. The gas turbine is connected to one side of the electrical generator via a gearbox and the steam turbine to the other. The gearbox is shown diagrammatically mounted onto front of generator to give the 6000 to 3600 rpm speed reduction between the gas turbine and generator. Vertically offset parallel shaft gears are used to allow for the differing centerline heights of the gas and steam turbines.

The generator is scaled to 300 MW size from a Brush 100 MW machine

The steam generator (HRSG) shown is based on the direct cycle design and measures approximately 5-m diameter and 14 m tall. The steam turbine comprises a high pressure (HP) turbine and twin LP turbines. Steam, having passed through the HP turbine is re-circulated through the HRSG before passing to the LP turbine. The length of the steam turbine has been estimated at approximately 10 m.

4.8.5 Water Systems

Steam from the LP turbine outlet passes through a condenser with the outlet water being re-pressurized before returning to the HRSG. The condenser shown is a “Basco” type shell and tube cross flow design but no analysis has been done to determine the size. The cooling water system includes 20-off Baltimore AirCoil (BAC) closed circuit coolers, water tank and treatment/pump house.

4.9 Technology Readiness and Development Requirements for Key Technologies

For each key technology for both the direct and indirect cycles, a summary table is given that shows current estimated Technology Readiness Levels (TRL) using the definitions given in INL/EXT-08-14251 (from which Figure 4-15 was taken). The reasoning behind the TRL estimates is given along with the anticipated development work required to advance the TRL.

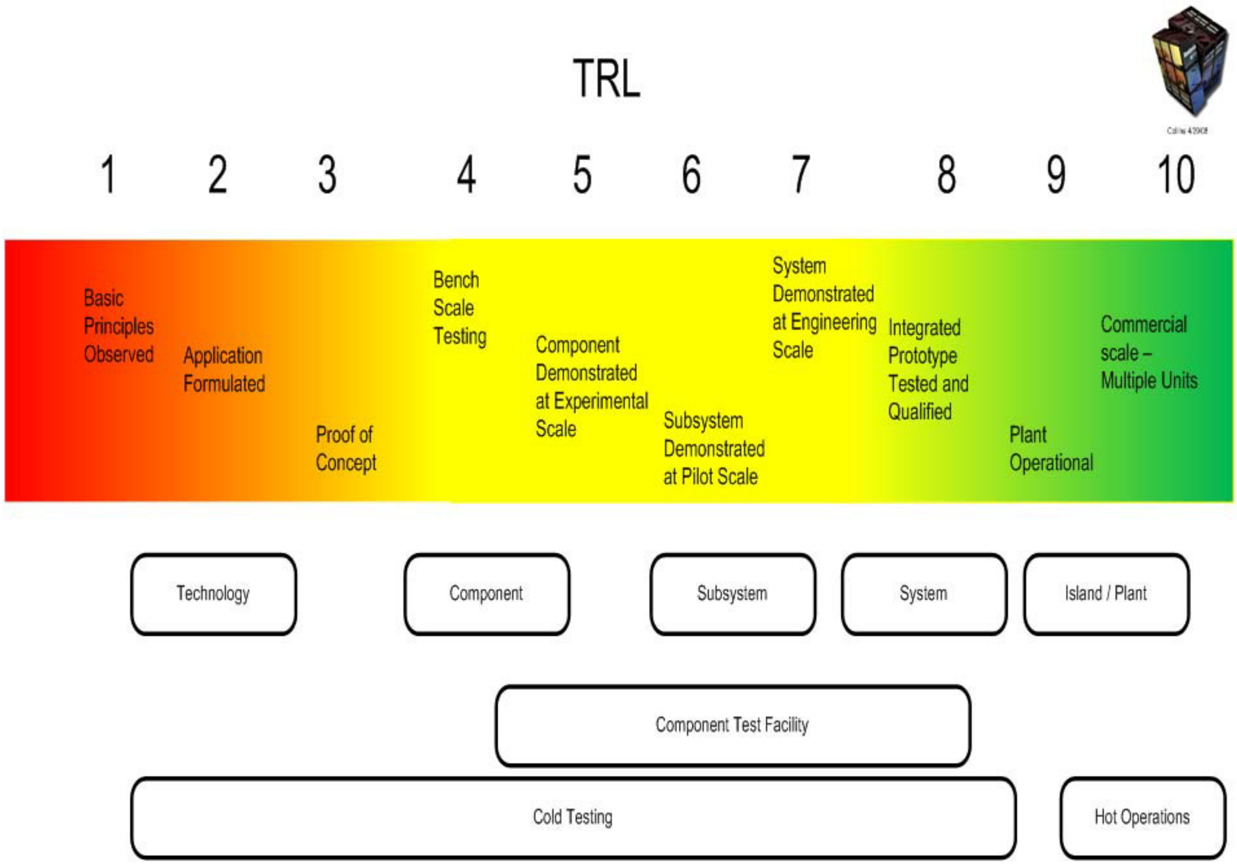


Figure 4-14. Summary definition of TRLs

Table 4-3. Turbomachinery Aerodynamics TRLs

	Direct Combined Cycle	Indirect Combined Cycle
Current TRL level	**3 to 3.5**	**5**
Reasoning	Helium working fluid a long way from current experience	Nitrogen/Helium mix is much closer to air in properties
Requirements to develop technology to higher TRL levels	Closed loop single row cascade testing with Helium would enable TRL4. To get higher is very difficult – best strategy would be to test whole PCS at reduced pressure with large non-nuclear heat source.	Because turbomachinery is not in primary cycle, some development would be possible in the pilot plant. Compressor and turbine could be stripped down and rebuilt with modified blading.

Table 4-4. Turbomachinery Mechanical TRLs

	Direct Combined Cycle	**Indirect Combined Cycle**
Current TRL level	**3.5**	**4**
Reasoning	Turbomachinery will be exposed to radioactive contaminants (especially silver and cesium). Also, the high pressure Helium environment may cause embrittlement.	Amount of Helium is less. No radioactive contamination. The effects of oil build up in the circuit need to be understood.
Requirements to develop technology to higher TRL levels	Exposing material samples to high pressure Helium environment and performing tensile tests would verify effects. Nuclear contamination is harder to assess – will not know concentrations until reactor is operating for some time.	Exposing material samples to high pressure Helium/Nitrogen mixture environment and performing tensile tests would verify effects.

Table 4-5. Electrical Generator TRLs

	Direct Combined cycle	**Indirect Combined cycle**
Current TRL level	**3**	**10**
Reasoning	High pressure Helium cooled. Exposed to high temperatures and harsh environment – novel insulation may be required	Commercial off the shelf.
Requirements to develop technology to higher TRL levels	Insulation tests at high temperatures. Scale rotor tests in high pressure helium to verify windage losses.	

Table 4-6. Power Electronics TRLs

	Direct Combined Cycle	Indirect Combined Cycle
Current TRL level	8	Not required
Reasoning	Commercially available, but all 'one-offs' rather than off the shelf.	
Requirements to develop technology to higher TRL levels		

Table 4-7. Duct Work TRLs

	Direct Combined Cycle	Indirect Combined Cycle
Current TRL level	6	6
Reasoning	Losses in Helium pipework – theoretically calculated, but not experimentally verified	Losses in Nitrogen/Helium pipework – theoretically calculated, but not experimentally verified
Requirements to develop technology to higher TRL levels	Simple flow tests at correct Reynolds Number.	Simple flow tests at correct Reynolds Number.

Table 4-8. Steam Generator TRLs

	Direct Combined Cycle	Indirect Combined Cycle
Current TRL level	8	8
Reasoning	AGR experience shows manageable, but not Commercial off the shelf.	AGR experience shows manageable, but not Commercial off the shelf.
Requirements to develop technology to higher TRL levels		

Table 4-9. Steam Plant TRLs

	Direct Combined Cycle	Indirect Combined Cycle
Current TRL level	**10**	**10**
Reasoning	Commercial off the shelf	Commercial off the shelf
Requirements to develop technology to higher TRL levels		

Table 4-10. Active Magnetic Bearings and Catcher Bearings TRLs

	Direct Combined Cycle	Indirect Combined cycle
Current TRL level	**4.5**	**Not required**
Reasoning	Challenging combination of environment, loads and rpms. Not commercially available for this type of application	Not required.
Requirements to develop technology to higher TRL levels	Small scale rig tests followed by full scale tests.	

4.10 Direct Versus Indirect Cycle - Risk Assessment

The proposed indirect combined cycle and the direct combined cycle are compared and discussed in this section and the next. The comparison has been done from a risk point of view in this subsection and from a cost point of view in Section 4.11.

A detailed comparison between the indirect combined cycle and the direct combined cycle to evaluate best commercial option has shown that a significant risk reduction will be achieved for the indirect cycle compared to a direct cycle. A direct cycle is still seen to be an achievable option for long term future applications when development of high risk and high cost components such as magnetic bearings are more mature and the risks associated reduced.

4.10.1 Advantages and Risk Reduction with the Indirect Cycle

For the indirect cycle the secondary loop is not in direct contact with the fuel and there is no or only very small risk of contamination. It is therefore assumed that the turbomachinery can be placed outside the containment building and a more suitable working fluid for the turbomachinery can be used. This allows the turbomachinery to adopt a much smaller more conventional arrangement using conventional oil lubricated bearings instead of electromagnetic bearings, and conventional gearbox instead of power electronics. The following advantages with the indirect cycle compared to the direct cycle were identified:

Turbomachinery Close to Trent Style

One advantage of the indirect cycle is the possibility of using Nitrogen (with 20% Helium by mass) as the working fluid in the secondary loop. Because of its gas properties (close to properties for air) nitrogen will give a much smaller device with relatively low development risk due to well proven design tools. Furthermore nitrogen is a cheap gas that could potentially very easily be produced on-site.

Conventional Bearings instead of Electromagnetic Bearings

The development of a viable electromagnetic bearing including catcher bearings and adequate stiffness control required for a direct cycle is an area of great concern. The cost of such a system is also believed to be significant. For an indirect cycle the turbomachinery can be placed outside the containment area allowing conventional oiled bearings to be used. Rolling element a bearings are common in both the aerospace and energy sector and existing off the shelf solution significantly reduces the risks. It should be pointed out that using oil lubricated bearings is likely to cause small oil contamination in the secondary cycle. Some form of filtering would probably be needed in the secondary cycle however this is at this stage not seen to cause any major problems.

More Frequent Maintenance Intervals

The extent of radioactive contamination of serviceable turbomachinery components for the direct cycle is unclear; however, there is no doubt that the indirect cycle with a turbomachinery outside the containment area can be accessed for maintenance more easily and more frequently. This has two main benefits: more frequent planned maintenance intervals can be allowed and quicker less costly unplanned maintenance will be achievable.

Reduced Size of Containment Building

For the indirect cycle the turbomachinery and the HRSG can be placed outside the containment building. However, instead the IHX will be placed inside the containment area. It is likely that a decrease of the containment building size, leading to a cost reduction can be made for the indirect cycle.

Conventional Gearbox instead of Power Electronics

One of the key commercial risks for the direct cycle includes the anticipated very high cost of the power electronics. For the indirect cycle conventional gearbox can be used reducing costs significantly.

Double-ended Generator

The two separate generators (gas turbine and SG) used for the direct cycle can be replaced in an indirect cycle by a single ‘double ended generator’ where the gas turbine drives one end and the steam turbine drives the opposite end via a SSS clutch. This will reduce the cost of electrical transformers, switch gear etc. and also increases inertia connected to the gas turbine, which would help in preventing overspeed for electrical dropped load events.

More Design and Off-Design Flexibility

The indirect cycle has the flexibility to be designed for a more optimum reactor inlet temperature. Optimum reactor inlet temperatures are close to the upper limit for each reactor outlet temperature (i.e. close to reactor ΔT of 360°C). Furthermore, the indirect cycle has more off-design flexibility to vary steam flow to maintain reactor inlet temperature and maintain better off-design efficiency.

4.10.2 Disadvantages and Additional Risks for the Indirect Cycle

Although the risks are significantly reduced for an indirect cycle for the reasons given above, it should be acknowledged that there are additional disadvantages and risks for the indirect cycle associated with the IHX. The following disadvantages with the indirect cycle compared to the direct cycle were identified:

Increased ΔP in IHX and Decreased Turbine Inlet Temperature Reduce Performance

Introduction of an IHX have two disadvantages on cycle performance. The first one is the penalty associated with the decreased turbine inlet temperature and the second one is the pressure drop across the IHX. The efficiency drop for the indirect cycle compared to a direct cycle with same reactor outlet temperature is approximately 1.6%pts.

Cost and Risks Associated with the Introduction of an IHX

The introduction of the IHX for the indirect cycle does have a cost and risk associated. Although the turbomachinery and the HRSG can be placed outside the containment building the IHX will be placed within that area. More detailed design and sizing of the IHX is required to determine the full cost implication. The main risks identified for the IHX are; leakage of primary or secondary fluid, material damage due to nitriding, failure due to a pressure differential across the IHX. The implication of this has not been fully assessed.

4.10.3 Risks that Apply to both the Indirect and the Direct Cycle

There are still risks associated with transient that apply to both the direct and indirect cycle. The bypass valve, which needs to pass 45% of the main flow in order to reduce the net power on the gas turbine shaft, will need additional investigation. The practicalities of the bypass valve passing such a large flow, while also being able to exert fine control over the speed of the shaft when it is not synchronized, will need careful consideration. The design of such a valve is at this stage believed to be possible however the difficulties with the design should not be ignored.

Furthermore the probability of component failure (shaft failure, blade-off, disk-off failures etc) is still the same for both the indirect and the direct cycle. For the indirect cycle a component failure could damage the IHX. For the direct cycle same failure would cause damage to reactor. The impact, cost and plant down time, of the latter one is likely to be higher.

It has been assumed that the PCS can be placed outside the containment area for the indirect cycle; this is believed to be a reasonable and likely assumption. If this for any reason would not be possible and the PCS would have to be placed within the containment building then the risks would remain similar as for the direct cycle

4.11 Direct versus Indirect Cycle - Cost Comparison

A preliminary cost comparison of different cycle configurations for the PCS has been made. This comparison has been done using information available at the time, and it is recommended that the comparison be refined once more detailed designs have been established. The cost comparison shows cost benefit to the indirect combined cycle compared to the direct cycle and also compared to a pure steam cycle. It should be kept in mind when examining the results below that it is likely that in the long term the cost of electromagnetic bearings and power electronics can be reduced significantly, which will bring the cost of electricity (COE) for the direct cycle down. However, Rolls-Royce believes that the significant risk reduction of an indirect cycle compared to a direct cycle should be weighted higher than any cost benefit of the direct cycle.

NOTE: To enable the cost comparison, a number of assumptions had to be made. There is high uncertainty in the estimates and they should only be used to identify relative cost trends between the different plant configurations, and not as absolute costs.

4.11.1 Cost Comparison for Different Working Fluid Mixtures

By introducing approximately 20% helium by mass to the nitrogen in the secondary circuit, an overall cost reduction of the PCS can be seen. This benefit arises because of the contradictory nature of the dependence of the IHX and turbo-machinery components. A cost comparison of the turbomachinery and the IHX for pure nitrogen, a nitrogen/helium mix, and pure Helium was done. The comparison is based on the size and cost of the turbo-machinery, the IHX, and the steam plant. As illustrated in Figure 4-16, the results show that there is a cost benefit to the nitrogen/helium mixture.

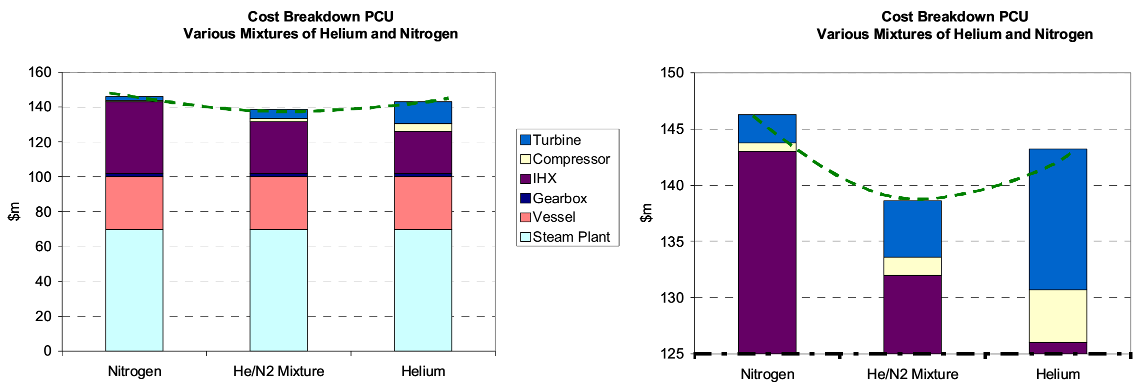


Figure 4-15. Capital cost of PCS for different helium/nitrogen mixtures

4.11.2 Cost Comparison for Different PCS Configurations

A cost comparison was then done between the indirect combined cycle gas turbine (CCGT), a direct CCGT, an indirect gas turbine cycle with pre-cooler and recuperator, and a direct gas turbine cycle (OKBM). Both of the indirect cycles have a Helium/Nitrogen mixture as the working fluid while the direct cycles are constrained to pure helium as the working fluid. As illustrated in Figure 4-17, the results show that both indirect cycles have lower capital costs than the direct cycles. This is mainly due to the high cost of the electromagnetic bearings, the power electronics, and the more costly turbomachinery required for using pure helium as the working fluid.

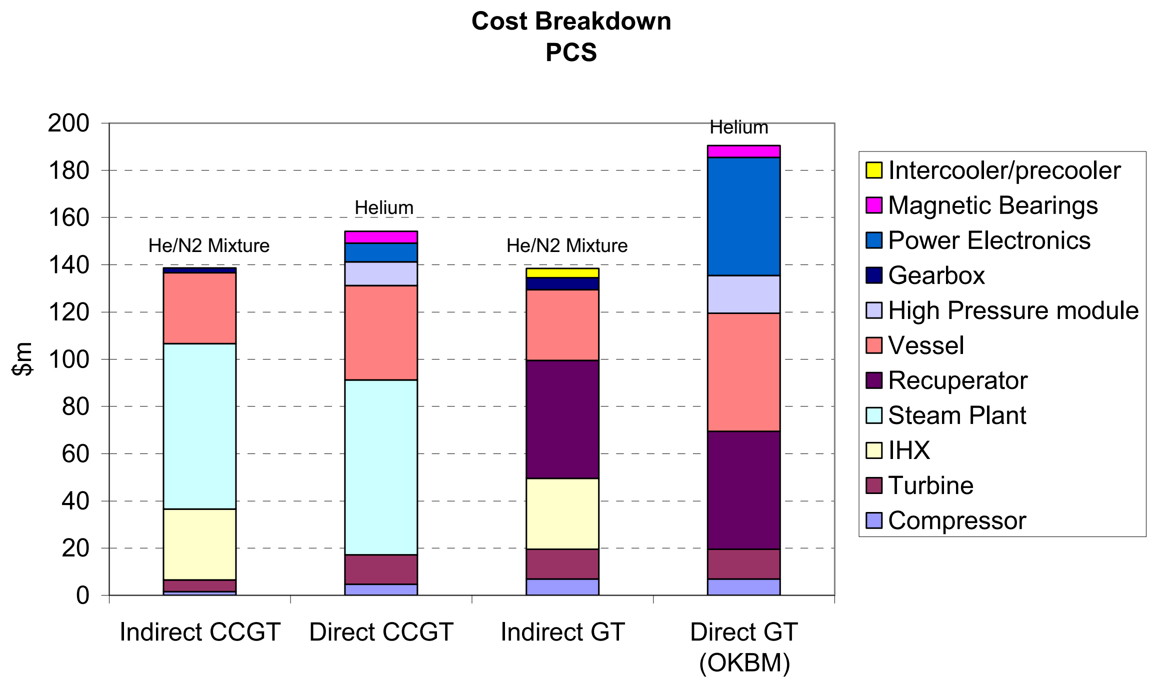


Figure 4-16. Capital Cost Comparison for PCS for various Plant Configurations

The above configurations all have different plant efficiencies, which must to be taken into account when comparing the costs.

The cost comparison shown in Table 4-11 is based on the COE associated with the capital cost of the PCS (IHX included for indirect cycles) and does not include the costs associated with the reactor and Reactor Building. Furthermore, all cases assume plant availability of 90%, but do not include any cost for maintenance. The payback period was set to 30 years with a discount rate of 10% (bank practice is to use a rate of 10 or 12 percent to calculate the net present value of a project).

Table 4-11. Cost comparison of PCS for Various Plant Configurations

	Direct CCGT 850°C	Indirect CCGT 850°C	Indirect CCGT 950°C	Direct GT (OKBM)	Indirect GT	Pure Steam
Plant Efficiency	50.2%	48.6%	49.3%	48.0%	46.0%	42.6%
COE	Ref.	-3.0 %	-4.4%	+16%	+1.7%	+2.5%

The PCS capital cost for a pure steam plant was estimated at $120M, which derives from removing the gas turbine from the indirect cycle and scaling up the steam plant. A pure steam plant was modeled using IPSEpro in the same way as the indirect CCGT cycle was modeled, but with the gas turbine removed. For a steam temperature of 580°C at 170 atm pressure, the maximum plant net efficiency is 42.6%. This was considered as the fairest comparison with the other cycles.

The cost comparison shown in Table 4-11 shows that the indirect CCGT has the best COE when comparing the electricity output with the cost of the PCS. Although the direct CCGT has better efficiency, the higher capital cost results in a COE higher than the indirect CCGT; it does not compensate for the higher capital cost compared to the indirect CCGT.

4.11.3 Whole Plant Cost Comparison and Parametric Study

When the total plant COE is calculated, including cost of reactor, Reactor Building, operation and maintenance, etc. (these costs were available for this study), the plant efficiency will have a far greater impact on the COE. This section discusses the main factors that will impact the total plant COE.

The cost model used for comparison of various PCS options is an Excel-based tool developed within Rolls-Royce. The tool compares the cost of various electricity generation sources such as coal fired, nuclear, and CCGT plants, and also renewable energy sources such as on-shore and off-shore wind. Part of the tool could be used to compare the various NGNP options in this study. The inputs required can be divided into three different components: capital, finance and operating costs. Capital and financing costs make up the project cost. Capital cost here includes the reactor installed cost (including land, infrastructure, buildings, site works, licenses etc) and the installed cost for the pure Power Conversion System, i.e., the PCS design installed cost. Financing costs will depend on the assumed discount rate. Operating costs are O&M costs, fuel cost, and any connection charge to the grid.

Four different case studies where done for different financial scenarios. The costs are proposed for nth-of-a-kind plant. The following assumptions were made:

- The cost of the PCS for the various configurations are the same as shown in Section 4.11.2
- The capital cost of the entire plant (excluding the PCS) is assumed to be fixed for a 600-MW plant, and therefore the cost per produced kilowatt of electricity will vary depending on plant efficiency. Plant costs between $500M and $1000M were assumed. Two cases have been run; one in the lower end of this range and one at the high end of the range.

- The fuel cost is assumed to be between $1700 and $3400 per kg of UO_2. The cost of the fuel includes processing, enrichment and fabrication of the uranium into fuel elements. It also includes management of radioactive spent fuel and the ultimate disposal of the spent fuel or the wastes separated from it.
- Plant efficiency for each configuration is based on results from modeling the specific thermodynamic cycle using IPSEpro. The plant efficiency for each cycle is shown in Table 4-11.
- Plant availability was assumed to be 90% for all plant configurations and is based on 20 days outage every 18 months due to refueling.
- O&M costs include predicted operation, and planned and unplanned maintenance. The O&M cost per kWh produced electricity for the indirect cycle has been assumed to be 2 c/kWh. O&M costs are assumed to be 20% higher for a direct cycle due to the higher component costs and risks and 20% lower for a pure steam cycle for the opposite reason. This was kept constant for all four cases.
- The discount rate is the percentage by which the value of a cash flow in a discounted cash flow valuation is reduced for each time period by which it is removed from the present. In this study, it has been set to 10% for two of the cases and 5% for the other two cases. A payback period of 30 years is assumed. It should be noted that this is not the same as the operational plant life, but is the financial project life.
- Lead time (time interval between the start of an activity or process and its completion, here the time between starting manufacturing and building the plant and its completion) has been estimated to be 48 weeks. The lead time has been kept constant throughout this study.

The results of the COE for each of the four cases are presented in Figures 4-18 through 4-21. The costs are split between equipment/installation costs, fuel cost, and O&M costs. Two different discount rates have been used. The final result is sensitive to the choice of discount rate and is often the most difficult and uncertain parameter to set. A discount rate of 10% is the more common to use; however, for comparison two cases with a lower discount rate of 5% were also run. For each of the two discount rates, one scenario for the upper range on capital and fuel cost and one scenario for the lower range of the capital and fuel cost were considered. The capital cost will have a much greater impact on the COE than the fuel cost and therefore the combination of high capital cost/low fuel cost and vice versa has not been presented.

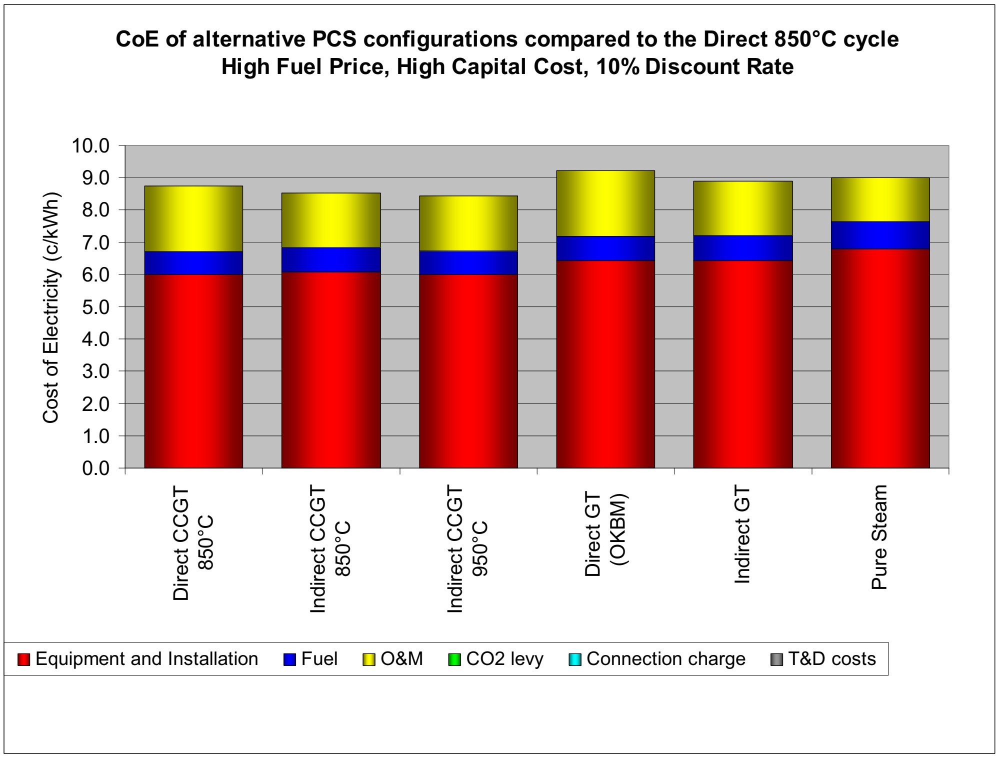


Figure 4-17. Estimated COE of configurations with high costs & 10% discount rate

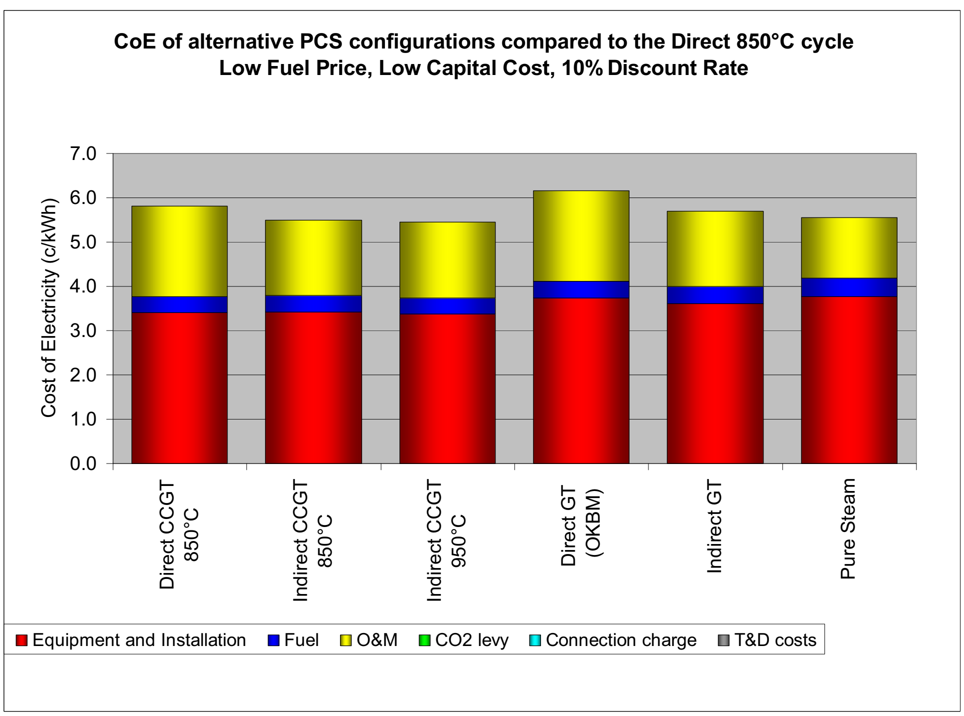


Figure 4-18. Estimated COE of configurations with low costs & 10% discount rate

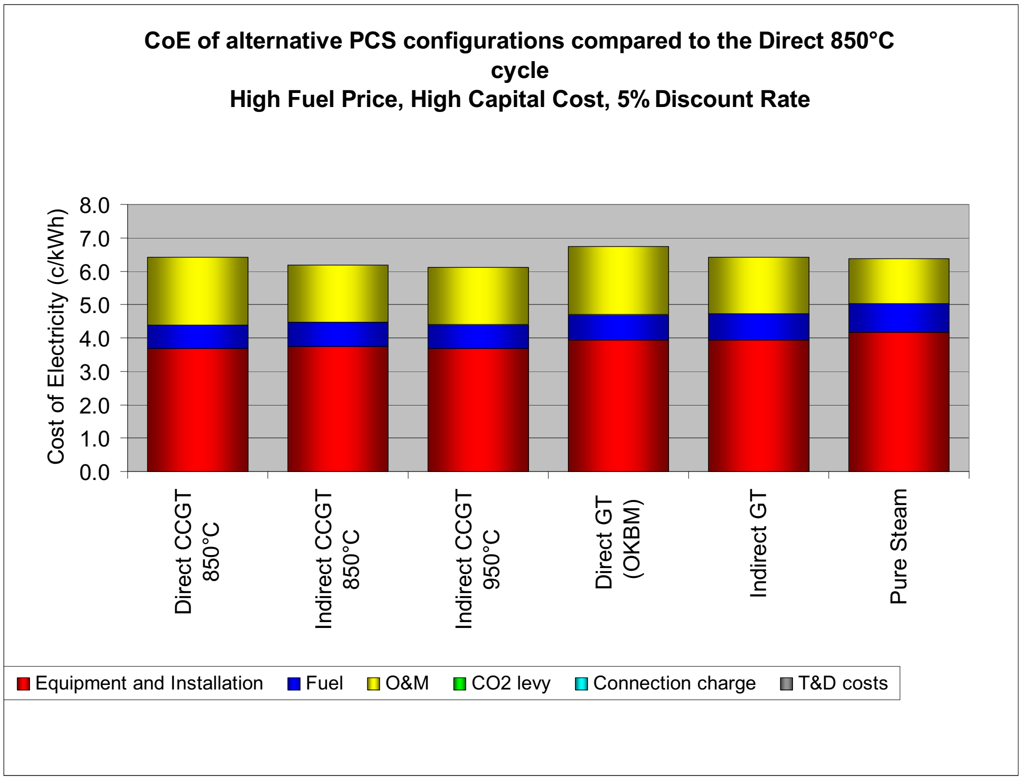


Figure 4-19. Estimated COE of configurations with high costs & 5% discount rate

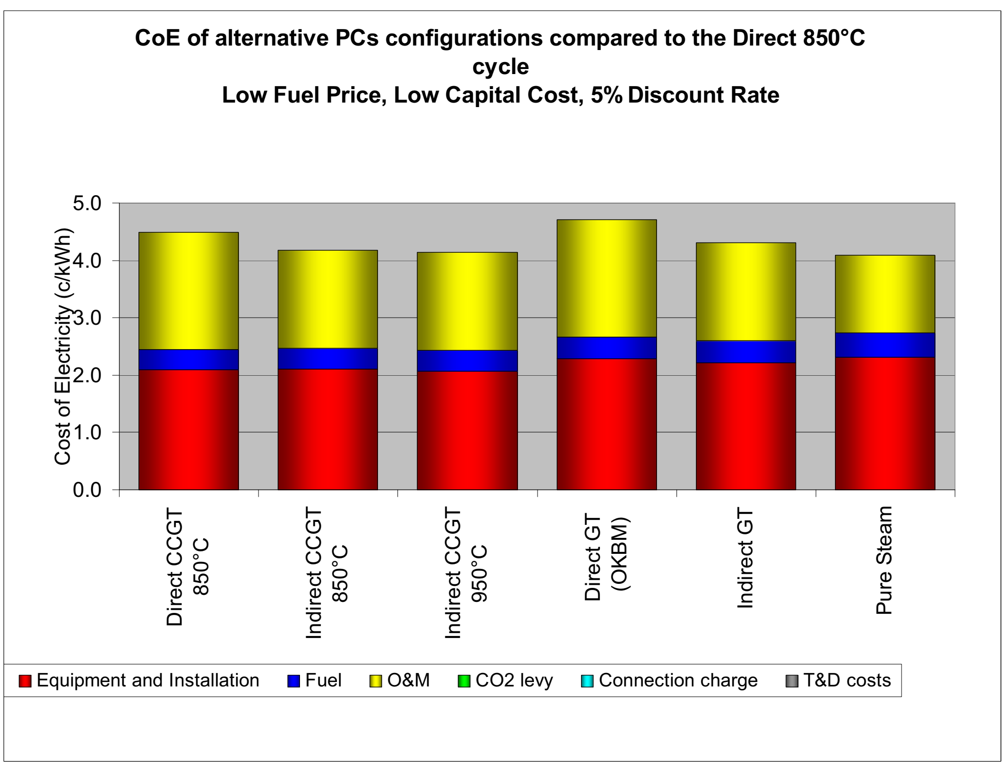


Figure 4-20. Estimated COE of configurations with low costs & 5% discount rate

Equipment and installation costs for a nuclear plant are known to be major cost drivers, and this is also the case for all of the configurations compared here. The second highest contribution to the COE is the O&M cost, while the fuel price represents a smaller fraction of the cost. The lower the capital cost and discount rate, the higher share the O&M costs represents. The COE for the different cases varies significantly. As the project develops, the certainties in the cost figures should increase, but it should also be acknowledged that the costs can vary significantly depending on the country and location of the plant. A summary of the four cases with the relative change to the direct CCGT cycle is shown in Figure 4-22.

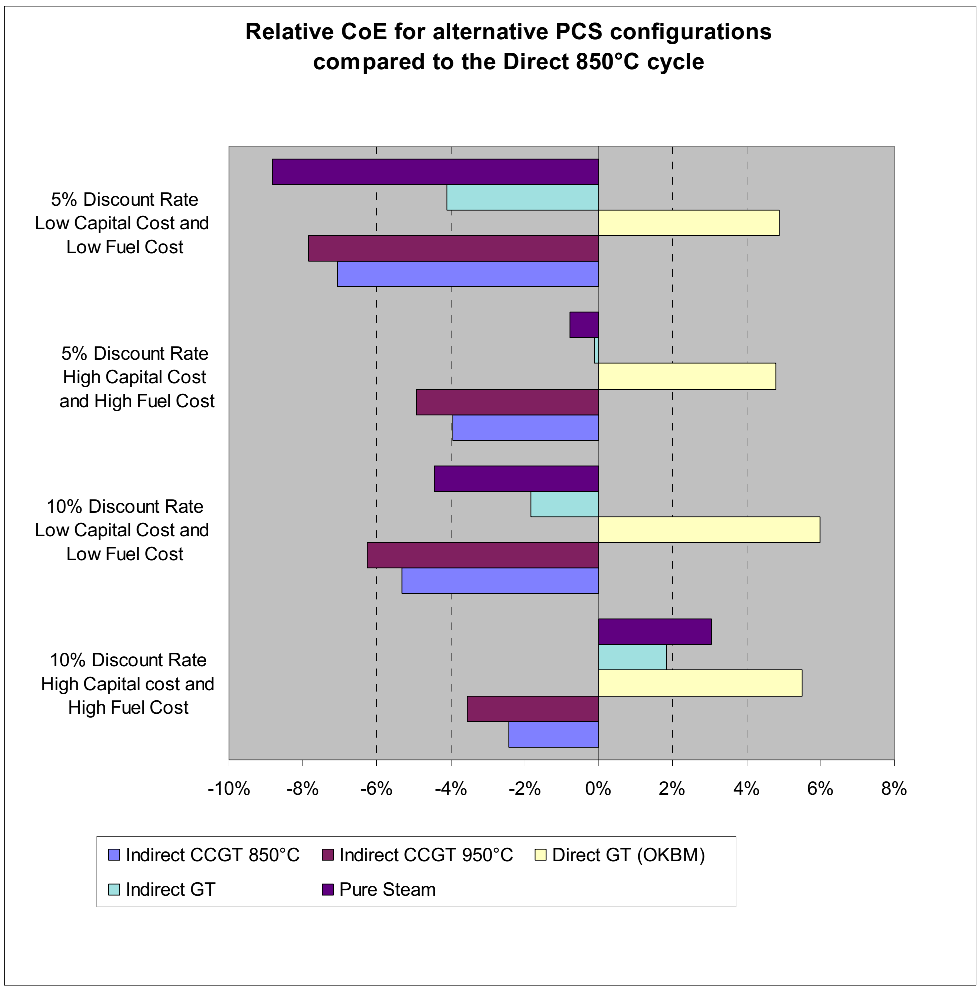


Figure 4-21. Whole plant cost comparison for various plant configurations

With the assumptions made, it shows that the indirect cycle still gives the lowest COE for three out of the four cases. If the capital cost is high, then high plant efficiency becomes more important. This is why the indirect and also direct CCGT have a lower COE than a pure GT or steam-cycle plant. For a low capital cost and low discount rate, the pure steam cycle has the lowest COE. The indirect CCGT still shows a very good COE in comparison with the direct CCGT and the GT cycles. When comparing the four cases, it can be observed that the indirect CCGT is less sensitive to change in capital cost and discount rate compared to the pure steam cycle. This is because the indirect cycle has a much higher efficiency, which makes it less sensitive to increased capital cost.

An uncertainty that has not been investigated is the plant availability for the above compared cycles. Plant availability has been assumed the same for all configurations. However the risks associated with the direct cycle is likely to reduce the plant availability, particular in the early development stage of these technologies.

4.12 Conclusions

An indirect combined cycle has been found to be the best choice for a medium-term commercial electricity plant. This cycle is shown to have the best efficiency of any indirect cycle and to achieve the lowest cost of electricity generation among the systems compared in this study[5].

The indirect combined cycle conforms to the requirements specified for NGNP, in that it is compatible with the provision of high temperature process heat and with provision of process steam, to support demonstration of hydrogen production technologies.

The direct Brayton cycle was the subject of a previous study. Its efficiency has been shown to be lower than that of the direct and indirect combined cycles, although only slightly lower than that of the indirect combined cycle. This is an elegant concept, for which the principal drawbacks were found to be development and maintenance risks (particularly the very challenging active magnetic bearing and large recuperator), together with some potentially high costs. In the longer term, if these major challenges can be overcome, the direct Brayton cycle may merit reconsideration because its lack of a steam cycle makes it simpler so there is less to fail.

[5] The indirect combined-cycle efficiency of 48.6% is obtained at a higher reactor outlet temperature (850°C) than is the steam-cycle efficiency of 42.6% (~680°C), and the higher reactor outlet temperature has significant implications on the cost and risks associated with the reactor. Consideration of the reactor costs and risks associated with increasing reactor outlet temperature was outside the scope of the current Rolls-Royce combined-cycle PCS study.

The direct combined cycle was also a subject of the previous study, achieving the highest efficiency of the systems considered here. Its greatest advantage over the direct Brayton cycle was considered to be its lower development risk (reduced risk active magnetic bearing and no recuperator requirement). However, the risk remains high in comparison with the indirect combined cycle. If future evolution of the technology base resolves the risk issues, the direct combined cycle may become a preferred option for longer-term commercial electricity generation plants.

The indirect Brayton cycle is outside the terms of reference of the present study and has not been given detailed attention. It has been shown to be a lower efficiency cycle than the direct cycles and the indirect combined cycle. Turbine risks are higher than for the indirect combined cycle, although much lower than for the direct Brayton cycle and might be resolved by future evolution of the technology base. Plant simplicity, there being no steam cycle is a benefit this cycle shares with the direct Brayton cycle.

All the cycles that incorporate a gas turbine have the potential for gaining improved performance from raised reactor operating temperatures (950°C and beyond), although some materials development and/or blade-cooling system would be required for the direct cycles.

The combined cycle systems are less susceptible than pure Brayton cycles to loss of output in off-design conditions such as reduced heat sink effectiveness.

The pure steam system is the lowest efficiency option of those considered for comparison, although there is limited scope for improvement by adoption of supercritical technology. A pure steam cycle is a disappointing match with the characteristics of a high temperature reactor, because the temperature limitations of the steam system prevent effective utilization of primary temperatures above about 650°C. Part-load conditions that produce steam at above design point temperatures may incur some materials development issues.

5 EVALUATION OF COMPACT IHX DESIGN ISSUES

Because of the importance of the IHX to any indirect PCS option, more detailed evaluations of compact IHX design issues remaining from the FY08-1 IHX and heat transport alternatives study [GA 2008a] were included as part of the current PCS alternatives study. The evaluations were performed by Toshiba Corporation and included calculations to determine the effect of heat transfer assumptions on IHX size and a structural analysis to estimate the effect of thermal stresses on IHX lifetime. Toshiba was also tasked to evaluate the impact of using an 80% nitrogen/20% helium mixture in the secondary loop on the size and cost of a PCHE-type compact IHX. The results of these evaluations are presented below.

5.1 Heat Transport System (HTS) Alternatives

The indirect PCS configurations considered in the current study are the serial configuration and parallel-loop configuration that were previously considered in the IHX and heat transport system alternatives study [GA 2008a] performed during the FY08-1 conceptual design studies. These configurations are shown in Figure 2-1 and Figure 2-2 of this report. The design conditions for PCS-side IHX in the parallel-loop configuration are listed in Table 5-1.

Table 5-1. PCS-side IHX Design Conditions

Parameters	Design Conditions
Heat Load, MW(t)	535
LMTD*, °C	186
Primary Side Fluid	Helium
Primary Side Flow Rate, kg/s	244.96
Primary Side Inlet / Outlet Temperature, °C	900 / 480
Primary Side Inlet / Outlet Pressure, MPa	7.0 / 6.95
Secondary Side Fluid	Helium
Secondary Side Flow Rate, kg/s	262.46
Secondary Side Inlet / Outlet Temperature, °C	308 / 700
Secondary Side Inlet / Outlet Pressure, MPa	7.1 / 7.05
Allowable Pressure Loss**, MPa	0.05

* LMTD = Log-Mean Temperature Difference
** Tentative condition.

5.2 Design Description

5.2.1 Detailed Evaluation of PCHE Module

5.2.1.1 Evaluation of Methodology to Size PCHE Module

In order to perform a more detailed evaluation of methodology to size the PCHE module, the heat transfer calculation was verified using the specifications for the PCHE module designed by Heatric Corp. Several equations to predict heat transfer were compared. These included:

- Kay's correlation for fully developed laminar flow
- Zigzag method recommended by General Atomics
- Colburn j-factor used for compact heat exchanger

The result of the zigzag method as shown in Table 5-2 best matches Heatric results. Therefore, Toshiba concludes that the zigzag method gives the most reasonable estimates of PCHE module size for the compact IHX[6].

Table 5-2. Comparison of Results of PCHE Module Sizing

Method	Dittus-Boelter and Kay's Correlation	Colburn j-factor	Zigzag Method of GA	Results of Heatric
Heat Transfer Coefficient of Hot Coolant, $W/(m^2 \cdot K)$	924.3	-	1,988	
Heat Transfer Coefficient of Cold Coolant, $W/(m^2 \cdot K)$	924.3	-	1,968	
Heat Transfer Area per Module, m^2	1,432	706	681	680

[6] It was further concluded that the Colburn j-factor used in the FY08-1 study gave conservative results with respect to IHX size.

5.2.1.2 Evaluation of Methodology to Estimate Pressure Drop

The total pressure drop was evaluated by the Weisbach's equation and the zigzag method. The result of the zigzag method as shown in Table 5-3 is in good agreement with Heatric results. Therefore, Toshiba decided to adopt the zigzag method for estimating pressure drops for the PCHE modules.

Table 5-3. Total Pressure Drop Given by Several Correlations

Flow Channel	Weisbach's Equation (kPa)	Zigzag Method of GA (kPa)	Result of Heatric (kPa)
Primary	12	31	32
Secondary	11	29	31

5.2.1.3 Resizing of PCHE Module

The PCHE module for the hot-stage and cold-stage IHXs in the serial configuration (Figure 2-1) and for the PCS-side and small IHXs in the parallel-loop configuration (Figure 2-2) were resized using the zigzag method. The results are given in Table 5-4. The basic dimensions (e.g., the number of channels, height of module, and so on) are not changed. The number of modules and the length of the modules were adjusted so as not to exceed a pressure drop of 40 kPa, which was selected as the maximum allowable pressure drop for the IHX given that the pressure drop increases when calculated using the zigzag method. The increased length of the module is modest, and there is no effect on the diameter of the IHX. However, the height of the IHX has increased due to the increased number of modules. Figure 5-3 shows an updated sketch of the PCS-side IHX. If the capacity of the helium circulator is increased to accommodate a larger pressure drop across the IHX, the height of the IHX can be reduced.

Table 5-4. Results of PCHE Module Re-Sizing

Configuration	Serial		Parallel Loop	
IHX	Hot-stage	Cold-stage	PCS-side	Small
LMTD, °C	46.1	116.8	185.6	43.7
Number of Modules	208 (192)	176 (160)	176 (160)	42 (36)
Channels per Plate	75			
Channels per Each Side	89			
Height of Module, mm	453			
Width of Module, mm	400			
Length of Module, mm	450 (433)	400 (372)	430 (417)	760 (755)
Edge Distance, mm	13			
Layer Thickness, mm	2.4			
Channel Radius, mm	1.5			
Channel Pitch, mm	3.9			
Channel Offset Pitch, mm	12.7			
Channel Offset Height, mm	2.286			
Zigzag Angle, degree	108.5 (108)			
Flow Area per Module, m^2	23.6×10^{-3}			
Heat Transfer Area per Module, m^2	18.5 (17.0)	16.4 (13.2)	13.7 (9.7)	37.6 (39.5)
Effective Heat Transfer Length, mm	359 (331)	319 (256)	266 (188)	730 (767)
Heat Transfer Core Length, mm	291 (268)	259 (207)	216 (152)	593 (620)
Pressure Drop of Primary Side, kPa	48 (32)	46 (32)	35 (31)	27 (16)
Pressure Drop of Secondary Side, kPa	40 (40)	40 (37)	39 (35)	26 (33)
Note: The numbers in ( ) are the values obtained in the FY08-1 IHX study [GA 2008a] using the Colburn j-factor heat transfer correlation				

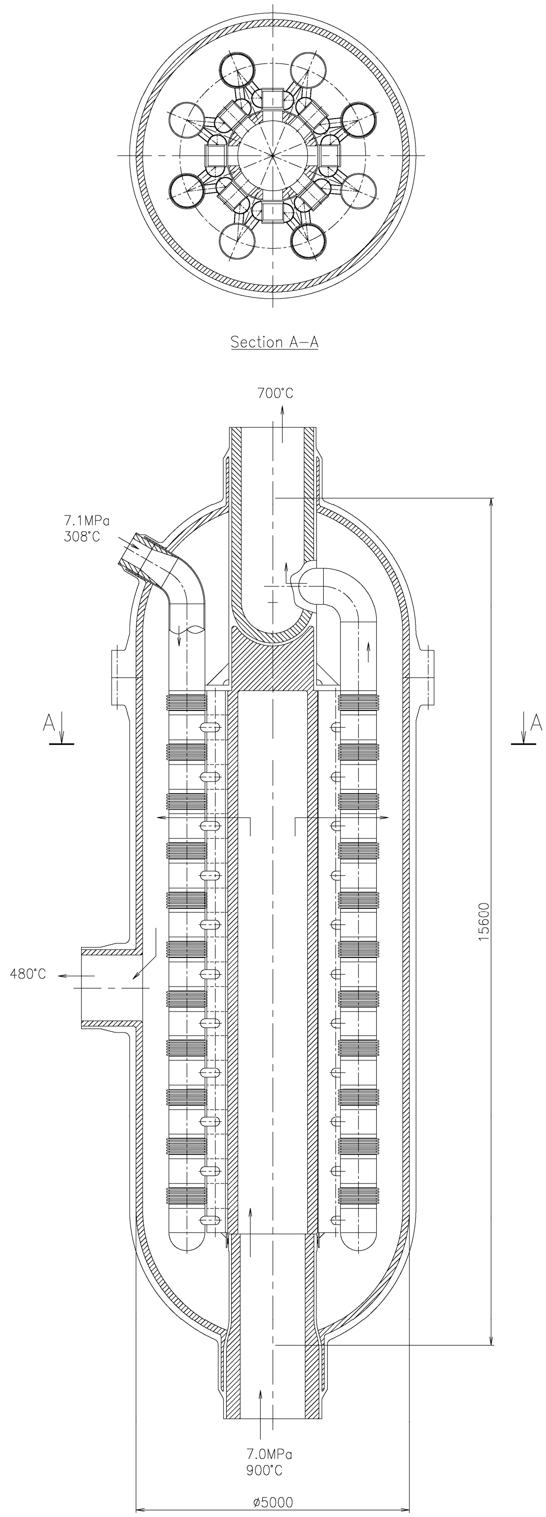


Figure 5-1. Conceptual drawing of PCS-side IHX

5.2.1.4 Effect of 80 wt% Nitrogen / 20 wt% Helium Mixture on PCHE Module

The effect of using 80 wt% nitrogen/20 wt% helium as the working fluid in the secondary loop (relative to using 100% helium as the working fluid) on the size and cost of the PCHE-type IHX was roughly assessed by Toshiba to assist with the comparison of direct and indirect combined-cycle PCS options. The results of the PCHE module sizing for each IHX are shown in Table 5-5. The number of modules increased by 16 for both the hot-stage and the cold-stage IHXs, but the number of module for the PCS-side and the small IHX did not change. The length and weight of the PCHE modules increased for all of the IHXs. The overall diameter of all of the IHXs increased as did the overall height of the hot-stage and cold-stage IHXs. The estimated maximum increase in cost (for the IHX hot-stage IHX) is estimated to be of the order of $30M. The increased cost associated with use of an 80 wt% nitrogen/20 wt% helium mixture in the secondary loop would be less for the other IHXs.

Table 5-5. Results of PCHE Module Sizing for 80 wt% Nitrogen / 20 wt% Helium

Configuration	**Serial**				**Parallel Loop**			
IHXs	Hot-stage		Cold-stage		PCS-side		Small	
Secondary Coolant	Helium	Mixture	Helium	Mixture	Helium	Mixture	Helium	Mixture
Number of Modules	208	224 (+16)	176	192 (+16)	176	176	42	42
Height of Module, mm	453		453		453		453	
Width of Module, Mm	400		400		400		400	
Length of Module, mm	450	495 (+45)	400	440 (+40)	430	475 (+45)	760	880 (+120)
Increased Weight of PCHE Module, ton		26		22		12		8
Increased Diameter of IHX, mm		90		80		90		240
Increased Height of IHX, mm		906		906		-		-

5.2.2 Stress Analysis to Estimate Effect of Thermal Stresses on IHX Lifetime

A preliminary stress analysis was performed to estimate the effects of thermal stresses on IHX lifetime using the methodology described below.

5.2.2.1 Analysis Model

The stress analysis of a PCHE module channel and the stress evaluation were conducted using the design criteria of ASME Code, Sec. III, Subsection NH and [ORNL, 2004]. The stress analysis included both a structural analysis to account for pressure loads and a thermal stress analysis to account for thermal loads. The subjects of the analyses were the PCS-side IHX and the hot-stage IHX. The primary inlet/secondary outlet side of the PCHE module was selected as the evaluation point because it is the location having the most severe conditions. The analysis model is shown in Figure 5-4. ABUQUS version 6.5 was used as the analysis software.

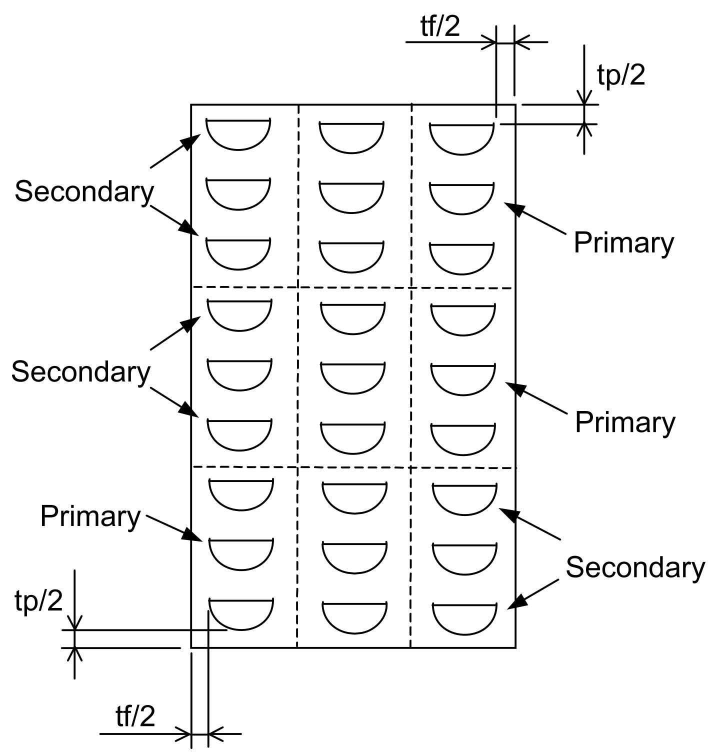


Figure 5-2. Analysis model

5.2.2.2 Boundary Conditions

The pressure conditions of the primary channel and secondary channel are 7.0 MPa and 7.1 MPa, respectively.

5.2.2.3 Results and Evaluation

The result of the structural analysis is shown in Figure 5-5. The maximum value of the Mises stress distribution was 7.0 MPa at the thinnest part of the wall between the primary and secondary channels. The results of the primary stress evaluation of the PCS-side IHX at rated operation with Level A and B service loadings are shown in Table 5-6. The primary stress is low enough to meet the allowable stress value at nearly 810°C for a 60-year lifetime by considering the external pressure loaded on the outer surface of PCHE module, which was mistakenly not taken into account in the FY08-1 study because the stress was evaluated in only 2-dimensional cross-section inside the PCHE module.

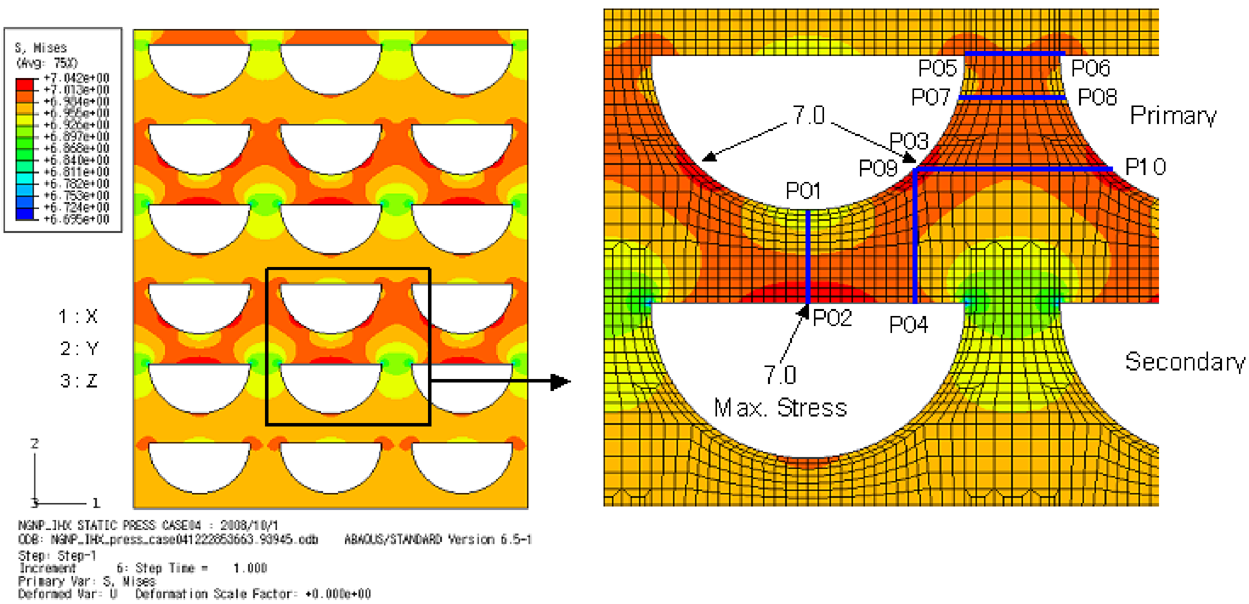


Figure 5-3. Mises stress distribution by pressure load in PCHE module (MPa)

Table 5-6. Load Controlled Limits (Pm) of PCS-side (MPa)

Levels A and B Service Limits
Pm

	Cross Section	P01-P02	P03-P04	P05-P06	P07-P08	P09-P10
Pressure Loads	σ x	-7.0	-7.1	-7.0	-7.0	-7.1
	σ y	-7.0	-7.0	-7.0	-7.0	-7.0
	σ z	0.0	0.0	0.0	0.0	0.0
	τ xy	0.0	0.1	0.0	0.0	0.0
Pressurel Loads in Axial Direction	σ x	0.0	0.0	0.0	0.0	0.0
	σ y	0.0	0.0	0.0	0.0	0.0
	σ z	-7.0	-7.0	-7.0	-7.0	-7.0
	τ xy	0.0	0.0	0.0	0.0	0.0
Combination of Stress Components	σ x	-7.0	-7.1	-7.0	-7.0	-7.1
	σ y	-7.0	-7.0	-7.0	-7.0	-7.0
	σ z	-7.0	-7.0	-7.0	-7.0	-7.0
	τ xy	0.0	0.1	0.0	0.0	0.0
Principal Stresses	S1	-7.0	-7.1	-7.0	-7.0	-7.1
	S2	-7.0	-6.9	-7.0	-7.0	-7.0
	S3	-7.0	-7.0	-7.0	-7.0	-7.0
Stress Intensity	Pm	0.1	0.2	0.1	0.1	0.1
Allowable Limits	Smt	15	15	15	15	15

For the PCS-side IHX, the result of the temperature distribution analysis is shown in Figure 5-4. The maximum calculated temperature was 806°C at the side surface of the primary channel. The results of the thermal stress analysis are shown in Figure 5-5. The maximum stress was 25 MPa at the plane surface of the secondary channel. Based on the strain limit and the creep-fatigue evaluation, it is concluded that the PCHE modules have the potential to satisfy the design criteria for a 60-year service lifetime as shown in Tables 5-7 and 5-8. However, this conclusion is based only upon consideration of normal operation and cold shutdown conditions.

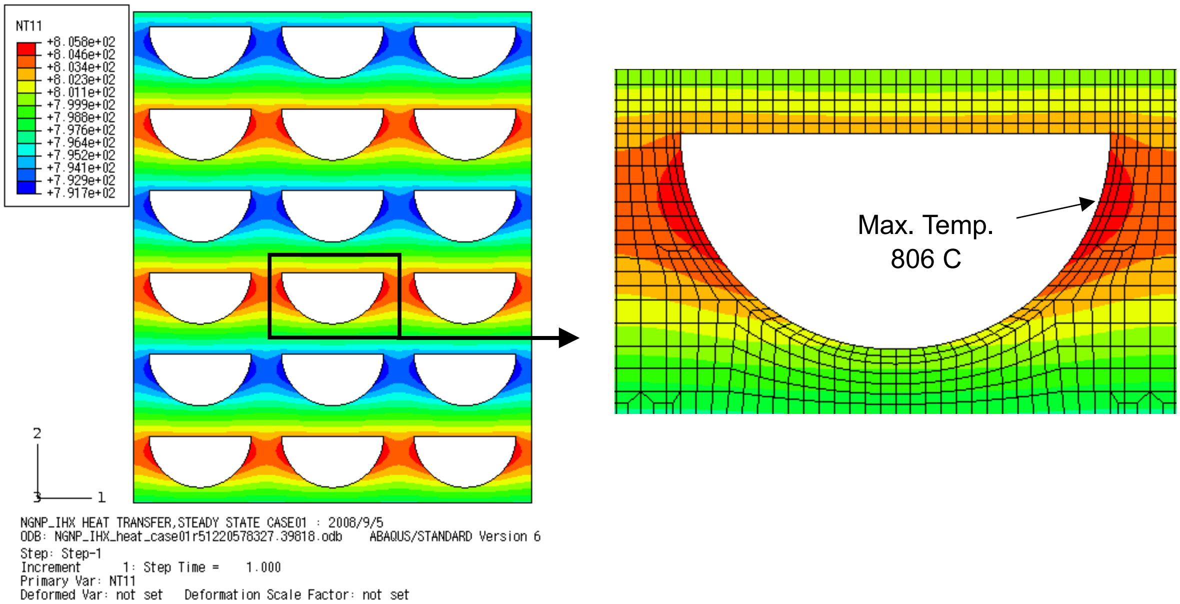


Figure 5-4. Temperature distribution in PCS-side PCHE module (°C)

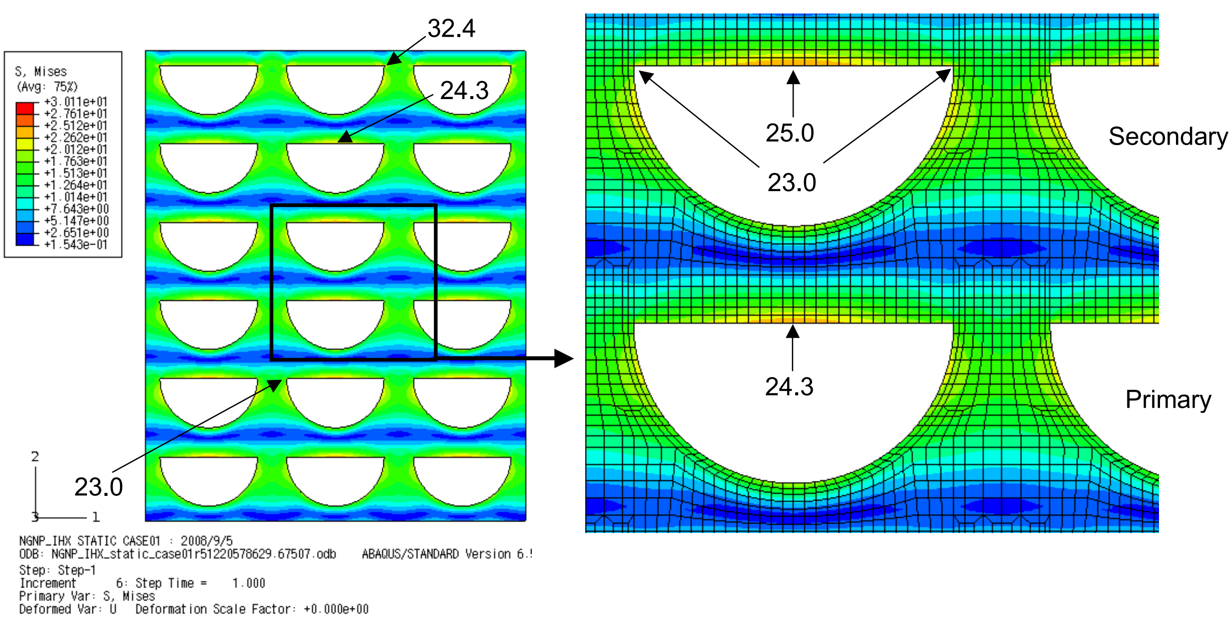


Figure 5-5. Mises stress distribution by thermal load in PCS-side PCHE module (MPa)

Table 5-7. Strain Limits on PCS-side (MPa)

Evaluation Points	$(P_L+P_b+Q)_{RANGE}$*	3Sm( )
P01	19	257
P02	29	257
P03	18	257
P04	19	257
P05	18	257
P06	18	257
P07	18	257
P08	18	257
P09	16	257
P10	16	257

*: Maximum Stress Range of Primary plus Secondary Stress Intensity under Level A and B Services

Table 5-8. Creep-Fatigue Evaluation (Partial)

IHXs	Σ(ni/Ndi)	D	Start-up / Shut-down Cycle n	Design Allowable Cycle Nd	Temp. °C	Service Lifetime hr	Evaluation Point
PCS-side	0.003	1	240	10^5	810	525,600	P02

5.2.3 Updated Cost Estimate for Full-size IHX

The estimated costs for all of the IHXs considered in this study increased relative to the costs estimated for the FY08-1 IHX study because use of the zigzag method to re-size the IHXs resulted in increased IHX pressure drops that were offset by increasing the PCHE module length and the number of PCHE modules for each IHX. (In other words, the IHXs were re-sized to maintain the same pressure drop). If an increase in pressure drop, which would increase the required power rating of the helium circulator power, can be tolerated, then the impact of using the zigzag method on the size and cost of the IHX would be reduced.

The impact of using an 80 wt% nitrogen/20 wt% helium mixture as the working fluid in the secondary on the IHX would be to further increase the size and cost of the IHX due to the decrease in heat transfer performance.

Contrary to the IHX service-lifetimes estimated in the FY08-1 study and reported in [GA 2008a], the results of the refined PCHE module stress analyses (for normal operation and cold shutdown conditions only) performed in the current study suggest that the service-lifetime of each IHX could potentially be 60 years. So, based on stress considerations only[7], the service-lifetimes of the hot-stage IHX, PCS-side IHX, and the small IHX may potentially be longer than 20 years as previously-estimated. If the replacement frequency of the IHX is reduced, this would clearly have a significant impact on the NGNP life-cycle cost.

5.3 Summary

The results of the IHX evaluations performed as part of the current PCS alternatives study are as follows:

- The zigzag method of calculating PCHE module heat transfer and pressure drop gives estimates that are in best agreement with Heatric Corp. PCHE module specifications.
- The results of the refined PCHE module stress analyses performed in the current study suggest that the service-lifetime of all of the IHX designs considered could potentially be 60 years. However, these stress calculations were for normal operation and cold shutdown conditions only.
- The effects of thermal and environmental aging on the IHX will need to be accounted for in more rigorous analyses and could potentially reduce the service lifetime of the IHX. Accumulation of graphite dust in the PCHE modules could also potentially reduce the IHX service lifetime.
- The estimated costs for all of the IHXs considered in this study increased relative to the costs estimated in the FY08-1 IHX study because use of the zigzag method to re-size the IHXs resulted in increased IHX pressure drops that were offset by increasing the PCHE module length and the number of PCHE modules for each IHX. (The IHXs were re-sized to maintain the same pressure drop).

[7] The effects of thermal and environmental aging on the IHX would have to be accounted for in a more rigorous analysis and could potentially reduce the service lifetime of the IHX. Accumulation of graphite dust in the PCHE modules could also potentially reduce the IHX service lifetime.

- The impact of using an 80 wt% nitrogen/20 wt% helium mixture as the working fluid in the secondary on the IHX would be to further increase the size and cost of the IHX due to the decrease in heat transfer performance.

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