Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code

Nuclear Science and Engineering Division
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May 10, 2022
ABSTRACT

This report presents the load following analysis of the “Holos-Quad” 10MWe micro-reactor design developed by HolosGen LLC. This analysis is focused on the Holos-Quad micro-reactor ability to match the changing grid demand at 10%/min rate. The control mechanisms for the plant are identified, simulated, and compared. Based on the comparison results, control strategy for load following of the Holos-Quad micro-reactor is developed. The control strategy and load following capabilities are demonstrated in a full-range down-and-up reactor power transient from 100% to 0% and back to 100% load at 10%/min rate. All calculations are carried out with Argonne’s Plant Dynamics Code.
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1 Introduction: Analysis Goals and Definitions

The Holos-Quad micro-reactor design, developed by HolosGen LLC, is equipped with a 22 MWth (Mega-Watt thermal) core and an integral power conversion system converting the core thermal energy into approximately 10 MWel (Mega-Watt electric). This design can be configured to support a wide range of applications. It is an innovative high-temperature gas-cooled reactor concept using TRI-structural ISOtropic particle fuel (TRISO). The fuel is distributed in graphite hexagonal blocks, cooled with helium in a direct Brayton cycle independently executed by four Subcritical Power Modules (SPMs) fitted into a hardened 40-foot container, whose dimensions are in compliance with ISO shipping container requirements. In FY2019 HolosGen LLC was awarded by the Department of Energy Advanced Research Project Agency-Energy (DOE ARPA-E) under the MEITNER funding program to conduct technical and economic feasibility and performance analyses [1]. As part of the MEITNER award, the Argonne National Laboratory (Argonne) contributed expertise through two specialized teams: The “Design Team” and the “Resource Team”. The Design Team was dedicated to validate feasibility of the Holos-Quad core and to optimize its core design through neutronics analyses. The Resource Team was dedicated to feasibility verification via high-fidelity codes of Holos-Quad thermal-hydraulic, heat transfer, shielding, and structural aspects.

This report summarizes the activities conducted to analyze Holos-Quad thermal-hydraulic performance by the Argonne Resource Team. The “Holos-Quad” micro-reactor, is characterized by a 22 MWth graphite-moderated high-temperature core directly coupled to a Brayton cycle, utilizing helium as working fluid to convert the core thermal energy into approximately 10 MWel. Figure 1 illustrates the Holos-Quad 10 MWel micro-reactor formed by coupling four identical SPMs, each equipped with an independent closed-loop Brayton cycle. The Holos-Quad micro-reactor design is characterized by many innovative features, including (but not limited to) horizontal arrangement of a reactor core located between the compressors and a turbine, with components layout similar to those of a turbo-jet engine, wherein the thermal energy from the combustion of jet fuel is replaced in the Holos-Quad by nuclear heat with a closed-loop Brayton cycle. The composition of the micro-reactor core includes four Subcritical Power Modules (SPMs) each with its own power closed-loop Brayton cycle, and placement of all of the reactor systems structure and components (SSCs) within the volume characterized by a 40-ft ISO container, to leverage standard transport platforms and lower deployment cost. This micro-reactor configuration enables independent shielded transport of each SPM spent (1/4) core through dry-transport and storage casks as illustrated in Figure 1.

One of the Holos-Quad design goals is to demonstrate computationally an ability of the micro-reactor to execute “power maneuvering”, which is referred to in this report as load following. The primary goal of this design load following feature is to match the changing demand from the electrical grid. Because the micro-reactor is developed for full-power operation at the design conditions (10 MWel), the purpose of the load following analysis is to demonstrate that the micro-reactor can operate at power levels below 100% nominal. As part of the design requirements, the load following goals are specifically defined as:

- Ability to change electrical power output from 100% all the way to 0%, and viceversa;
- Utilize a minimum power change rate of 10%/min.
Therefore, the primary goal of the analysis presented in this report is to demonstrate the Holos-Quad design ability to change power from the nominal design condition at full 100% power all the way to zero power in 10 minutes (600 seconds). However, in order to show these capabilities, the specific ways to achieve such power maneuvering, i.e. find and analyze the plant control mechanisms, need to be identified, simulated, and investigated. This part of the analysis is referred to as control strategy development.

Note that here and in the rest of the report, the “power level” refers to the net electrical output from the plant. As it will be shown in the analysis, this output could be different from the reactor core power level. The net reactor power output, in addition to the reactor core power, accounts for the efficiency of the components forming the energy conversion system, all mechanical losses (such as friction), generator effectiveness (losses), and all internal power requirements, such as those for the cooling water pumps. In a simple form, the power output is the net electrical output at the grid connection terminals.

Figure 1. 10 MWe Holos-Quad micro-reactor fitting a 40-ft transport container.
2 PDC Model for the Holos-Quad Micro-Reactor

The analysis presented in this report is carried out with the Plant Dynamics Code (PDC) [2]. PDC was developed at Argonne originally for steady-state and transient analysis of supercritical carbon dioxide (sCO₂) Brayton cycles utilizing intermediary fluids, where the core cooling fluid may be different than the fluid executing thermal conversion to electrical power. For the purposes of the Holos-Quad load following analysis, the code was modified to allow simulation of gas-cooled reactors with direct helium Brayton cycles [3]. The PDC includes two major parts: steady-state and transient. The steady-state PDC capability was first used to develop, analyze, and optimize the steady-state performance of the Holos-Quad helium Brayton cycle components at the design (full power) conditions [4]. The transient capabilities of PDC were utilized to carry out the load following analysis presented in this report.

Figure 2 shows the design conditions for the Holos-Quad micro-reactor obtained with PDC. Note that the results in this figure are for one (out of four) subcritical power modules (SPMs). Since the SPMs are mechanically and hydraulically independent (they are coupled via core neutronics), in these analyses it is assumed that all SPMs behave identically. Therefore, the analyses focus on the behavior of a single SPM. As shown in Figure 2, each SPM is characterized by a thermal power of 5.5 MW at full design conditions, and produces 2.45 MW of net electrical output, with the net cycle efficiency of 44.6%. All transients presented in this report start at time=0 at the design conditions reported in Figure 2.

Figure 2. Holos-Quad Design Conditions for One SPM.
Several generations of the Holos-Quad core were designed within this project before converging to the current full-scale Gen 2+ design that is detailed in this report [5]. “Gen 2+” represents the most optimized design configuration achieved as a result of the work conducted under the ARPAE award. In this configuration, stationary and simplified SPMs are represented by sealed fully integrated power modules to ease assembly and spent SPM removal. In Gen 2+ the Holos-Quad SPMs are stationary during operations and can be independently decoupled from other SPMs to ensure irradiated SPMs can be disposed of via shielded dry casks with dimensional constraints compliant with ISO containers’ dimensions for transport to spent fuel storage facilities. For each design configuration, the size and layout of components within the volume represented by a 40-foot shipping container was modified to satisfy Brayton cycle, dimensional, heat transfer and cost constraints. The PDC enabled down-selection of optimal configurations under various design performance constraints and supported the design evolution from Gen 0 to Gen 2+. The micro-reactor design configuration adopted in the analyses described in this report is the “Gen 2+ Holos-Quad” design with water-cooling circulation through the Inter-Cooler (IC) and the Cooler (Cool) heat exchangers during power operations. Main aspects of the Gen 2+ SPM design are shown in Figure 3.

The PDC model of the Holos-Quad reactor is shown in Figure 3, including the components, piping, valves, and other controls. A detailed discussion of the cycle arrangement and control mechanisms will be provided in this and following chapters. Table 1 defines the components in the PDC model and in Figure 3, while Table 2 provides the nomenclature used in the PDC modeling and elsewhere in this report, including the result plots. Figure 3 also defines the flow nodes (pipe indeces in green boxes) that will be used later in this report for the PDC model description.
Figure 3. Holos-Quad Model in PDC.
Table 1. PDC Modeling: Components

<table>
<thead>
<tr>
<th>PDC Model</th>
<th>Component</th>
<th>PDC Model</th>
<th>Component</th>
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</thead>
<tbody>
<tr>
<td>Rx</td>
<td>¼ reactor core*</td>
<td>Turb</td>
<td>Turbine</td>
</tr>
<tr>
<td>LPC</td>
<td>Low pressure compressor</td>
<td>HPC</td>
<td>High pressure compressor</td>
</tr>
<tr>
<td>Rec</td>
<td>Recuperator</td>
<td>Cool</td>
<td>Cooler</td>
</tr>
<tr>
<td>IC</td>
<td>Intercooler</td>
<td>Tank</td>
<td>Inventory tank</td>
</tr>
<tr>
<td>Gen</td>
<td>Generator</td>
<td>Motor</td>
<td>Compressor motor</td>
</tr>
<tr>
<td>sp</td>
<td>Valve</td>
<td>mx</td>
<td>Water pump</td>
</tr>
<tr>
<td>I</td>
<td>Flow split</td>
<td></td>
<td>Flow mix</td>
</tr>
<tr>
<td></td>
<td>Pipe index</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*RHX (reactor heat exchanger) is also used in this report

Table 2. PDC Modeling: Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Meaning</th>
<th>Abbreviation</th>
<th>Meaning</th>
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</thead>
<tbody>
<tr>
<td>TBP</td>
<td>Turbine bypass</td>
<td>(A)</td>
<td>Automatic control</td>
</tr>
<tr>
<td>CBP</td>
<td>Cooler bypass</td>
<td>(M)</td>
<td>Manual control</td>
</tr>
<tr>
<td>INV</td>
<td>Inventory</td>
<td>I</td>
<td>Inlet (port)</td>
</tr>
<tr>
<td>INVI</td>
<td>Inventory inlet (valve)</td>
<td>O</td>
<td>Outlet (port)</td>
</tr>
<tr>
<td>INV0</td>
<td>Inventory outlet (valve)</td>
<td>HI, HO, CI, CO</td>
<td>Hot/cold inlet/outlet (ports)</td>
</tr>
<tr>
<td>TIN</td>
<td>Turbine inlet (valve)</td>
<td>I1, O1</td>
<td>Primary inlet/outlet (ports)</td>
</tr>
<tr>
<td>CIN</td>
<td>Compressor inlet (valve)</td>
<td>I2, O2</td>
<td>Secondary inlet/outlet (ports)</td>
</tr>
</tbody>
</table>

A unique feature of the Holos-Quad design is a module-type arrangement where the majority of the plant components are integrated and enclosed in a single vessel (shell). This configuration eliminates the traditional balance of plant coupling various components and is somewhat similar to the integrated balance of plant adopted by tubo-jet engines and gas turbines. The components inside the vessel are connected by ducts (flow paths) which are not necessarily cylindrical pipes. Still, because PDC was developed for a traditional cycle analysis, the underlined assumption in the code is that the Brayton cycle components are connected by pipes. Therefore, an effort was made to, while still using the piping arrangement, represent the SPM flow paths with “equivalent” pipes. Table 3 shows the input for the piping assumed in the current calculations, including the connected components and ports (e.g., “CO” represents cold outlet port), “Count” for number of parallel runs (in this design, all are single pipes), length (L), inner diameter (ID), thickness (t), number of 90° bends (N_{bend}), bend radius-to-pipe-diameter ratio (r_{b/D}), and heat loss multiplication factor (k_{HL},...
discussed below). This input was prepared in consultation with HolosGen engineers to represent Gen 2+ piping arrangement (the design of the flow-paths and piping arrangements is under optimization as the design evolves into future generations). Table 3 also shows the Gen 2+ piping arrangement and components locations, specifically with respect to connection of bypass and other control lines (see Figure 3 and Tables 1 and 2 for definition of pipes listed in Table 3).

In further transient calculations, it was found that the numerical solution is more stable when the reactor inlet and outlet plena are represented as volumes rather than pipes (PDC has provision of volume on each component inlet and outlet, e.g., for heat exchanger plena). Therefore, the reactor inlet and outlet pipes (#14 and #1 in Table 3) are included in the model mostly to represent the calculated pressure drop (~1 kPa) from inlet and outlet form losses, respectively, with the pipe dimensions (ID) selected to match those pressure drops.

### Table 3. Input for HolosGen Cycle Piping

<table>
<thead>
<tr>
<th>No.</th>
<th>Input</th>
<th>Output</th>
<th>Count</th>
<th>L, m</th>
<th>ID, m</th>
<th>t, mm</th>
<th>N_bend</th>
<th>r/D</th>
<th>k_HL</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Rx</td>
<td>CO</td>
<td>Turb</td>
<td>I</td>
<td>0.5</td>
<td>0.13</td>
<td>25</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Turb</td>
<td>O</td>
<td>Rec</td>
<td>HI</td>
<td>6</td>
<td>0.2</td>
<td>25</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Rec</td>
<td>HO</td>
<td>TBPmx</td>
<td>HI</td>
<td>1</td>
<td>0.25</td>
<td>25</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>TBPmx</td>
<td>O</td>
<td>INVmx</td>
<td>HI</td>
<td>1</td>
<td>0.25</td>
<td>25</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>5</td>
<td>INVmx</td>
<td>O</td>
<td>CBPsp</td>
<td>I</td>
<td>1</td>
<td>0.25</td>
<td>25</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>6</td>
<td>CBPsp</td>
<td>O1</td>
<td>Cool</td>
<td>HI</td>
<td>1</td>
<td>0.25</td>
<td>25</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>7</td>
<td>Cool</td>
<td>HO</td>
<td>CBPmx</td>
<td>HI</td>
<td>1</td>
<td>0.5</td>
<td>25</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>8</td>
<td>CBPmx</td>
<td>O</td>
<td>LPC</td>
<td>I</td>
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<td>25</td>
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<td>1</td>
</tr>
<tr>
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<td>O</td>
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<td>HI</td>
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<tr>
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<td>HPC</td>
<td>I</td>
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<td>3</td>
<td>1</td>
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<td>INVsp</td>
<td>I</td>
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<td>INVsp</td>
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<td>TBPsp</td>
<td>I</td>
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<td>13</td>
<td>TBPsp</td>
<td>O1</td>
<td>Rec</td>
<td>Cl</td>
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<td>1</td>
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<tr>
<td>14</td>
<td>Rec</td>
<td>CO</td>
<td>Rx</td>
<td>Cl</td>
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<td>0.12</td>
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<td>0</td>
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<tr>
<td>15</td>
<td>CBPsp</td>
<td>O2</td>
<td>CBPmx</td>
<td>I2</td>
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<td>O2</td>
<td>TBPmx</td>
<td>I2</td>
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<td>2</td>
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<tr>
<td>17</td>
<td>INVsp</td>
<td>O2</td>
<td>Tank</td>
<td>I</td>
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</tr>
<tr>
<td>18</td>
<td>Tank</td>
<td>O</td>
<td>INVmx</td>
<td>I2</td>
<td>1</td>
<td>1</td>
<td>0.01</td>
<td>1</td>
<td>2</td>
</tr>
</tbody>
</table>

The PDC calculations assume that all pipes are thermally isolated, with no heat loss to the environment. This assumption is indicated in Table 3 by selecting the heat loss multiplication factor $k_{HL} = 0$ for all pipes. This assumption will not affect the results of load following calculations due to the rather short time scale of the transients (within one hour) under consideration in these analyses, and because the flow conditions (temperature) would not significantly deviate from steady-state values. In addition, the largest heat losses are usually encountered in the pipes on the hot side of the cycle. In the Holos-Quad design, however, these “pipes” will not necessary experience large heat losses, as they are characterized by internal flow paths (inside a vessel). In the Gen 2+ design configuration, the “turbine return pipe” (Flow Return Channel in Figure 3), coupling the turbine to
the recuperator (#2 in Table 3), is characterized and modeled as an actual pipe. As this pipe may run outside or inside the SPM shell (e.g., within the reactor reflector region), in this analysis it is provided with an effective insulation. Still, even in the absence of heat loss to the environment, PDC would calculate the heat transfer between the working fluid (helium) and the pipe wall. But again, this heat transfer is not expected to be a significant factor for load following calculations, for the reasons discussed above.

Important for the transient calculations is the helium inventory in the cycle and its distribution among the components and pipes. Table 4 shows the helium mass (inventory) in the cycle at steady state design conditions (see previous figures and tables in this chapter for pipe numbering). These masses are calculated by PDC based on the component and piping dimensions (input) and local densities (PDC results). As Table 4 demonstrates, the majority of the helium mass is located in the reactor component. Because this inventory is initially at high cycle pressure (~7 MPa), this result will affect how helium mass is redistributed in the cycle when the temperature and pressure conditions change during load following.

### Table 4. Working Fluid Inventory

<table>
<thead>
<tr>
<th>Inside components</th>
<th>Inside pipes</th>
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<tbody>
<tr>
<td>M_Turb</td>
<td>Pipe 1</td>
</tr>
<tr>
<td>M_LPC</td>
<td>Pipe 2</td>
</tr>
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<td>M_HPC</td>
<td>Pipe 3</td>
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<td>M_Rx</td>
<td>Pipe 4</td>
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<tr>
<td>M_Rec</td>
<td>Pipe 5</td>
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<tr>
<td>M_Cool</td>
<td>Pipe 6</td>
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<td>M_IC</td>
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<td>Pipe 18</td>
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<table>
<thead>
<tr>
<th>kg</th>
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</tbody>
</table>

TOTAL 5.56 kg
3 Holos-Quad Plant Control Mechanisms

For innovative concepts as that developed by HolosGen, there is no "established" solution for approach to load following and plant control. Therefore, for this work an approach was taken to investigate all feasible control mechanisms that could be used to achieve load following goals. Not all of the plant control mechanisms identified are intended to be implemented in the final design; - the selection of the recommended control mechanisms is executed in the control strategy development chapter of this report. The control mechanisms identified for the Holos-Quad Gen 2+ design are shown in Figure 4 and include:

- Throttling valves: at turbine and compressor inlets,
- Turbine bypass loop,
- Inventory control circuit,
- Compressor shaft speed control, and
- External controls: reactor power and water flow rate.

Figure 4. Holos-Quad Plant Control Mechanisms.

Note that the turbine bypass line in Figure 4 is located somewhat away from the turbine itself. This was done due to the Holos-Quad-specific design Gen 2+ configuration. In particular, as discussed in previous chapter, the lines connecting the cycle components in Figure 4 are not necessarily “pipes” in the actual design. For example, due to integral concept of HolosGen’s SPMs, all the components on the main axis, - generator, turbine, reactor, recuperator, two compressors, and motor, - are located inside a single volume (SPM shell), which also provides the flow paths.
between these components. The absence of traditional pipes may represent some difficulties for (but necessarily eliminates the possibility of) connecting the bypass lines between these components. On the other hand, the line to and from the two cooling heat exchangers (IC and Cool) are external to the SPM and are represented by actual pipes. In addition, placing the bypass line on the low-temperature side of the recuperator heat exchanger (Rec) will reduce the design requirements for the valve itself, since it does not need to operate at high temperatures. For these reasons, the bypass line in Figure 4 and everywhere else in this report includes “low temperature (LT)” designation in its name. Also note that because of this selected location of the bypass line, it’s actually closer to the compressor than to the turbine. In some designs where similar concepts were proposed, it might be called “compressor recirculation” path. Regardless, the principle is the same with the goal of diverting a controlled flow away from the turbine. And so this mechanism is still called “turbine bypass” in this report. Lastly, note that the in the Holos-Quad design the turbine throttling valve is not located on a pipe, and therefore this component will likely not to be represented by an actual valve, but rather by a flow restriction mechanism as part of a wide duct. Still, for the purposes of plant response simulation this component can be represented by an actual valve.

The compressor speed control option in Figure 4 is a somewhat unique capability of the Holos-Quad micro-reactor design. Because the compressors and turbine are physically located on the opposite sides of the core, they are placed on mechanically decoupled and independent turbomachinery shafts. These shafts are mechanically coupled to high-speed motor (compressor side) and generator (turbine side) without gear-boxes and all rotary components are magnetically levitated via active magnetic bearings (no metal-to-metal contact) to eliminate the need for lubrication and filtration systems. Thus, the compressor shaft speed and the turbine shaft speed can be independently varied via electronic motor drives (within the motor and generator rotational speed capabilities) and presents a novel, additional and independent way of controlling helium flow rate in the cycle. In the analysis presented in this report, only freedom in the compressor speed was investigated, as the turbine speed was assumed to be fixed in transients.

The control principles (i.e., how these controls affect the system behavior and the plant output) and their implementation in the PDC are discussed in the next chapter, which is dedicated to simulation of load following by each of the control mechanisms illustrated in Figure 4. The exception is made to the external controls, for reactor power and water flow rate through the IC and Cool heat exchangers, which are designed to maintain conditions (outlet helium temperature), rather than for power maneuvering. These two auxiliary controls are discussed below.

### 3.1 Reactor Modeling and Power Control

The newly implemented reactor component model in the PDC is described in Reference [3]. For the Holos-Quad design, the model’s option considers the coolant flowing inside the tubes within coolant channels and separate fuel channels, shown in Figure 5. The coolant tubes are formed by SiC sleeves, thermally coupled to graphite elements by means of lead films filling the gap (“lead gap”) formed by the outer walls of the SiC sleeves and the inner walls of the cooling channels within the graphite-moderator core matrix. The coolant is in the “tube” element of the model, which combines the SiC sleeves and the lead gap. Because the sleeve wall thickness is much greater than the lead buffer’s, SiC properties are used for the combined tube material. The fuel channels are represented by uniform mixture of graphite and TRISO particles, with 40% fuel (particles) volume
fraction. The PDC reactor module models a single average coolant channel. The number of fuel channels (per one coolant channel) is calculated to preserve total number of fuel channels in one SPM. The heat transfer equations from fuel to coolant are provided in Reference [3].

Figure 5 shows radial (cross-sectional) structure of the Holos-Quad reactor component Rx, as modeled in the PDC. Ten nodes are used axially, along the coolant channel length.

The PDC reactor module doesn’t employ reactor kinetics equations. Instead, the reactor power is assumed to be an external input. The total reactor power is divided by the number of axial nodes to calculate heat deposition in each axial region (thus, the power is assumed to be axially uniform). Although there are several option in PDC for how the reactor power can be calculated in transient, in the model described in this report, the reactor power is set by an automatic control of core-outlet coolant temperature. In all the simulations presented in this report, the control settings were configured to maintain nominal 850 °C for the core outlet temperature. In PDC, the automatic control is based on proportional, integral, and differential (PID) controllers, which monitor the set temperature, compare it to the target value at each time step, and adjust the controllable parameter (reactor power in this case) accordingly. There is also a set of limitations for each controller in the code. For the reactor power, arbitrary limits can be set for the reactor power control to follow design requirements as for example:

- The maximum power change rate is limited to ±50% per second, and
- The reactor power is limited to 110% of nominal core power.

The PID coefficients for the reactor power control were optimized for the Holos-Quad Gen 2+ design configuration. The optimization was done to obtain a fast but stable system response to a pre-defined step change in the target temperature from 850 °C to 840 °C (-10°C change). Figure 6 demonstrates the reactor power and outlet temperature response to such requested change with the optimized PID coefficients. This response was considered satisfactory since the controller is able to adjust the target temperature relatively quickly with only slight overshoot. It is noted, however, that the results in Figure 6 are only obtained for this artificial change in target temperature. The
ability of the controller to maintain the reactor-outlet temperature will be determined based on the actual transient results presented in the rest of this report.

![Figure 6. Optimized Reactor Power Control Response to -10°C Change in Target Temperature.](image)

It is important to note that for the purpose of load following, and with this setup for the reactor power control, the reactor power is calculated by the code in each transient. Under this control approach, the reactor will follow the heat removal demand from the cycle by trying to maintain the set reactor-outlet temperature. There are multiple ways the reactor can be operated during a load following transient. For example, the setup of fixed core-outlet temperature for this controller is not necessarily an optimal method for all power levels. This setup is adopted for simplicity, as the reactor power will be automatically calculated by the code, assuring at the same time that the maximum coolant temperatures (at the core outlet) are maintained at the design conditions to minimize thermal cycling of the structures subjected to the highest-temperature.

In the transients, the code will calculate the required reactor power (from the PID control action). This is the power “deposited” in the fuel channels. Propagation of power (heat) from fuel to coolant will be calculated by the code with appropriate thermal conductivities and thermal inertias (masses and heat capacities) of fuel, matrix, tubes, and coolant materials (structures shown in Figure 5). As the results in Figure 6, for example, show that the resulting reactor thermal response could not be instantaneous and represents the real thermal inertia of the graphite components forming the Holos-Quad design. Specifically, as shown in Figure 6, the reactor power reaches minimum value well before the core outlet temperature stops decreasing. The delayed reactor response will be important in some transients presented in this report and might be one of the limiting factors on how fast the plant output level can be changed during load following.

### 3.2 Water Flow Rate Control in Coolers

Somewhat similar to the reactor power control on the high-temperature side of the cycle, the water flow rate control is used to maintain the temperatures at the low-temperature side of the cycle, in particular at the compressors inlet. The controllable parameter in this case is the water pump head, which is used by the code to calculate the water flow rate in the cooler and intercooler (Cool and IC in Figure 3). The controls are set to maintain the design value of 40 °C at the inlet of low-
pressure and high-pressure compressors (LPC and HPC in Figure 3). The limits for this controller can be arbitrarily set as follows:

- Maximum pump head change rate set at ±10% per second, and
- The pump head is limited between 0.5% and 200% of its nominal value.

Similar to the reactor power, the water flow rate controller PID coefficients were optimized to obtain a satisfactory response to a pre-defined step change in the target temperature. Figure 7 shows an example of such optimized response for the cooler and low-pressure compressor. In this case, the target temperature at the compressor inlet was set to change instantaneously by +5 °C. Similar response was obtained for the intercooler water flow and the high-pressure compressor. Again, the determination of whether the controller response is adequate will be made based on the response in the actual transient, i.e. if the automatic control will manage to maintain the design conditions during load following simulation.

![Figure 7. Water Flow Rate Control PID Check.](image)
4 Load Following by Individual Controls

In this chapter, the load following by each individual control in Figure 4 is simulated, calculated, and analyzed. This is achieved by acting on one control at a time as it is introduced into the PDC transient simulation with a goal of matching the grid demand. The action could be automatic or manual, depending on how each control is setup in the code.

For load following from 100% to 0% at 10% per minute rate, the grid demand table is provided as the input to PDC. That input is shown in Table 5 and represents linear reduction in grid demand at 10%/min rate (10% reduction every 60 seconds). Note in Table 5 that the PDC input also includes operation at zero net output for 300 seconds (5 minutes) at the end of the transient to allow for system stabilization. Also note that in this chapter only power-down transient, i.e., power reduction, will be investigated. The power-up transient with power increase will be analyzed in following chapters.

The goal of this portion of the analysis is to determine the control action by each individual control such that the net output from the plant would match the specified grid demand shown in Table 5. Power “matching” will be determined from plots analysis, rather than through absolute matching for actual values at each time step.

Table 5. Input for Load Following Grid Demand

<table>
<thead>
<tr>
<th>Time, s</th>
<th>Load, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>100</td>
</tr>
<tr>
<td>60</td>
<td>90</td>
</tr>
<tr>
<td>120</td>
<td>80</td>
</tr>
<tr>
<td>180</td>
<td>70</td>
</tr>
<tr>
<td>240</td>
<td>60</td>
</tr>
<tr>
<td>300</td>
<td>50</td>
</tr>
<tr>
<td>360</td>
<td>40</td>
</tr>
<tr>
<td>420</td>
<td>30</td>
</tr>
<tr>
<td>480</td>
<td>20</td>
</tr>
<tr>
<td>540</td>
<td>10</td>
</tr>
<tr>
<td>600</td>
<td>0</td>
</tr>
<tr>
<td>900</td>
<td>0</td>
</tr>
</tbody>
</table>

As discussed above, the external controls for the reactor power and the water flow are turned on all the time to maintain the corresponding conditions at the reactor outlet and compressor inlets.
For all the load following transients analyzed in this report, the ability of the reactor to satisfy the load following goals will be determined by comparing the plant output to grid demand, given that the following limitations are not encountered:

- **Compressor surge (stall) should be avoided.** Compressor stall is encountered when the pressure ratio across the compressor is so high that the compressor could no longer deliver any flow. In surge, negative flow through the compressor driven by such high pressure ratio starts occurring. The stall flow is the minimum flow that can be maintained in a compressor. The code is setup to calculate the margin (on flow) to the surge/stall conditions. This margin should be maintained positive at all times.

- **Turbine and compressors choke should be avoided.** For a turbine, choke conditions happen when a local flow velocity is equal to the sonic velocity and the flow could no longer increase regardless of the pressure ratio. For a compressor or turbine, “choke” conditions are defined in PDC when pressure ratio is equal to unity. The choke flow is the maximum flow rate through a compressor or turbine. Similar to surge condition, the margin to choke should be maintained in positive region at all time.

- **External controllability should not be lost.** For example, the reactor-core power should not be reduced to zero (or, more accurately, to the decay heat level) in order to maintain controllability of the reactor outlet temperature. Similar limitation applies to the water flow controls. Note, however, that this limitation is not as strict as the previous two - losing active control will result in deviation of temperatures from the design conditions, which is undesirable but not necessarily prohibitive.

Also note that absent from the list above is any structural mechanical consideration for the transients. The analysis presented in this report only deals with the thermal hydraulic aspects of load following. Questions whether the reactor structures would survive such transients, and if they do, then for how many cycles, can be answered with dedicated structural mechanical analyses, which are beyond the scope of the present work. In lieu of such analyses, significant changes in components temperatures (for thermal loading indication) or very fast changes in temperatures (for thermal stresses) will be identified and marked as potential problems for more detailed future analyses.

### 4.1 Turbine Bypass

The turbine bypass (TBP) control (Figure 8) works by diverting some of the flow away from the turbine. As less flow is going through the turbine, this component produces less work, thus resulting in lower power output from the cycle. In addition, the bypass flow is diverted back to the compressor, increasing the compressor power demand, and further reducing useful output from the plant. If no action is taken by the reactor, it continues to operate at full power level. And because less energy is being converted into useful output in the turbine, more energy will need to be removed from the cycle in the cooling heat exchangers (Cool and IC). Although the turbine bypass is one of the easiest control mechanism to implement, it usually represents the least effective control mechanism because it works by artificially reducing turbine work while inducing increased compressor work.
As discussed above, and as shown in Figure 8, the preferred Holos-Quad design option for the turbine bypass hardware is at the locations characterized by the low temperature side of the recuperator. PDC calculations executed by placing the turbine bypass on the high temperature side of the recuperator showed very similar results for the load following transient. As the turbine bypass control method is the least effective option for load following, and by considering that engineering actuators operating at very high-temperature represents high-cost and high risks, this analysis is focused on the low temperature (LT) version of this control mechanism as shown in Figure 8.

![Figure 8. Turbine Bypass (Low Temperature) Control.](image)

Since the turbine bypass acts on flow and pressure difference (and not on temperature change), the effects of this control action is expected to manifest very quickly as the perturbation of the bypass valve opening and closing action propagates through the system with the speed of sound. That is, it takes a fraction of a second for the turbine and the rest of the system to “feel” the effects of this control action. For these reasons, the turbine bypass control is used in the PDC as a fast and precise control of power output, to fine-tune the net plant output to the required grid demand. The turbine bypass control is automatically used in PDC simulations for cycle power control, unless user specifically disables this option. Therefore, for the simulation of load following with the turbine bypass control, no additional input (or code modifications) were needed, as the code is already setup for automatic action for this control. The only Holos-Quad design-specific possible change would be tuning of the controller’s PID coefficients. However, for this analysis, it was found that no tuning is needed as the default coefficients already provided sufficiently good response for the system.

The detailed PDC results of the load following simulation with the turbine bypass control are shown in Figure 9 over the next several pages. These PDC results are shown as a function of time.
for the 900 seconds total simulation time which included 600 seconds of power reduction followed by 300 seconds of stable operation at zero output (see Table 5). Since all the transients simulated in this chapter are going to be for the same setup – load following with the same external input – the system response would be somewhat similar for all control actions.

As plot in Figure 9a shows, that the external grid demand (purple “W_grid” curve from Table 5) is matched very closely by the calculated net power output to the grid (orange “W_2_grid” curve behind the W_grid curve as shown by the markers, confirming the very good matching of the grid demand). Power reduction to zero is achieved by close to linear decrease in turbine output. This power decrease is the result of the turbine by-pass (TBP) control valve opening action (see “Valves Control Action” plot d), with the valve position calculated by the code from the automatic control action. As a result of this action, the flow rate through the turbine is decreased while flow through the compressors is increased (see “Flow Rates: Turbomachinery” plot k), leading to a reduction in turbine and net power output. In this simulation, no action on the shaft speed control is taken, so both turbine and compressor shafts operate at the design 100% speed.

The action from the reactor power and water flow control is calculated automatically as described in the previous chapter. Note that the “Heat Balance in RHX” plot b for the reactor power shows two curves: one (blue “Q_Rx” curve) is the actual power input to the fuel (calculated by automatic control of reactor-outlet temperature); the other (red “Q_BC” curve) is for the power actually delivered to the Brayton cycle, i.e., heat balance on helium coolant. The difference between those two plots shows the thermal inertia of the reactor structure. For this particular transient, this difference is not very large (compared to other transients presented below) because there is very little temperature change in this control action which is mostly driven by flow change. The reactor heat balance plot b shows that by the end of the transient, the reactor still operates at 64% level, even though the net output from the plant is zero at these conditions. As demonstrated by the comparison with other controls shown in following chapters, this is the highest reactor power level in this transient, signifying the least efficient reactor operation under this control. As a result of the reactor power control action, the reactor-outlet temperature (see “Temperatures: Rx” plot n) is maintained very closely to the 850 °C target. Note that the reactor-inlet temperature does change in this simulation. This is a result of the turbine-outlet temperature changes (see “Temperature: Turbomachinery” plot m), as less energy is being extracted from the working fluid in the turbine. This temperature increase in the turbine outlet also leads to the rising temperatures at the hot end of recuperator (see “Temperatures” Recuperator” plot o).

On the low temperature side of the cycle, the compressor-inlet temperatures are maintained by water flow rate control (see “Temperature: Cooler” plot p, “Temperatures: Intercooler” plot q, and “Cooling Fluid Flow Rate” plot e). As more heat needs to be removed in the coolers, the control action calculates the need to increase water flow rates. Eventually, a limit (of 200% on pump head) is hit for both cooler, after which the flow rate could not be further increased. The temperature plots show that once this happens, the cooler outlet (compressor-inlet) temperatures start to increase. However, since the power level is relatively small at these conditions, the temperature increase is not significant (less than 5 °C). Still, this increase will contribute to further decrease in the cycle efficiency under this control mechanism.

The results in Figure 9 show that this control action has a benign effect on the turbomachinery limits. The compressor stall margin is actually increasing, as flow rates in compressors are
increasing. The choke margins increase in turbine and decrease in compressors, but remain far from the stall-surge and choke limits.

Figure 9 shows that this control action also has some effect on the system pressures (see “Pressures: Turbomachinery” plot r). As the bypass valve opens, a new flow path is opening, reducing hydraulic resistance of the entire loop. As a result, the system pressures converge somewhat, with high pressures decreasing and low pressures increasing. However, the pressure change under this control action is rather small (as it will be shown by comparison with other control options).

Figure 9 also shows the detailed results for reactor and recuperator temperatures and temperature change rates (see plots v through ab). In these plots, internal temperatures at all axial nodes are shown. For the reactor, fuel, matrix, and tube components the temperatures are presented at 10 axial nodes. For the recuperator, the wall temperatures at the 10 axial nodes are shown. This data is used to compute the temperature change rates, in degrees Centigrade per second, for the reactor and recuperator structures. Figure 9 also includes the curves for maximum temperature change rates among all axial nodes for reactor and recuperator. The maximum rates shown will be compared with the results obtained when analyzing other control mechanisms.
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May 10, 2022
Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code
May 10, 2022
Figure 9. PDC Results: Load Following by LT Turbine Bypass.
Once the results for the turbine bypass control are obtained, the automatic action for this control is disabled in the PDC to execute calculations of other control mechanisms. This is done by overwriting the automatic action with a manual action, which maintains the TBP valve in fully closed position all the time.

### 4.2 Turbine Throttling

The turbine throttling control (Figure 10) consists of a single valve located upstream of the turbine. In this report and in the PDC model, the control and its valve are also referred to as turbine inlet or “TIN”. The valve, which is usually fully open at the design conditions, when partially closed restricts the flow rate through the stem and in the turbine, thus reducing the useful work produced by the turbine. This control action also increases pressure resistance of the entire thermal-hydraulic circuit, inducing higher work by the compressors (thus reducing flow rate delivered by the compressors).

As discussed in Chapter 2, in the Holos-Quad configuration there is no actual pipe connecting the reactor and the turbine. Therefore, the turbine throttling would not involve a traditional valve, but rather an adjustable flow restrictor, represented, for example, by movable vanes at the turbine first stage inlet. The detailed design of this mechanism is beyond the scope of this analysis. However, for the PDC simulation, the actual geometry of a valve (or a flow restrictor device) does not matter, as the code only needs to know the open flow area fraction to calculate the “valve resistance” [2]. For the purpose of present discussion and for the code implementation, this control mechanism involves a turbine throttling (inlet) valve.

![Figure 10. Turbine Throttling Control.](image-url)
The PDC has a provision of automatic power control by the turbine throttling valve. This feature is similar to the automatic control for turbine bypass, except the control action is applied to the turbine inlet valve the effect is opposite – closing this valve decreases the output (for TBP, this action results in opening the valve). Therefore, the turbine throttling calculations presented here are executed in an automatic control mode, and no other adjustments to the PDC input were needed (except for switching to power control by turbine inlet valve). A limit of ±10%/s was applied to the maximum valve opening and closing rates.

The results of load following with the turbine throttling valve are shown in Figure 11. The grid demand is matched by the generator output all the way to 0% (plot b). The main control action is demonstrated in the “Valve Control Action” plot f, in the orange “f_op_TIN” curve, which shows open fraction of the turbine inlet (throttle) valve. Notice that the valve action does not start from 100% open. In these calculations, it was determined that closing a fully open valve has very little effect initially on the valve pressure drop and, therefore, on the entire system response. For these reasons, a small pressure drop to the valve (0.004 MPa) was introduced at steady-state conditions to slightly “pre-close” the valve to avoid a delay in the system response. This pressure drop is achieved with valve open to 80% and provides an optimum between steady-state performance (losses) and speed of valve action during load following. Closing the valve from 80% proved to deliver much faster response to matching grid demand.

Figure 11c also shows that the compressor stall margins are a concern for this control mechanism at low loads. The low pressure compressor stall margin is violated at 520 s at loads around 13%. The HPC margin is also small, less than 5%, by the end of the transient. Even though the results in Figure 11c show that the PDC calculations proceeded beyond the compressor stall, while it is not desirable to operate the compressors at these conditions.

Another unique result of load following with the turbine throttling is shown in the “Pressures: Turbomachinery” plot p. When the valve is partially closed, a pressure drop across the valve is induced. Because the valve is located at the turbine inlet where pressure is approximately 7 MPa at design conditions, closing the valve decreased the pressure after the valve (at the turbine inlet), but at the same time increases the pressure before the valve (all the way to the HPC outlet). The results in Figure 11 show that with this control, the high cycle pressure increases to 7.6 MPa (from 7.0 MPa at design). Therefore, if this control is to be implemented, all high-pressure components, including compressor, recuperator, reactor core, and corresponding piping or ducts will need to be designed for pressures exceeding 7 MPa.

The rest of the system response to this control action will be compared to other controls in Section 4.6.
Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code
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**a**
HEAT BALANCE IN RHX

**b**
TURBINE AND COMPRESSORS WORK AND GENERATOR OUTPUT

**c**
COMPRESSORS STALL MARGIN

**d**
TURBINE AND COMPRESSORS CHOKE MARGIN

**e**
SHAFT SPEED

**f**
VALVES CONTROL ACTION

**g**
COOLING FLUID FLOW RATE

**h**
INSTANTANEOUS EFFICIENCIES
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REACTOR TEMPERATURES: FUEL [BY NODE]

REACTOR TEMPERATURES: MATRIX [BY NODE]

REACTOR TEMPERATURES: TUBES [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: FUEL [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: MATRIX [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: TUBES [BY NODE]

REACTOR MAX TEMPERATURE CHANGE RATES

RECUPERATOR TEMPERATURES: WALL [BY NODE]
Figure 11. PDC Results: Load Following by Turbine Throttling.

4.3 Compressor Throttling

The compressor throttling control action is similar to that of turbine throttling, except the valve is located at the low pressure compressor inlet (Figure 12). This location is beneficial for the Holos Quad design for two reasons. First, the valve is located on the low-pressure low-temperature side of the thermodynamic cycle, which reduces the engineering and manufacturing requirements for the valve design. Second, in the Holos-Quad configuration, the line connecting the cooler and the LPC can be represented at least in part by an actual external pipe (as opposed to a duct with complex geometry connecting the reactor and the turbine inside the SPM shell), which further simplify the valve design, procurement, installation, and operation. Therefore, the compressor throttling location shown in Figure 12 would be a preferred choice for this type of control action, provided that the transient results show that it is capable of providing load following capabilities.

Figure 12. Compressor Throttling Control.
For the PDC implementation, the valve was simply moved (compared to previous setup with the turbine throttling) to the compressor inlet pipe. No other changes to the Holos-Quad model was needed, except the reduction of steady-state valve pressure drop to 0.002 MPa, to maintain approximately the same relative pressure drop in the low pressure (3.5 MPa vs 7 MPa) line. The automatic control action methodology remains the same as before.

The load following results with the compressor throttling valve are shown in Figure 13. In general, the results are similar to the turbine throttling control action methodology described in previous paragraphs, and the system demonstrates it follows the grid demand closely. There are however, two important distinctions from the turbine throttling results. First, the compressor throttling results in Figure 13 show noticeably larger margins to compressors stall (plot c), without any operational conditions exceeding these margins. Second, the high cycle pressure does not increase nearly as much as it occurred with the turbine throttling control. The results in Figure 13p show a significant increase in the turbine outlet (LPC inlet) pressure. However, this pressure starts from the low value of 3.5 MPa and never approaches the cycle max pressure of 7 MPa. The turbine inlet (and reactor) pressure still shows some increase in Figure 13p; however this increase is less than 0.1 MPa and does not represent concerns (as these components are usually designed to operate within 10%-15% excess margin). For these reasons, the compressor throttling control mechanism remains the preferred option for the throttling control action for the Holos-Quad design.

Other results in Figure 13 will be discussed in following sections by comparison with other controls mechanisms.
Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code
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TEMPERATURES: RECUPERATOR

TEMPERATURES: COOLER

TEMPERATURES: INTERCOOLER

PRESSURES: TURBOMACHINERY

REACTOR TEMPERATURES: FUEL [BY NODE]

REACTOR TEMPERATURES: MATRIX [BY NODE]

REACTOR TEMPERATURES: TUBES [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: FUEL [BY NODE]
Figure 13. PDC Results: Load Following by Turbine Throttling.

4.4 Inventory Control

The inventory control (Figure 14) works by removing controlled amounts of helium mass from the thermodynamic cycle and (temporarily) storing the removed mass of working fluid in an external tank. As working fluid mass is removed from the cycle, helium density is reduced across the closed-loop. Because the turbomachinery tends to operate at constant volumetric flow rate (to conserve velocity triangles in the blade rows), the reduction in density is compensated in the compressors by a proportional reduction in mass flow rates. For ideal gases, such as helium, the change in density, pressures, and flow rates is very close to linear with mass reduction, meaning
that temperatures tend to remain constant during this control action. The only impact induced by changed helium density from this control methodology results in reduced turbomachinery efficiencies because not all losses scale linearly with density and flow. Also for an ideal gas, since specific heat is (mostly) independent of pressure, change in pressure and density does not affect enthalpy change, resulting in linear reduction in heat and work balances with flow only. As a result, inventory control action tends to maintain relatively high efficiency (of individual components and for the entire thermodynamic cycle) over a wide range of control actions.

To facilitate the helium mass removal from the cycle, the inventory control lines are usually connected to high- and low-pressure pipes for mass removal and return, respectively. As shown in Figure 14, the inventory control tank in the Holos-Quad design is connected to the HPC outlet pipe (highest cycle pressure) and to the cooler inlet (close to LPC inlet, which represents the lowest pressure in the cycle).

For all the benefits represented by adopting the inventory control methodology, one disadvantage is that the control action resulting in opening and closing the inventory control valves in and out the tank (Figure 14), tends to disturb the fluid conditions at either the compressor inlet or the compressor outlet. As such, this control action cannot be executed at a high rate (fast valves actuation), as larger disturbance in pressure conditions in the proximity of the compressor components could lead to a compressor stall. For these reasons, the inventory control is usually not used for fast and precise control of the turbine power (in contrast with fast valve actuation possible for turbine bypass), but rather for slow and long-term control at more or less steady-state conditions. Consequently, in the PDC simulation, inventory control is not intended to provide a power control function for the thermodynamic cycle. Rather, the user input translates into the amount of mass removed from the cycle (added to the tank) as a function of generator load. The assumption is that in fully automatic mode, any remaining difference and fine-tuning of power output will be done by
a faster turbine bypass control. Therefore, the simulation of inventory control for load following described in this analysis is executed in semi-automatic mode. First, a guess of helium mass needed to be removed from the cycle was taken. Then, for each mass change, the actual power output from the plant, as calculated by PDC, is recorded, and the inventory control table is adjusted. The process is repeated until the grid demand is matched for all points in the input in Table 5. Because inventory control action is quasi-linear, this process converged in only 2 or 3 iterations.

It is also important to note that the inventory control system is simulated in the PDC exactly as shown in Figure 14. That is, it only consists of the inventory tank, connecting pipes, and the two inlet and outlet valves. Specifically, no charging pumps or compressors regulating the operations of the working fluid tank are added to the simulation. In this report, this arrangement is referred to as “passive” inventory control, wherein the control action is actuated by leveraging a pressure difference, thus it does not require a dedicated active charging pump/compressor operation. A consequence of this approach is that for the inventory control to be functional, the tank pressure should always be between the high and low cycle pressures. This limitation will be discussed in greater details in following portions of this report. For the simulation of down transient in load following discussed in this chapter, the tank pressure needs to remain below the high cycle pressure. To ensure that, in the calculations presented here, the tank pressure was set to a low value at steady state (0.1 MPa), and the tank volume was artificially set to a very large value (100 m$^3$) so that the tank pressure remains low with mass addition to the tank. The tank temperatures is assumed to be at 30 °C at the design condition. Note that this input only affects the conditions in the tank. The conditions in the cycle and overall cycle response are only defined by the amount of helium mass removed from the cycle, and not where this mass is stored externally.

The results of the PDC calculations for load following with the inventory control are shown in Figure 15. As shown in all plots in Figure 15, it is immediately noticeable that the calculations did not proceed all the way to 900 seconds and were terminated at 480 s, or at 20% load. The limitation is induced by the reactor power, shown in the blue curve in plot a of Figure 15. Because of high efficiency of the inventory control, the PDC results show almost linear reduction in reactor power with time (or plant output). As time elapsed to 480 s, the reactor power is reduced below 0.5 MW, or 10% of the nominal value. This means that by the time the load is reduced to 20%, the reactor power is at least approaching the decay heat level (as described in Section 3.1, PDC does not simulate reactor decay heat), thus the ability to actively control reactor power would be lost (or severely degraded) at these conditions. For these reasons, PDC calculations were stopped at 480 s. Although technically PDC could proceed further, after this time it would produce unrealistic results for the reactor power. Also noticeable from this plot is the significant difference (up to 1 MW) between heat generation in the fuel (blue curve) and heat removed by the helium (red curve), indicating a significant thermal inertia of the reactor core during this aggressive power change.

Figure 15 demonstrates that most of the results show linear behavior during inventory control action. This applies to heat removal in the reactor core by the working fluid helium (plot a), turbine and compressor power (plot b), mass change in the thermodynamic cycle (mass addition by transfer from the thermodynamic cycle to the tank via controlled actuation of the inventory control valves, plot e), flow rates (plot j), and pressures (plot q). All these results are explained by the minimal change in cycle temperatures due to ideal-gas behavior of helium under this control action. As a result, the instantaneous thermodynamic cycle efficiency is calculated to remain above 40% until 420 s, or 30% load.
The results in Figure 15 (plots e, q, and r) also show that in order to reduce the plant output to 20%, significant change in pressures (densities) is needed. The correspondingly required helium mass removal is approximately 3.7 kg, out of 5.6 kg initial helium inventory mass as shown in Table 4. The system pressures are reduced from 7 MPa to less than 2 MPa and from 3.5 MPa to less than 1 MPa on the low cycle pressure side. This significant change in pressures is not a concern for structural integrity, since the pressures are reduced below the design values.

Although the helium temperatures do no change much in this simulation, PDC still calculates noticeable change in reactor temperatures (plots s through u), especially for the fuel as a result of significant power reductions. As shown in plot z, for the recuperator, the results show very little change in the wall temperatures of this component.

The results shown in Figure 15 will be further discussed in Section 4.6 by comparison with other controls mechanisms.
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Figure 15. PDC Results: Load Following by Inventory Control.
4.4.1 Inventory Control Tank Requirements

The results of load following calculations presented in previous section of this report were obtained by assuming a very large inventory tank, to avoid limitations from the tank pressure. This assumption does not affect the cycle results. That is, exactly the same results as summarized by the plots shown in Figure 15 would be obtained with a smaller tank, except for the tank pressure line. In a real plant, the tank volume will be limited, as a result the increase in tank pressure could be significant. By utilizing the results indicating the amount of helium mass to be removed (Figure 15e) from the thermodynamic cycle (to be stored in the tank), calculations were carried out to determine a realistic size of the inventory storage tank. The results of these calculations are presented in Figure 16. The initial tank pressure (prior to helium mass addition) is assumed to be 0.1 MPa, as in the cycle calculations shown in Figure 15. The results of the required tank volume (per SPM) are plotted against the net plant output for several values of maximum tank pressure. The input for these calculations is the additional mass the tank needs to store as a function of plant output (see ”Mass Change in INV Tank” curve in Figure 15e). For these calculations, there is no consideration of how a controlled mass of helium will be delivered from the cycle to the tank; it only factors the relationship between mass, pressure, and volume of the tank itself. The higher the tank pressure is allowed to be (e.g., by dimensional and materials constraints), the smaller the tank volume. The results in Figure 16 show, for example, that if the tank is designed to a maximum pressure of 2 MPa, a tank volume of 1.2 m³ would be required to store the mass removed from the thermodynamic cycle for the control system to actuate from 100% power to satisfy an electric demand corresponding to 20% of the nominal load. Again, those results are per individual SPM. Therefore, four of these tanks would be required to equip the four SPMs forming the Holos-Quad micro-reactor.

![Figure 16. Inventory Tank Volume Requirements.](image)
The helium mass delivery mechanisms engineered to transfer controlled amounts of helium mass from the thermodynamic loop to the tank will be dictated by the inventory control system as a whole system implemented for the Holos-Quad design. Options for actuation of inventory control systems and arrangements are discussed next.

4.4.2 Passive Inventory Control

The inventory control arrangement shown in Figure 14 represents the simplest setup for this control mechanism, with only one tank (which can be formed by coupling multiple industrial tanks to satisfy volume and pressure requirements) and two valves. In this passive approach, the flow to and from the tank is driven by the pressure differential between the high and low pressures in the thermodynamic cycle and tank pressures. For this approach to work, the tank pressure should always be between the high and low cycle pressures. In particular, at steady-state design condition, the tank pressure needs to be above the low pressure of 3.5 MPa and below the high pressure of 7 MPa. Assuming 3.5 MPa for steady-state tank pressure, the previous results from Figure 16 are re-calculated. These new results are shown in Figure 17. This time, the plot shows the tank pressure, with the tank volume as a parametric value. All curves in Figure 17 start from the 3.5 MPa at full load and all the pressures increase as mass is being removed from the cycle and is added to the tank as load decreases. Again, the larger the tank volume the smaller the pressure increase. Also plotted in Figure 17, in dashed lines, are the system high and low pressures from the results in Figure 15q. As the mass is removed from the cycle, both system pressures decrease. Around 60% load, the tank pressure lines go above the high cycle pressure value. Beyond this point, the passive inventory control system would no longer execute its control functions. The exact cross-over point depends on the assumed tank volume, but it is always in the range of 50-65% load for the tank volumes considered in this analysis. If the inventory control volumes tank(s) are located in the same ISO container (to avoid helium lines going outside the container), the total available volume in the container would be around 4 m³, or 1 m³ for each SPM. The results in Figure 17 show that with this volume, the inventory control range is limited to 65%-100% load. But, in order to maintain some margin in pressures to drive the flow into the tank, 70%-100% range is probably the most realistic range for the Holos-Quad design with this passive arrangement.
Figure 17. Inventory Tank Pressures – Passive Approach.

4.4.3 Active Inventory Control

In order to eliminate the limitation represented by the passive inventory control system described in Section 4.4.2, and overcome the thermodynamic cycle high and low pressure difference, an active system arrangement with at least one charging pump/compressor can be adopted. This active control system can be implemented with various layout configurations. Figure 18 shows two of multiple possible implementation options. In Figure 18 left, two dedicated compressors denoted as Pump 1 and Pump 2, one on each symmetrical inlet/outlet of the inventory tank, transfer controlled amounts of helium mass to and from the tank, regardless of the tank and thermodynamic cycle pressures. In Figure 18 right, two tanks, one high pressure (H.P.) and one low pressure (L.P.) are employed, with a charging pump to transfer controlled amounts of helium mass from one tank to another and the thermodynamic cycle. In this approach, the L.P. tank operates always below the high cycle pressure (e.g., at atmospheric pressure or vacuum at the design conditions), while the H.P. tank would always operate at pressures above low cycle pressures (e.g., at 4 MPa at design condition). The pump components would be operated to set and maintain controlled pressures.

Regardless of specific arrangements used to implement the active inventory control system, the main goal is to overcome the pressure limits. Under the active inventory control approach, the inventory control range would not be limited by the tank pressures and it could be used over the entire 20%-100% range obtained in the transient calculations results shown in Figure 15 (with the range limited by the reactor power). The disadvantage of the active control approach, is represented by the requirements for additional equipment (pump(s) and/or tank(s)) which would increase both the capital cost of the plant and the complexity of plant control for load following.

The two approaches for the passive and active inventory control will be re-examined in following sections of this report, when comparing the performance of various load following control mechanisms.
For the operations change from zero to full design require any additional equipment, as the (variable speed) compressor motor the plant output. Unlike any other control presented so far, the compressor speed control does not be used to reduce or increase the helium flow rate in the cycle and lower or increase the plant output. Unlike any other control presented so far, the compressor speed control does not require any additional equipment, as the (variable speed) compressor motor in the Holos-Quad design is already part of the compressor electronic equipment to drive the compressors for speed change from zero to full speed during plant startup, as well as plant full power and load following operations.

In the PDC, the shaft speed (as a table of shaft speed versus time) is an independent user input. For the analysis that follows, the shaft speed was adjusted every 60 seconds to match the required plant output reported in Table 5. The speed control adjustment was executed manually, and the iterative approach to match the grid demand was similar to the inventory control table input.

4.5 Compressor Speed Control

A novel and unique Holos-Quad micro-reactor feature is represented by the mechanical decoupling of the compressor and turbine. This approach eliminates the traditional turbomachinery speed constraints. In the Holos-Quad SPM design, the compressors and turbine shafts are independent and located on the opposite sides of the reactor core and the rotors speeds for these components is regulated via electronic drives. This configuration offers an opportunity to independently vary the compressor shaft-rotor speed. As the compressor speed is electronically controlled by means of a direct-drive high-speed motor, the compressor shaft-rotor variable speed (Figure 19) can be used to reduce or increase the helium flow rate in the cycle and lower or increase the plant output. Unlike any other control presented so far, the compressor speed control does not require any additional equipment, as the (variable speed) compressor motor in the Holos-Quad design is already part of the compressor electronic equipment to drive the compressors for speed change from zero to full speed during plant startup, as well as plant full power and load following operations.

In the PDC, the shaft speed (as a table of shaft speed versus time) is an independent user input. For the analysis that follows, the shaft speed was adjusted every 60 seconds to match the required plant output reported in Table 5. The speed control adjustment was executed manually, and the iterative approach to match the grid demand was similar to the inventory control table input.
described in previous section. All other control mechanisms, except for the external reactor power and water flow controls, were disabled in this simulation.

**Figure 19. Compressor Speed Control.**

The PDC results for load following with the compressor shaft speed control are shown in Figure 20. As shown in plot b the grid demand is matched all the way to zero by reducing the compressor speed to approximately 50% (plot c). This control action reduces the helium flow rate from 4.1 kg/s to 2.5 kg/s (plot h). As shown in plot n as the compressor slows down and the cycle pressures converge. However, due to significant initial coolant inventory in the reactor on the high pressure side, this pressure convergence is not symmetrical and the low pressure changes are noticeably higher than the high pressure. The results in Figure 20e also show a potential problem with the compressor speed control on the compressor operating range, as the LPC stall margin limit is violated at 520 s, i.e., at 13% load.

The compressor speed control action has a significant effect on the cycle temperatures. Reduction of pressure rise across the compressors also decreases the pressure drop across the turbine. As less energy is extracted in the turbine, while the turbine inlet temperature is maintained at 850 °C by the reactor power control, the turbine-outlet temperature increases. As shown in plot i, this increase in turbine-outlet temperature is significant, from 600 °C at design to more than 800 °C by the end of the transient. This temperature change also propagates through the recuperator (plot k) and eventually to the reactor inlet, where the helium temperature increases from below 600 °C to more than 750 °C (plot j). Consequently, there is a significant temperature change for the reactor’s fuel, matrix, and tubes, especially on the cold end (plots o through q). The temperature change resulting from utilizing the compressor speed control as a method to execute load following, will be compared to other controls in Section 4.6.
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May 10, 2022

HEAT BALANCE IN RHX

TURBINE AND COMPRESSORS WORK AND GENERATOR OUTPUT

SHAFT SPEED

COOLING FLUID FLOW RATE

COMPRESSORS STALL MARGIN

TURBINE AND COMPRESSORS CHOKE MARGIN

INSTANTANEOUS EFFICIENCIES

FLOW RATES: TURBOMACHINERY
Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code
May 10, 2022

Figure 20. PDC Results: Load Following by Compressor Speed Control.

4.6 Comparison of Controls

The transient result presented in previous sections showed that all of the control mechanisms analyzed were able to meet the load following goals, with few limitations on the low end of the grid demand. A comparison between the individual control results with data from previous figures will be shown in this section through a series of comparative plots. On each plot, a value of interest (such as cycle efficiency) at partial loads will be compared for different control mechanisms. To simplify the comparisons, the plots in this section will be made against the grid demand (and not time). However, it is important to note that the results presented here are still the transient results obtained with 10%/min grid demand change.

In the series of figures presented in this section of the report, the results for each individual control mechanism shown in Figure 4 will be shown with color coded lines as shown in Figure 21.

Figure 21. Control Mechanisms – Color Coded.
Figure 22 shows the comparison of cycle efficiency at partial loads (grid demands below the maximum nominal reactor power). In this analysis, the cycle efficiency is defined as the ratio of net generator output to heat input to helium in the reactor; for the transient calculations, both these values are taken instantaneously at each time step, thus this metrics is called instantaneous cycle efficiency. Since this parameter demonstrates how efficient the Holos-Quad micro-reactor operates at partial loads, this is one of the most important metrics for control comparison during load following. The results illustrated in Figure 22 clearly show the benefits of the inventory control, compared to other control options. As shown, the cycle efficiency remains above 40% while following the load all the way to 30% load. The turbine bypass, as well as turbine and compressor throttling, are the least efficient control mechanisms, while the compressor speed control shows performance somewhere in between those of the inventory and other controls.

![Figure 22. Comparison of Controls: Cycle Efficiency.](image)

Figure 23 shows the comparison of the different control mechanisms analyzed in terms of reactor (fuel) power. In general, the power is related to cycle efficiency (higher cycle efficiency means lower reactor power for the same net output). However, the plot in Figure 23 also includes the effect of thermal inertia of the reactor structures, as the values were obtained in transients. In this plot, inventory control results in the most significant reactor power variation, and the lowest reactor power at low loads. Compressor speed control requires as much reactor power variation initially, with less variation towards the low loads.
Figure 23. Comparison of Controls: Reactor Power.

Figure 24 compares the various control mechanisms in terms of the thermodynamic cycle pressures. The high cycle pressures are similar for all controls, except for the inventory control case, and remain close to the design value. However, both turbine and compressor throttling control actions result in increasing pressure above 7 MPa, especially for the turbine throttling. Results for the low pressure side of the cycle are significantly different for all controls. While this pressure is reduced with the inventory control, and to some degree, with both throttling actions, the low pressure increases for turbine bypass and compressor speed controls.
Figure 24. Comparison of Controls: Cycle Pressures.

Figure 25 collects the results for flow rates through the reactor and turbine. The general trend for all controls is a reduction in flow rate with load variation. However, the inventory control shows the most reduction in flow rate, almost linear with load. This helps this control mechanism to maintain temperatures and thus high efficiencies at partial loads.
Figure 25. Comparison of Controls: Flow Rates.

Figure 26 shows the results for stall margins for the low- and high-pressure compressors. As shown, there is a significant difference between control actions. Only for the turbine bypass, both margins increase from the steady-state conditions (as the compressor flow increases under this control) and thus do not present challenges for partial load operation. Also, for compressor throttling, both margins are maintained at sufficiently high values, such that this control does not present challenges for compressor operation. For all other controls, one (or both) margins are either reduced significantly or violated. With turbine throttling, the induced high-pressure negatively impacts the compressor margins, while a positive margin (although small) is maintained for the low-pressure compressor. For the compressor speed control, the results are reversed, and the LPC margin limitation is violated at low loads (<20%). For the inventory control, although the two margins show positive values at very low loads, both margins might be violated if the control were extended to zero loads. The results in Figure 26 present the most significant limitation on the range of individual controls for load following obtained in this analysis. A general recommendation from this comparison analysis is that compressor speed, turbine throttling, and inventory controls should not be used below 20% loads.
Figure 26. Comparison of Controls: Compressors Stall Margin.

Figure 27 shows the results for temperatures at the recuperator hot side inlet. Consistent with all other cycle temperatures, the inventory control shows the smallest variation in this temperature, thus demonstrating another benefit of this control mechanism. All other control mechanisms actions result in an increase of the recuperator temperature above the steady-state design value. The most significant increase is calculated for the compressor speed control. The results shown in Figure 27 indicate the recuperator hot side inlet temperature. The temperatures at the recuperator cold side outlet and the reactor inlet are very close and show similar trends.
In addition to the values of interest presented above, a comparison between the results for individual controls is also made in terms of temperature change rates for the reactor structures and recuperator walls. These results are presented in this section. Note that in this case, the results are plotted versus time, rather than load, since these results are intrinsically of transient nature and are better shown on time scale. Figure 28 compares maximum temperature change rates (temperature change per unit time) for the reactor structures: fuel, graphite matrix, and coolant tubes (sleeves). The results in these plots are the maximum values among all axial nodes. Also, the absolute values of these rates are plotted in Figure 28, and thus it does not distinguish between temperature rise and temperature decrease. The absolute value is taken to allow selection of maximum value among all axial nodes. Figure 29 shows similar plots for the maximum rate change among the recuperator wall axial nodes.

Results in Figure 28 demonstrate that the compressor speed control results in consistently higher temperature change rates for the reactor structures. This is consistent with other temperature results shown in previous analyses, for example in Figure 27. Another interesting observation from Figure 28 is that inventory control action results in the highest fuel and matrix temperature change rates, at least initially. This is due to the most aggressive reactor power variation required for this control to maintain high plant efficiency (reactor power changes almost as much as plant output). Therefore, fuel elements structural integrity needs to be analyzed carefully for load following with this control. The coolant tubes change rates, on a contrary, are among the lowest among all control, which is a result on little coolant temperature variation under the inventory control. The reactor-side temperature change rates for other (throttling and bypass) controls are among the smallest ones, which is consistent with the fact that these are the least efficient controls and the reactor power is maintained at relatively high level at partial loads, thus resulting in low thermal transients for the reactor structures.
Figure 28. Comparison of Controls: Reactor Temperature Change Rates.
Figure 29 shows the maximum rate changes for the recuperator channel wall (plates). This plot confirms previous results for this component, where the compressor speed control results in faster thermal transients. So much so that the rates for the compressor speed control, reaching to and exceeding 0.3°C/s are the highest rates recorded among all controls and all considered components (reactor and recuperator) in this analysis. Note that the “jig-saw structure” of the compressor speed results in Figure 29 is due to manual adjustment of the control at 60, 120, etc. seconds. Because the control action was only changed at those points, there is a change in the control action slope, which is reflected in the slope changes in Figure 29. Similar behavior could also be observed for the other control mechanisms with manual adjustments (inventory), although to a smaller degree.

Figure 29 also demonstrates the benefits of the inventory control for thermal transients, as this control shows the smallest temperature variation for the recuperator wall. As with many other results in this section, other throttling and bypass controls show the intermediate results for the recuperator temperature change rates.

![RECUPERATOR WALL MAX TEMPERATURE CHANGE RATES](image)

**Figure 29. Comparison of Controls: Recuperator Temperature Change Rates.**

The results of load following by individual control mechanisms presented in this section are summarized in Table 6, including the rank for efficiency at partial loads, and achieved load range with the factors limiting that range. Table 6 also lists additional concerns and benefits identified for each control during the analysis.
Table 6. Summary of Load Following by Individual Controls
(listed in order of decreasing efficiency at partial loads)

<table>
<thead>
<tr>
<th>Control</th>
<th>Inventory</th>
<th>Compressor Speed</th>
<th>Turbine Throttling</th>
<th>Compressor Throttling</th>
<th>Turbine Bypass</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency rank</td>
<td></td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>Load range</td>
<td>100%-20%</td>
<td>100%-15%</td>
<td>100%-15%</td>
<td>100%-0%</td>
<td>100%-0%</td>
</tr>
<tr>
<td>(100%-65%)*</td>
<td></td>
<td>100%-15%</td>
<td></td>
<td>100%-0%</td>
<td>100%-0%</td>
</tr>
<tr>
<td>Limiting factor(s)</td>
<td>Reactor power, (tank volume)*</td>
<td>LPC stall</td>
<td>HPC stall</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Other concerns</td>
<td>Reactor fuel and matrix temperature variation; Additional external system needed</td>
<td>Recuperator and reactor temperature increase</td>
<td>Implementation; Pressure increase</td>
<td>-</td>
<td>Water flow rates in coolers</td>
</tr>
<tr>
<td>Other benefits</td>
<td>Little temperature variation outside reactor</td>
<td>No extra equipment needed</td>
<td>Might be needed for turbine protection anyway</td>
<td>Simple to implement</td>
<td>Fastest control</td>
</tr>
</tbody>
</table>

*with passive system.

Based on the results in Table 6 as well as all other results in this section, the following conclusions are made:

- **No control shows sufficient performance and/or range to be able to provide load following on its own.** A combination of controls (control strategy) will be required – this is discussed in the next chapter.
- Inventory control is the best choice for load following, obtaining both highest plant efficiency and the smallest temperature variation at partial loads. Thus, *inventory control is recommended as the preferable control at higher loads.*
- Active inventory control is recommended since it extends the range of the preferable control mechanism significantly.
- Compressor speed control provides second-best efficiency, but also results in significant temperature change in and around the recuperator. *This control could be a viable supplement to inventory control at low loads.*
- Compressor throttling, although one of the least efficient controls, is also the only one that does not show any significant limitations. Thus, *compressor throttling control is another viable option for low loads.*
• Although the least efficient control, turbine bypass could still be implemented to provide fast tuning of generator output over the entire range.

• All control actions result in temperature variations in some structures. Therefore, structural analysis will be needed to confirm the Holos-Quad micro-reactor capability for load following and to define the load following frequency and/or allowable number of load following cycles. This analysis is beyond the scope and will be proposed in future work.
5 Control Strategy

The results in previous chapters showed that the preferable control mechanism – inventory – could not be used to provide load following over the entire range. Therefore, if this control is to be used, it needs to be supplemented by other controls. Thus a control strategy, meaning a combination of controls, needs to be developed so that the micro-reactor:

a) is capable of providing control over the entire 0%-100% range, and

b) maximizes the plant efficiency at partial loads.

In previous chapters, it was also shown that the range of the preferred inventory control depends on the inventory control system arrangement. Use of a passive system, where the flow to and from the tank is driven by pressure difference alone, resulted in limiting the inventory control to the range of 100%-70% loads. On the other hand, implementing active means in the inventory control system, such as additional pumps, allowed to extend the control range down to 20%. An economic analysis would be needed to compare the benefits of the extended inventory range, and thus higher efficiencies at middle loads, against the additional cost and operational complexity of active components. The analysis presented in this chapter considers both these options. Still, due to significant benefits of the inventory control over other mechanisms for the plant efficiency, the active control arrangement is currently selected as a preferred option and will be investigated first.

5.1 Option A: With Active Inventory Control

As demonstrated in Section 4.4.3, active inventory control arrangement (Figure 18) eliminates the restrictions represented by the inventory tank volume and therefore would allow using the inventory control from 100% down to 20% loads, where the limitation on reactor power below 20% load is encountered. Because the inventory control showed benefits for both plant efficiency and temperature variations, this control mechanism could be used over the entire range of its applicability, i.e., between 100% and 20% loads. Below 20% load, other controls should be used.

The results in previous section identified both compressor speed and compressor throttling as viable options for low loads. To compare the performance of these controls at low loads and to select the best options, calculations of load following transients were carried out with both these controls, from 20% to 0% load (100%-20% is achieved by inventory control as described in Section 4.4). Figure 30 shows the results of this analysis, in plots indicating control performance versus net plant output. Similar to the plots shown in previous section, the results are shown here in a static fashion; however, these results were obtained in transient simulations with 10%/min grid demand change.

The results in Figure 30 confirm that the compressor speed control provides higher efficiency than the compressor throttling control (plot a). But they also show that implementation of the compressor speed control results in a significant increase in the recuperator (and reactor-inlet) temperatures (plot b). Even for a relatively small change in load, from 20% to 0%, an increase of more than 100 °C, above the design values, is calculated for these temperatures. For all other results in Figure 30, the performance of the compressor speed and compressor throttling controls is similar. Because plant efficiency is not expected to be very important at loads below 20%, the compressor
throttling control for loads below 20% is selected, for the benefits of smaller temperatures increase in the recuperator\(^1\).

Therefore, for this option with an active inventory control, the selected control strategy consists of:

- Adopting inventory control between 100% and 20% loads,
- Adopting compressor throttling between 20% and 0% loads.

Also, the compressor throttling control will be used in automatic mode as secondary control in the 100%-20% range to provide fine-tuning of grid demand matching. This control action is expected to be minimal in that range, since the primary inventory control has already showed good matching.

The transient calculations with this control strategy will be demonstrated in Chapter 6 for combined simulation of both down and up transients.

\(^1\) The Holos-Quad can be coupled to batteries or other electrical loads that require a minimum load corresponding to at least 20% of the micro-reactor nominal power.
Figure 30. Control Strategy Options with Active Inventory Control.
5.2 Option B: With Passive Inventory Control

In passive inventory control arrangement (Section 4.4.2), the inventory tank pressure needs to be maintained between the high and low pressures of the cycle. During the inventory control action, when helium mass is removed from the cycle and is transferred or added to the tank, the cycle pressures decrease, while the tank pressure increases. Consequently, the results in Section 4.4.2 showed that with a realistic tank volume, the inventory control range in this arrangement is limited to 100% to 65%. However, in order to maintain operational stability, a minimum pressure difference between the high cycle pressure and tank pressure needs to be maintained (i.e., pressure difference could not be zero for practical reasons). To maintain this margin, the inventory control is set to operate from 100% to 70% (rather than 65%) loads in this scheme.

Below 70%, the primary choice is again between the compressor speed and the compressor throttling controls. In previous section (with active inventory), the preference was given to the compressor throttling due to expected less stringent requirements on plant efficiencies below 20% load. In this case, however, the plant needs to be operated at loads up to 70%, which is expected to be more frequent and thus cycle and plant efficiency is expected to play a more important role. For these reasons, the compressor speed control for the range below 70% load is selected. Because the compressor speed control action is expected to increase the recuperator temperatures above the nominal design level, it is desirable to limit the resulting temperature rise. The exact value of an allowable (or acceptable) temperature increase during partial load operation will be determined in future work by executing high-fidelity structural analyses. In the analysis presented in this report, a temporary limit of +100 °C increase in the recuperator hot side inlet temperature (to 720 °C) is adopted to define the allowable extension of the compressor speed control. The calculations presented below have showed that this limit is encountered at 30% load. Therefore, the compressor speed control will be used in the 70%-30% load range, and the compressor throttling control will be used below 30%.

Therefore, with the passive inventory control arrangement, the following control strategy is adopted:

- Use inventory control from 100% to 70% loads,
- Use compressor speed control between 70% and 30% loads, and
- Use compressor throttling control below 30% loads.

Similar to previous calculations, compressor throttling control will be used in an automatic mode at all loads (100%-0%) to provide accurate matching of the grid demand.

The results of the PDC transient calculations with this control strategy are shown in Figure 31. Again, even though the transient results are presented in a steady-state fashion, as a function of grid demand, the actual results were obtained in a load following transient with 10%/min grid demand change. Compared to results with an active inventory control in Figure 30, the cycle efficiencies (plot a) are lower for loads below 70% (results above 70% loads are identical to previous ones as the same control mechanism is used). The cycle efficiency reduces even further below 30% loads, where the least efficient control mechanism of compressor throttling is used. As a result of such lower efficiency, the reactor power variation (plots c and d) is noticeable smaller with this control strategy – the reactor power reduces to 2 MW, compared to 0.5-1 MW in Figure 30.
The results in Figure 31b also show that although the recuperator inlet temperature was set to be limited to +100 °C increase with the compressor speed control, using the compressor throttling control below 30% loads only slightly decreased the rate of the temperature increase, and did not eliminate it. As a result, the recuperator inlet temperature increase overall from 620 °C to 770 °C. Again, structural analysis will be needed to see if such temperature increase is acceptable. Alternatively, the recuperator will need to be designed for temperatures higher than nominal design value, say for 800 °C.

Another noticeable results in Figure 31 are the compressor stall margins (plots g and h). Even though the PDC predicts that positive margins to stall will be maintained during the entire transient, those margins are reduced to a small value, 5% or lower, which presents a concern for the compressor operation. More detailed compressor aerodynamic analysis will be needed to confirm safe compressor operation in these regimes.

Overall, comparing the results in Figure 31 to those in Figure 30 confirm the benefits of an active inventory control, in terms of higher efficiency, lower temperature, and larger compressor margins. Therefore, the active inventory control system in the Holos-Quad microreactor design is recommended for implementation.

In Figure 32, the temperature change rates for the recuperator and reactor structures are presented. Similar to results in Chapter 4, only maximum absolute values among all axial nodes are shown in this figure. Consistent with previous results, the maximum rates are recorded for the compressor speed control, although the rates with the compressor throttling control are not significantly smaller with exception for the fuel temperature change rates.
Figure 31. Control Strategy Options with Passive Inventory Control.
Figure 32. Temperature Change Rates with Passive Inventory Control.
6 Load Following Demonstration

In this chapter, the control strategies developed in Chapter 5 will be demonstrated for the full transient simulation. This demonstration will be executed for the full range of load change, from full power to zero. Unlike all previous calculations in this report considering power-down the Holos-Quad micro-reactor, this demonstration will also include load increase from zero back to full power to analyze any transient effects of this power-up regime. Therefore, for the PDC transient calculations in this Chapter, the following external input is imposed for the grid demand (see Figure 33) as follows:

1. Decrease from 100% to 0% at 10%/min rate (over 600 s),
2. Operation at 0% for 5 minutes (600 s – 900 s),
3. Increase from 0% to 100% at 10%/min rate (over 10 minutes, 900 s – 1500 s), and
4. Operation at 100% for 5 minutes (1500 s – 1800 s).

![GRID DEMAND INPUT](image)

**Figure 33. Grid Demand Input For Full Transient Simulation.**

The entire transient considered in this analysis elapses for 30 minutes, and in this time period the reactor will be brought from full power to 0% and then back to full power. For these reasons, the simulated transient will be called “down-and-up” transient, attributing to the power decrease followed by power increase operating conditions. Of these 30 minutes, only 20 minutes will be the actual power change periods; the rest will be stabilization periods at zero and full loads.

The control strategy developed in the previous chapter for the “down” transient will be used without any alterations for the “up” part of the transient as well. For example, with both options, inventory control will be the primary control mechanism for both 100%-to-70% loads in the “down” part and for the 70%-to-100% loads in the “up” part.
Aside from the grid demand input and the specifics of control strategy implementation, no other changes to the PDC input files are introduced here. Both external controls, the reactor power for the reactor-outlet temperature and the water flow for the IC and Cool heat exchangers impacting the compressor-inlet temperatures, will be operating as before, with targets of maintaining 850 °C in the core and 40 °C in the coolers at all times for these temperatures. The fine-tuning of the generator output to match the specified grid demand will be done by the compressor throttling control, even in the regions where it is not a primary control for load following. All control mechanisms not selected for the control strategy, like turbine bypass, will remain inactive in the simulations presented in this chapter.

The results in this chapter will be presented in reversed order, with the passive inventory control option presented first. This is done for a simple reason that it is believed that the results for the preferred Option A with the active inventory control will be the reference results for the Holos-Quad load following analysis summarizing all the work described in this report.

### 6.1 Option B: Passive Inventory Control

The control strategy developed for this option described in Section 5.2 includes the inventory control in the 100%-70% range, the compressor speed control in the 70%-30% range, and the compressor throttling below 30% loads.

The PDC results of down-and-up transient simulation with this control strategy are shown in Figure 34. As shown in plot a, the requested load demand is matched by the net generator output during the entire transient for both the power reduction and power increase phases. This matching is achieved by controls of inventory, compressor shaft speed, and the compressor throttling valve. Also, the external controls of the water flow rate in the IC and Cool heat exchangers and the reactor power remain active to maintain boundary conditions (40 °C and 850 °C, as discussed above). The limits on external controls, turbomachinery margins, and pressures are satisfied confirming the Holos-Quad micro-reactor load following capabilities with this control strategy. In general, the system response is rather symmetric and the results for the up phase are similar to those for the down phase (analyzed in Chapter 5)). Differences related to the transient effects, such as thermal inertia of the reactor structures for reactor power, and inertia of entire system, are best shown in the compressor inlet valve action plot c.

The system pressure plot k shows the effect of the inventory control (initial decrease and final increase), the compressor speed change (pressure convergence and divergence in the intermediate stages), and the compressor throttling valve action (divergence of “turbine-out” and “LPC-in” curves in the middle). The results also confirm that the inventory tank pressures are maintained between high and low cycle pressures at all time, thus retaining the inventory control functionality with passive means.

The temperature plots m through p show that the boundary conditions of the reactor outlet (turbine inlet) and the compressors inlet (coolers outlet) are maintained accurately by the external controls at the 850 °C and 40 °C set temperatures, respectively - the actual change in the reactor-outlet temperature is within 2 °C over the entire transient. At the same time, even though the reactor-core outlet temperature is uniform, there is a significant change in the reactor-core inlet temperature, for the reasons discussed in previous chapters. The temperature change is also showing in the temperatures of other components, all the way down to the Cooler inlet temperatures (see plots n
through k) - these temperatures decrease in the central portions of the plot due to the recuperator behavior at reduced pressures and flow rates.
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TURBINE AND COMPRESSORS WORK AND GENERATOR OUTPUT

HEAT BALANCE IN RX

VALVES CONTROL ACTION

SHAFT SPEED

COOLING FLUID FLOW RATE

INSTANTANEOUS EFFICIENCIES

COMPRESSORS STALL MARGIN

TURBINE AND COMPRESSORS CHOKE MARGIN
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ANL/NSE-21/32
Load Following Analysis of the Holos-Quad 10MWe Micro-Reactor with Plant Dynamics Code
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TEMPERATURES: TURBOMACHINERY

TEMPERATURES: PIPE WALLS

TEMPERATURES: PIPE WALLS

TEMPERATURES: PIPE WALLS

REACTOR TEMPERATURES: FUEL [BY NODE]

REACTOR TEMPERATURES: MATRIX [BY NODE]

REACTOR TEMPERATURES: TUBES [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: FUEL [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: MATRIX [BY NODE]

REACTOR TEMPERATURE CHANGE RATES: TUBES [BY NODE]
Figure 34. PDC Results: Full Range Load Following with Passive Inventory Control.

Figure 34 also shows the detailed temperature profiles inside the reactor (plots s through u) and recuperator (plot aa), along with temperature change rates at several axial locations (plots v through x and z). The reactor temperatures follow the general behavior of increasing inlet temperatures during low power operation. In addition, the fuel temperature decreases at the hot end (reactor-core outlet) due to reduction in reactor power. The recuperator temperatures increase at the hot end (hot side recuperator inlet) and are rather stable on the cold end (hot side recuperator outlet) during the entire transient.

One of the interesting results from this simulation shown in plot b in Figure 34 is that while the reactor power is reduced during the down transient, in the up transient, it does not exceed the steady-state limit, even when the plant output returns to 100% at 1500 s. This results can be better explained by Figure 35, which shows the same data, but in steady-state mode. Figure 35 shows the regions of different control ranges, as well as the results for the down and up phases of the transient. Even though, again, most of the results are symmetrical for the power-up and power-down phases (e.g., cycle pressures are almost identical), there are few noticeable differences. Specifically, since the recuperator temperatures increase during the down transient, they need to decrease during the up transient. Still, due to thermal inertia of the recuperator (and other structures, such as the reactor-core), this decrease is somewhat delayed during the power-up phase. Effectively, the recuperator hot end (and the reactor cold end) structures act like a thermal storage during this transient: they accumulate thermal energy as they are being heated in the power-down transient and then release that stored energy during the power-up phase. As a result, the reactor power does not need to be as high in the power-up phase as it is in the power-down phase. Another consequence of this stored...
energy is clearly shown in Figure 35a as the increase in instantaneous cycle efficiency during the power-up transient. Because with the thermal energy stored in the system the reactor needs to provide less heat to the cycle for the same net output, the instantaneous cycle efficiency is noticeably higher during the power-up operation. In fact, the results show cycle efficiency of approximately 47%, even exceeding the design values during this stage (since this is instantaneous cycle efficiency, higher values do not mean that the plant would be more efficient if operated in this regime for a longer term; this is just a transient effect). The results in Figure 34b show that this stored energy is just sufficient to preclude the reactor power to increase above the nominal values during the power increase phase.

Overall, the results in this section show stable and predictable behavior of the system during load following for both the down and up power transients. All system parameters are calculated to return to full power levels after the transient, confirming that all conservation equations are satisfied in the PDC calculations.
Figure 35. Comparison of Down and Up Phases, Passive Inventory Control.
6.2 Option A: Active Inventory Control

In this section, the same transient defined in Figure 33 in Chapter 6 is simulated with an active inventory control system. As discussed in Section 4.4.3, active inventory control setup allows for extension of the inventory control action down to 20% load. The control strategy defined for this case was described in Section 5.1 and relies on the inventory control between 100% and 20% loads and the compressor throttling control below 20% loads. The compressor throttling control is also used in the 100%-20% range for fine and automatic adjustment of the generator output to match grid demand.

Note that for the simulation presented in this chapter, an active inventory control setup is still simulated with a passive arrangement, as this is currently the only option implemented in the PDC. To eliminate the limitations of the tank volume for active inventory, a relatively large tank volume was used in the current simulation (as in Chapter 4). For the power-up transient, the tank pressure was artificially increased to a high value of 10 MPa (with a code modification) before starting the simulation, to ensure that the tank pressure is set above the cycle pressures. As discussed in previous chapters, the details of how the helium mass is removed and added to the cycle and where it is stored have no effect on the cycle conditions and thus on the results of this transient simulation.

Figure 36 shows the results of the full range load following simulation with the Holos-Quad SPM operated with an active inventory control arrangement. Although some results are similar (or identical) to those shown in Figure 34 (for example, for the generator output, which matches grid demand in both cases), there are still some important differences to point out.

The first difference is that under the active inventory control, the reactor (fuel) power does exceed the steady-state value of 5.5 MW (per SPM) when the load demand returns to 100% at 1500 s (plot b). The increase is rather small, below 106%, so it didn’t trigger the reactor power limit adopted for this simulation (110%). It is also for a short time period, less than 1 minute above 100%, so under these conditions it is not expected to raise concerns for reactor operation or safety. This difference from previous results is explained by the fact that the compressor speed control is not used in this simulation. As previous results have shown, the compressor speed control resulted in the highest increase in the recuperator (and the reactor cold end) temperatures. In the absence of this control action, the amount of thermal energy stored in the system is lower in the current simulation. Therefore, the reactor-core needs to operate at higher levels during the power-up transient phase. This consequence is also reflected in the cycle efficiency plot a in Figure 37, where the difference between power-down and power-up phases is less pronounced.

Other than the “106% core power issue”, all other controls performed as expected as shown in Figure 36. Plot c shows the inventory control inlet valve open during the power-down transient from 100% load to 20% load. During the power-up transient, the inventory outlet valve opens to transfer helium mass back to the cycle. The compressor inlet (throttling) valve is active all the time, but goes through significant adjustment only when inventory control is not used, i.e. below 20% loads. The water flow for both IC and Cool heat exchangers (plot e) is reduced as the reactor power decreases.

The results for turbomachinery limiting conditions show that positive margins are maintained for both the compressor stall (plot g) and turbine choke (plot h) limits. However, the stall margins for both the LPC and HPC are reduced to a small value, less than 5%, so a more detailed aerodynamic analysis of compressor behavior is recommended for future work dedicated to confirm the compressor operability in this regime.
The temperature results in Figure 36 plots m, o, and p confirm that the external controls are able to maintain the target temperatures at the reactor outlet (at 850 °C) and at the compressor inlets (at 40 °C). Other cycle temperatures show some variation, but mostly during the compressor throttling control action at low loads in the middle of the transient. Besides, these variations are smaller than with the passive inventory control simulation analyzed in Section 6.1.

The system pressure results shown in Figure 36 plot q mostly reflect the pressure reduction, with subsequent increase, from the inventory control action. Also, in the middle of the transient, the diverging “Turbine-out” and “LPC-in” curves in plot q demonstrate the pressure drop from the closing compressor throttling valve. For this simulation, the inventory tank pressure results reflect the assumption that the active system was approximated with a passive configuration; the vertical line in plot r shows an artificial increase in the tank pressure before the power-up transient phase.

Figure 36 shows the results for detailed temperature profiles for the reactor-core (plots s through u) and recuperator (plot z) structures, along with the temperature change rates (plots v-y and aa-ab). These change rates will be compared below to the results of simulations with the passive inventory control option from Section 6.1.

Figure 37 compares the results for the power-down and power-up transient cases, as well as for different control mechanisms. Similar to the previous case, the transient is fairly symmetric, except for the reactor temperatures and power (plots c and d), resulting cycle efficiency (plot a), and the recuperator temperatures (plot b). Different from the previous results is that the cycle efficiency only slightly increases above the nominal value in the power-up transient. Again, this is a consequence of less thermal energy stored from the power-down transient with this control strategy.
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i. Helium Mass Addition to Inventory Tank

j. Flow Rates: Turbomachinery

k. Flow Rates: Control

l. Temperatures: Turbomachinery

m. Temperatures: RX

n. Temperatures: Recuperator

o. Temperatures: Cooler

p. Temperatures: Intercooler
Figure 36. PDC Results: Full Range Load Following with Active Inventory Control.
Figure 37. Comparison of Down and Up Phases, Active Inventory Control.
In summary, the results in this section show that the primary goal to demonstrate the load following capabilities of the Holos-Quad micro-reactor at 10%/min rate has been achieved. The control mechanisms to execute load following operations at the 10%/min rate were identified, and a control strategy to successfully achieve these goals has been developed. The results of transient calculations with the PDC show that the load following goals are satisfied without encountering limitations imposed on the calculations.

Table 7 compares the recorded maximum temperature change rates from the two previous simulations in this chapter for the reactor and recuperator structures. Note that this table only lists the maximum and minimum values from the results shown in Figures 34 and 36, and does not include the time-length for which these peaks were sustained. The general result is that the control strategy with passive inventory control results in higher temperatures peaks (as expected, and mostly from the compressor speed control action). Detailed structural analysis is needed to determine whether these rates (plus actual durations from previous figures) are acceptable for the Holos-Quad reactor.

<table>
<thead>
<tr>
<th></th>
<th>Max/Min ΔT/Δt, °C/s</th>
<th>Option A: Active Inventory</th>
<th>Option B: Passive Inventory</th>
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<tr>
<td>Reactor</td>
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</tr>
<tr>
<td>Fuel</td>
<td>+0.18 / -0.15</td>
<td>+0.19 / -0.21</td>
<td></td>
</tr>
<tr>
<td>Matrix</td>
<td>+0.12 / -0.08</td>
<td>+0.18 / -0.25</td>
<td></td>
</tr>
<tr>
<td>Tubes</td>
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<td>+0.20 / -0.28</td>
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</tr>
<tr>
<td>Recuperator</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Wall</td>
<td>+0.40 / -0.35</td>
<td>+0.33 / -0.38</td>
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</table>
7 Supplementary Analyses

The analyses presented in this chapter lay beyond the scope of the Holos-Quad load following analysis. These additional investigations leverage data obtained from the load following transient results presented in this report and provide an overview of how the transient analyses results can be used to investigate reactor kinetics, structural and other Holos-Quad design aspects. All the material in this chapter use the results from full range load following demonstration in Section 6.2 and Figure 36.

7.1 Reactivity Requirements

The load following analysis investigated in this report did not include considerations for the reactor kinetics. As described in Section 3.1, the required reactor power (to maintain reactor-outlet temperature in transients) was calculated purely from the thermal hydraulic perspective. Neutronics transient analysis is required to investigate whether the power control mechanisms and power variations under load following operations could be achieved by varying the core reactivity in combination with actuation of the thermal-hydraulic power control mechanisms envisioned in the Holos-Quad design.

The primary goal of the analysis presented in this section is to calculate the reactivity requirements to achieve the power-up and power-down reactor power variations analyzed in previous Chapters. The input data for these calculations is represented by the reactor power (“Q_RHX_Rx” curve from plot b in Figure 36). This is the total power that is generated in the fuel during the transient. This data is used to calculate:

- Decay heat,
- Fission power,
- Delayed neutron precursor concentrations, and
- Net reactivity.

Most of these calculations follow the point kinetics formulations used in Argonne SAS4A/SASSYS-1 code [6] for analysis of fast reactors (kinetics formulations do not depend on the reactor type and thus are applicable to the Holos-Quad micro-reactor design). The calculations presented in this section are executed with Microsoft Excel spreadsheet and did not require specialized software.

The decay heat is calculated using 23-group standard ANS curve representation for thermal fission of U-235. Each of delayed neutron groups are calculated using the known total power history at each time step. The decay heat is calculated as the sum of these groups. The fission power is calculated as the difference between the total power and the decay heat. The results of these calculations are presented in Figure 38. Again, the top curve for the Total Power is identical to the reactor power in Figure 36, except that all results in in Figure 38 are normalized to the nominal power of 5.5 MW.
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Figure 38. Reactor Power Calculations.

The delayed neutron precursor concentrations are calculated using the point kinetics equations from SAS4A/SASSYS-1 with 6 groups based on kinetics data ($\beta_i$, $\lambda_i$, and $\Lambda$) provided for the HolosGen design (see Table 8 below):

\[
\frac{d\phi}{dt} = \phi(t) \frac{\rho_{\text{net}}(t) - \beta}{\Lambda} + \sum_i \lambda_i C_i(t)
\]

\[
\frac{dC_i}{dt} = \frac{\beta_i \phi(t)}{\Lambda} - \lambda_i C_i(t)
\]

The input for these calculations is the fission power, $\phi(t)$, at each time step from in Figure 38. Using the same point kinetics equations, the net reactivity, $\rho_{\text{net}}$, is also calculated at each time step. Note that in these calculations, the reactor power is known and the point kinetics equations above are solved for reactivity, rather than power. In these reversed solution, there is no need to solve the differential equation for power, since the reactor power and its derivative (change over time step size) are known, and the equation is simply solved for net reactivity at each time step. The results of this step, in terms of the net reactivity, are shown in Figure 39 below in the black “Net” curve. Note that this is the net reactivity the reactor need to have at each time in order to meet the power requirements in Figure 38.

Next, the reactivity calculations are extended to include various reactivity feedbacks. For these calculations, the following reactivity feedbacks are considered:

- Fuel temperature feedback,
- Graphite matrix temperature feedback,
Xenon reactivity feedback.

The net reactivity is calculated [7] as a sum of reactivity feedbacks from fuel, matrix, and xenon, plus the external reactivity (from control rods or drums):

\[
\rho_{\text{net}} = \alpha_f(T_f - T_{f0}) + \alpha_m(T_m - T_{m0}) + \frac{\sigma_X}{\beta}(X - X_0) + \rho_{\text{ext}}
\]

where:

\( \alpha_f, \alpha_m = \) fuel and matrix reactivity coefficients,
\( T_f, T_m = \) fuel and matrix (core-average) temperatures,
\( X = \) xenon concentration,
\( \sigma_X = \) xenon absorption cross-section.

The fuel and matrix reactivity coefficients, as well as flux-averaged xenon absorption cross section, were provided from neutronics calculations. The fuel and matrix temperatures are calculated as the average values from all the axial nodes in the PDC calculations in Figure 36 (since all nodes are equally spaced, no volume averaging was needed). The temperatures at each time step are compared to initial values to provide temperature change for reactivity calculations in the above equation.

Xenon concentration is calculated using the xenon and iodine (a precursor to xenon) mass balances in a reactor [7]:

\[
\frac{dI}{dt} = \gamma_I \frac{n(t)}{n_b} \phi_0 - \lambda_I I \\
\frac{dX}{dt} = \gamma_X \frac{n(t)}{n_b} \phi_0 + \lambda_I I - \lambda_X X - \sigma_X X \frac{n(t)}{n_b} \phi_0
\]

where:

\( \gamma_I, \gamma_X = \) iodine and xenon yields per fission,
\( \lambda_I, \lambda_X = \) iodine and xenon decay constants,
\( \frac{n(t)}{n_b} = \) normalized neutron flux (fission power),
\( \phi_0 = \) average neutron flux magnitude.

Again, the yields and flux were provided from neutronics calculations. Table 8 list the numeric values of the constants used in the calculations. The equations for iodine and xenon concentrations were solved in Excel in a matter similar to one used for delayed neutron precursor concentrations.
Table 8. Data for Reactivity Calculations

<table>
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<th>Parameter</th>
<th>Value</th>
<th>Units</th>
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</thead>
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<tr>
<td>$\alpha_f$</td>
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<td>pcm/K</td>
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<td>$\alpha_m$</td>
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<td>$\sigma_X$</td>
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<table>
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<th>Delayed Neutron Precursor Group</th>
<th>$\beta_i$</th>
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</tr>
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<td>3</td>
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<td>4</td>
<td>2.48E-03</td>
<td>3.03E-01</td>
</tr>
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<td>5</td>
<td>1.02E-03</td>
<td>8.50E-01</td>
</tr>
<tr>
<td>6</td>
<td>4.29E-04</td>
<td>2.86E+00</td>
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</table>

The results of the reactivity calculations are summarized in Figure 39. The black Net line is the one calculated form the reversed point kinetics equations using the known power from PDC load following calculations. This is what net reactivity needs to be in order to provide reactor power calculated by PDC. The brown and green curves are the reactivity feedback for Fuel and Matrix, respectively, which are calculated based on the PDC results for temperature changes and the reactivity coefficients. The purple line is Xenon reactivity. It is clearly seen in Figure 39 that this feedback is very small; it’s about an order-of-magnitude smaller than those for fuel and matrix. This is due to the fact that the entire load following transient is completed in half an hour, while the time constants for iodine and xenon are of the order of 10 hours. Therefore, those concentrations change very little during this relatively short transient.

The red External reactivity curve is the difference between the net reactivity and all the reactivity feedbacks combined. This if what control mechanisms (drums and/or rods) need to provide to achieve the load following goals. As Figure 39 shows, this net reactivity is not very large, with the maximum magnitude of variation less than 1$ (or 640 pcm). This reactivity is much less than the drums reactivity reserve (6,272 pcm [8]), such that the reactor power change requirements calculated by PDC in the previous chapter can easily be provided by the Holos-Quad reactivity control mechanisms.
Figure 39. Reactivity Calculations.

### 7.2 SPM Control Options

The calculations presented in this report were executed for a single Holos-Quad Subcritical Power Module (SPM), under the assumption that four coupled SPMs are independent and identical.

At the same time, identical (or uniform) SPM control is not the only option for Holos-Quad design. Because the concept has four parallel units, there is a theoretical possibility to operate the reactor in a modular fashion. That is, instead of reducing power uniformly in all SPMs, it can be done in a one-by-one power mode. For example, to get to 75% power, reduce power (to zero) in one module only, while keeping all others at 100%. Then, shutdown the active SPM, switch to the next one for power control, and so on.

This control option is more complicated than the uniform approach analyzed previously. However, for the additional analysis presented in this section, a preliminary investigation of modular approach to Holos-Quad operation module-by-module or “modular control approach” mode was carried out to explore whether there could be potential benefits of such reactor operation. The goal here is to determine which mode is more effective, i.e. provides higher efficiency at partial loads: uniform or modular.

To investigate the potential of the modular control approach, the following assumptions have been made to assess potential benefits of modular Holos-Quad micro-reactor operation for high-performance load following.

- Reactor modules can be operated independently, up to complete shutdown of one module while keeping others at 100%. This assumption, will need confirmation from neutronics calculations. Although an earlier indication suggests that “neutronically” it would not be possible to completely shut down one SPM (one-quarter of the Holos-Quad reactor) while
the remaining SPM operate at or close to full power. This operational configuration, if neutronically feasible would be subject to reactivity variation as a result of fuel burn-up, therefore, it might be feasible for a portion of the Equivalent Full Power Years (EFPYs) the coupled Holos-Quad core features when all SPMs are operated identically. Therefore, in this study burn-up considerations are neglected.

• The shutdown module produces zero power and electricity. That is, decay heat and any other power production in that module is ignored, as well as any power requirement to keep it in hot shutdown mode. Most likely core portions of the shutdown SPM will continue to fission (and produce power above decay heat) as in the Holos-Quad Gen 2+ the SPMs are stationary with respect to one another, therefore neutrons leaking from the full-powered SPMs will induce power generation in the quarter core maintained in shut-down conditions by actuation of the reactivity control mechanisms of this individual SPM.

• SPM power conversion cycles are fully independent of one another, e.g. no limits on compressor speeds between the units, no influence of inventory control system action, even when that system is shared between the units, etc. The Brayton cycle components forming each SPM are independent with independent control systems, generators and compressors’ motors.

• SPMs are identical, meaning that the load following performance in one SPM obtained in all previous analysis is the same for all other active SPMs.

• All calculations are executed in steady-state mode, meaning that this analysis does not consider transient effects for the modular control operation.

The calculations presented in this section are executed with Microsoft Excel using the results from the active inventory control strategy calculation in Section 4.4.3 for power-down transient only (the control strategy for the power-up transient simulation in Chapter 6 was the same). There were no new transient calculations to support this analysis. From the simulation described in Sections 5.1 and 6.2 the only results used in the modular control analysis is the cycle efficiency at partial load, whose results are shown in Figure 40 and will be used as input data for this analysis. Note that the efficiencies in Figure 40 represent the reference case for the current analysis, with uniform load following for all four SPMs, because in this mode the SPM and plant efficiencies are identical. For further comparison with other options, the average cycle efficiency for the entire load range (the area below the curve in Figure 40) is 38.2% for the uniform control. Also note in Figure 40 a significant change in cycle efficiency trends when the switch is made from the inventory control to the compressor throttling at 20% load.
To calculate the plant efficiency in a modular approach, a load schedule for each of four SPMs has been developed. In this schedule, power in a single SPM is reduced linearly, while other SPMs are kept at 100%. This schedule is demonstrated in the top plot of Figure 41. For the total plant outputs between 100% and 75%, power in the first SPM is reduced from 100% to 0%. Then, that SPM is shutdown, and the power in the second SPM is reduced to provide total plant output between 75% and 50%, and so on.

Based on this load schedule, the net plant efficiency is calculated at each load level. In this case, the efficiency for an “active” SPM (SPM #1 for 100%-75% range, for example) is taken from Figure 40, for the SPM load level. All other (non-active) SPMs are either maintained at 100% (44.6% efficiency at design level) or not included in calculations if they are shut down at 0%. For example, at 80% grid output in Figure 41, SPM #1 operates at 20% with efficiency of 37%, while SPMs 2-4 are at 100% with 44.6% efficiency. Because all SPMs are identical and produce the same power at 100%, the net plant efficiency is a simple average among the non-shutdown SPMs. For the example above, the average efficiency is 42.7% at 80% load. The full results of these calculations at every load level are shown in the bottom plot Figure 41. Note the jumps in net plant efficiency at 75%, 50%, and 25% loads, as the active (and less efficient) SPMs are shut down and are excluded from further calculations. Figure 41 also includes the results for the uniform control, which are identical to those in Figure 40.

The results in Figure 41 show that the modular control mode is more efficient than the uniform approach at some loads, but also it is less efficient at other loads. Overall, though, the average efficiency over the entire range for the modular approach is 41.0%, which is higher than 38.2% for the uniform approach. Therefore, the results at this stage confirm that the modular approach in Figure 41 is beneficial for the purposes of increasing the plant efficiency at partial loads.

At the same time, the results in Figure 41 show that the modular approach is less efficient at some points, notably at 75%, 50%, and 25% loads, before the active SPMs are shutdown. This is due to significant efficiency decrease for SPM loads below 20% in Figure 40, when the compressor throttling control is utilized. To limit the decrease in cycle efficiency at lower loads, an alternative approach to modular schedule is proposed.
In this option, shown in Figure 42, power in each active SPM is reduced to 20% only and operation below 20% is avoided. In Figure 42, plant operation between 100% and 80% is identical to the previous results in Figure 41, and SPM #1 is brought from 100% to 20% loads. However, that reduction in power does not continue all the way to 0%. Instead, once SPM #1 power reaches 20%, the power in SPM #2 starts to decrease. When that power reaches 80%, the first two SPMs collectively produce a power of one full SPM (20%+80%). Since the other two SPMs operate at 100%, the average plant output at this point is equivalent to 75%. Once that 75% level is reached, the first SPM is shut down, while the power in SPM #2 is brought back to 100%. This schedule allows to avoid operation of an active SPM below 20%. The same approach is repeated at 50% and 25% grid levels.

Because the decrease in SPM cycle efficiency between 100% and 80% is much less than that between 20% and 0%, the results in Figure 42 show that with this optimized SPM control, the net plant efficiency at 7%, 50%, and 25% loads does not decrease as much as in previous case in Figure 41. The net plant efficiency is close to the uniform approach, but jumps back to the design efficiency once an active SPM is disconnected. As a result, the net plant efficiency for this optimized modular
operation remains above that of the uniform approach for the majority of the loads. Combined with the benefits below 25% load, where a single SPM operates at 100%-0% loads (compared to all SPMs below 25% loads in uniform scheme), the average cycle efficiency for the entire range for this approach is calculated to be 42.0%, thus demonstrating benefits of this control schedule compared to both uniform and linear modular approaches investigated. Moreover, implementing this optimized SPM schedule avoids drops in cycle efficiencies and allows for operation with high plant efficiency over a very wide range of loads. For example, the net plant efficiency remains above 40% for loads from 100% all the way to 8%.

Note that the SPM schedule in Figure 42 is only one example of how the schedule could be optimized. It might be possible to increase plant efficiency even further, by using three SPMs simultaneously, for example, to avoid any SPM operation below 30% loads, and so on.

Figure 42. Improved Option for Modular Load Following.
Under this preliminary analysis of the Holos-Quad micro-reactor operated with modular-control, the most efficient approach might involve a combination of uniform and modular control approaches. This hybrid approach (shown in Figure 43), takes advantage of higher cycle efficiencies at higher loads, a uniform reduction in power for all four SPMs would be implemented for loads between 100% and 75%. In this range, the results would be identical to uniform ones shown in Figure 40. Since no SPM is operating at a particularly lower power, there would be no efficiency penalty in these ranges, such as those observed in Figure 40 and Figure 41. Once 75% power level is reached, however, one SPM would be shutdown, while the remaining three would be brought back to full power. This part is identical to the original modular approach shown in Figure 41. From that point onward, the remaining three SPMs would uniformly reduce power until 50% is reached and the second SPM is shutdown. And so on. Note in Figure 43 that in this scheme the rate of uniform power change increases as less units remain to contribute to total power. For example, the three units in the 75%-50% range need to provide more power variation than for four units in the 100%-75% range. The resulting plant efficiency plots in Figure 43 clearly show the benefits of this hybrid approach for the net plant efficiency, as the new “hybrid” curve stays above the uniform curve for all loads. As a result, **the average plant efficiency over the entire range is 42.6%**, while operated under load following which is the highest value recorded in this section, and is not much lower than the design point of 44.6%.

The results of the investigation presented in this section confirm the potential benefits of modular (or hybrid) Holos-Quad SPM control compared to a uniform control scheme (all SPMs controlled identically). Therefore, as the long-term benefits of operating at high efficiency while satisfying load following requirements translate into a lower fuel consumption (which is a strong cost-driver) it is recommended for future work to continue investigation of the Holos-Quad modular control schedules. Still, it is reminded that these results are only preliminary and under simplifying assumptions, one of which is that the SPMs can be controlled neutronically independent of each other.

It is also important to point out that the results in this section were obtained under a steady-state approach only, and no transient effects were analyzed. Specifically to the load following goals of 10%/min rate, the modular approach can present a challenge in a transient sense. Because only one active module is used at any time, in order to achieve 10%/min rate for the entire plant, the rate for individual SPMs need to be four times as high, at 40%/min. For example, reducing the overall plant output from 100% to 80% in a uniform approach would be done in 2 minutes. For the two modular schemes considered here, going to 80% net output would require SPM #1 power reduction from 100% to 20%. If it is to be done during the same 2 minutes time frame, that would effectively require power maneuvering in this SPM at 80%/2min = 40%/min rate. Although transient calculations need to be carried out in order to see if it is possible to execute such a control scheme, the results obtained so far with 10%/min rate already show significant reactor power variation, such that 40%/min rate is going to represent several challenges. In addition, the optimized modular approach in Figure 42 as well as the hybrid approach in Figure 43 require an instantaneous change in SPM power. Obviously, this could not be achieved in a real transient simulation. Therefore, it is concluded that the modular approach to SPM power control can be implemented only in a true steady-state fashion, with very slow change in power output. Still, the potential benefits of these approaches to maximize the plant efficiency demonstrated in this section warrant further investigation of these options for the Holos-Quad design.
Figure 43. Hybrid Modular-Uniform Load Following.
8 Summary and Conclusions

The purpose of the work described in this document was to investigate the load following capabilities of the Holos-Quad micro-reactor design developed by HolosGen LLC. In the project goals, load following is defined as an ability of the reactor to change electrical power output from full design power all the way to zero power at 10%/min rate.

The calculations presented in this report were carried out with Argonne’s Plant Dynamics Code (PDC). The code was modified during the project to be able to model the Holos-Quad Subcritical Power Modules (SPMs), including addition of helium properties and a reactor module. In the initial analysis, PDC was used to establish Holos-Quad Gen 2+ Brayton cycle design and optimization at full power conditions (design point). The PDC transient capabilities were then used for simulation of load following transients.

In order to investigate the load following and control of the Holos-Quad micro-reactor design, the control mechanisms were first identified for the plant and were simulated in PDC. A wide range of control mechanisms were considered, including turbine and compressor throttling control, turbine bypass control, inventory control, and compressor speed control. The latter is unique to the Holos-Quad design, as the compressor is mechanically decoupled and independent of the turbine, such that their speeds can be changed independently of one another. In the Holos-Quad SPMs design the compressor is driven by a direct-drive high-speed motor (no gear box), as a result its operation is independent of the grid frequency. However, the PDC compressor and turbine modules deal with shafts whose rotational speed is locked to multiples of the grid frequencies. In addition to the controls for power output, external controls for reactor power and control for water flow rate in the cooler and intercooler heat exchangers were simulated for active adjustment of the reactor outlet and compressor inlet temperatures, respectively. All control mechanisms are simulated in PDC with proportional, integral, and differential (PID) controllers. The PID coefficients were optimized for the Holos-Quad design-specific features when it was necessary. The control action for load following was either automatic or manual, depending on how each particular control mechanism was modeled in the PDC.

As a first step in the analysis, load following cases were simulated with each individual control mechanism. These results showed that, in general, all control mechanisms were capable of providing the Holos-Quad design with the load following capabilities at the target 10%/min power rate. Some issues, such as approaching compressor stall or reducing reactor power to a decay heat level, were identified at low loads (20% or below) for some of the control mechanisms analyzed. Then, the results of the transient calculations for each control mechanism were compared for the figures of merit, such as plant efficiency at reduced loads, temperature increase in recuperator, increase in cycle pressures, and so on. Comparison analysis showed that the inventory control is a preferable control mechanism for load following operations of the Holos-Quad micro-reactor, providing both the highest efficiency and smallest temperature change at partial loads. The second-most efficient control was the compressor speed, although it also resulted in the highest temperature increase in the cycle. The throttling and bypass controls produced the least efficient control methods, with a moderate change in cycle temperatures. Among the throttling and bypass control mechanisms, the compressor throttling control was preferable for efficiency, temperature, and pressure variations.
Although the inventory control showed clear benefits in the control comparison results, its operability range was limited. Two variations of the inventory control system were considered: passive and active inventory control with the utilization of a working fluid inventory tank. In a simple passive inventory control arrangement, flow to and from the inventory tank is driven by a pressure difference. However, during the inventory control action the tank pressure increases while the system pressures decrease, such that the range of this control was limited to only 100%-65% loads. The tank pressure limitation are avoided by utilization of an active inventory control arrangement, where flow is actively driven by one or more dedicated pumps (or compressors). Although the active inventory control system is more complicated to build and operate, it allows to extend the range of the most efficient inventory control down to 20% load. In this case, the lower 20% load limit was limited by the reactor power, which was reduced to the decay heat level in the simulation. Both passive and active inventory control options were investigated in this analysis.

Based on the results of control mechanisms comparison, options for the control strategy for the Holos-Quad micro-reactor operated under load following were developed. With an active inventory control system, the strategy uses inventory control from 100% to 20% loads, while the compressor throttling control mechanism is utilized for loads below 20%. With a passive inventory system, the strategy relies on the inventory control between 100% and 70% loads, compressor speed control between 70% and 30% loads, and compressor throttling control below 30%.

The developed control strategies were tested via PDC under full range loads with power down- and-up transient simulations. In these transients, the grid demand was simulated to decrease from 100% to zero (power-down), followed by an increase from 0% back to 100% (power-up). Both power-down and -up ramps were at 10%/min rates. The results for both options, with active and passive inventory controls and corresponding control strategies, showed that the Holos-Quad reactor was able to follow the grid demand. Therefore, these results achieved the goals of demonstrating the load following capabilities of the Holos-Quad micro-reactor over the entire load range at the change rate of 10%/min (~1MW/min).

In addition to the main analysis, two supplementary studies were carried out based on the results of the load following transient simulation. The first supplementary analysis estimated the core reactivity requirements to satisfy the thermal power variations required during the load following transients. A preliminary conclusion from this supplementary study is that the Holos-Quad core reactivity control mechanisms provide sufficient margins to satisfy load following requirements. The second supplementary study investigated options for independent control of four identical Subcritical Power Modules (SPM) and their behavior under load following. The analyses reported in the main body of this report assumed uniform (all at once - SPMs operated simultaneously) control for identical design and behavior of all SPMs. In this second supplementary analysis, options of modular control (one-by-one) involving individual SPM power maneuvering were investigated. The modular control approach has a potential to provide higher net plant efficiency at partial loads. A hybrid uniform-and-modular control approach further maximized plant efficiency under load following. However, this second supplementary study assumed that the Holos-Quad SPMs can neutronically be controlled independently of each other. These supplementary analyses are only preliminary and additional more detailed analysis are recommended in future work.
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