A Reference 2400 MW(t) Power Conversion System Point Design for Molten-Salt-Cooled Fission and Fusion Energy Systems

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ABSTRACT

This report describes three 2400-MW(t) reference point designs for the molten-coolant gas cycle (MCGC), based on the current power conversion unit (PCU) designs for the GT-MHR. A high-temperature helium cycle with a 900°C turbine inlet temperature achieves a thermodynamic efficiency of 54% and net electric output of 1300 MW(e), while a low temperature helium cycle with a 600°C turbine inlet temperature achieves a thermodynamic efficiency of 45%. A third reference case, using nitrogen with 10 weight percent helium with a 900°C turbine inlet temperature also achieves a thermodynamic efficiency of 54%, with a PCU volume 46% larger than the helium-cycle.

The reference MCGC designs have substantial maturity because the helium-cycle turbo-machinery, intercoolers, and recuperators are close derivatives of the GT-MHR PCU. The MCGC flow configuration keeps the entire pressure vessel boundary in contact with compressor outlet gas, keeping the vessel temperatures below 150°C. The nitrogen MCGC compressor inlet pressure is 0.7 MPa, compared to the 0.1 MPa that combustion turbines must work at to take in air and to discharge combustion products and pollutants back to the atmosphere. The nitrogen (and helium) cycles thus operate at a much higher average gas pressure, and thus higher power density compared to current large combustion turbines. This fact creates the theoretical potential for future advanced high-temperature fission and fusion reactors to have comparable, or lower, total capital costs compared to current natural-gas combined cycle power plants.

INTRODUCTION

This report describes three point designs for the molten coolant gas cycle (MCGC) [1], derived from the GT-MHR power conversion unit. Figure 1 compares the size of a high-temperate helium MCGC point design (900°C turbine inlet temperature, 1300 MW(e) output) to the turbine building of a 1380 MW(e) steam Rankine cycle of a modern light water reactor (LWR). With similar power output, the MCGC system is clearly more compact, and thus provides the potential for major reductions in the turbine building volume, and power conversion system capital cost, for future high-temperature nuclear energy systems, both fission and fusion.

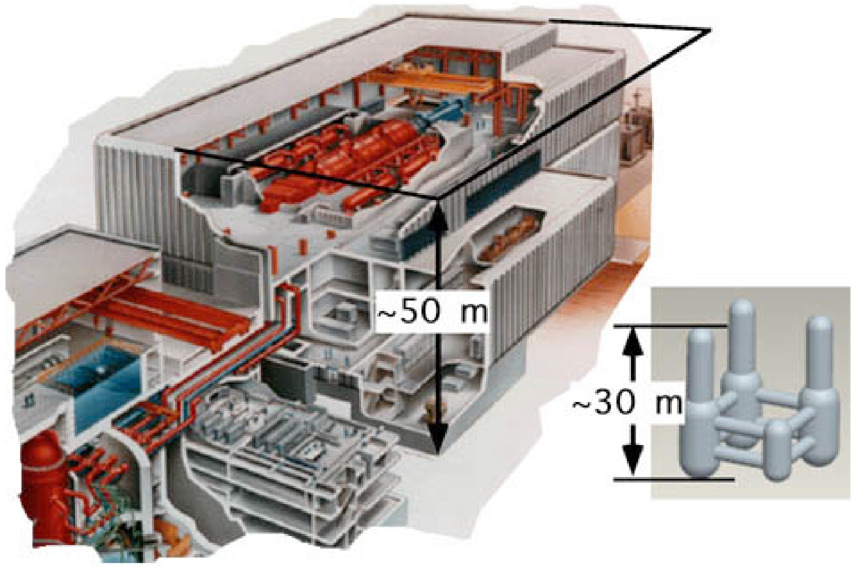

Fig. 1 Size comparison between the power conversion units of a high-temperature, 1300 MW(e) MCGC, and the 1380 MW(e) turbine building of the Advanced Boiling Water Reactor (courtesy GE Nuclear Energy). The MCGC turbine building will also require space for a crane, turbine lay-down space, cooling water circulation equipment, switchgear and high-pressure gas storage. The MCGC requires 1100 MW(t) of cooling water capacity, compared to 2800 MW(t) for the steam cycle.

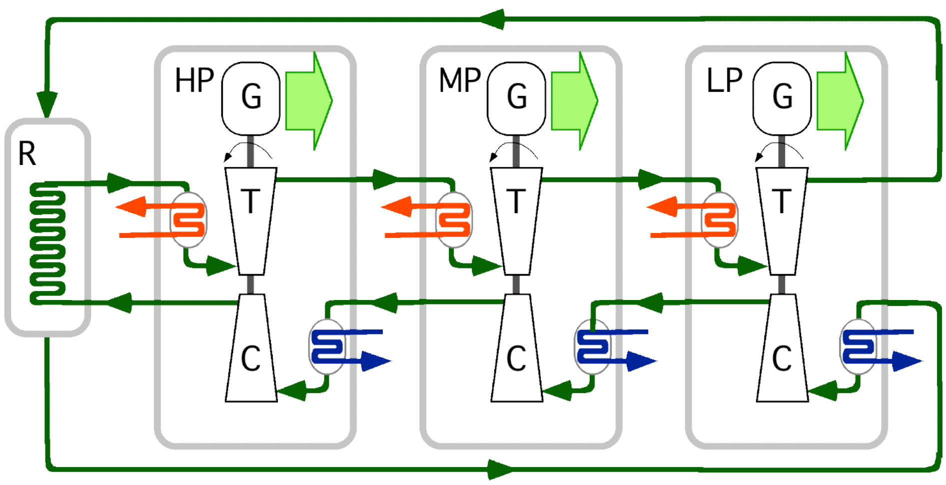

Fig. 2 Schematic flow diagram for the reference three-expansion-stage MCGC, using three PCU modules (HP, MP, and LP) each containing a generator (G), turbine (T), compressor (C), and heater and cooler heat exchangers, with a recuperator (R) located in a fourth vessel.

Figure 2 provides a schematic diagram of the MCGC flow configuration. The 2400 MW(t) high-temperature MCGC design uses three PCU's and a turbine inlet temperature of 900°C, to achieve a thermal efficiency of 54% (1300 MW(e)). The low-temperature design uses four PCU's, with a turbine inlet temperature of 600°C, to achieve a thermal efficiency of 44% (1050 MW(e)). To achieve the same thermodynamic efficiency as the high-temperature design, the nitrogen-based design has a total volume 46% greater than the helium-based design. For this report, only helium was considered for the low-temperature design. Designs at intermediate turbine inlet temperatures are also possible, and can be assumed to achieve performance intermediate between these high and low temperature reference designs

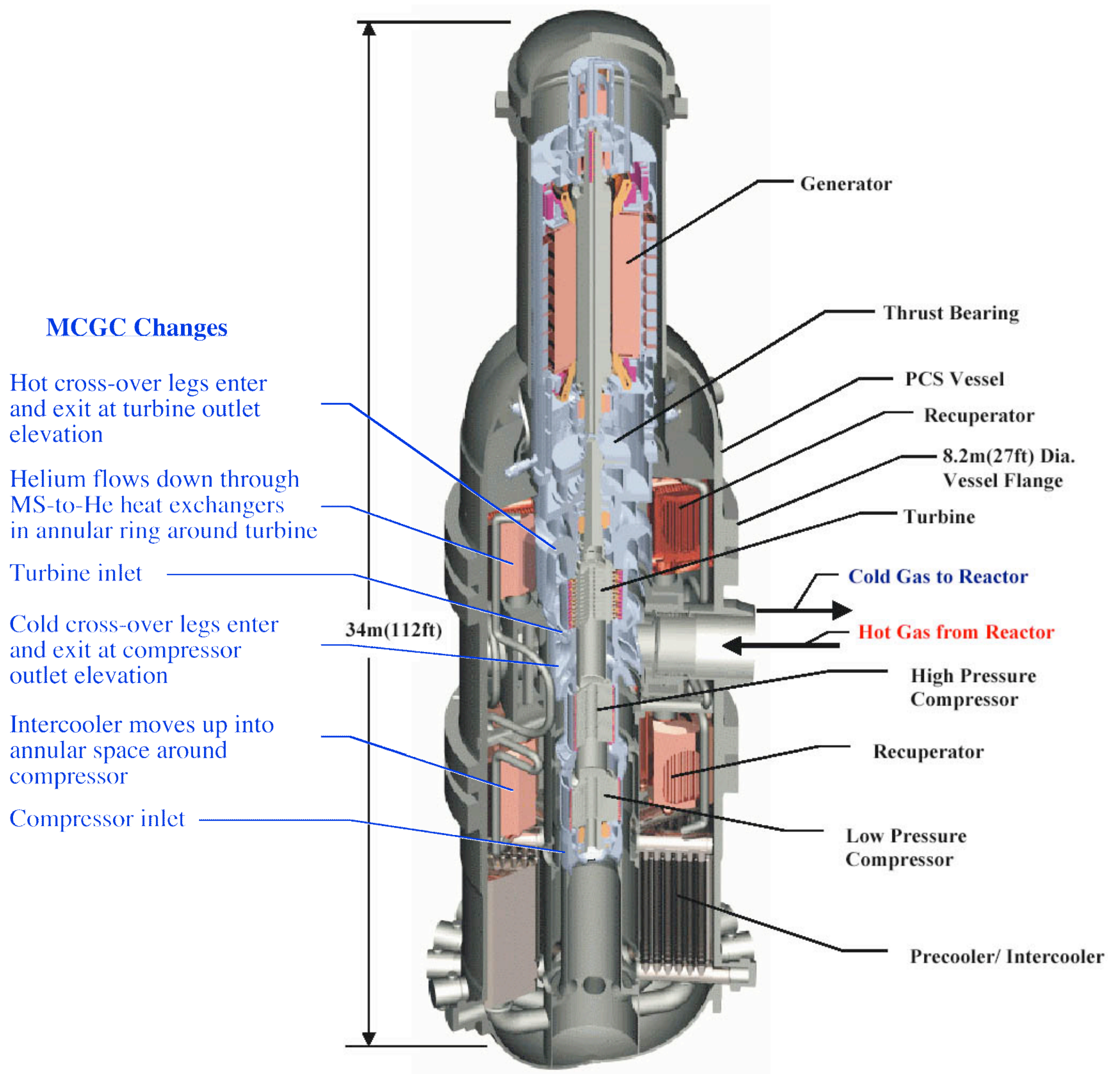

Fig. 3 Cross section of the current GT-MHR PCU, with changes required for MCGC indicated on left.

The General Atomics GT-MHR PCU, shown in Fig. 3, is currently the only closed helium cycle system that has undergone detailed engineering design analysis, and that has turbomachinery which is sufficiently large to extrapolate to a >1000 MW(e) MCGC power conversion system. Analysis presented here shows that, with relatively small engineering modifications, multiple GT-MHR PCU's can be ganged together to create a MCGC power conversion system in the >1000 MW(e) class. To do this, compact salt-to-helium heat exchangers (power densities from 80 to 120 MW/m^3) are inserted the annular space around the turbines, currently occupied by the upper set of recuperator heat exchangers in the GT-MHR design (Fig. 3), and the MCGC recuperator is moved to a separate pressure vessel. As discussed in this report and shown in Figure 1, the resulting configuration is quite compact, and results in what is likely the minimum helium duct volume possible for a multiple-reheat system.

The GT-MHR PCU produces 285 MW(e) with a 850°C turbine inlet temperature, a PCU power density of 230 kW(e)/m^3. Based on the same turbomachinery parameters, and using three PCU's with size similar to the GT-MHR PCU, the high-temperature MCGC is predicted to achieve a power density of 360 kW(e)/m^3. Because the MCGC power conversion system can be located in conventional structures, rather than within the nuclear safety envelope, this suggests that the MCGC PCU capital cost, per kW(e), will be around half that of the GT-MHR PCU. Figure 1 suggests a quite favorable cost scaling relative to LWR power conversion systems as well. The low-temperature MCGC, with a 600°C turbine inlet temperature, achieves a PCU power density of 250 kW(e)/m^3, suggesting a capital cost comparable to the GT-MHR PCU.

The following sections describe in greater detail the physical arrangement of the MCGC PCU's, the motivations for adopting a vertical turbomachinery design, and the results of calculations for the MCGC reference design, including flow parameters and heat exchanger sizes.

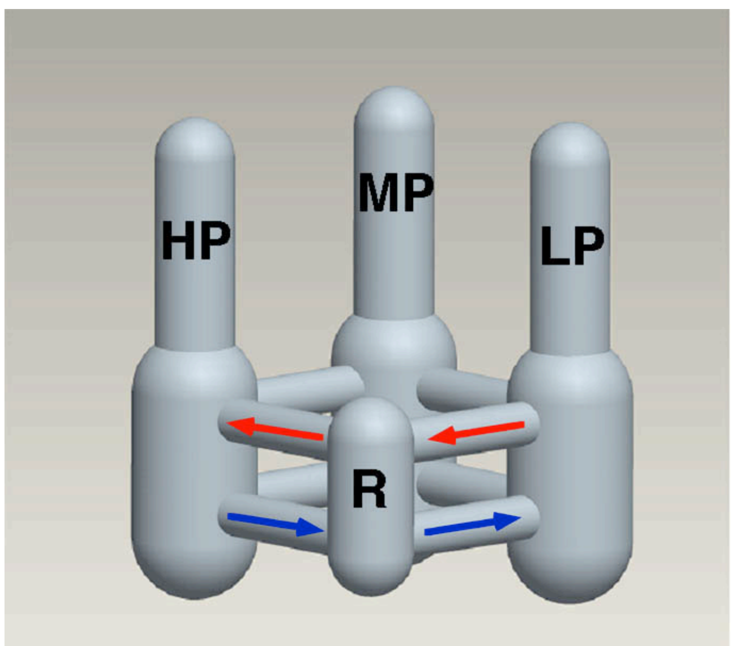

Fig. 4 Hot and cold leg configurations for the MCGC based with three (HP, MP, and LP) PCU's and a separate recuperator vessel (R).

MCGC SYSTEM CONFIGURATION BASED ON THE GT-MHR PCU DESIGN

The reference high-temperature MCGC configuration uses three MCGC power conversion unit (PCUs), each connected by an upper hot leg and a lower cold leg, as shown in Fig. 4. 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 Fig. 3, the hot legs connect the PCU vessels at the elevation of the turbine outlets. Flow is collected from the turbine outlet diffuser and crosses at ~650°C in the hot leg to the next PCU vessel. This hot-leg flow enters the top of an annular ring of liquid-silicon infiltrated (LSI) carbon-carbon composite heat exchangers and flows downward, to be heated to 900°C, 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, based on the LSI composite plate-design shown in Fig. 5, indicates that they should fit without problems in the annular volume around the turbine, currently occupied by the upper recuperator bank of the current GT-MHR PCU design.

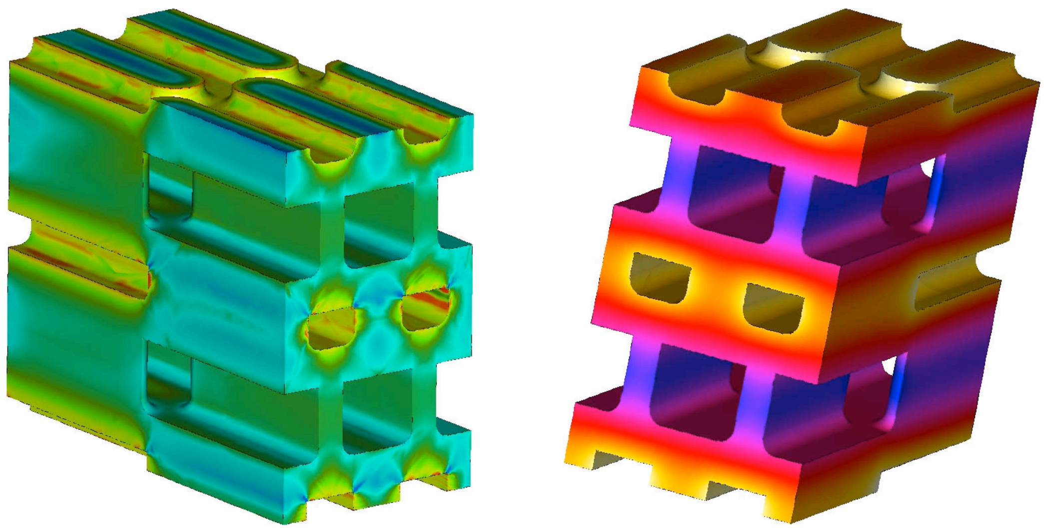

Fig. 5 Stress and temperature distributions in a plate type LSI composite heat exchanger currently under development at UC Berkeley [2,3].

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 at ~140°C crosses in 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. It is doubtful that one will be able to identify a more compact configuration for the MCGC than this multiple-shaft design with annular rings of heaters around each tubine, because all flows enter at the optimal elevation in the vessels and the hot gas flow path is extremely short.

Table 1 MCGC design parameters.

	High Temp. Helium MCGC	Low Temp Helium MCGC	High Temp. Nitrogen-Helium MCGC	Reference GT-MHR PCU
Number of PCU's	3	4	3	1
Working Fluid	Helium	Helium	Nitrogen-helium mixture (10 weight percent He)	Helium
Gas Mass Flow Rate (kg/s)	596	818	1934	317
Turbine Inlet Temperature	900°C	600°C	900°C	848°C
Turbine Outlet Temperature	650°C	463°C	650°C	508°C
MS Inlet Temperature	920°C	620°C	920°C	N/A
MS Outlet Temperature	860°C	570°C	860°C	N/A
Compressor Inlet Temp.	35°C	35°C	35°C	26.4°C
System Pressure	10 MPa	10 MPa	10 MPa	7.24 MPa
Cycle Pressure Ratio	7.04	6.32	14.3	2.69
Turbine Efficiency	0.93	0.93	0.93	0.93
Compressor Efficiency	0.88	0.88	0.88	0.88
Recuperator Effectiveness	0.95	0.95	0.95	0.95
Generator Efficiency	0.98	0.98	0.98	0.98
Pressure Loss Fraction	0.04	0.06	0.06	0.013
Overall Cycle Efficiency	0.54	0.44	0.54	0.46
Power Density (kW(e)/m^3)	360	250	260	230

MCGC THERMODYNAMIC ANALYSIS

Peterson [1] provides equations for analyzing the thermal efficiency of the MCGC. Using these equations, a parametric search was used to identify promising design parameters for the high and low temperature MCGC systems. It was found that a relatively high total pressure ratio was desirable, because it did not have a strong effect on the cycle efficiency and it resulted in a relatively low turbine outlet temperature and a significantly smaller recuperator volume. As shown in the temperature-entropy diagram in Fig. 6, the turbine outlet temperature is sufficiently low (650°C for the high-temperature cycle) for conventional materials to be used for the recuperator and the hot cross-over leg ducts.

Table 1 shows the resulting design parameters for the MCGC systems. Table 2 shows the heat exchanger parameters for the high-temperature helium MCGC. These heat exchangers are described by Peterson [2, 3], and use a compact offset-fin plate configuration with 1-mm thick plates, 1-mm high molten-salt fins, and 2.0-mm high helium fins. Here the small volume of the heaters is notable, showing that they can be fit with relative ease into the annular volume around the turbines in each PCU.

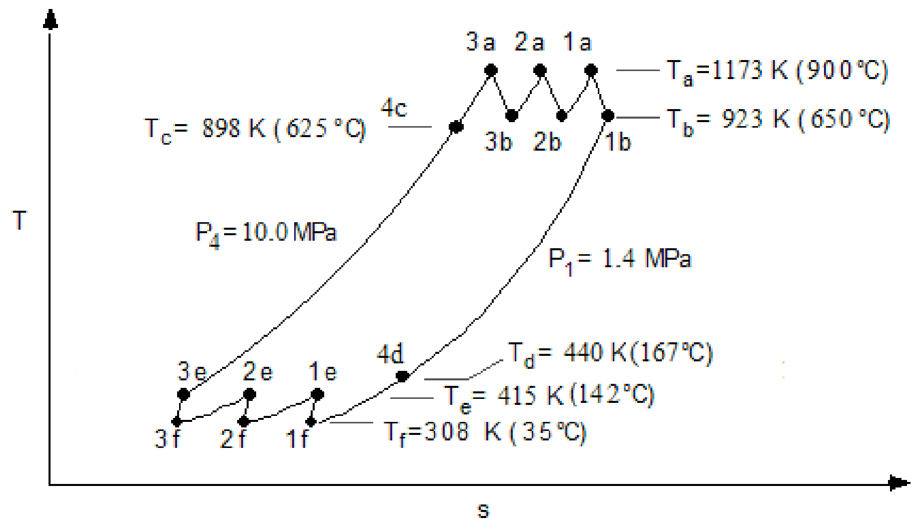

Fig. 6 Temperature-entropy diagram for the MCGC high-temperature reference case.

Table 2 High temperature helium MCGC heat exchanger design parameters

	Recuperator	MS to Helium Heaters		
		High Pressure Heater	Middle Pressure Heater	Low Pressure Heater
Power [MW]	1495	852	774	774
$T_{max,}$ [°C]	650	920	920	920
$T_{min,}$ [°C]	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 Parts, [-]	0.0072	0.0028	0.0025	0.0023

Table 3 shows the turbomachinery parameters for the high-temperature helium MCGC. 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. The generators for the MCGC PCU's will be somewhat taller than the GT-MHR generator, due to their larger power output. But the main MCGC PCU vessels will be shorter than the GT-MHR, because the coolers move upward, and the volume of each MCGC PCU will be similar or smaller than the GT-MHR PCU.

Table 3 High temperature helium MCGC preliminary turbomachinery 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

VERTICAL VERSUS HORIZONTAL MCGC TURBOMACHINERY

The GT-MHR PCU's use vertical turbomachinery, and the NGNP may end up demonstrating the design and reliability of large, vertical turbomachines, including the control of vertical turbine rotor dynamics and the performance of axial catcher bearings. The other option for the MCGC involves a horizontal configuration for the turbomachinery. But there would be several negative impacts from a horizontal orientation that would balance potential simplification of turbomachinery bearing systems.

For horizontal turbomachinery it would be much more difficult to configure the MCGC heaters and coolers into compact annular volumes around horizontal turbomachinery to provide a uniform flow into the turbine and compressor inlets, and to minimize the hot-gas flow length. It would also be more difficult to configure the salt-to-helium heat exchangers to drain their molten salt by gravity.

The vertical orientation permits a very short vessel flange perimeter length compared to the horizontal orientation. Because high-pressure flanges tend to be massive structures, the smaller flanges of the vertical vessels reduce the pressure boundary mass and cost, for the same vessel volume, as well as the helium leakage rate. For molten salt reactors and fusion power applications, the reduced flange perimeter should also simplify tritium management and control. For horizontal turbomachinery it would likely be difficult to minimize the helium duct volume to the small value possible with the cross-over-leg configuration of the vertical PCU's (Fig. 3), so the total pressure-vessel volume of a horizontal turbomachine design would also likely be larger, further increasing the mass of the pressure boundary materials.

Vertical closed-cycle helium turbomachinery is likely to be developed and deployed before horizontal helium turbomachinery, because the first application of a closed helium cycle will most likely be to direct-cycle power conversion for gas cooled reactors (potentially the NGNP). Thus at the time of deployment of the MCGC, engineering design and manufacturing capabilities are likely to be further advanced for vertical machinery than for horizontal. As with aero-turbines, there exist strong drivers to reduce the mass of vertical helium turbomachinery that have better potential to ultimately make vertical machinery less expensive, and resource intensive, than horizontal machinery.

The MCGC PCU turbomachinery will be only slightly larger in diameter, and shorter, than the GT-MHR PCU turbomachinery. Substantial optimization of the GT-MHR PCU design has occurred during GA's collaboration with the Russians to design a plutonium-burning version of the GT-MHR, which has greatly reduced the mass and improved the rotational dynamics of the turbomachinery. Because the MCGC turbomachinery will be located away from the plant's nuclear island, structures costs will be lower.

MULTIPLE-REHEAT NITROGEN GAS CYCLE

A brief summary for the thermal design of a 2400 MW(t) multiple-reheat nitrogen Brayton cycle is given in this section. To increase the gas thermal conductivity, 10 weight percent helium, or 44 mole percent helium, is added to the nitrogen. Table 1 lists the primary design parameters. Comparing to the pure helium high temperature MCGC, the optical nitrogen cycle pressure ratio is as two times larger. The nitrogen system volume and mass are about 46% larger than for helium, so there is a clear economic attraction to using helium as the working fluid due to its better thermophysical properties.

Table 4 High temperature nitrogen MCGC heat exchanger design parameters

	Recuperator	MS to Nitrogen Heaters		
		High Pressure Heater	Middle Pressure Heater	Low Pressure Heater
Power [MW]	1489	852	774	774
$T_{max,}$ [^oC]	650	920	920	920
$T_{min,}$ [^oC]	142	624	650	650
Core Volume, [m^3]	93	11	14	17
Flow Length, [m]	0.74	0.67	0.53	0.42
Total Frontal Area, [m^2]	125	17	26	41
Thermal Density, [MW/m^3]	16	77	56	45
Fractional Pressure Losses for the Counter Flow Parts, [-]	0.012	0.004	0.0036	0.0034

Table 4 shows the heat exchanger parameters for the high-temperature nitrogen MCGC. Comparing to Table 2, the nitrogen MCGC thermal densities are much smaller than the helium ones due to the worse thermophysical properties of nitrogen.

Table 5 shows the turbomachinery parameters for the high-temperature nitrogen MCGC. Due to the large mass flow rate, only a supersonic blade design is possible, as used in modern combustion turbines. The tip diameters for turbomachines are much larger than for helium turbines.

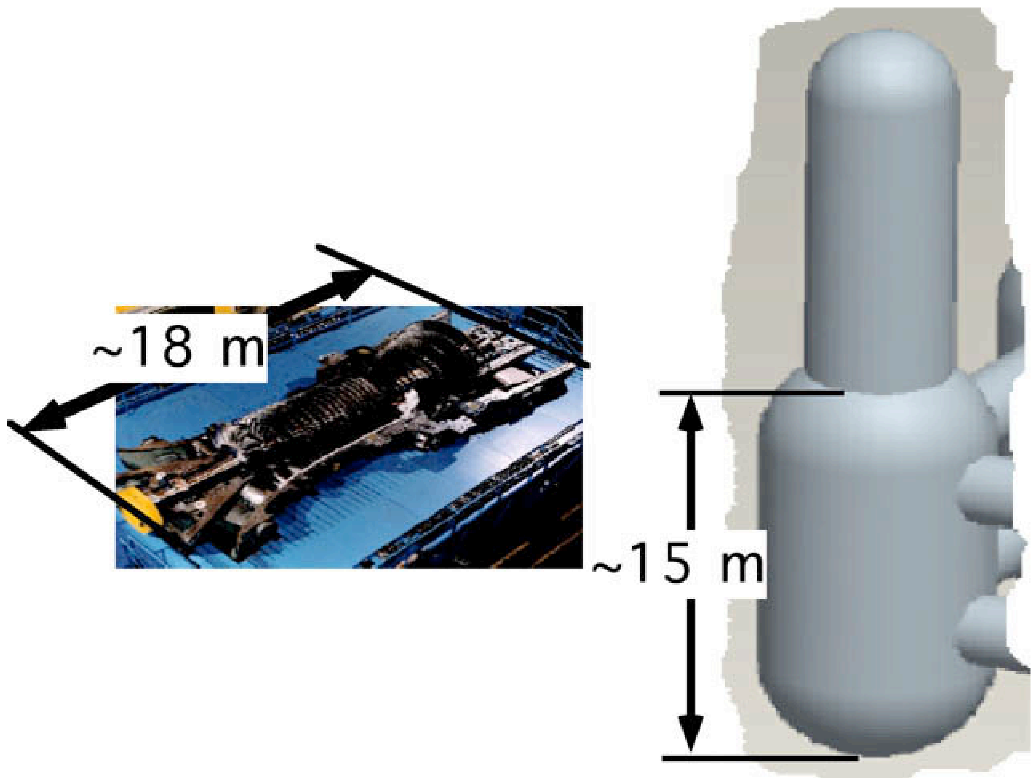

Fig. 7 Comparison of the size of a large 480 MW(e) GE 9H compressor and turbine (left, 1430°C, 685 kg/sec air) to a single nitrogen MCGC PCU (right, 435 MW(e)). The nitrogen MCGC PCU includes its heater and intercooler. The GE 9H photo does not show the exhaust gas recovery heat exchanger, steam bottoming condenser

Current large combustion turbines are designed to operate in horizontal orientation. The required nitrogen MCGC mass flow is approximately 3 times that of existing large combustion turbines. However, the nitrogen MCGC operates with a compressor inlet pressure of 0.7 MPa, while combustion turbines are forced to operate with an inlet pressure of 0.1 MPa. Because power density scales with the gas density, the nitrogen MCGC turbo-machinery achieves much higher power density than current combustion turbines, and the physical dimensions of the 1300 MW(e) nitrogen MCGC turbo-machinery are similar to those for a 400 MW(e) combustion turbine. Figure 7 contrasts the physical dimensions of a large combustion turbine (the GE 9H) to a nitrogen MCGC PCU.

The higher gas density of the MCGC, relative to current combustion turbines, also reduces pressure losses in recuperators, so the MCGC adapts well to a recuperated Brayton cycle. Combustion turbines, however, must commonly use complex and expensive steam bottoming cycles to achieve thermodynamic efficiencies above 50%.

Table 5 High temperature nitrogen MCGC preliminary turbomachinery design parameters

	Compressors			Turbines		
	HP	MP	LP	HP	MP	LP
Power (MW)	331	331	331	774	774	774
Inlet Temp. (°C)	35	35	35	900	900	900
Pressure Ratio	2.48	2.48	2.48	2.43	2.43	2.43
Number of Stages	7	7	6	4	4	4
Adiabatic Efficiency	0.88	0.88	0.88	0.93	0.93	0.93
Exit Dynamic Pres. over Sys. Pressure	0.11%	0.27%	0.74%	0.09%	0.23%	0.55%
Max Tip Diameter (m)	2.5	2.5	2.5	3	3	3
Tip Speed (m/s)	470	470	470	559	556	560
Min. Hub/Tip Ratio	0.56	0.56	0.57	0.56	0.57	0.56
Overall Length (m)	5	5	5	5	5	5

In summary, to obtain similar thermodynamic efficiency, it appears that nitrogen-based systems will have somewhere around 40% larger volume than a helium-based systems, and higher capital cost, due to the less optimal thermodynamic properties of nitrogen compared to helium.

CONCLUSIONS

The GT-MHR PCU provides a useful reference for designing and evaluating multiple-reheat Brayton cycles. With relatively modest modifications, this PCU design can provide a high power density, and high thermal efficiency, using heat provided by a molten-salt coolant.

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