ASSESSMENT OF GAS COOLED FAST REACTOR WITH INDIRECT SUPERCRITICAL CO_2 CYCLE

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Various indirect power cycle options for a helium cooled gas cooled fast reactor (GFR) with particular focus on a supercritical CO_2 (SCO_2) indirect cycle are investigated as an alternative to a helium cooled direct cycle GFR. The balance of plant (BOP) options include helium-nitrogen Brayton cycle, supercritical water Rankine cycle, and SCO_2 recompression Brayton power cycle in three versions: (1) basic design with turbine inlet temperature of 550°C, (2) advanced design with turbine inlet temperature of 650°C and (3) advanced design with the same turbine inlet temperature and reduced compressor inlet temperature. The indirect SCO_2 recompression cycle is found attractive since in addition to easier BOP maintenance it allows significant reduction of core outlet temperature, making design of the primary system easier while achieving very attractive efficiencies comparable to or slightly lower than, the efficiency of the reference GFR direct cycle design. In addition, the indirect cycle arrangement allows significant reduction of the GFR "proximate-containment" and the BOP for the SCO_2 cycle is very compact. Both these factors will lead to reduced capital cost.

KEYWORDS : Supercritical Carbon Dioxide, Indirect Power Cycle, Gas Cooled Fast Reactor

1. INTRODUCTION

The gas cooled fast reactor (GFR) with helium Brayton direct cycle is the reference design under consideration for Generation IV service because of its simplicity, high achievable efficiency and synergism with its thermal counterpart, as a very high temperature reactor for electricity generation and hydrogen production. On the other hand, GFR fuel requires much higher heavy metal loading than the particle fuel for thermal gas cooled reactors and thus gives less freedom in the application of several layers of coatings, which provide effective barriers to fission product release. The development of a robust fuel that meets extremely stringent integrity and leak tightness requirements at high operating temperatures will be challenging, especially for the direct cycle where minimum or no contamination of turbomachinery is desirable. Therefore, investigation of an indirect cycle as a backup to the reference direct cycle design is of high interest. The indirect cycle also provides benefits of reduced containment cost (because the power cycle can be located outside the containment), easier maintenance of turbomachinery, reduction of LOCA initiators, and the possibility to use reheat, which is not practical for a direct cycle. This paper summarizes results of studies on the performance of a helium-cooled GFR coupled to a supercritical CO_2 (SCO_2) cycle carried out at MIT and CEA within the framework of the International Nuclear Energy Research Initiative (I-NERI).

The reason for selecting a supercritical cycle for the balance of plant (BOP) is the potential for high efficiency at significantly lower temperature than for the Brayton helium cycle due to low compression work near the critical point. Among several different fluids, CO_2 was selected as the most promising candidate because of the moderate value of the critical pressure, its stability, relative inertness (for the temperature range of interest), sufficient knowledge of the thermodynamic properties, non-toxicity, low cost and abundance.

A supercritical CO_2 cycle was first proposed in 1948 when Sulzer Bros patented a partial condensation CO_2 Brayton cycle [1]. More extensive studies of supercritical CO_2 cycles were carried out in the sixties by several investigators. In the U.S. Feher [2] proposed a cycle which operated entirely above the critical pressure of carbon dioxide with the compression process in the liquid phase below the critical temperature (critical point 7.377 MPa, 30.97°C) to minimize pump work. Angelino performed an extensive review of various arrangements of SCO_2

cycles [3] with respect to their thermodynamic efficiencies showing that the partial cooling cycle with improved regeneration and recompression cycles have the potential for the highest efficiencies. Extensive design work on a recompression cycle, including preliminary assessment of plant start-up and control, has been done by Brown Boveri-Sulzer Turbomachinery Ltd, Zurich [4].

The interest in the SCO_2 cycle has been recently revived in conjunction with its application to Generation IV reactors at the Massachusetts Institute of Technology (MIT) [5,6] and the Tokyo Institute of Technology (TIT) [7,8]. The TIT work is currently focused on a partial pre-cooling cycle with the lowest cycle pressure well below the critical pressure, which provides benefits from a smaller compression work and allows one to operate at a lower maximum pressure. MIT is developing the recompression SCO_2 cycle recommended by Feher. However, condensation of CO_2 was eliminated and the pump was replaced by a compressor since condensing CO_2 cycles require an available year-round supply of very cold cooling water (10 – 15°C), which is not available in all regions worldwide. This cycle can achieve attractive cycle efficiencies at relatively low temperatures (~45% at 550°C) due to significantly reduced compression work above the critical point. On the other hand, the SCO_2 cycle requires significantly higher pressure (20MPa versus 8MPa for helium), which necessitates thicker vessels but results in very compact turbomachinery [9,10]. Because of its high efficiency, compactness and simplicity, the recompression cycle option was selected as the reference BOP for the indirect cycle GFR. The same recompression cycle was also chosen as a reference power cycle conversion system for liquid metal-cooled fast reactors [11].

The indirect GFR cycle study, presented in this paper, is based on a two-step approach. First an optimization of a direct SCO_2 recompression cycle was carried out as reported in Section 2. This study was used as a basis for the performance evaluation of the indirect GFR cycle which is reported in Section 3 (second step). The calculated dependence of the direct SCO_2 cycle efficiency on the heat source pressure drop allowed linkage of both studies. An entire re-optimization of the indirect cycle was therefore not necessary, the pressure drop experienced by SCO_2 in the heat source being replaced, in the indirect cycle optimization process, by the one experienced in the intermediate heat exchanger (IHX).

2. DIRECT RECOMPRESSION SCO_2 CYCLE

A schematic of the reference SCO_2 recompression cycle is shown on Figure 1. Large property changes near the critical point challenge the recuperator design because they can result in a pinch point. The recompression cycle avoids this problem by dividing the recuperator into low- and high- temperature parts, each having different flow rates to cope with the large variation of heat capacity of the cooler fluid stream. Thus, only a fraction of the fluid flow is compressed to high pressure in the main compressor (points 1-2) to be preheated in the low temperature recuperator (points 2-3). Then the fluid is merged with the rest of the fluid flow from the recompressing compressor (point 3). The total fluid flow is then preheated in the high temperature recuperator (points 3 – 4) to enter the heat source, which in an indirect cycle is an intermediate heat exchanger (IHX). The heated fluid from the IHX proceeds to the turbine, where it expands (points 5 – 6) and generates energy. The hot stream from the turbine transfers heat in the high (points 6 – 7) and low (points 7 – 8) temperature recuperators, to the cooler high pressure side fluid flow. The flow from the low temperature recuperator is split (point 8) so that a fraction of the stream is recompressed to high pressure (points 8 – 3), while the other fraction proceeds though the precooler to the main compressor (points 8 – 1). Highly efficient and compact heat exchangers are required to minimize the cost and space requirements because the SCO_2 cycle is highly recuperative. HEATRICTM printed circuit heat exchangers (PCHE) [12] were selected as the reference heat exchangers for this application.

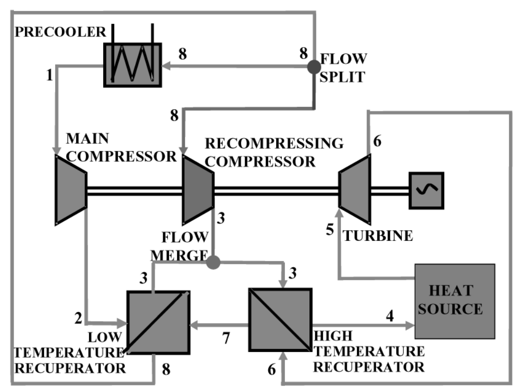

Fig. 1. Layout of Recompression SCO_2 Cycle*

The SCO_2 power cycle was designed for a power rating of 600MWth (i.e. equal to the power of indirect He/SCO_2 GFR cycles presented in Section 3) and two possible turbine inlet temperatures - 550°C (basic design) and 650°C (advanced design). The basic design allows reduction of core temperatures, thereby reducing the challenges for fuel and core materials development, while the advanced design strives for high efficiency. It is noted that the British Advanced Gas (CO_2)-cooled Reactors (AGRs) operate at a core outlet temperature of 650°C, but at much lower pressure of about 4MPa compared to 20MPa for the SCO_2 design. Because of this much higher pressure, the SCO_2 cycle with turbine inlet temperature of 650°C is designated as the advanced design.

The sizing of the SCO_2 cycle is performed in such a manner that maximum efficiency is reached at minimum cost since large components, in particular heat exchangers lead to higher cycle efficiencies due to lower pressure drops, but increase cost. A computer code has been developed and the optimization performed as follows (see Refs. 10 and 13 for more details):

1. Main compressor inlet temperature was selected as a compromise between the cycle efficiency and prevention of operation below the critical temperature of CO_2 because of uncertainties with compressor behavior below this point. The closer the temperature to the critical temperature, the higher the efficiency, hence 32°C was selected. A CEA study, reported in Section 3.5, explored the region below the critical temperature to further increase efficiency.
2. Main compressor outlet pressure was selected based on cycle efficiency sensitivity to pressure and economic considerations. The analyses showed a sharp efficiency decrease for operating pressures below 20 MPa but very small increase for pressures above 20 MPa. Because the economic penalty for pressure above 20MPa would begin to dominate benefits of the small efficiency increase and because it is desirable to remain within the experience base of supercritical steam cycle pressure, 20MPa to 25MPa was selected as the operating pressure of the SCO_2 cycle. The CEA study, reported in Section 3.5, explored the impact of a maximum operating pressure of 25 MPa on the cycle efficiency.
3. Total volume of the cycle heat exchangers (recuperators and precooler) has been selected to yield minimum specific capital cost ($/kWe). This volume was found to be about 120m^3 for a target plant capital cost of 1000$/kWe
4. Given the optimum total volume, the individual heat exchangers were optimized to yield the highest cycle efficiency. This involved optimum split of the total volume among individual heat exchangers and identification of the optimum length and face area of each heat exchanger. The optimum volume apportionment between high- and low- temperature recuperator and precooler was found to be 53, 46, and 21m^3 with corresponding active lengths of 1.75, 2.05 and 1.1 m, respectively [13].

All heat exchangers were of HEATRICTM's printed circuit type with straight channels, semicircular channel diameter of 2mm, plate thickness of 1.5mm and pitch of 2.4mm and fully countercurrent flow. Heat exchanger sizes, operating pressure and main compressor inlet temperatures are the same for both the basic and advanced designs. Operating conditions of both designs for compressor inlet temperature above the critical value are summarized in Table 1 and the statepoints for each design are given in Table 2. The net efficiency, η_{Net}, was evaluated from the following equation:

$$\eta_{net} = \frac{\eta_G\left(1-\xi_{SY}\right)\left[W_{net}\left(\eta_M-\xi_P\right)-\frac{W_{CM}}{\eta_M^2}-\frac{W_{CR}}{\eta_M}+W_{CT}\right]-W_P}{Q_{th}}$$

where : η_M is the coupling mechanical efficiency (99%) (one coupling between each turbine or compressor body, i.e. 3 couplings)
η_G is the generator efficiency (98%)
ξ_{SY} is the switchyard loss (0.5%)
ξ_P accounts for system parasitic losses (2%)
W_{NET} is the net turbine work
W_{CT} is the total work of compressors
W_{CR} is the work of the recompressing compressor
W_{CM} is the work of the main compressor
W_P is the pumping power for water cooling
Q_{th} is reactor thermal power (added heat rate)

Turbomachinery efficiencies used in the cycle calculation were those obtained from the detailed design at MIT of the turbine and compressors. A fixed pressure drop of the heat source of 130kPa was used in the cycle optimization. In the indirect cycle, this pressure drop depends on the design of the IHX. As mentioned in the introduction, to avoid reoptimization of the entire cycle, the dependence of cycle efficiency on heat source pressure drop was calculated and used in performance evaluation of the indirect GFR cycle presented in Section 3. This dependence is fairly linear and can be accurately described by the equation $\eta = \eta_0 - 0.002\,\Delta p_{IHX}$, where η_0 is cycle efficiency for zero IHX pressure, Δp_{IHX}= 0kPa.

The attractive feature of the SCO_2 cycle is extremely compact turbomachinery. The dimensions, number of stages and efficiencies of the turbine [9] and compressors [14] are summarized in Table 3. It can be noted that all of the blading sections of the turbomachinery for a 300MWe unit can fit in a home-size refrigerator. This is because of small volumetric flow rates due to high compressor inlet and outlet pressures.

It is of interest to compare the net efficiencies of the above SCO_2 cycles to the efficiency of a direct helium Brayton cycle, such as the GT-MHR. To make a consistent comparison the net efficiency of a helium cycle with 1 intercooler and 2 compressors and turbine inlet temperature of 850°C (same as GT-MHR) was calculated, using the same procedure, and found to be 46.3%*. This is 2.5% higher than the net efficiency of the basic design at 550°C core outlet temperature and about 1.5% less than the advanced design at core outlet temperature of 650°C.

* GT-MHR net efficiency is reported as 48% [15]. This efficiency was reproduced by our calculations when neglecting water pumping power and parasitic system losses. Higher GT-MHR efficiency may indicate that the parasitic system losses assumed in our calculations may be conservatively high.

Table 1. Key SCO_2 Cycle Parameters

Parameter	Design Basic	Design Advanced
Cycle Thermal Power (MWth)	600	600
Thermal Efficiency (%)	47.2	51.3
Net Efficiency (%)*	43.9	47.8
Compressor Outlet Pressure (MPa)	20	20
Pressure Ratio	2.6	2.6
Heat Source Pressure Drop (kPa)	130	130
Turbine Inlet Temperature (°C)	550	650
Compressor Inlet Temperature (°C)	32	32
Cooling Water Inlet Temperature (°C)	27	27
Mass Flow Rate (kg/s)	3246	3027
Recompressed Fraction	0.41	0.41
Total Heat Exchanger Volume (m^3)	120	120
Turbine Efficiency (%)	94.2	94.2
Main Compressor Efficiency (%)	91.1	91.1
Recomp. Compressor Efficiency (%)	90.5	90.5

* Includes 2% generator losses and 1% mechanical losses per coupling, cooling water pumping power, 0.5% switchyard losses and 2% other system losses accounting for component cooling, coolant leakage, core bypass and heat losses.

Table 2. Statepoints for Basic and Advanced SCO_2 Cycles

Point	Basic design P	Basic design T	Advanced design P	Advanced design T
	MPa	°C	MPa	°C
1	7.692	32.00	7.692	32.00
2	20.000	60.91	20.000	60.91
3	19.989	157.26	19.989	159.11
4	19.959	391.94	19.948	481.83
5	19.829	550.00	19.818	650.00
6	7.892	435.45	7.919	527.15
7	7.806	167.29	7.804	168.31
8	7.703	69.47	7.702	70.89

Table 3. Turbomachinery Parameters

	Turbine	Main comp	Rec. comp.
Number of stages	4	7	8
Pressure ratio*	2.5	2.6	2.6
Efficiency (%)*	94.2	91.1	90.5
Length (m)	0.71	0.7	0.4
Max. tip diameter (m)	1.2	0.5	0.9
Rated Flow Rate (kg/s)	3246	1915	1331

* Total to total values

3. PERFORMANCE POTENTIAL OF HELIUM COOLED GFR COUPLED TO SCO_2 CYCLE

An indirect cycle is penalized by lower efficiency due to its reduced turbine inlet temperature as a result of the temperature difference across the IHX and by the power consumption of the primary helium circulator, which for gas coolant is significant. An important consideration is the additional cost of the intermediate heat exchangers and circulators. Smaller IHX volumes are preferable with respect to their capital cost but exhibit higher temperature difference and pressure drops. Higher temperature difference increases thermal stresses and wall temperatures reducing the allowable stress of the material. This leads to thicker walls and higher costs. Also, the high pressure drops increase the cost of the circulators. Clearly, this is a multiple-parameter problem, for which optimization is required for the best selection of the parameters for the primary system and IHX.

The goal of the optimization is to minimize the capital cost of the plant on a $/kW$_e$ basis. For a given reactor power output, turbine inlet temperature and CO_2 inlet temperature into the IHX obtained from the optimized SCO_2 cycle per description in Section 2, the optimum parameters to be identified are: (1) reactor inlet and outlet temperatures and corresponding reactor mass flow rate, IHX dimensions and plant efficiency that accounts for the electricity consumption of primary system circulators (with assumed efficiency of 85%). Note that plant efficiency significantly affects specific capital cost because it determines electrical power output.

3.1 GFR Primary System Description

The evaluation of the performance of the GFR in the indirect cycle arrangement was carried out on the 600MWth GFR design with low-pressure drop core, (plate-type core) developed at CEA. The primary system layout is shown in Figure 2, where the larger coaxial duct connects the reactor vessel with the IHX and the smaller coaxial duct connects the vessel with the shutdown/emergency cooling heat exchanger. More details on the GFR design, strategy for decay heat removal and reactor building layout are given in [16]. For the purpose of this study, primary loop geometry is important to determine pressure drop around the helium loop and thus the required circulator power.

The helium loop consists of the IHX, helium circulator mounted on the bottom of the IHX (IHX and circulator are not shown), cold leg formed by the annular space in the coaxial duct, reactor vessel downcomer, lower plenum, distribution plate, bottom reflector, active core, top reflector, plenum above the core and hot leg (inner pipe in the coaxial duct), which connects the vessel with the IHX. The flow path is indicated by arrows on Figure 2. The key dimensions used in the analysis are summarized in Table 4.

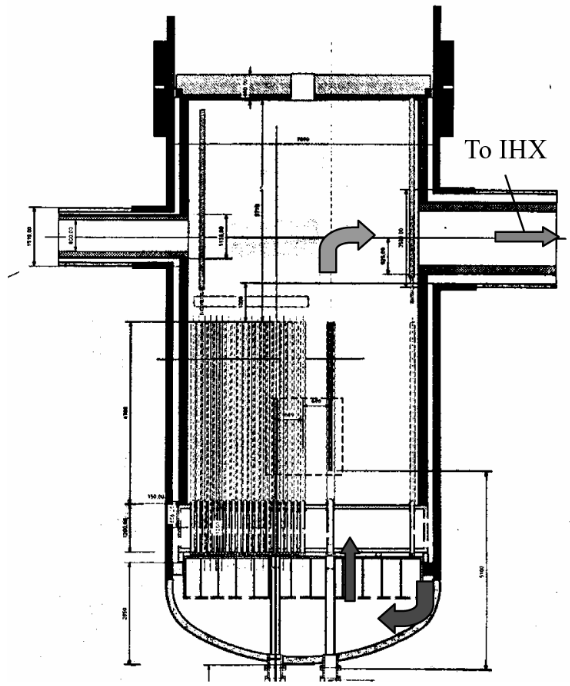

Fig. 2. Schematic of GFR Vessel with Core (from [20])

Table 4. Key Dimensions of Primary Loop

Component name	Length (m)	Flow Area (m^2)	Hydraulic diameter (m)
Inlet duct	3.0	3.90	1.3650
Downcomer	3.8	5.50	0.6000
Inlet plenum	1.0	12.57	4.0000
Distribution Plate	0.3	3.0	0.7000
Bottom reflector	1.0	1.41	0.0127
Core	1.95	1.41	0.0127
Top reflector	1.0	1.41	0.0127
Outlet plenum	3.5	12.57	4.0000
Outlet duct	3.0	0.50	0.8000

3.2 GFR Methology for Selection of Optimum Parameters

IHX dimensions depend on core flow rate and inlet and outlet temperatures, hence they need to be reoptimized during calculations. A HEATRICTM type IHX was used with straight channels and semi-circular channel diameter of 2mm (channel pitch and plate thickness were determined from 1D stress analysis). The IHX inlet and outlet temperatures on the CO_2 side were used from the optimized cycle designs in Table 2. Note that for the selected turbine inlet temperatures of 550 and 650°C the IHX inlet temperatures are determined by the SCO_2 cycle thermodynamics. In the search process for the optimum reactor inlet and outlet temperatures a wide range of possible IHX temperatures was considered. For every set of inlet and outlet temperatures the IHX was sized and the pressure drops on both helium and CO_2 sides were calculated. Using the calculated IHX CO_2 pressure drop, Δp_{IHX}, the new cycle efficiency was calculated from the relation $\eta = \eta_0 - 0.002\,\Delta p_{IHX}$ (see Section 2.1). Next, the required pumping power was determined based on the calculated pressure drop around the primary loop. Note that the circulator pumping power is converted ultimately to heat added to the primary system, hence the reactor thermal power, which enters efficiency calculations, will be reduced for fixed total power. The indirect cycle efficiency is calculated by dividing net electric power by reactor thermal power, where the net electric power is the cycle electric power minus the electric pumping power). This efficiency is then used for the specific cost calculations. The cost of the IHX is calculated based on its weight using cost of 30 $/kg for stainless steel HEATRICTM heat exchangers [17]. The cost of the circulator is scaled based on the pumping power requirements and the IHX vessel cost is scaled based on the IHX volume. The baseline cost data for these components ($17,421,051 for circulator and $11,224,237 for IHX vessel in 1992$) were taken from General Atomics cost estimates for 450MWth Modular High-Temperature Gas-cooled reactors [18]. These costs were scaled with respect to differences in power rating and volume and adjusted for inflation. The reference direct cycle plant cost was assumed to be 1000$/kWe – a target for new nuclear power plants in deregulated electricity market. More details of the economic assessment are given in Ref. 13.

Sizing of the IHX was based on detailed heat transfer calculations using 30 axial nodes. In addition, a simplified 1-D stress analysis was also performed to determine the plate thickness and channel pitch necessary to withstand the pressure difference between the SCO_2 side (20MPa) and the helium side (7MPa). At elevated temperatures, creep becomes the primary mechanism of concern for stress analysis, because it limits IHX lifetime. The prime currently available structural material for high temperature reactors, alloy 800, was used for the basic design. It is noted that the longest time to creep-driven rupture available for this material is 10^5 hours, which constitutes an IHX lifetime of about 12 years. This is a relatively short lifetime so a better material with higher stress rupture strength will have to be developed if a 30 year lifetime is specified, to reduce the number of replacements during a 60-year plant life. For the advanced SCO_2 design, it becomes very difficult to design the IHX even with a lifetime of 10 years because of the low allowable stress rupture strength of Alloy 800 at higher temperatures (IHX helium inlet temperatures of 800°C would be needed). Therefore, the calculations were performed with the stress rupture strength of alloy 800 shifted by 100°C (see Figure 3) assuming that advanced materials will become available. This is also the reason we designated the SCO_2 case having a turbine inlet temperature of 650°C "an advanced design".

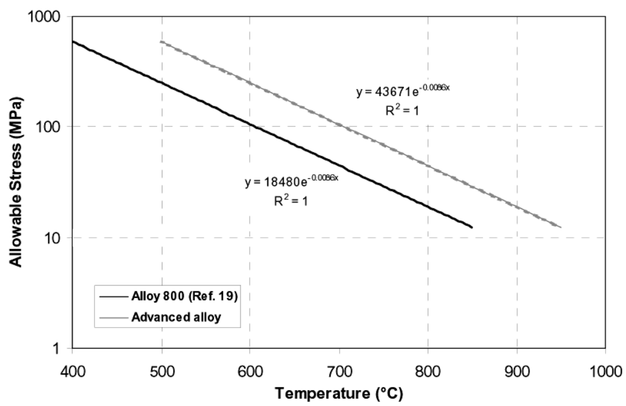

Fig. 3. Allowable Stress for 10^5 Hours Lifetime

3.3 GFR Coupled to Basic SCO_2 Design at 550°C

First, the performance of an indirect helium to supercritical CO_2 cycle with turbine inlet temperature of 550°C (basic design) is evaluated. Figure 4 shows the indirect cycle cost in $/kW$_e$ relative to the direct SCO_2 cycle (at 1000$/kWe) as a function of the reactor inlet temperature for different reactor outlet temperatures.

It is noted that the cost study is by far not exhaustive since it includes only the additional costs of IHX, circulator and IHX vessel. In reality, additional piping, possibly smaller containment size and different inlet and outlet core temperatures will also affect the cost. Because these effects could not be accurately quantified at this preliminary stage of investigations, they were not incorporated in the cost evaluations. However, preliminary containment sizing was performed by General Atomics and it was concluded that no appreciable containment size reduction was possible since the size was to a large extent determined by the layout of shutdown cooling heat exchangers. Hence, Figure 4 should be viewed as a guidance tool for the selection of a good design point for the intermediate heat exchangers and core inlet/outlet temperatures. It shows that a reactor outlet temperature of at least 700°C is necessary in order to maintain the cost increase due to the additional costs of IHX, IHX vessel and circulators below 10%. This corresponds to about 10 to 25m^3 of IHX active volume, depending on the core inlet temperature. In addition, core inlet temperature should not be above 450°C. The minimum cost increase of 9% was achieved at a reactor core outlet temperature of 760°C (compared with 550°C for a direct supercritical CO_2 cycle) and inlet temperature of 440°C. The comparison with the reference direct helium Brayton cycle is also of interest. Studies in Ref. 13 showed that the plant with basic SCO_2 direct cycle was about 4% cheaper than the plant with direct helium Brayton cycle at 850°C. Hence, considering the above 9% cost increase for a plant with indirect helium/SCO_2 cycle versus that with direct SCO_2 cycle, a cost increase of about 5% versus the reference helium direct cycle can be expected.

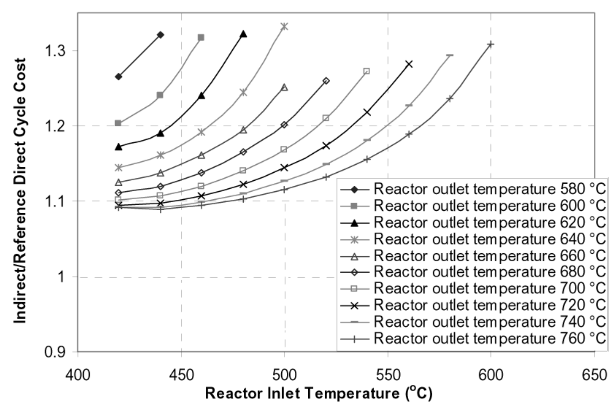

Fig. 4. Relative Increase of Capital Cost of Indirect Cycle Versus Basic SCO_2 Direct Cycle*

Figure 5 shows how the indirect cycle efficiency decreases as the reactor inlet temperature increases for constant core outlet temperature: hence as the core temperature rise decreases. The net efficiency of the direct supercritical basic CO_2 cycle is 44.0% for a GFR core pressure drop of 60kPa. The smallest efficiency penalty from the transition to the indirect cycle is at the core inlet/outlet temperatures of 440°C/760°C, where indirect cycle efficiency reaches 42.7%. Therefore, if the intermediate heat exchangers are carefully optimized and the GFR core is designed for low pressure drop, the efficiency of the indirect helium/SCO_2 cycle is not significantly lower than the efficiency of a direct SCO_2 cycle at the same turbine inlet temperature. It is also noted that the efficiency optimum of the inlet/outlet temperatures corresponds to the cost optimum. This is not surprising considering the fact that the additional costs of an indirect cycle are only a small fraction of the total plant capital cost.

In conclusion, compared to the direct SCO_2 cycle of the basic design the operation of the indirect cycle with a reactor core outlet temperature of 760°C does not introduce a significant cost increase due to the cost of the key additional hardware and the efficiency reduction in comparison with the SCO_2 direct cycle is only 1.3%. More importantly, the efficiency reduction is also small compared to the helium direct cycle at the same core outlet temperature of 760°C, which has net cycle efficiency of 43.0%. This is made possible by the very low-pressure-drop GFR core, the design of which was driven by the desirability to remove decay heat by natural circulation. Also it should be noted that reducing the reactor outlet temperature down to about 700°C introduces a miniscule cost increase, which indicates that a wider range of operating temperatures is economically attractive.

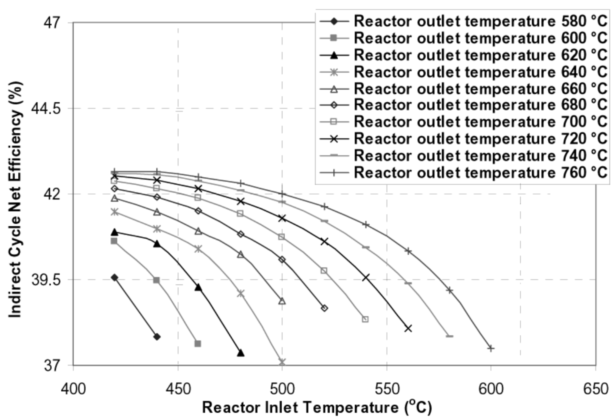

Fig. 5. Indirect Cycle Efficiency Map for Various Core Inlet and Outlet Temperatures*

3.4 GFR with Advanced SCO_2 Design at 650°C

The basic design with turbine inlet temperature of 550°C provides the benefit of a reduction of core outlet temperature from 850°C to 700°C while still achieving attractive net plant efficiency of 42.2%. However, this efficiency is appreciably lower when compared to that of the reference helium direct cycle at a core outlet temperature of 850°C, which has a net plant efficiency of 45.9%, based on the same efficiency calculation methology and assuming the same losses. To achieve higher efficiencies comparable to those of the reference helium direct cycle, a higher turbine inlet temperature for the SCO_2 cycle is necessary. Therefore, the advanced SCO_2 cycle with turbine inlet temperature of 650°C was coupled to a GFR and the same optimization study was performed as for the basic SCO_2 cycle. Results of the economic evaluation for temperatures of interest are shown in Figure 6. Two sets of curves are shown: the upper couple of curves was generated for Alloy 800 as IHX material, the lower two curves were obtained for an advanced alloy with higher allowable stress rupture strength (see Figure 3) assuming its cost to be the same as for Alloy 800. The advanced alloy was used because it was not possible to design IHX that would satisfy the allowable stress rupture strength of Alloy 800 at temperatures above 750°C and achieve a reasonable lifetime. Such an advanced material has to be developed, tested and incorporated into ASME standards.

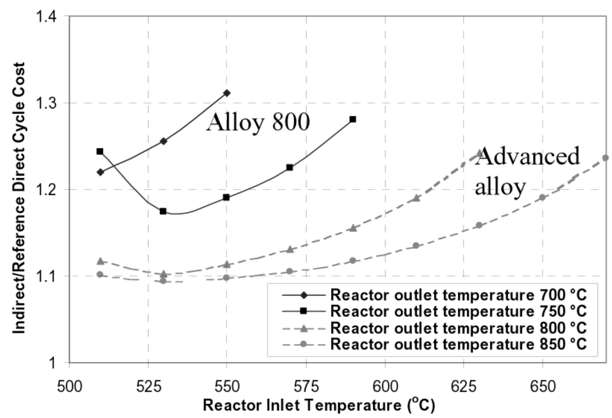

Fig. 6. Relative Increase of Capital Cost of Indirect Cycle Versus Advanced SCO_2 Direct Cycle

The optimum IHX volume of 20 to 45m^3 is almost twice as large as for the basic SCO_2 cycle design. The pumping power is about the same as in the basic case, which indicates that the pumping power is more important than the IHX volume. Also increasing the turbine inlet temperature by 100°C yielded an increase of the core outlet temperature by about 100°C as well. About the same 10% cost increase for both the basic and the advanced designs versus the reference direct plant comes from the assumption of the same specific cost of 1000$/kWe for the reference direct plants for 550 and 650°C. Therefore, the results only indicate the potential of an indirect cycle versus direct cycle at a given temperature and should not be cross-compared at different temperatures. No conclusion can be drawn whether the basic or advanced case has better economics.

For alloy 800, the core outlet maximum temperature at which an IHX could be designed with reasonable lifetime was 750°C. The minimum cost increase of 17% occurs at core inlet/outlet temperatures of 530/750°C. If lower cost increase is desirable, core outlet temperatures above 750°C are necessary and an advanced alloy with higher allowable stress rupture strength used. If the cost of such material would be comparable to that of alloy 800, the cost increase could be reduce to about 10% at a core outlet temperature of 800°C, as can be observed from the lower curve set.

The efficiency map in the core inlet/outlet temperature space for both alloy materials is shown in Figure 7. It can be observed that alloy 800 cannot reach the efficiency of the direct helium Brayton cycle of 46.3%. If an advanced alloy can be developed, the maximum net efficiency of 46.3%, which matches the target efficiency of 46.3%, is achievable at 50°C lower core outlet temperature than the 850°C for the direct Brayton cycle. Only a small efficiency gain is possible by increasing core outlet temperature to 850°C, hence the core inlet/outlet temperatures of 530/800°C have been selected as an optimum for the advanced SCO_2 cycle design.

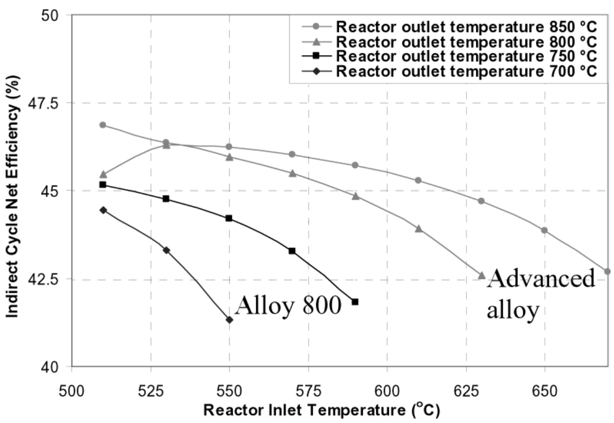

Fig. 7. Indirect Cycle Efficiency Map for Two Core Inlet and Outlet Temperatures

3.5 GFR with Advanced SCO_2 Design at 650°C and Reduced Heat Sink Temperature

In the framework of GFR exploratory studies, CEA has launched a consistent study about various power conversion cycles of potential interest for GFR design using the CYCLOP computer tool [20]. Similar assumptions have been adopted to the largest extent possible in order to have a fair comparison. In all cases, helium was the primary coolant. direct, indirect and combined cycles were studied with different types of fluids: N_2, CO_2, H_2O. For these latter fluids, supercritical options were considered with similar maximum values: 25 MPa, 650°C. A sea water heat sink with a low temperature of 15°C and a maximum inlet core temperature of 480°C were assumed. Some preliminary results are summarized in Table 5.

The following comments can be made concerning the He/SCO_2 cycle:

- The 15°C sea water heat sink chosen implies that the main compressor inlet temperature will be below the critical value of 30.97°C (about 21°C, for a 6°C pinch point). As mentioned in Section 2, a CO_2 sub-critical temperature could lead to design problems (cavitation). It has yet to be confirmed whether this is very different from condensate extraction pumps used in steam cycles.

Table 5. Three Potential GFR Energy Conversion Cycles

	Direct He	Indirect He/N2-He	Indirect He/SCO_2
Core outlet P (MPa), T (°C)	7 / 850	7 / 850	7 / 680
Core inlet T (°C)	480	480	461
Primary flowrate, (kg/s)	310	310	526
Turbine inlet P, T	7 / 850	6.5 / 820	25 / 650
Turbine outlet P, T	2.5 / 500	1.8 / 460	6 / 468
Net efficiency %	48.2	46.6	46.2

- An outlet core temperature of 680°C was imposed in order to have a significant advantage in term of core design independently of potential penalties in terms of IHX design (decrease of the primary to secondary temperature difference leading to an increase of the required heat transfer area).
- The obtained cycle parameters result from an optimization process (in particular the inlet core temperature).

The interest in SCO_2 cycles for balance of plant is confirmed, i.e., attractive efficiency for lower outlet core temperature. Compared to “Advanced I” case (all indirect He/SCO_2 cycle designs are summarized in Table 6 of Section 3.5), this lower core temperature implies a greater coolant mass flow rate and therefore a greater blower power. In spite of this additional energy consumption, the cycle efficiency remains about the same due to the colder heat sink and higher maximum pressure set. Another interesting issue of the He/SCO_2 cycle studied by CEA is the optimum core inlet temperature for which a maximum value of 480°C was imposed during the optimization process to limit the vessel wall temperature. In standard Brayton cycles, the thermodynamic optimum is above this 480°C temperature whereas in the SCO_2 indirect cycle, the optimum found is slightly below this limit which could ease the vessel design.

Another objective of the CEA study has been to define the global layout of an indirect cycle for a GFR based on SCO_2 in order to compare it to the direct cycle lay out. For the conditions reported in Table 5, preliminary pre-sizing of the SCO_2 main power conversion cycle components has been performed. Five to six stages for the turbine (tip diameter of 1.3 m) and 16 to 19 stages for the recompressing compressor (tip diameter of 0.8 m) were obtained. The main compressor, which operates in the liquid phase and behaves more like a pump, was not studied. However, the compactness of the conversion cycle is confirmed even with a higher pressure ratio. As far as the power conversion system is concerned a possible turbine hall lay out is given in Figure 8. MIT studies complemented by CEA analyses have led to this pre-sizing.

The overall system layout is shown in Figure 9. This is a very preliminary arrangement showing the compactness of the power conversion system. It is worth recalling that for reactor decay heat removal, the strategy chosen is to use a "close-containment" in order to keep some gas backup pressure after a loss of coolant transient. The size of this close-containment is driven by the size of primary components. Here, the size of the "close-containment" was not changed, in fact the intermediate heat exchanger and the helium blower which replace the PCS are, quite likely, more compact. This means the possibility of reduction of the "close-containment" size, and thus of the capital cost. The IHX size was estimated roughly at this stage, having in mind, that in this arrangement (Table 6), the minimal difference of temperature between the core outlet (680°C) and the SCO_2 turbine inlet (650°C) was selected. The potential advantage for the core design is significant enough to accept penalties in the IHX sizing.

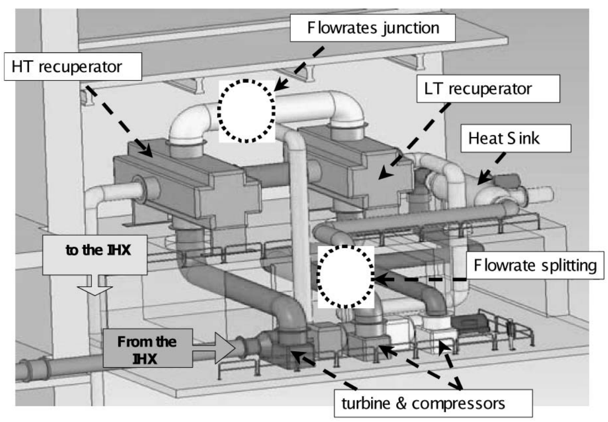

Fig. 8. Power Conversion System Hall

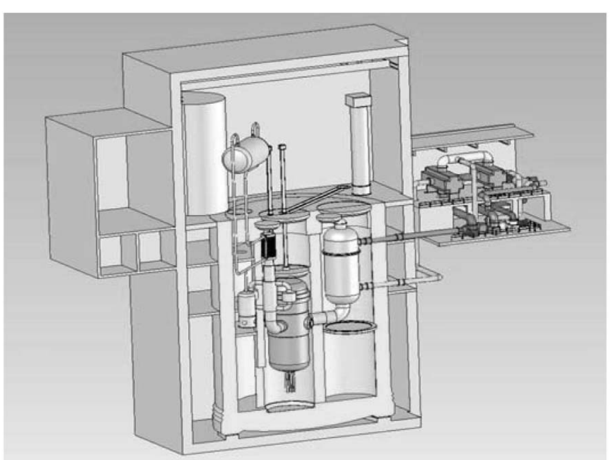

Fig. 9. GFR System Layout with SCO_2 Indirect Cycle

Table 6. Summary Table of Key Parameters

	Basic	Advanced I	Advanced II
Turbine inlet temperature (°C)	550	650	650
Compressor inlet temperature (°C)	32	32	21
SCO_2 IHX inlet temperature (°C)	396.5	488.8	414.5
Core inlet temperature (°C)	440	530	461
Core outlet temperature (°C)	700	800	680
Core flow rate (kg/s)	385	388	526.
Helium pressure (MPa)	7	7	7
Net plant efficiency (%)	42.2	46.3	46.2
IHX volume (m^3)	16.4	33.7	-
Circulator power consumption (MW)	17.9	17.4	46.

4. CONCLUSIONS

The GFR indirect cycle with SCO_2 power conversion system is an attractive alternative option to a GFR direct cycle, since it allows the achievement of appealing plant efficiencies at moderate core outlet temperatures. In addition, separation of the balance of plant from the primary helium coolant eliminates contamination of the turbomachinery in case of fission product leakage from the fuel, making maintenance easier. On the other hand, specific capital cost is, quite likely, higher than that of the reference helium direct cycle because of the additional hardware and lower efficiency. Therefore, the indirect cycle is primarily attractive as a backup to the direct cycle for the first GFR units before fuel performance data are established. The preliminary studies presented here showed that a GFR with core outlet temperature of 700°C (150°C lower than the reference helium cycle) coupled to the SCO_2 basic cycle at a turbine inlet temperature of 550°C can achieve an attractive net efficiency of almost 42.2%. Coupling the GFR to a higher performance SCO_2 cycle (turbine inlet at 650°C and 20 MPa or 25MPa) can increase the efficiency to 46.3%, but requires development of IHX structural materials with significantly higher stress rupture strength than the currently available alloy 800 to achieve an acceptably long lifetime for the IHX. The net plant efficiency using the reference helium direct Brayton cycle with core outlet temperature of 850°C would be between 46% to 48%, i.e., just slightly above the net efficiency of advanced SCO_2 designs.

Compared to a helium direct cycle option, there is another significant advantage in using indirect cycles for the GFR: the reduction of primary component size makes easier the design of the "close-containment", a key component responsible for maintaining the backup pressure in a loss of coolant accident, which allows significant reduction of the power demand of the decay heat removal systems.

In addition to the SCO_2 component design issues, it is worth mentioning that operating issues, such as start-up and load variations should be also studied. In particular, for load following or cogeneration applications, it is necessary to identify adequate means to maintain high efficiency at reduced load.

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