Interstage Heating and Cooling Options for High Temperature Helium Brayton Cycles

Haihua Zhao and Per F. Peterson
U.C. Berkeley
Report UCBTH-05-002

May 10, 2005

Abstract

This report summarizes work to examine interstage heating and cooling options for multiple reheat helium Brayton cycle power conversion system (PCS) designs. Previous work has demonstrated that thermal powers in the range of 2400 MWt can be readily derived from combining multiple single-expansion units derived from the 600-MWt, vertical shaft gas-turbine module helium reactor (GT-MHR). This PCS design approach was optimized for the 2400 MWt Advanced High Temperature Reactor (AHTR) system, which uses liquid salt as coolant. Because this original AHTR PCS used an integrated design where all the heat exchangers except for the recuperator are put into PCU vessels, the resulting multiple-reheat PCS is very compact and has very high power density.

This new study further extends the earlier design to examine horizontal, distributed PCS configurations. Four variants of horizontal shaft designs were developed for analysis. The power densities, specific steel inputs, and specific helium inventories for all these designs are compared. It was found that differences in the figures of merit are not sufficiently different to deselect any systems, that is, that both horizontal and vertical configurations can achieve attractive efficiency and power density. Therefore, detailed studies and other considerations must be provided to further compare these designs.

Modular designs in the 600 MWt class are also considered in this study. Although the power density and specific steel input for the 600 MWt design are lower than those large systems, the difference again is not so large that smaller designs have a substantial cost penalty relative to large systems. The results show that the multiple-reheat indirect cycle can achieve the same thermodynamic efficiency as the non-reheat Brayton cycle commonly used for direct-cycle power conversion. Because the multiple-reheat cycle can still reject heat easily to low-quality heat sinks, it provides a potentially attractive option for indirect-cycle power conversion for 600 MWt-class gas-cooled reactors.

1 INTRODUCTION TO MULTIPLE REHEAT AND INTERCOOLING BRAYTON CYCLE SYSTEMS

Objectives and Background

Closed gas-turbine Brayton power conversion systems can have lower capital costs than comparable steam-turbine systems, due to their higher power density and thermodynamic efficiency. Multiple reheat and intercooling can further improve thermal efficiency than Brayton cycle without reheat. This study developed several conceptual design(s) for cost effective IH/IC approaches for high temperature helium Brayton cycles, and identified advantages, issues, and experiments to demonstrate key technologies. This report summarizes the study and presents a few representative point designs for multiple-reheat Brayton cycle power conversion systems using liquid salts or liquid metals for high temperature reactor systems. By utilizing reheat, these multiple-reheat gas cycles have the potential for substantially higher thermal efficiency than current gas cooled reactors, if used with comparable turbine inlet temperatures [Peterson, 2003a]. Multiple reheat stages allow larger pressure ratios and more closely approach Carnot efficiency. The larger pressure ratio reduces the recuperator size substantially, which is one of the largest equipment items for Brayton cycle systems without reheat. The multiple-reheat systems also eliminate the need for steam generators required for Rankine cycles. For liquid coolants, the elimination of steam generators removes the potential for chemical reactions between steam and the liquid salt or liquid metal, and greatly simplifies the control of tritium.

Figure 1 is a schematic T-s diagram for one high temperature reference design optimized for use with liquid salt heat transfer fluids as the heat source, the very high temperature liquid-coolant gas cycle (AHTR-VT), which illustrates the basic concept of the multiple-reheat cycle. Using multiple reheat, multiple intercooling and recuperation, the overall thermal efficiency can approach Carnot cycle efficiency. With multiple reheat stages, the average heat input temperature is close the highest heat source temperature; by multiple intercooling, the average heat rejection temperature is close the heat sink temperature. The optimal cycle efficiency for the reference design shown in this figure is 54% at a turbine inlet temperature of 900ºC (T_a in Figure 1). The pumping power for reheat and intercooling can be kept low by using a liquid to deliver heat to the power-cycle gas. For liquid salt or liquid metal heat transfer fluids, both the pipe size and pumping power needed for reheat are much smaller than for gas, which makes reheat technically and economically attractive.

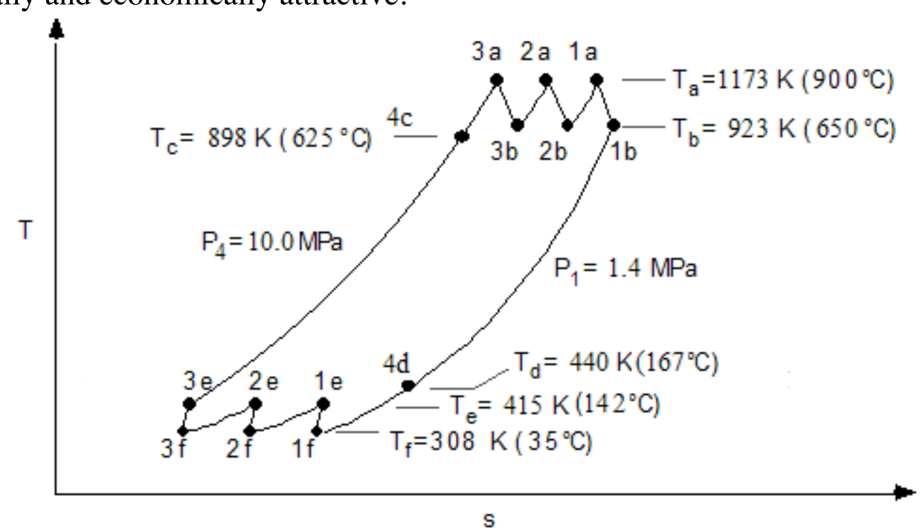


Figure 1: Temperature-entropy diagram for the multiple-reheat very-high-temperature reference case.

Multiple Reheat and Intercooling Brayton Cycle Thermal Efficiency Calculation

As shown in Figure 1, the multiple-reheat turbines are sized to provide approximately equal pressure expansion ratios. The gas entering each turbine is heated to approximately the same inlet temperature, using the intermediate coolant in a counter-flow heat exchanger. Likewise the compressors are sized to provide approximately equal compression ratios. The gas entering each compressor is cooled to approximately the same temperature, using water in a counter-flow heat exchanger.

To evaluate the multiple-reheat cycle thermal efficiency, the efficiencies of the turbine and compressor are defined as [Peterson, 2003a]

$$\frac{W_t}{M} = \left(T_a - T_b\right) = \eta_t\left(T_a - T_{bs}\right) \tag{1}$$

$$\frac{W_c}{M} = \left(T_e - T_f\right) = \frac{\left(T_{es} - T_f\right)}{\eta_c} \tag{2}$$

where W is the power, M the mass flow rate times specific heat, T absolute temperature, η_t the turbine efficiency, and η_c the compressor efficiency. The total pressure ratio for the cycle is then

$$\frac{P_{n+1}}{P_1} = \pi_c\left(\frac{T_{es}}{T_f}\right)^{\frac{n\gamma}{\gamma-1}} = \left(\frac{T_a}{T_{bs}}\right)^{\frac{m\gamma}{\gamma-1}} \tag{3}$$

where P is pressure, γ the gas constant, assumed to be constant, n the number of compression stages, m the number of expansion stages, and

$$\pi_c = 1 - \frac{\Delta P}{P_{n+1}} \tag{4}$$

corrects for system pressure losses ΔP. Then for specified turbine inlet and outlet temperatures, and compressor inlet temperature, the compressor outlet temperature T_e can be determined from Eqs. (1-3) as,

$$\frac{P_{n+1}}{P_1} = \pi_c\left(1 + \eta_c\left(\frac{T_e}{T_f} - 1\right)\right)^{\frac{n\gamma}{\gamma-1}} = \left(1 - \frac{1}{\eta_t}\left(1 - \frac{T_b}{T_a}\right)\right)^{-\frac{m\gamma}{\gamma-1}} \tag{5a}$$

$$T_e = T_f\left\{1 + \frac{1}{\eta_c}\left[\pi_c^{-\frac{\gamma-1}{n\gamma}}\left(1 - \frac{1}{\eta_t}\left(1 - \frac{T_b}{T_a}\right)\right)^{-\frac{m}{n}} - 1\right]\right\} \tag{5b}$$

The cycle efficiency η_{MR} can be determined from the heat added Q_H and the net work produced W_n,

$$\frac{Q_H}{M} = m(T_a - T_b) + \Delta T_r \tag{6}$$

$$\frac{W_n}{M} = \left[m(T_a - T_b) - n(T_e - T_f)\right] \tag{7}$$

$$\eta_{MR} = \frac{W_n}{Q_H} = \frac{1 - \dfrac{n}{m}\left(\dfrac{T_e - T_f}{T_a - T_b}\right)}{1 + \dfrac{\Delta T_r}{m(T_a - T_b)}} \tag{8}$$

$$\eta_{MR} = \frac{W_n}{Q_H} = \frac{1 - \dfrac{T_f}{(T_a - T_b)\eta_c}\dfrac{n}{m}\left[\pi_c^{-\frac{\gamma-1}{n\gamma}}\left(1 - \dfrac{1}{\eta_t}\left(\dfrac{T_a - T_b}{T_a}\right)\right)^{-m/n} - 1\right]}{1 + \dfrac{\Delta T_r}{m(T_a - T_b)}} \tag{9}$$

where the average temperature drop across the recuperator ΔT_r is related to the recuperator effectiveness η_r as

$$\Delta T_r = (T_b - T_c) = (T_d - T_e) = (1 - \eta_r)(T_b - T_e) \tag{10}$$

Multiple Reheat and Intercooling Brayton Cycle Design Procedures

Using these equations, a parametric search was used to identify promising design parameters under these design constrains. Two groups of important design parameters are used to determine the cycle efficiency. The first group is fixed, including

- thermal power 2400 MW,
- turbine inlet temperature 900°C,
- recuperator effectiveness 0.95,
- compressor efficiency 0.88,
- turbine efficiency 0.93,
- generator efficiency 0.986, and
- system pressure 10 MPa.

The second group is adjustable, including

- numbers of expansion and compression stages,
- turbine outlet temperature or total pressure ratio (defined as the ratio of the maximum pressure over the minimum pressure in the cycle), and
- the system fractional pressure loss coefficient.

The total pressure ratio can be derived from other parameters if the turbine outlet temperature is chosen as a variable; so is for the turbine outlet temperature if the total pressure ratio is chosen as a variable. As a first order of approximation, one can use the following estimation for the total fractional pressure loss (total pressure loss over system pressure):

$$\pi_c = 1 - (c_r + c_T \cdot m + c_C \cdot n), \tag{11}$$

For the first iteration in solving for cycle efficiency, c_r=0.01 is used for the fractional pressure loss for recuperator, c_T=0.01 fractional pressure loss for one stage of heating and expansion (heater, turbine, and related ducting loss), and c_C=0.005 fractional pressure loss for one stage of cooling and compression (cooler, compressor, and related ducting loss). These parameter values are only used as the initial estimation of system fractional pressure loss. After detailed design for major components such as turbomachinary, heaters, coolers, recuperator, and ducting system is finished, the fractional pressure loss value is updated according to each component's pressure loss.

The design procedures include the following steps: firstly the numbers of expansion and compression stages are specified, and then turbine outlet temperature according to maximum heat source temperature, total pressure ratio, mass flow rate and recuperator power; basing on these parameters, each component is designed, such as heaters, turbomachines, coolers, recuperator and ducts; finally, the system pressure loss is calculated and the design process repeated if there exists large difference between the calculated fractional pressure loss value and the estimated value according to equation (11). Due to relatively detailed design for components such as heat exchangers, turbomachinary, and duct systems, relatively accurate total pressure loss can be obtained, which results in more credible net efficiency estimation than for thermal dynamic analyses that do not include detailed pressure drop evaluations.

The following section describes how to select these parameters:

1. **Select the ratio of expansion stages over compression stages:** as shown in Figure 2, more stages of compression corresponding to one stage of expansion could increase efficiency. But too many stages of compression result in too complex and expensive systems. Each turbine along with two compressors can increase net electricity production by around 5% relative to the case of one turbine along with one compressor. Therefore, two stages of compression are chosen for each stage of expansion, which is also the ratio of compression to expansion stages used in the GT-MHR PCU design.
2. **Select the number of expansion stages:** with the ratio of expansion stages over compression stages is determined, the number of expansion stages can be selected. Also from Figure 2, thermal efficiency increases with the increase of the number of expansion stages until that too many expansion stages result in too much pressure loss. Too many stages of expansion will result in very large total pressure ratio as shown in Figure 3. Very large pressure ratio will make the recuperator smaller, but will complicate the designs of the recuperator (very large pressure difference between hot side and cold side) and low pressure turbine (low power density and large blade diameter), therefore at most 4 stages of expansion is feasible. For many cases, 3 expansion stages may be the best choice with 6 stages of compression.
3. **Select the turbine outlet temperature:** turbine outlet temperature must be lower than the turbine inlet temperature and higher than the salt freezing temperature. As shown in Figure 4, within a range, thermal efficiency is not very sensitive to the turbine outlet temperature. Higher turbine outlet temperature means smaller total pressure ratio (shown in Figure 5). As shown in Figure 6, mass flow rate increases

with turbine outlet temperature. Larger mass flow rate requires larger turbomachinary and larger duct diameters. Higher turbine outlet temperature also means larger and more expensive recuperator because turbine outlet temperature decides the high temperature construction material choice (shown in Figure 7, relative recuperator power is defined as the ratio of recuperator power over the total thermal power). Balancing all these factors, 650°C is chosen as the turbine outlet temperature so that conventional materials could be used for the recuperator and the hot cross-over leg ducts.

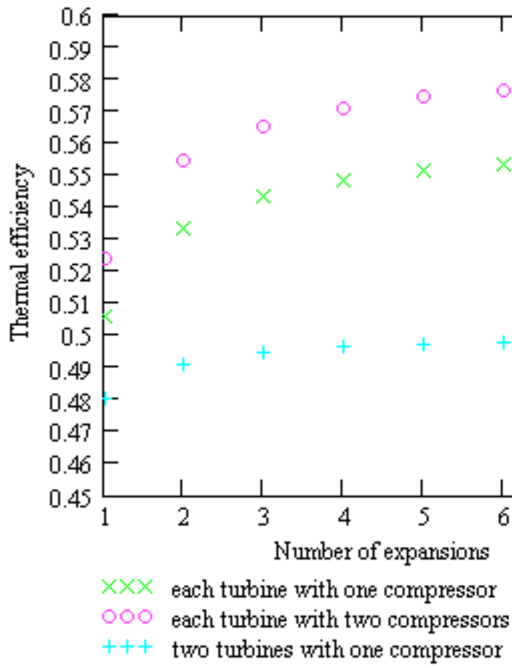


Figure 2: Thermal efficiency variation with the numbers of expansion stages and compression stages.

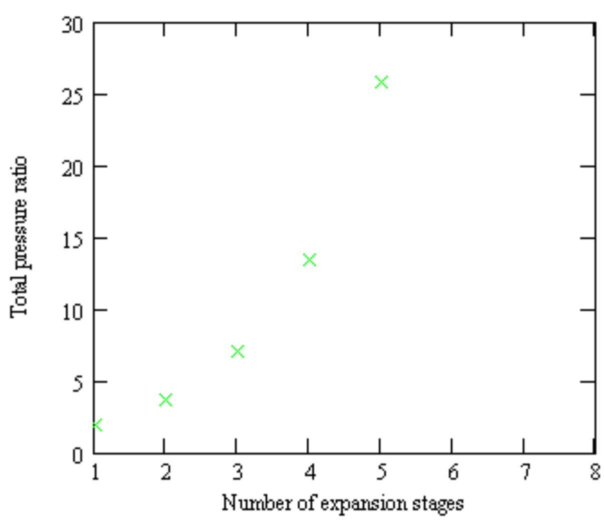


Figure 3: Total pressure ratio variation with the number of expansion stages.

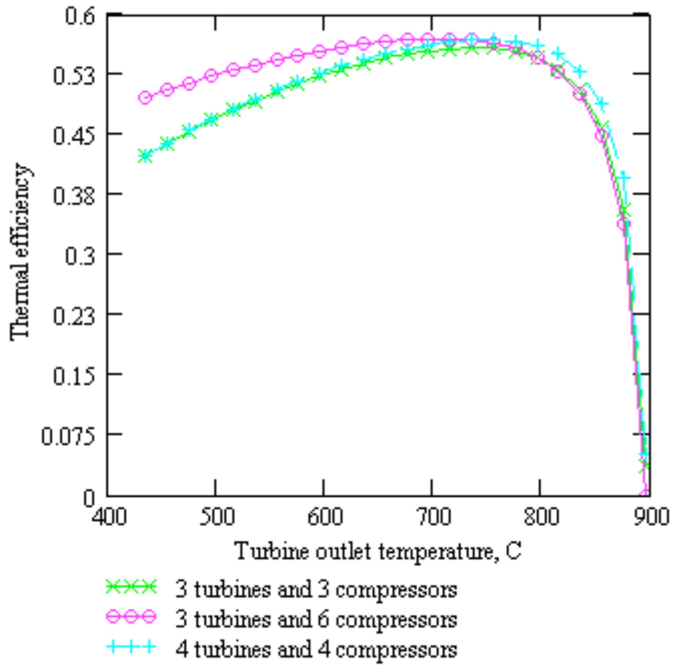


Figure 4: Thermal efficiency variation with turbine outlet temperature.

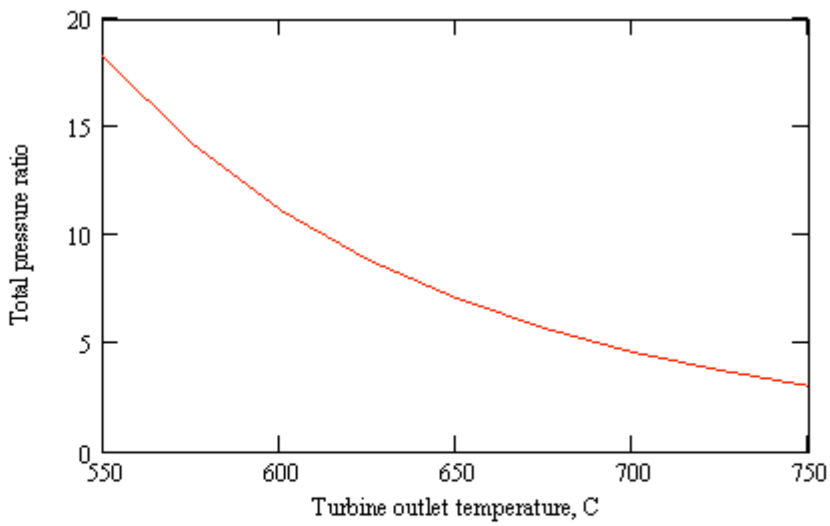


Figure 5: Total pressure ratio variation with turbine outlet temperature.

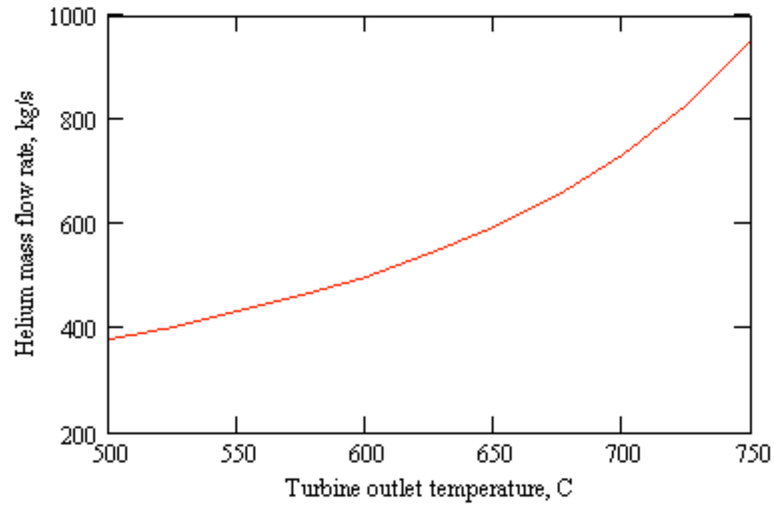


Figure 6: Helium mass flow rate variation with turbine outlet temperature.

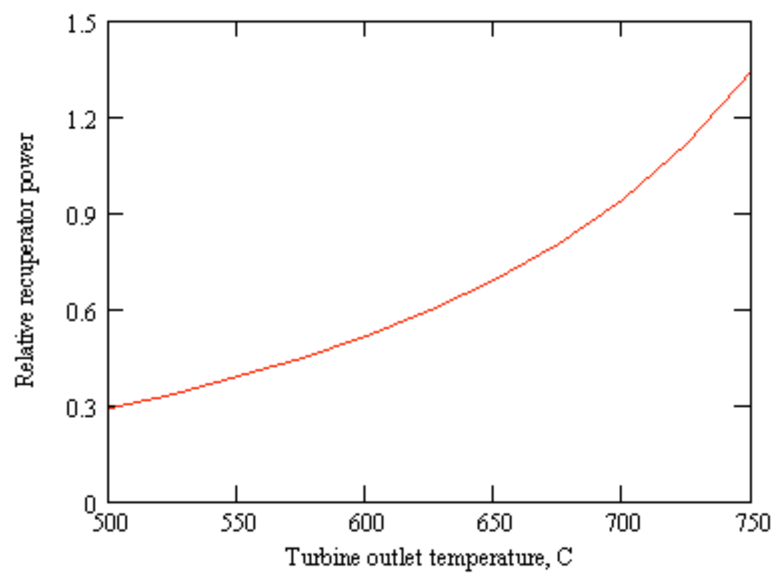


Figure 7: Recuperator power, relative to total thermal power, variation with turbine outlet temperature.

PCU Design Considerations

There are several major design choices to make when considering a multiple-reheat helium Brayton power conversion system [Peterson, et al., 2004]:

- horizontal shaft versus vertical shaft
- single shaft versus multiple shafts
- integrated system versus distributed system
- high temperature heat exchangers selection
- turbine design choice

- active pressure vessel cooling

Vertical versus Horizontal Turbomachinery

Turbomachinery can be oriented either vertically or horizontally. The orientation affects the compactness of the system, the optimal design of ducting between turbomachinery and heat exchangers, and specific aspects of turbomachinery design. In existing industrial practice for steam turbines and open-cycle gas combustion turbines, horizontal orientations are most commonly found. For hydroelectric turbines, the vertical orientation is most common. For high-pressure direct gas cycles, such as GT-MHR, designers have commonly evaluated and selected vertical turbomachinery orientations, to provide a reduced PCS footprint area and building volume, and to simplify the ducting arrangement to modular recuperator and intercooler heat exchangers which commonly optimize to an annular, vertical configuration. However, the PBMR has recently switched from vertical to horizontal orientation for its turbomachinery, and also switched from a multiple-shaft to a single-shaft turbo-compressor-generator design. Depending on which type of system is selected for upcoming VHTR (Very High Temperature Reactor) projects, the construction and operating experiences from early VHTR PCU demonstrations (for example, a high temperature Brayton cycle, coupled to a gas-cooled VHTR, without reheat) will provide main basis for designing and constructing a large multiple reheat high temperature helium Brayton power conversion system.

The selection of vertical versus horizontal turbomachinery raises design issues in several areas, which must be considered in detailed design evaluation:

- Bearing design
    - Magnetic bearings and catcher bearings
    - Dry-gas seals and oil lubricated bearings
- Rotor dynamics
- Ancillary equipment
    - Turning gears
    - Gas seals
    - Flexible couplings

Single Shaft versus Multiple Shafts

In single-shaft systems, compressors, turbine, and generator are integrated to a single shaft, which may also include flexible couplings or reducing gears. In multiple-shaft systems, the design consists of several independent turbomachinery-generator modules arranged in an efficient cluster. Each PCU module typical produces equal electrical power. Flow passes through each module and then through an independent recuperator module. The pressure ratio across each module is same and the pressure ratio across the total system is the product of the module pressure ratios. Within each module, there is one turbine, one heater, two compressors, two coolers and one generator. For these multiple-shaft configurations, the compressor load prevents over-speed transients upon loss of generator load or field.

Single-shaft systems are potentially easier to control and might have smaller footprints than horizontal multiple-shaft systems. When the total power is large, the PCU pressure vessel becomes very large. This is difficult to construct, especially for an integral vertical single-shaft system. There exist other challenges for single-shaft configurations such as isolating different pressure zones and long shaft rotor dynamics. For a multiple-reheat system, each stage of expansion and compression operates at a different pressure level. It is highly desirable to separate these different pressure zones so that the pressure boundary can operate at the local compressor cooler inlet pressure and temperature, and the internal hot-duct and turbine-shroud components can operate with minimal pressure differentials to reduce long-term creep deformation.

Integrated System versus Distributed System

PCS components can be located inside a single pressure vessel (e.g. GT-MHR), or can be divided among multiple pressure vessels (e.g., PBMR). This is a major design choice, with important impacts in several other areas of design and maintain. This study developed point designs to compare distributed and integrated PCS designs, and finds that the power density and efficiency of the integral and the distributed designs can be comparable. The distributed designs require substantially larger volumes for connecting ducts, which is expected due to the greater distances between components. The integrated system tends to have larger volumes dead volumes inside pressure vessels. Careful designs therefore can bring the difference in the power density to a small range, but the integrated system usually has a somewhat larger power density and efficiency. Integrated systems usually have somewhat smaller footprint and smaller building volume than a distributed system.

The distributed and integrated PCS configurations generate different issues for surveillance and maintenance activities. The reduced footprint of the integrated systems, and the more compact configuration of their PCS, reduces the laydown space for maintenance activities. Thus integrated designs require very careful attention to design to permit surveillance and maintenance, and careful review of these designs.

High Temperature Heat Exchangers Selection

All the high temperature heat exchangers are assumed compact heat exchangers to increase power density. Heaters are the most difficult components to design and manufacture due to their operation at the highest operating temperature and pressure difference. Ceramic compact plate fin heat exchangers and metallic PCHE type heat exchangers are considered for long-term application and near-term application, respectively. For heat transfer with liquid coolants like liquid salts, heaters will be immerged in high-pressure helium environment to reduce helium flow misdistribution and improve thermal effectiveness. Coolers and recuperator designs can be essentially identical to those for the GT-MHR PCU.

Compact plate heat exchangers are already commonly used for heat transfer at lower temperatures. Of great interest for advanced high temperature Brayton cycles is the potential to fabricate compact plate type heat exchangers that would provide very high surface area to volume ratios and very small fluid inventories while operating at high temperatures. Both metal and non-metal heat exchangers are being investigated for high-temperature, gas-cooled reactors for temperatures to 1000ºC. A recent high temperature heat exchanger study for nuclear hydrogen production [Peterson, et al. 2003a, 2003b] has suggested that carbon-coated composite materials such as liquid silicon infiltrated (LSI) and polymer infiltrated (PI) chopped fiber carbon-carbon preformed material potentially could be used to fabricate high-temperature plate fin heat exchangers.

LSI and PI carbon-carbon composites can maintain nearly full mechanical strength at high temperatures (up to 1400°C), have low residual porosity and are compatible with liquid salt and high-pressure helium. These materials are relatively simple to fabricate and have relatively low cost. Surfaces to be exposed to liquid salts would be coated with carbon, using chemical vapor infiltration (CVI) or chemical vapor deposition (CVD). Such methods have been developed at Oak Ridge National Laboratory (ORNL) for coating carbon-carbon composite plates for fuel cells [Besmann, T. M., 2000]. ORNL has subjected samples treated by CVI of carbon to 100 MPa stresses in bi-directional bending of plates. These samples were then tested for hermeticity by pressurizing one side with 206 kPa of hydrogen and measuring the through-thickness gas leakage rate, and it was found that excellent permeation resistance could be achieved. Recently UC Berkeley performed high-pressure helium permeation tests on a CVD SiC and Pyrolytic Carbon coated LSI composite sample. The coated sample kept excellent hermeticity up to 280 MPa maximum tensile stress before breaking, which is much higher than 9 MPa maximum tensile stress (under 10 MPa He) typically found from our stress analysis for heat exchanger unit cells. This test directly verified that CVD carbon coated SiSiC material can keep helium hermeticity under high pressure and stress well beyond the working pressure and stress expected for typical high temperature reactor power systems. From the perspective of protecting the substrate material from the liquid salt, some porosity of the carbon layer could be acceptable, as is found for nuclear graphites. Liquid salt does not wet graphite, which means fouling deposits are difficult to form in liquid salt side channels.

To fabricate compact plate-type heat exchangers, one side of each plate is die-embossed or milled, to provide appropriate flow channels, leaving behind fins or ribs that would provide enhanced heat transfer, as well as the mechanical connection to the smooth side of the next plate. For green carbon-carbon material, milling can be performed readily with standard numerically controlled milling machines. Alternatively, plates can be molded with flow channels, as has been demonstrated for carbon-carbon composite plates fabricated at ORNL for fuel cells. For assembly, the ends of the fins and other remaining unmachined surfaces around the machined flow channels would be coated with phenolic adhesive, the plate stack assembled, header pipes bonded and reinforced, and the resulting monolith pyrolysed under compression. Then liquid silicon or polymers would be infiltrated to reaction bond the plates and headers together, forming a compact heat exchanger monolith.

Figure 8 illustrates discontinuous fin geometry for liquid salt-to-helium compact heat exchangers. The cross-sectional area of the fins and the thickness of the remaining plate below the machined channels would be adjusted to provide sufficient strength to resist thermal and mechanical stresses. For the salt-to-helium heat exchangers in the power conversion system, mechanical stresses are relative large. For the case in which the heat exchanger is immersed into a helium environment, detailed stress analysis has indicated that the stresses are dominantly compressive and can be accommodated with relative ease. Figure 9 shows a preliminary draft plate design for the compact offset fin plate heat exchangers. Helium plates and liquid salt plates are alternatively joined together to form a heat exchanger module. Pressure losses were estimated for the liquid salt side in an actual size heat exchanger module with similar flow distribution structure to that shown in Figure 9. The result shows that flow nonuniformities in both transverse and stack height directions are very small. This type of distribution design has a relative high salt pressure loss. However, due to the small volumetric flow rate of liquid salt, the total pumping power for liquid salt is acceptable and does not affect the net plant efficiency. This design is applicable to compact counterflow heat exchanger, where there is a substantial difference between the volumetric flow rates of the two fluids. Applications can include liquid-to-gas heat exchangers and gas-to-gas heat exchangers where there exists a large difference in the volumetric flow rates. The helium side plate has several equal spacing flow dividers to reduce flow misdistribution. Initial mechanical stress analysis shows that the stress on the helium divider in the distribution region is very low. Most of it comes from the compressive force of 10 MPa helium on the divider itself. Further thermal stress analysis is needed to study the thermal response under transient conditions.

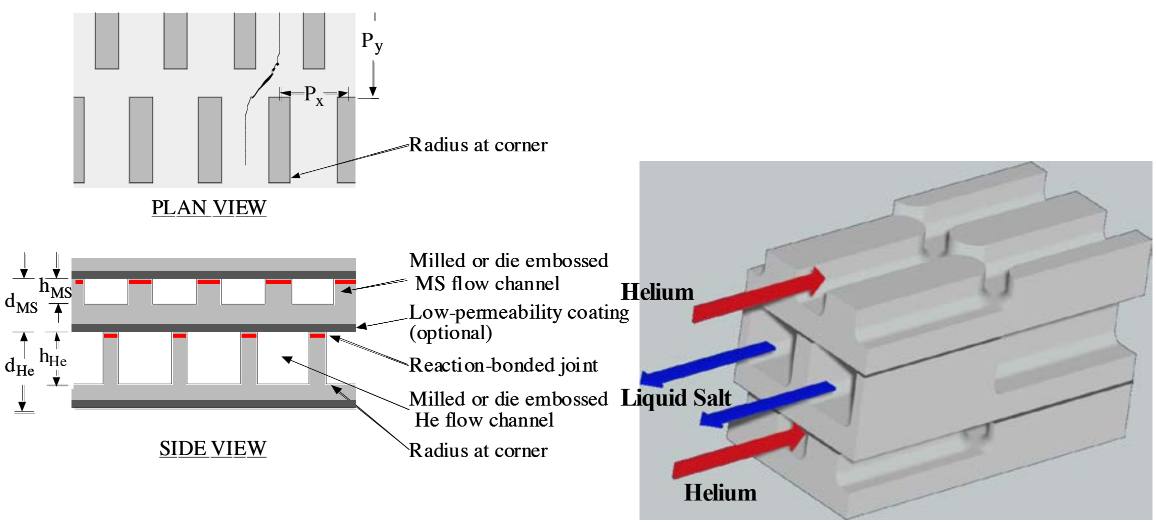


Figure 8: Cut away view through a plate showing alternating liquid salt (top and bottom arrows) and helium (middle arrows) flow channels. Dark bands at the top of each fin indicate the location of reaction-bonded joints between each plate.

Figure 10 shows the idea to arrange heater modules into the annual space around turbine in a vertical integral PCU system. Helium flows along circumferential direction through heaters. Between two rows of heater modules, there are plates to separate hot outlet helium from a heater and warm inlet helium into another row of heaters. The inlet helium flow into the space among heater modules from the top and the hot helium is collected at the bottom then goes to turbine inlet. In distributed PCUs, heaters are located in independent vessels. Heater modules are arranged in similar way as integrated system with smaller vessel diameter and higher height as shown in Figure 11.

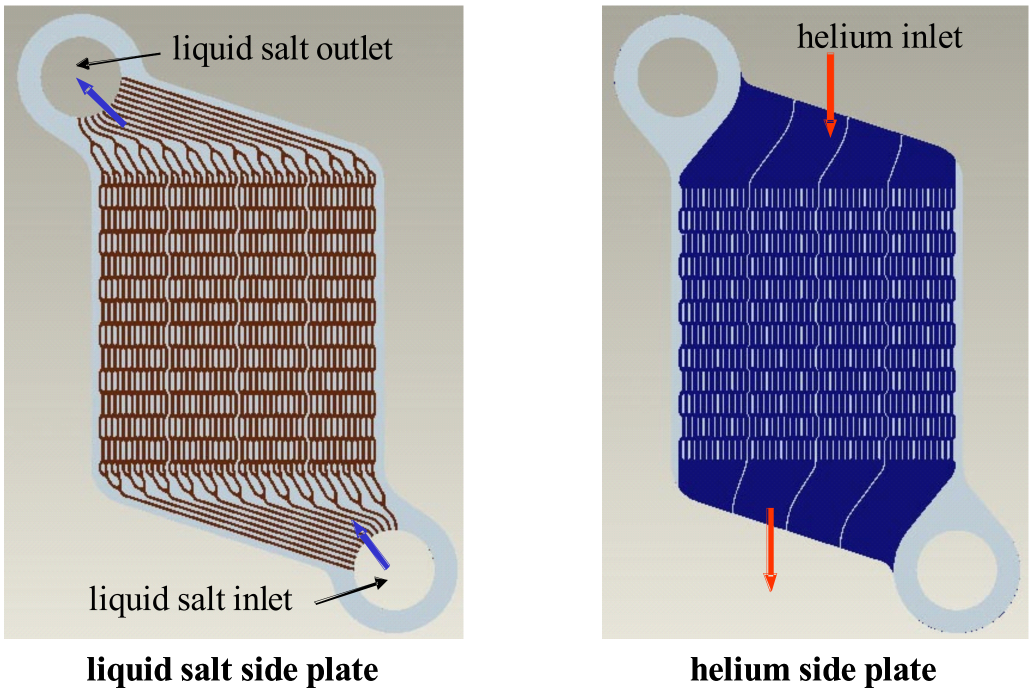


Figure 9: Plates design for compact offset fin plate heat exchanger.

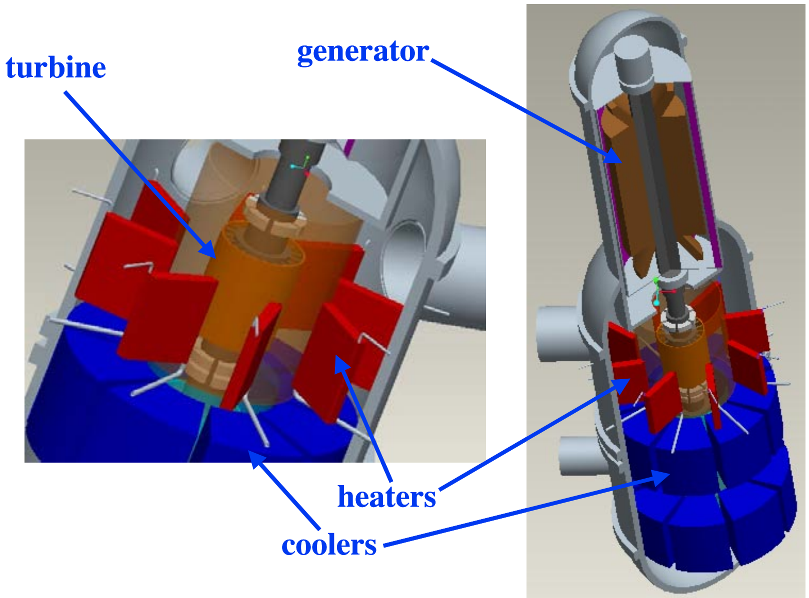


Figure 10: Schematic showing arrangement of major PCU components in integrated system.

Although ceramic heat exchangers have very good potential to be used in long-term advanced Brayton cycle PCS systems, they still need significant development work. One of near-term choice for high temperature salt-to-helium heaters are PCHE heat exchangers such as Heatric heat exchangers. PCHE are constructed from a process of chemically etching metal plates and diffusion bonding (heatrics.com) as shown in Figure 12. UCB has designed a PCHE type high temperature metal heat exchanger for the NGNP IHX application (Peterson, et. al., 2004) as shown in Figure 13 to transfer heat from primary helium to intermediate liquid salt side. Similar modules can be used as heaters in a distributed PCS system. PCHE type heaters for multiple-reheat cycle have lower power density than ceramic compact heaters (50 MW/m^3 versus 100 MW/m^3). Heater modules can also be arranged in the annual space in a vertical vessel. Helium flows into the heaters from outside and hot helium is collected in the vessel center.

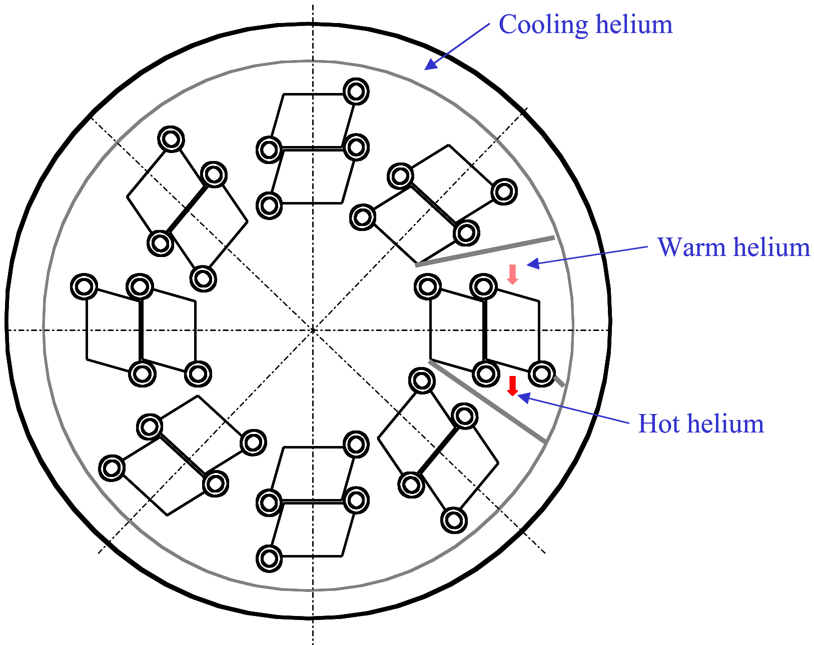
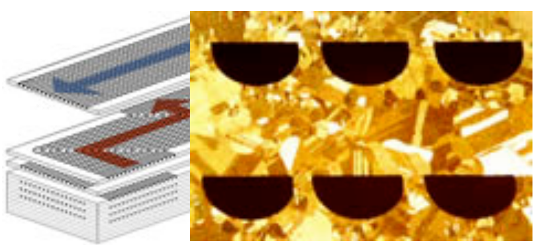


Figure 11: Schematic showing arrangement of heater modules in an independent heater vessel for a distributed system.



Flow channels in a PCHE Plates are stacked then diffusion bonded

Figure 12: Heatric heat exchanger (Credit of Heatrics)

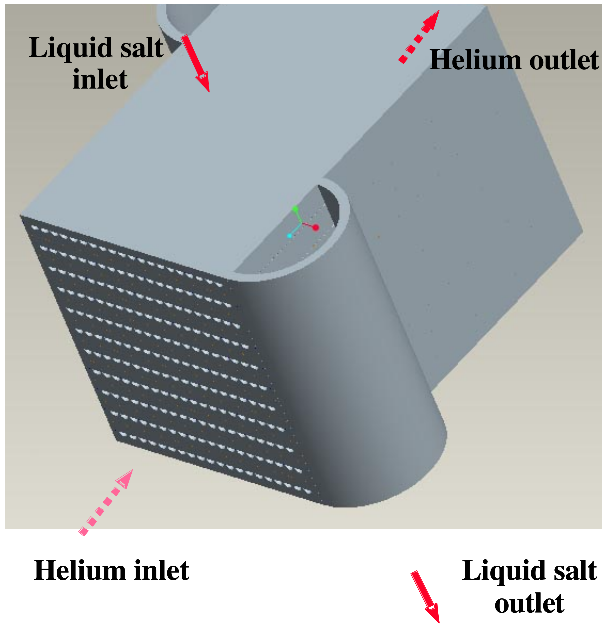


Figure 13: PCHE type HX module design for NGNP IHX. Liquid salt side will subject to the ambient helium pressure. No manifold is needed for helium flow distribution.

Turbine Design Choices

Depending upon the manufacturer, the upper temperature limit for the use of conventional, uncooled turbine blades is between 800°C and 900°C. In combustion turbines, higher temperatures are achieved by using blade cooling, where cool gas is injected through flow passages in the blades. However, for helium and helium-nitrogen mixtures, the gas thermal conductivity is up to 5 times greater than for pure nitrogen or air. In this case, blade cooling encounters difficulty because the temperature drop from the gas to the blade surface decreases and the temperature drop inside the blade increases, increasing blade thermal stresses. Thus the demonstration of turbine operation at temperatures above 900°C will require the development of advanced coatings for turbine blades, or ceramic composite blades. Carbon composites, which cannot be used in high-temperature combustion turbines due to oxidation, could be an option for the closed-cycle inert-gas operation of the multiple-reheat systems.

Turbine blade root stress is proportional to the density of blade material. If conventional metal blade is assumed, split flow and double split flow gas paths may need to be used for the intermediate pressure (IP) and low pressure (LP) turbines respectively in order to limit blade root stresses to thresholds that are achievable with uncooled metal blades. With a single flow LP turbine, the nominal blade root stress would have increased by a factor of 3.4. If carbon composite material is used for blade, split flow for turbines can be avoided. The resulting systems have much less complexity and much smaller PCU volumes. Therefore, ceramic composite turbine blades, except where specially noted (one case is considered in this study for a near-term case with metal heaters), are assumed to be used to avoid using split turbines.

Active cooling

One of major choice for pressure boundary design is between vessel materials that can operate at high temperature and conventional LWR vessel materials that require active cooling by compressor outlet flow. These choices affect the PCS power density and efficiency. Because the multiple-reheat cycle system tends to optimized at high pressure, all the pressure vessels and ducts are actively cooled so that high system pressure is feasible and high pressure vessel material cost is kept at reasonable level. In practice, for multiple-reheat closed gas cycles, it is easy to develop configurations where the pressure boundary operates at the compressor outlet temperature, and all hot components are submerged inside the cold boundary.

2. MULTIPLE-REHEAT CYCLE SYSTEM CONFIGURATION BASED ON THE GT-MHR PCU DESIGN – INTEGRATED VERTICAL MULTIPLE-SHAFT SYSTEM

UCB has developed multiple-reheat systems based on the GT-MHR PCU design (LaBar, 2002). These systems use multiple vertical shafts, integrated PCUs (three PCUs), and a separate recuperator vessel. This section describes this study and its results.

Deriving Multiple-Reheat PCUs from the GT-MHR PCU design

The General Atomics GT-MHR PCU is shown in Figure 14, and is currently among the few closed helium cycle systems that have undergone detailed engineering design analysis, and is the only system that has turbomachinery which is sufficiently large to extrapolate to a >1000 MW(e) multiple-reheat cycle power conversion system. Analysis presented here shows that, with relatively small engineering modifications, multiple GT-MHR PCU's can be connected together to create a multiple-reheat cycle power conversion system for >1000 MW(e) class high-temperature power plants. The resulting power conversion system is quite compact, and results in what is likely the minimum helium duct volume possible for a multiple-reheat system. To do this, compact offset plate fin type salt-to-helium heat exchangers (power densities from 80 to 120 MW/m^3) are inserted in the annular space around the turbines as shown in Figure 10 and 14, currently occupied by the upper set of recuperator heat exchangers in the GT-MHR design, and the multiple-reheat cycle recuperator is moved to a separate pressure vessel. High density difference between liquid and helium allows a large frontal area for helium flow (as in a typical car radiator), giving high effectiveness and small helium pumping power. Locating heaters in annular arrangement around turbines gives very short hot-gas flow path. Because they are submerged in the helium environment, the heaters are loaded primarily in compression, a potential advantage for ceramic heat exchangers.

Figure 15 provides a schematic diagram of the reference multiple-reheat cycle flow configuration and Figure 16 shows hot and cold leg configurations for the reference multiple-reheat cycle using three PCU modules (high pressure (HP), middle pressure (MP), and low pressure (LP)) and a separate recuperator vessel (R). An upper hot leg and a lower cold leg connect each pair of PCUs. A separate recuperator vessel is also connected to the low-pressure and high-pressure PCU's with similar hot and cold legs. As shown in Figure 14, the hot legs connect the PCU vessels at the elevation of the turbine outlets. Flow is collected from the turbine outlet diffuser and crosses the hot leg to the next PCU vessel. This hot-leg flow enters the top of an annular ring of compact heat exchangers and flows downward, to be heated to turbine inlet temperature, and then is ducted directly into the next turbine inlet, resulting in a very short hot-gas flow path. Current calculations for the frontal area, flow path length, and volume of these heaters indicate that they can fit without problems in the annular volume around the turbine, currently occupied by the upper recuperator bank of the current GT-MHR PCU design.

Likewise, the cold legs connect the PCU's at the elevation of the compressor outlets. Flow is collected from the compressor diffuser, and approximately 90% of the flow crosses the cold leg, and enters the top of an annular ring of coolers to flow downward, to be cooled and then go directly into the next compressor inlet. Approximately 10% of the cold flow is bypassed upward to flow through an annulus around the hot-leg duct, so the hot leg pressure boundary is maintained at the same temperature as the cold-leg boundary to minimize thermal stresses due to the PCU vessels being connected at two elevations by cross-over legs. The cold cross-over leg eliminates the vessel volume and pressure drop that would be required to bring 100% of the cold flow to the hot-leg elevation to flow across in an annular duct, as is done with direct-cycle gas-cooled reactors. With this configuration, for the recuperator the low-pressure turbine discharges its gas into a hot leg going over to the top of the recuperator vessel, and the low-pressure gas flows down through the recuperator and then returns to the low-pressure compressor in the low-pressure cold leg. Likewise, the discharge from the high-pressure compressor flows across in the high-pressure cold leg to the recuperator vessel, and flows upward through the recuperator to be heated, and then across in the high-pressure hot leg to the high-pressure PCU.

In designing closed helium cycles, a major cost driver is the volume of ducting required to transfer helium between equipment, because it affects the cost of the pressure boundary. This multiple-shaft design with annular rings of heaters around each turbine provides a highly optimized configuration, because all flows enter at the optimal elevation in the vessels and the hot gas flow path is extremely short.

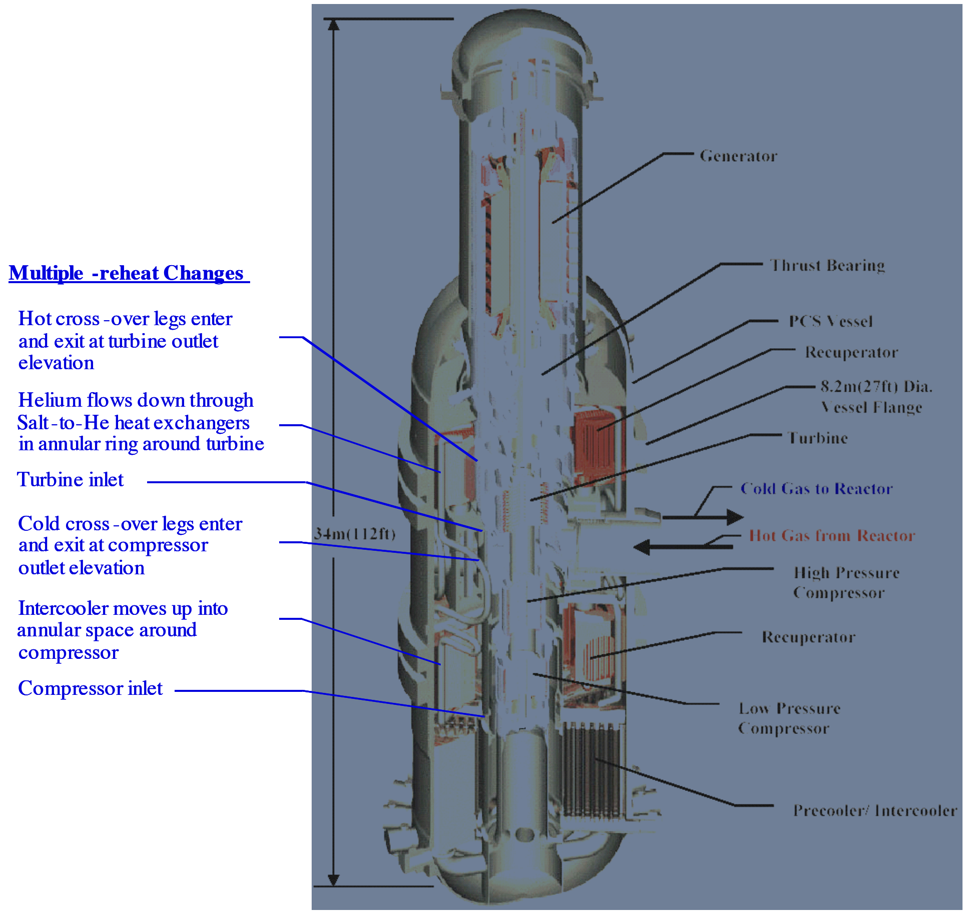


Figure 14: Cross section of the current GT-MHR PCU, with changes required for the multiple-reheat cycle indicated on left.

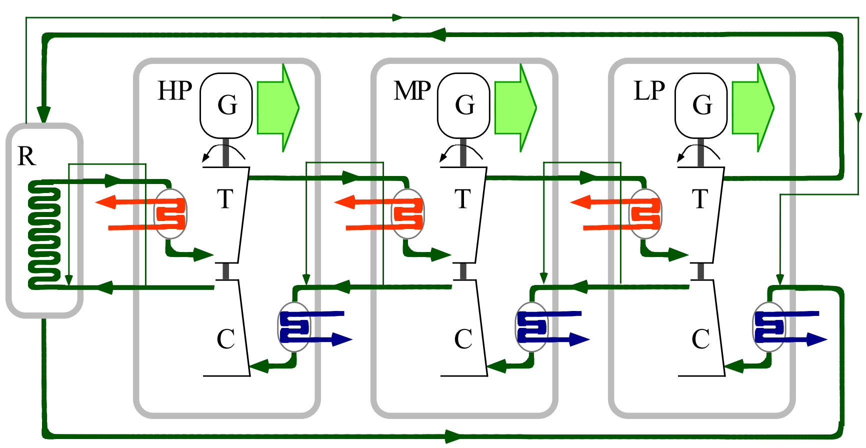


Figure 15: Schematic flow diagram for the reference three-expansion-stage multiple-reheat cycle, using three PCU modules (HP, MP, and LP) each containing a generator (G), turbine (T), compressor (C), and heater and cooler heat exchangers, with a recuperator (R) located in a fourth vessel.

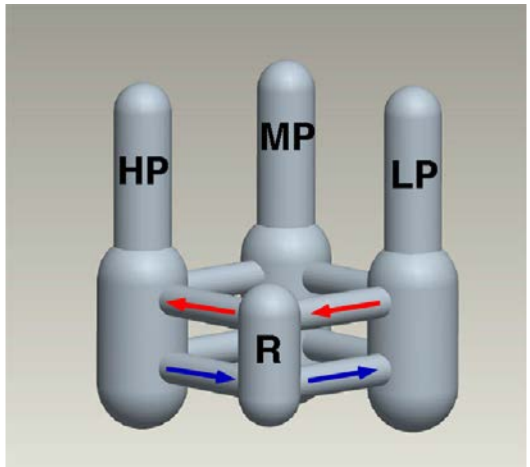


Figure 16: Hot and cold leg configurations for the multiple-reheat cycle based with three (HP, MP, and LP) PCU's and a separate recuperator vessel (R).

Main Design Parameters and Results

Table 1 summarizes the power conversion thermal dynamics design parameters for three peak temperature options and the reference GT-MHR design, and Table 2 gives the power conversion design main system sizes and power density. All the salt multiple-reheat cycles use three PCUs and each has one turbine and two compressors. For turbine inlet temperatures of 900ºC (AHTR-VT), 750ºC (AHTR-MT), and 675ºC (AHTR-LT), the net thermal efficiencies are 56%, 51%, and 48% and corresponding PCU power densities are 560kW(e)/m^3, 490kW(e)/m^3, 460kW(e)/m^3, respectively. Even the low temperature option (LT) has higher efficiency than GT-MHR (46%) although GT-MHR uses much higher turbine inlet temperature (848ºC).

The AHTR-LT has a PCU power density two times as the GT-MHR PCU. The much higher power densities for multiple-reheat cycle systems are due to several reasons: higher system pressure, higher thermal efficiency due to multiple reheat and intercooling, smaller recuperator specific power, and more compact and shorter helium flow path arrangement. The multiple-reheat cycle systems have similar or smaller PCU vessel diameters as GT-MHR and the multiple-reheat cycle PCU vessels are shorter than GT-MHR PCU vessel. The multiple-reheat cycle turbines and compressors are slightly larger than GT-MHR due to higher powers. Table 3 shows the heat exchanger designs for the AHTR-VT system. These heat exchangers are described in section 1, and use a compact offset-fin plate configuration with 1-mm thick plates, 1-mm high liquid-salt fins, and 2.0-mm high helium fins. Here the small volume and very short flow length of the heaters are notable, showing that they can be fit with relative ease into the annular volume around the turbines in each PCU as shown in Figure 10. Table 4 shows the turbomachinery parameters for the high-temperature helium multiple-reheat cycle. The diameter of the turbomachinery is quite similar to the GT-MHR PCU (1.7-m tip diameter for compressor, 2.0-m for turbine), and the length of the turbomachine rotor is somewhat shorter. Turbine blades are assumed to be ceramic material. The generators for the multiple-reheat cycle PCU's will be a little larger than the GT-MHR generator, due to their larger power output. But the main multiple-reheat cycle PCU vessels will be shorter than the GT-MHR, because the coolers move upward, and the volume of each multiple-reheat cycle PCU will be similar or smaller than the GT-MHR PCU.

Multiple reheat cycle PCUs use much less metal per unit electricity output than GT-MHR PCU (7.5 MT/MWe(ave)) as shown in Table 2. AHTR-VT only needs half of specific metal as GT-MHR PCU needs. This may imply a 50% even more capital cost saving for AHTR-VT PCUs system relative to GT-MHR PCU, because both of systems operate under similar high temperature and have similar size, while AHTR-VT is indirect cycle and GT-MHR is direct cycle. The same material for nuclear island usually costs twice more than non-nuclear application. Multiple reheat cycle systems need less than half of helium mass per MWe than GT-MHR. This will reduce the cost of helium storage and helium clean system, and also can ease the tritium management.

	AHTR –VT	AHTR –IT	AHTR-LT	GT-MHR
Primary Max. /Min. Temperature (°C)	1000/900	800/700	705/670	848/488
Intermediate Max. /Min. Temperature (°C)	920/860	770/690	690/620	N/A
Turbine Inlet/Outlet Temperature (°C)	900/650	750/570	675/495	848/508
Compressor Inlet/outlet Temperature (°C)	35/86	35/76	35/80	26.4/110.3
System Pressure (MPa)	10	10	10	7.24
Number of PCU's	3	3	3	1
Numbers of Turbines and Compressors	3/6	3/6	3/6	1/2
Helium Mass Flow Rate (kg/s)	594	818	824	317
Cycle Pressure Ratio	7.04	4.82	5.54	2.69
Pressure Loss Fraction	0.07	0.08	0.08	0.07
Overall Cycle Efficiency	0.566	0.515	0.48	0.46
Electrical Power (MW)	1357	1235	1151	285
The Ratio of Recuperator Power Over Electrical Power	1.2	1.6	1.5	2.2

Table 1: 2400 MWt AHTR multiple-reheat cycle system design parameters and comparison with GT-MHR PCUs

	AHTR –VT	AHTR –IT	AHTR-LT	GT-MHR
PCU Total Height (m)	32	32	32	38
Main PCU Vessel Diameter (m) (HP, MP and LP)	6 / 6 / 7.4	6.3 / 6.3 / 7.5	6.3 / 6.3 / 7.5	7.2
Generator Vessel Diameter (m)	4.8	4.7	4.6	4.4
Max. Turbine Tip Diameter (m)	1.956	2.103	2.111	1.783
Max. Compressor Tip Diameter (m)	1.857	2.016	2.016	1.684
Heater Core Power Density (MW/m^3) (HP, MP and LP)	123 / 97 / 84	81 / 64 / 57	74 / 59 / 52	N/A
PCUs Power Density (kW(e)/m^3)	560	490	460	230
Specific Metal Mass for PCUs (MT/MWe) (Capacity factor: 0.9)	3.7	4.3	4.5	7.5
Specific Helium Inventory (kg/MWe)	5.6	7.4	7.7	15.9

Table 2: 2400 MWt AHTR multiple-reheat cycle system size parameters and comparison with GT-MHR PCUs

	Recuperator	Liquid Salt to Helium Heaters		
		High Pressure Heater	Middle Pressure Heater	Low Pressure Heater
Power (MW)	1495	852	774	774
T_{max} (ºC)	650	920	920	920
T_{min} (ºC)	142	625	650	650
Core Volume (m^3)	35	7.0	7.9	9.1
Flow Length (m)	0.51	0.45	0.37	0.31
Total Frontal Area (m^2)	70	15	21	30
Thermal Density (MW/m^3)	43	122	98	85
Fractional Pressure Losses for the Counter-Flow Region	0.0072	0.0028	0.0025	0.0023

Table 3: 2400 MWt AHTR helium AHTR-VT system heat exchanger design parameters

	Compressors			Turbines		
	HP	MP	LP	HP	MP	LP
Power (MW)	330	330	330	774	774	774
Inlet Temp. (°C)	35	35	35	900	900	900
Pressure Ratio	1.94	1.94	1.94	1.92	1.92	1.92
Number of Stages	19	19	17	13	13	13
Adiabatic Efficiency	0.88	0.88	0.88	0.93	0.93	0.93
Exit Dynamic Pres. over Sys. Pressure	0.57%	0.54%	0.56%	0.55%	0.55%	0.53%
Max Tip Diameter (m)	1.86	1.86	1.86	1.93	1.96	2.00
Tip Speed (m/s)	350	350	350	363	369	378
Min. Hub/Tip Ratio	0.85	0.77	0.67	0.79	0.70	0.57
Overall Length (m)	4.9	4.9	4.7	4.8	4.9	5.0

Table 4: 2400 MWt AHTR helium AHTR-VT system preliminary turbomachinery design parameters

Estimating Material Consumption for PCUs

It is difficult to precisely account for the total material consumption without detailed engineering design finished. However, it is possible to estimate main material consumptions for PCS by including major PCS components such as pressure boundary (pressure vessels and ducts), generator, turbomachinary, and heat exchangers. There are two methods to estimate the material mass for components: direct calculation and scaling. For pressure boundary masses, almost all the pressure boundaries are in the form of cylinder vessels. The wall thickness for a cylinder vessel wall can be estimated by the following equation:

$$t = \frac{P \cdot D}{2\sigma_{allowable}}, \tag{12}$$

where t is the vessel thickness, P pressure, D vessel diameter, and $\sigma_{allowable}$ the allowable stress. Therefore, pressure vessel mass is proportional to the system pressure. Heaters and recuperator masses are estimated from component design calculations by assuming materials used. The masses for coolers, generator, and turbomachinary are scaled from GT-MHR design data, according to the following equation:

$$m_{MR} = m_{GT-MHR} \frac{Q_{MR}}{Q_{GT-MHR}}, \tag{13}$$

where m_{MR} is the multiple-reheat cycle component mass, $m_{GT\text{-}MHR}$ the corresponding GT-MHR component mass, Q_{MR} the multiple-reheat cycle component capacity, and $Q_{GT\text{-}MHR}$ the corresponding GT-MHR component capacity.

Table 5 summarizes the components mass and material assumed. Pressure boundary material accounts for near half of total material consumption for the PCS. Although high-temperature alloy 9Cr-1Mo-V is assumed for pressure vessel, conventional low-temperature steel can also be used because all the pressure boundaries are under active cooling with highest temperature less than 100°C. Therefore, unit mass pressure boundary cost should be less expensive than the GT-MHR pressure boundary (which is part of nuclear island boundary and operates at high temperature).

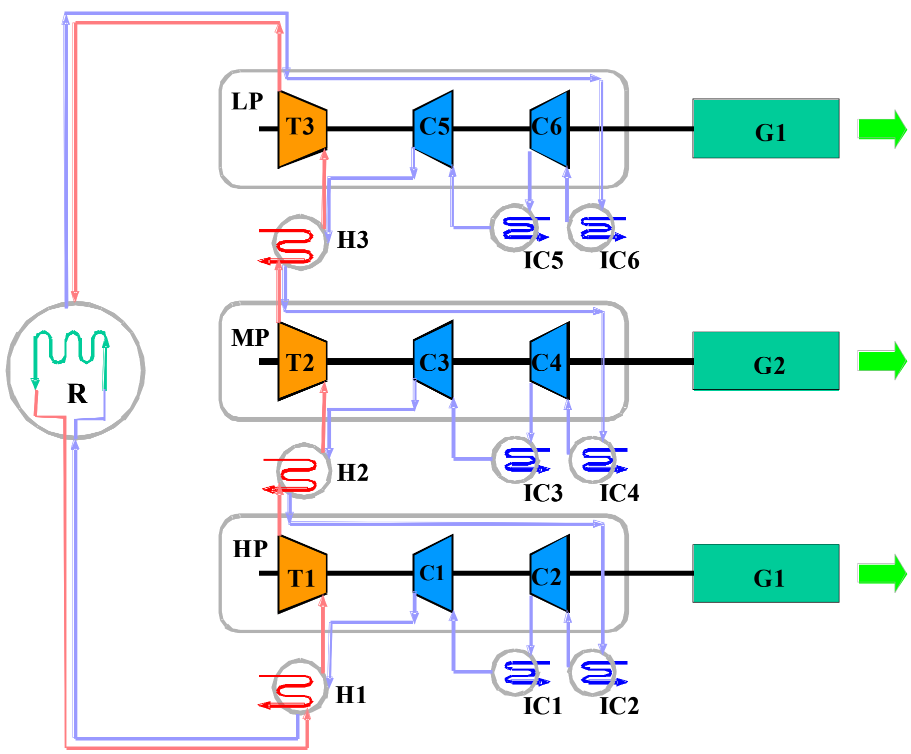
Power conversion system	Material	Metal mass, MT	Percent
Vessels and cross-ducts	9Cr-1Mo-V	2100	46%
Generators		1290	29%
Coolers	low chrome steel	650	14%
Turbomachines		240	5%
Recuperator	316L stainless steel	200	4%
Heaters	LSI C-C/SiC composite	40	1%
PCUs total		4520	100%

Table 5: 2400 MWt AHTR helium AHTR-VT system PCUs component mass and material

3. 2400 MWt HORIZONTAL-SHAFT MULTIPLE-REHEAT CYCLE SYSTEMS

This section summarizes four cases of horizontal-shaft designs for a 2400 MWt multiple-reheat cycle system, including two distributed multiple-shafts systems (one long term design and one near term design), one single horizontal shaft distributed system with one turbine vessel, and one single horizontal shaft distributed system with split-casing turbine vessels. In order to reduce flow misdistribution in the heat exchangers and directly use the current heaters and recuperator designs, all the heat exchangers are put into separated vertical vessels. In all of these designs, the pressure boundary operates at, or slightly above, the compressor outlet temperature. This is a very low temperature, and thus the pressure boundary can be made from inexpensive materials. There is also a pressure difference between the cold helium next to the pressure boundary and the hot helium in the hot ducts inside, so the hot ducts and turbine casings must operate with some pressure difference and stress too. However, insulation in these systems can allow the hot components to operate closer to the cold temperature than the hot temperature. In all modular single and multiple shaft designs except for one single shaft case, the pressure difference between the hot and cold fluids is minimized.

Figure 17 shows the schematic flow diagram for the distributed three horizontal-shaft multiple-reheat cycle, using three PCU modules (HP, MP, and LP) each containing a generator (G), turbine (T), compressor (C), and heater and cooler heat exchangers, and a recuperator (R) located in a separate vessel. The red arrows in the left part of this diagram form the hot flow loop (not a closed loop, but connected to the cold loop in recuperator), within which high temperature helium flows in hot ducts, turbines, and the recuperator. The arrangement pattern shown in the diagram is optimized to minimize the total length of hot ducts, which are much more expensive than cold ducts and generate heat losses. All the hot ducts are concentric ducts and cooled by cold helium which flows in the annulus outside hot inner ducts. The blue arrows form a larger cold loops. Figure 18 shows an example PCS layout for the distributed three horizontal-shaft multiple-reheat cycle. In this preliminary design, all the HP, MP and LP turbine vessels have flanges at the left ends, to permit the turbomachinery to be withdrawn horizontally from the vessels for maintenance. Alternatively, a split casing vessel could be chosen, but the much larger and longer vessel flange may make helium leakage a problem. The heat exchanger vessels are all configured in vertical orientations, to allow symmetric, level arrangement of the heaters in the vessel to allow uniform draining of the liquid heat transfer fluid. The tops of the vessels are flanged to allow vertical access for maintenance. The largest equipment in this system is LP turbine vessel, which has a diameter of 4.6 m and a length of 17 m. Therefore, all the components in this system could be factory-fabricated and assembled at the site to reduce construction time. One of potential concerns is helium leakage due to many connections and penetrations in the ducting system. However, the design minimizes the size of vessel flanges, and permits welded joints between assembled components, reducing the potential for leakage.



T: turbine, C: compressor, R: recuperator,
H: heater, IC: intercooler, G: generator
1 to 6: from high pressure to low pressure

Figure 17: Schematic flow diagram for the distributed three horizontal-shaft multiple-reheat cycle, using three PCU modules (HP, MP, and LP) each containing a generator (G), turbine (T), compressor (C), and heater and cooler heat exchangers, with a recuperator (R) located in a fourth vessel.

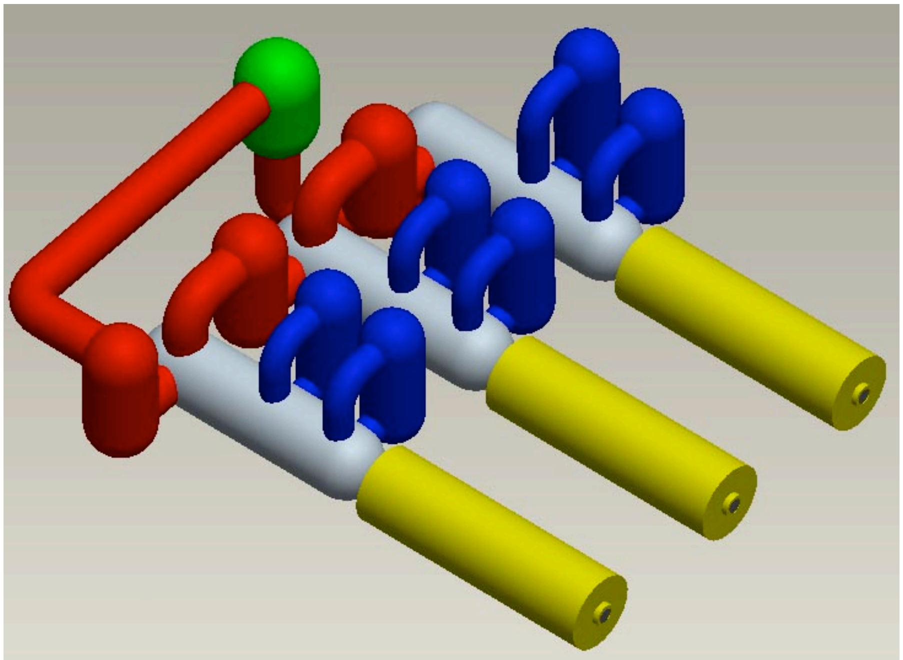
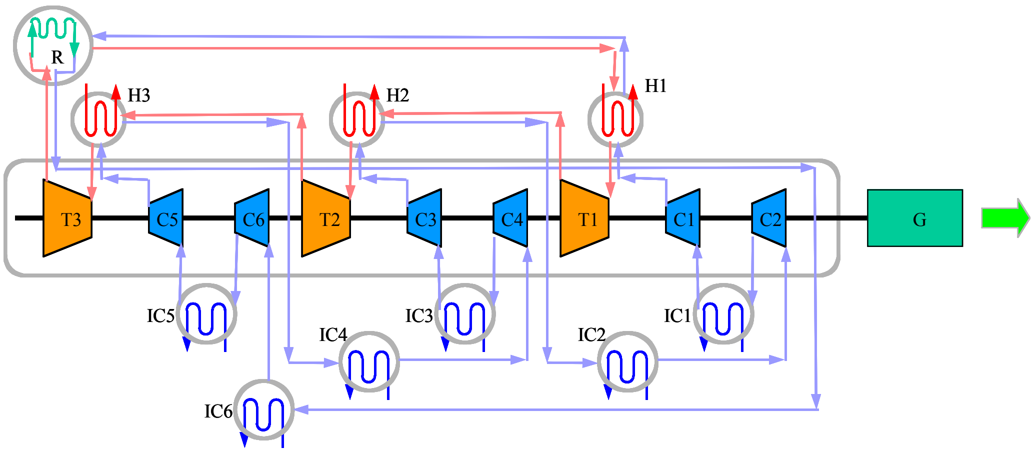


Figure 18: PCUs layout for the distributed three horizontal-shaft multiple-reheat cycle: red – hot concentric duct and heaters, blue – cold duct, gray – turbomachinery vessel, green – recuperator vessel.

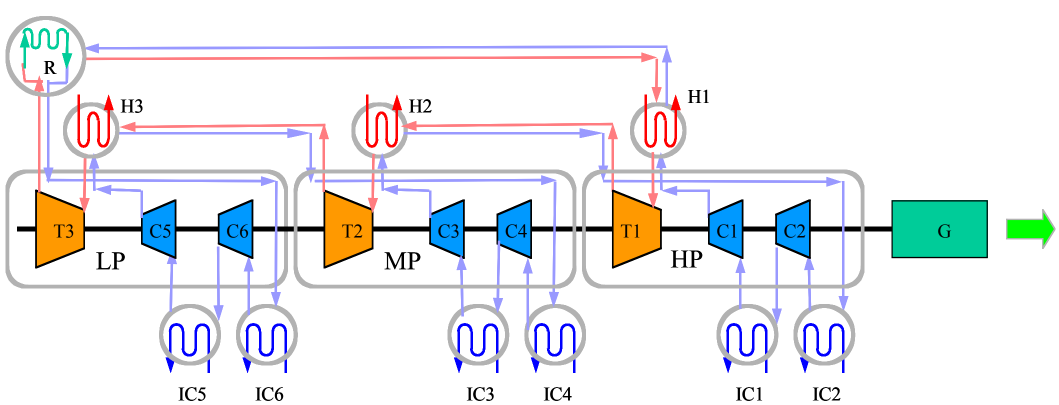
Figure 19 shows the schematic flow diagram for the distributed single horizontal-shaft multiple-reheat cycle with one turbine vessel. All turbomachinary are on the same shaft and within a big turbine vessel. The red arrows in the top part of this diagram form the hot flow loop (not a close loop, but connect to cold loop in recuperator), within which high temperature helium flow in hot ducts, turbines, and recuperator. All the hot ducts are concentric ducts and cooled by cold helium which flows in the annulus outside hot inner ducts. The blue arrows form a larger cold loop. The turbine vessel is cooled by the recuperator low pressure side exit flow. So the turbine vessel operates at low pressure. The maximum equipment size in this system is the turbine vessel, which has a diameter of 4.6 m and a length of 43 m. One of potential concerns is that HP and MP turbomachinary and inner ducts are not in pressure balance with the turbine vessel inside pressure (LP). Those hot components have to withstand the inner pressures. Overall, this design has inherent drawbacks which cannot be easily solved. Therefore, this case is presented only to show it is not good idea to arrange all the turbomachinary into a single pressure vessel, and instead it is better to provide separate vessels for the HP, MP and LP turbines and compressors, so the vessels can operate at different pressures.



T: turbine, C: compressor, H: heater, IC: cooler, R: recuperator, G: generator
1 to 6: from high pressure to low pressure

Figure 19: Schematic flow diagram for the distributed single horizontal-shaft multiple-reheat cycle with one turbine vessel, pressure decreases from 1 to 6.

Figure 20 shows a schematic flow diagram for a distributed single horizontal-shaft multiple-reheat cycle with three separate turbine vessels. All turbomachinary are on the same shaft but are located in three separate turbine vessels operating at different pressures. This arrangement solves the pressure mismatch problem shown in Figure 19. All the hot components now are cooled by cold flow with pressure close to the hot flow. The maximum equipment in this system is LP turbine vessel, which has a diameter of 4.6 m and a length of 17 m. In this configuration, split-casing vessels like those commonly used for steam turbine systems, like the Siemens turbine shown in Figure 20, must be used. Split-casing vessels have very large sealing boundaries so that controlling helium leakage may be a potential problem.



T: turbine, C: compressor, H: heater, IC: cooler, R: recuperator, G: generator
1 to 6: from high pressure to low pressure

Figure 20: Schematic flow diagram for the distributed single horizontal-shaft multiple-reheat cycle with three casing turbine vessels, pressure decreases from 1 to 6.

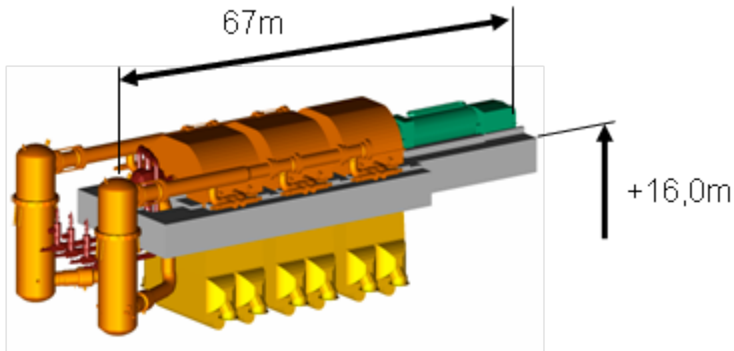


Figure 21: Turbo-generator set for the Fin 5 (EPR) - 1600 MW(e), with one HP turbine and three LP turbines.

Figure 22 shows a near-term multiple-shaft horizontal distributed multiple-reheat cycle PCUs, using PCHE heaters and metallic blade turbine. As discussed in previous sections, turbine-splitting and double turbine-splitting are necessary in order to use metal as turbine blade material at 2400 MWt. Therefore, the LP turbine vessel is much larger than HP turbine vessel. Turbine efficiency is 92%, which is lower than the value of 93% assumed for previous cases. Other aspects of the equipment configuration are same as the long-term multiple shafts horizontal distributed multiple-reheat cycle PCS shown in Figure 17.

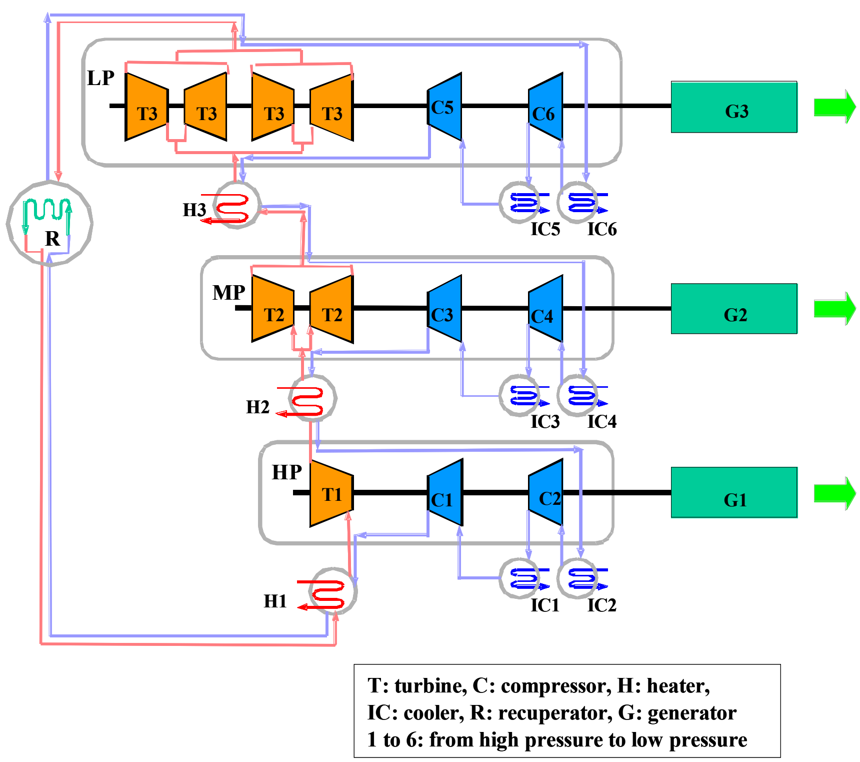


Figure 22: Near term multiple shafts horizontal distributed multiple-reheat cycle PCUs: use PCHE heaters and metal blade turbine.

Table 6 summarizes the design results for all of the four horizontal configuration results and the integrated vertical shaft result. Among all these configurations, the integrated multiple vertical-shaft system has the best efficiency and highest power density. The very short helium flow path results in minimized pressure loss, which results in higher efficiency. The compact configuration also has higher power density. All three long-term horizontal configurations have similar performance. The distributed single shaft with one turbine vessel has the drawback of unbalanced pressures for turbomachinary components and inner ducts.

More detailed analysis is needed to compare distributed multiple shafts configuration and the distributed single shaft with casing vessels. Because for all the long-term configurations, turbomachinary and heat exchanger designs are very similar, the only major difference is the configuration of the pressure boundaries. It is interesting that most of the steel for pressure boundaries for the integrated system is for pressure vessels while pressure boundaries steel consumption for the distributed systems is almost equally divided by pressure vessels and connecting ducts. The near-term distributed horizontal system has slightly lower thermal efficiency and much lower power density than the long-term systems, but is still much higher than the GT-MHR PCU. This shows the potential for large improvement from the transition from the non-reheat Brayton cycle to multiple-reheat Brayton cycle. Table 7 further summarizes the main advantages and disadvantages for all these system configurations. At this stage, it is difficult to select one to be the best potential system to further develop. But if we consider this choice in the context of broader R&D efforts on Brayton cycle for nuclear powers, we will have clearer pictures for future directions. Figure 23 compares four different power conversion systems. The top two, the GT-MHR and PBMR, are middle power level Brayton cycles without reheat. The bottom two are an integral vertical shaft configuration and another horizontal distributed shaft configuration, illustrate the change to large power level Brayton cycles with multiple-reheat. While the net power output increases by a factor of 5 to 6, the physical size of the systems are not greatly different. Multiple reheat Brayton cycles have better efficiency and have the advantage of scale than Brayton cycles without reheat. Therefore, multiple-reheat Brayton power systems represent the future direction for high-temperature reactors.

Figures 23, the systems on the left are vertical shaft and integrated systems; and the systems on the right are horizontal shaft and distributed systems. GT-MHR and PBMR are currently two advanced reactor systems under most detailed engineering design work. The success of these two programs will solve most technology challenges for the development of multiple-reheat systems and will set the technology base to choose directions for larger multi-reheat systems.

	Integrated multiple shafts	Distributed multiple shafts	Distributed single shaft: one turbine vessel	Distributed single shaft: casing turbine vessels	Distributed multiple shafts (split turbines and Heatric heaters)
Thermal efficiency	56.6%	55.5%	55.5%	55.5%	54.7%
Power density, MWe/m^3	0.56	0.51	0.49	0.51	0.35
Specific metal input, MT/MWe_{ave}	3.7	3.8	3.6	3.8	5.1
Specific helium inventory, kg/MWe_{ave}	6.3	5.8	6.0	5.5	7.4
PCUs building size (length*width*height), m*m*m	40*40*60	50*50*30	90*50*30	90*40*30	70*50*30
Pressure vessel mass, MT	1800	1050	600	1050	1540
Duct mass, MT	110	1140	1300	1130	1410
Total relative pressure loss, [-]	0.07	0.12	0.12	0.12	0.09
Largest vessel diameter (except for generator), m	7.4 (LPV)	4.6 (LPV)	4.6 (TBMV)	4.6 (TBMV)	5.0 (LPV)

Table 6: 2000MWt AHTR-VT PCUs design results

Multiple reheat cycle configuration cases	Main disadvantages	Main advantages
Integrated multiple shafts	• Vertical shafts, • Complex to maintain	• Shortest hot flow length, • Compact
Distributed multiple shafts	• Complex ducting, • Sealing difficulty	• Easy maintain, • Horizontal shaft,
Distributed single shaft: one turbine vessel	• Complex ducting, • Sealing difficulty, • Unbalanced pressures, • Complex to maintain	• Horizontal shaft, • Single tbm vessel
Distributed single shaft: casing turbine vessels	• Complex ducting, • Very hard to seal,	• Horizontal shaft, • Experiences on tbm casings
Distributed multiple shafts (split turbines and Heatric heaters)	• Complex ducting, • Sealing difficulty, • Lower power density and higher specific steel input	• Metal turbine blade, • Metal heaters, • Near term feasibility • Easy maintain, • Horizontal shaft

Table 7: Disadvantages and advantages comparison of 2000MWt AHTR-VT PCUs designs

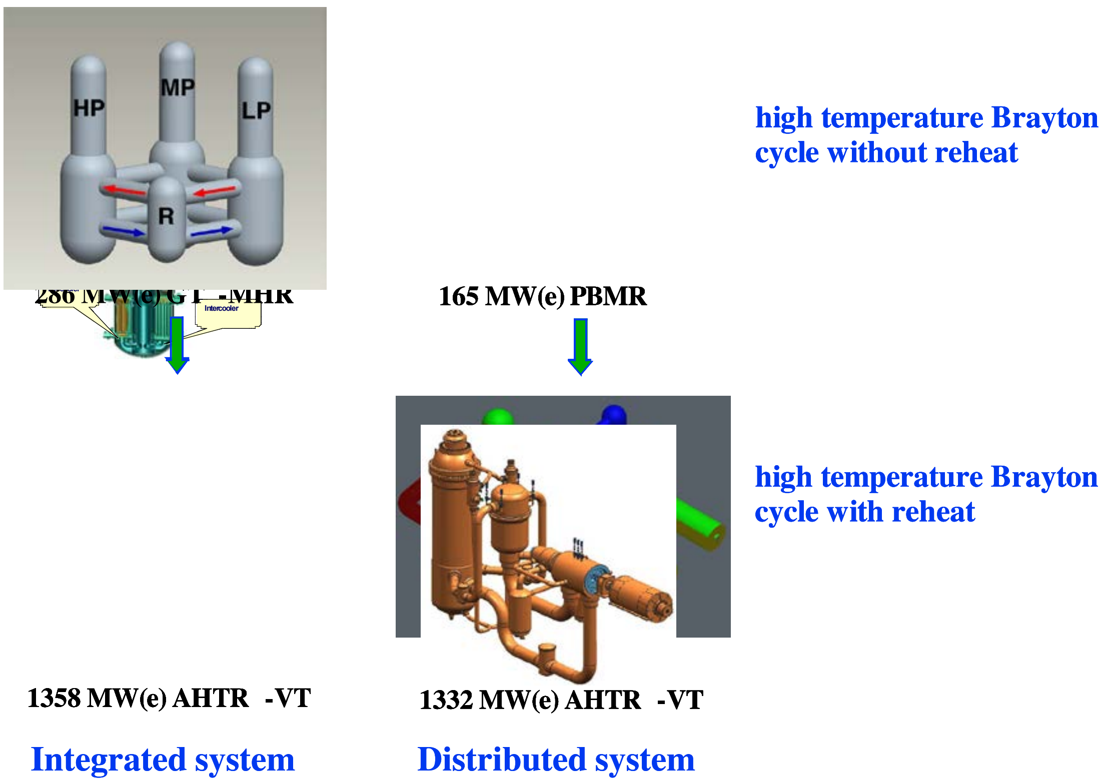


Figure 23: Scaled comparison of four Brayton power conversion systems. Note the small difference in size between the lower-power (165 to 286 MW(e)) and high-power (~1350 MW(e)) PCS options.

4. 600 MWt HORIZONTAL-SHAFT MULTIPLE REHEAT CYCLE SYSTEM

The designs presented earlier are for large 2400-MWt power conversion systems. Large power systems have the advantage of scale economy. However, middle power level systems are also needed in some situations. Moreover, one possible solution to avoid the difficulty of constructing very large systems is to use multiple smaller systems. For example, four 600-MWt multiple-reheat cycle systems can be used for power conversion for a 2400-MWt plant. It is interesting to compare the performance of the large system and smaller systems. Table 8 compares the design results for a 600 MWt system and a 2400 MWt system, both systems using distributed multiple horizontal shaft configurations (Figure 17). The 600-MWt system has lower power density and needs more steel per unit of electricity output than the 2400-MWt system. However, the difference is not sufficiently large to make the smaller systems excessively expensive to compete with large systems.

	600MW	2400MW
Thermal efficiency	55.6%	55.5%
Power density, MWe/m^3	0.39	0.51
Specific metal input, MT/MWe$_{ave}$	4.1	3.8
Specific helium inventory, kg/MWe$_{ave}$	6.3	5.8
PCUs building size (length*width*height), m*m*m	36*34*20	50*50*30
Pressure vessel mass, MT	430	1050
Duct mass, MT	200	1140
Total relative pressure loss, [-]	0.11	0.12
Largest vessel diameter (except for generator), m	3.5 (LPV)	4.6 (LPV)

Table 8: Comparison of 600MWt and 2400 MWt distributed multiple horizontal shaft design results

5. MULTIPLE REHEAT for GAS COOLED REACTORS

Multiple reheat options with gas cooled reactors having long been considered impractical due to the large pressure loss incurred to perform reheating. This section describes three options to implement reheat for a ~600-MWt high temperature gas cooled reactor. All the three designs used the 600 MWt GT-MHR reactor to facilitate comparison with direct single expansion Brayton power cycles. Because the heat source (reactor) has very large temperature change (from 488°C to 850°C), the number of expansion stages can only be two to maintain a practical total pressure ratio. For example, if three stages of expansion are used, the total pressure ratio is more than 20, which results in too low pressure for the low pressure turbine.

The first option uses a liquid salt intermediate loop to transfer heat from reactor to a two-PCU power conversion system. This type of liquid-salt intermediate loop is currently being designed for the purpose of thermochemical hydrogen production, so the extension to use for electricity production as well may be beneficial. Figure 24 shows the flow schematic. It has three flow loops: the helium primary loop, a liquid salt intermediate loop, and a helium Brayton cycle loop. The power conversion system is similar to the integrated multiple vertical-shaft AHTR-VT system described in section 2. The IHX is close to the reactor and within the nuclear island boundary with the reactor. The power conversion system is outside of the nuclear island and is constructed to meet normal industrial standards. The overall efficiency for the liquid salt loop multiple-reheat option is same as GT-MHR as shown in Table 9 (in the comparison, we assume that the GT-MHR also uses 35°C compressor inlet temperature, allowing heat rejection to low quality heat sinks like dry air cooling). Both of efficiencies are 47%. It appears then that the efficiency gain from reheat compensates the efficiency loss from decreasing turbine inlet temperature due to using the intermediate loop. The power density for the liquid salt loop multiple-reheat option is much higher than GT-MHR's because the 10 MPa system pressure is much higher than GT-MHR's 7.24 MPa system pressure. The liquid salt loop is very compact due to the high thermal capacity per unit volume of liquid salt. The salt loop also isolates the radioactive primary loop from the power conversion loop. Without radioactive contamination, the power conversion loop is much easier to maintain than the direct cycle GT-MHR. Overall, the liquid salt loop multiple-reheat system could potentially have lower overnight cost and operational cost than the direct cycle GT-MHR.

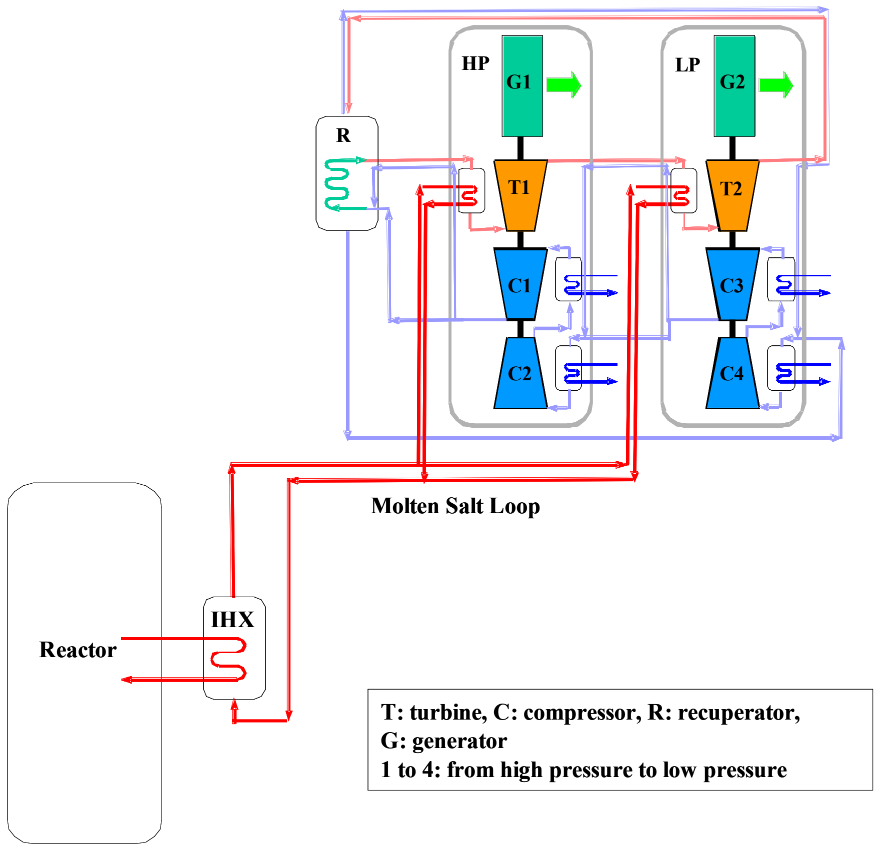


Figure 24: Schematic flow diagram for an indirect multiple-reheat cycle for GT-MHR with a liquid salt loop. Light blue lines represent cold helium flow; pink lines represent hot helium flow; narrow blue lines represent cooling bypass.

The second option uses IHXs as heaters to transfer heat from the helium primary loop to the power conversion loop. Figure 25 shows the flow schematic. There are two loops: the helium primary loop and the helium power conversion loops. Heaters (also IHXs) are located near the reactor to reduce nuclear building size and reduce the gas pumping power due to higher pressure in power conversion loop. Long ducts are used to connect heaters to turbines and recuperator. Therefore this case must use a distributed configuration. The PCU configuration is very similar as the distributed multiple horizontal-shaft AHTR-VT configuration described in section 3. Again, two stages of expansion and four stages of compression are used. Table 9 summarizes the design parameters. The overall efficiency is same as GT-MHR, so is power density. Higher system pressure helps to compensate the effect of added components in reducing power density. As a rule of thumb, same components within nuclear island tend to be one time more expensive than in non-nuclear system. Increased complexity in a reheat system may increase cost relative to a simple direct system. Overall, with similar power densities, this indirect cycle may have comparable overnight cost as the direct cycle GT-MHR system.

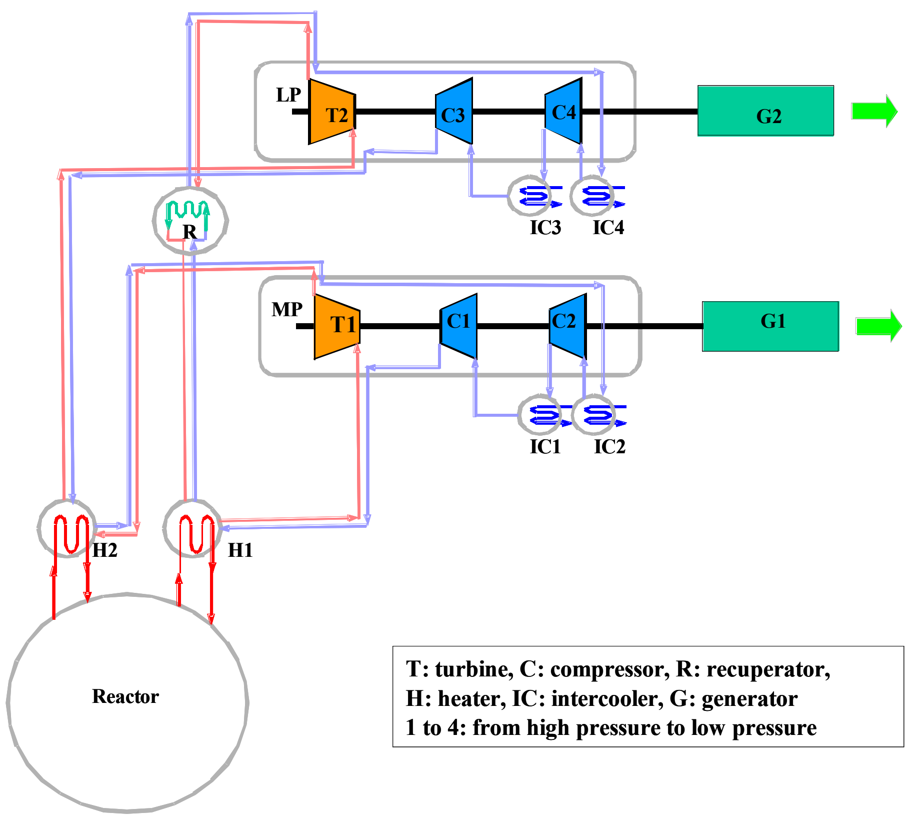


Figure 25: Schematic flow diagram for an indirect multiple-reheat cycle for GT-MHR with IHXs as heaters. Light blue lines represent cold helium flow; pink lines represent hot helium flow; red lines represent primary helium flow.

The third option is a reheat direct cycle shown in Figure 26. About half of flow (51%) from reactor outlet directly expands in the high-pressure turbine, and then gets reheated by another half of the reactor outlet flow (49%) before it expands in the low-pressure turbine. The PCU configurations are derived from GT-MHR PCU design. The HP PCU is basically same design, but smaller than a GT-MHR PCU; the LP PCU is also very similar to a GT-MHR PCU except that the recuperator in a GT-MHR PCU is replaced by the LP heater. Table 9 summarizes the design parameters. The overall efficiency for this case is slightly higher than single expansion GT-MHR. But it is expected that the power density is lower than the GT-MHR due to much increased complexity. In summary, even direct reheat cycle will not increase net efficiency significantly. This option may not be attractive due to system complexity and higher technical risk relative to direct single expansion GT-MHR.

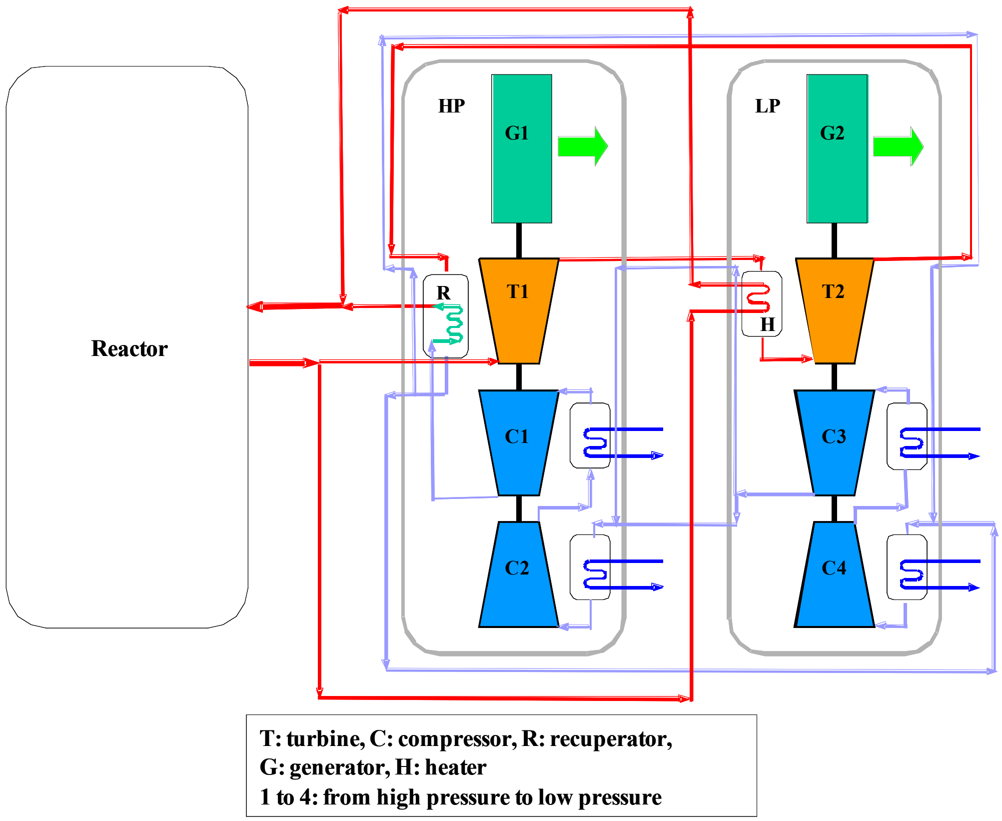


Figure 26: Schematic flow diagram for a direct multiple-reheat cycle for GT-MHR. Light blue lines represent cold helium flow; red lines represent hot helium flow; narrow light blue lines represent cooling bypass flow.

	Multiple Reheat Cycle			GT-MHR
	Liquid Salt-loop	He-loop	Direct	
PCU configuration	Integrated vertical shaft	Distributed horizontal shaft	Integrated vertical shaft	Integrated vertical shaft
Number of Loops	3	2	1	1
Primary Max. /Min. Temperature, °C	848/488	848/488	848/488	848/488
Intermediate Max. /Min. Temperature, °C	830/470	N/A	848/515	N/A
Turbine Inlet/Outlet Temperature, °C	800/460	810/470	848/480 and 800/480	848/508
Compressor Inlet/outlet Temperature, °C	35/118	35/118	35/118	26.4/110.3
System Pressure, MPa	10	10	7.24	7.24
Number of PCU's	2	2	2	1
Numbers of Turbines and Compressors	2/4	2/4	2/4	1/2
Helium Mass Flow Rate, kg/s	164	164	162	317
Cycle Pressure Ratio	8.02	7.83	7.8	2.69
Pressure Loss Fraction	0.05	0.06	0.08	0.07
Overall Cycle Efficiency	0.47	0.47	0.48	0.47*
Electrical Power, MW	292	294	295	285
The Ratio of Recuperator Power Over Electrical Power	0.95	0.97	0.98	2.2
PCUs Power Density, kW(e)/m^3	350	230	> GT-MHR	230
Specific Metal Mass for PCUs, MT/MWe (Capacity factor: 0.9)	6.3	7.7	> GT-MHR	7.5
Note:	Liquid salt loop transports heat from reactor to heaters	Heaters (IHXs) are close to reactor but far away from turbines	use 49% reactor outlet flow to provide reheat	* Commonly quoted 48% GT-MHR efficiency is for lower (26.4°C) compressor inlet temp.

Table 9: Multiple-reheat options for GT-MHR

6. CONCLUSIONS

Relatively detailed preconcept point designs for multiple-reheat Brayton power conversion systems have been developed using different equipment configurations. Multiple reheat systems can achieve very high thermal efficiency for high temperature liquid coolant heat sources, with the potential very low capital cost. They can also be adapted for indirect-cycle power conversion for gas-cooled modular reactors.

With similar components parameters and reasonable arrangement, different configurations such as horizontal or vertical shaft, integrated system or distributed system, have similar specific power densities and specific steel inputs. Because these high-level performance parameters are similar, further detailed design and comparison must be performed to select optimal system designs.

It is possible to design reheat cycles for a gas cooled high temperature reactor with similar thermal efficiency as the direct single-expansion Brayton cycle. For an indirect reheat cycle, the efficiency gain from reheat compensates the efficiency loss from the reduction of the turbine inlet temperature due to the use of the intermediate loop. The direct reheat cycle for a high temperature gas cooled reactor is not attractive due to increased complexity, higher technical risk, and lack of any large increase in thermal efficiency. Although the indirect cycle with reheat does not increase thermal efficiency compared to the direct cycle, using indirect reheat cycles may be attractive due to safety, operation, and maintaining benefits (such as the helium-loop case) or also due to potential economic benefit (liquid salt intermediate loop). Another option for maintaining high efficiency using an indirect gas power cycle is to use a steam-bottoming cycle; however, this approach then requires a high-quality heat sink to reject condenser heat to, while the indirect reheat Brayton cycle can reject heat to low-quality heat sinks.

Brayton cycle systems without reheat such as the GT-MHR and PBMR will provide technology and experience that can be the basis for the development of multiple-reheat power conversion systems. Near-term application of the multiple-reheat cycles is feasible with current technologies and can obtain high efficiency and high power density.

REFERENCES

Besmann, T. M., Klett, J. W., Henry, J. J. Jr., and Lara-Curzio, E., 2000. Carbon/carbon composite bipolar plate for proton exchange membrane fuel cells. *Journal of the Electrochemical Society*, **147** (11) 4083–4086.

Forsberg, C. W., Pickard, P. S., and Peterson, P. F., 2003. Molten-salt-cooled advanced high-temperature reactor for production of hydrogen and electricity. *Nucl. Technol.,* **144**, 289B302.

Forsberg, C.W., Peterson, P.F., and Zhao, H., 2004a. An advanced molten salt reactor using high-temperature reactor technology. *2004 International Congress on Advances in Nuclear Power Plants (ICAPP '04), Embedded International Topical Meeting, 2004 American Nuclear Society Annual Meeting, Pittsburgh, Pennsylvania, June 13-17, 2004.*

Forsberg, C. W., Peterson, P. F., and Pickard, P., 2004b. Maximizing temperatures of delivered heat from the advanced high-temperature reactor. *Proc., 2004 International Congress on Advances in Nuclear Power Plants (ICAPP '04), Embedded International Topical Meeting 2004 American Nuclear Society Annual Meeting, Pittsburg, Pennsylvania, June 13-17, 2004.*

Forsberg, C. W., Ingersoll, D. T., Ott, L. J., Williams, D. F., Renier, J. P., Wilson, D. F., Ball, S. J., Reid, L., Corwin, W. R., Del Cul, G. D., Peterson, P. F., Zhao, H., Pickard, P. S., Parma, E. J. and Vernon, M., 2004c. S*tatus of preconceptual design of the advanced high-temperature reactor*. ORNL/TM-2004/104, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Forsberg, C.W., Peterson, P.F., and Zhao, H., 2005. Sustainability and economics of the advanced high temperature reactor, *Journal of Energy Engineering*, to be published.

LaBar, M. P., 2002. The gas turbine–modular helium reactor: a promising option for near term deployment. *Proc., International Congress on Advanced Nuclear Power Plants, Embedded Topical American Nuclear Society 2002 Annual Meeting, Hollywood, Florida, June 9–13, 2002, GA-A23952.*

Peterson, P. F., 2003a. Multiple-reheat Brayton cycles for nuclear power conversion with molten coolants. *Nucl. Technol.,* **144**, 279–288.

Peterson, P.F., Forsberg, C., and Pickard, P., 2003b. Advanced CSiC composites for high-temperature nuclear heat transport with helium, molten salts, and sulfur-iodine thermochemical hydrogen process fluids. *Second Information Exchange Meeting on Nuclear Production of Hydrogen, Argonne National Laboratory, Illinois, USA, October 2–3, 2003.*

Peterson, P.F., Development of liquid-silicon-impregnated C/C-SiC composites for high-temperature heat transport, *U.C. Berkeley Department of Nuclear Engineering Report UCBTH-03-001, Oct. 1, 2003.*

Peterson, P.F., et. al., 2004. Next generation nuclear plant power conversion study: technology options assessment, *Generation IV Program.*

Peterson, P.F., Zhao, H., Huang, W., and Fukuda, G., Conceptual Design for a 50 MW(t) Metallic Intermediate Heat Exchanger for the Next Generation Nuclear Plant, *U.C. Berkeley Department of Nuclear Engineering Report UCBTH-04-001, Dec. 2004.*

Zhao, H., Fukuda, G., Abbott, R., and Peterson, P.F., 2004. Optimized helium-Brayton power conversion for fusion energy systems. *Fusion Science and Technology, Vol.* **47** *Apr. 2005.*