Preconceptual Engineering Services for the

Next Generation Nuclear Plant with Hydrogen Production

Preliminary Assessment of the GT-MHR Power Conversion System

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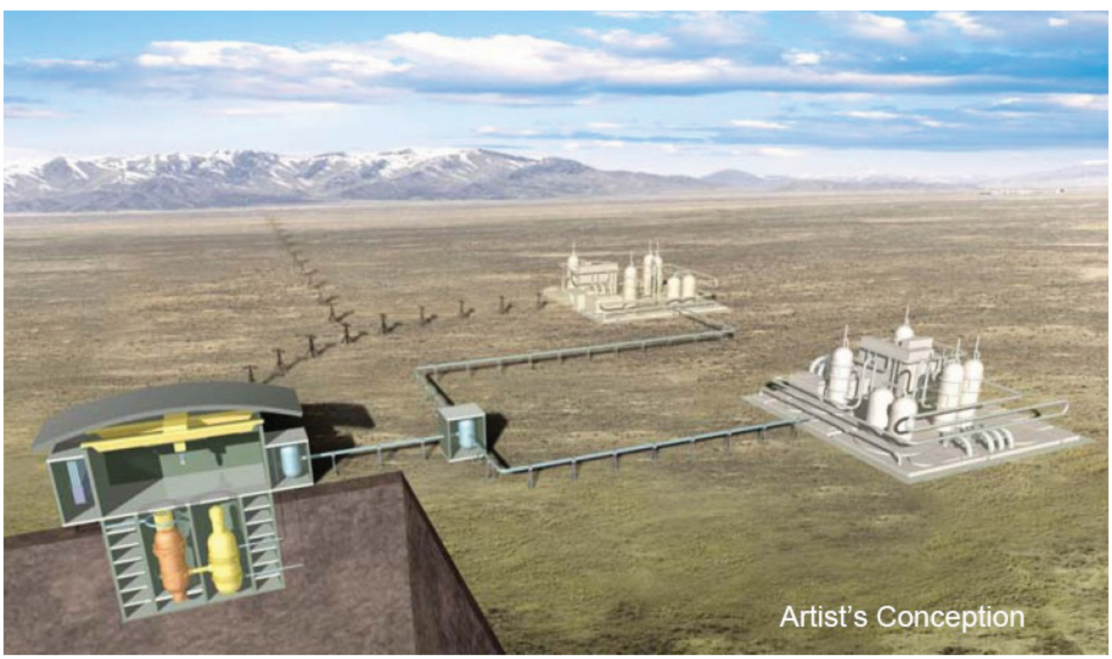
10 July 2007



Record of Change

Date	Version Number	Reason For Change
10 July 2007	1.0	Issued as part of first phase of preconceptual work

Executive Summary

As part of the initial design phase of the NGNP Project, the Battelle Energy Alliance, operator of the Idaho National Laboratory, contracted with three modular helium reactor technology development teams, including a team led by General Atomics, to provide preconceptual engineering services.

The GA team consists of Washington Group International, Toshiba Corporation, Fuji Electric Systems, the Korean Atomic Energy Research Institute, OKB Mechanical Design (OKBM) and Rolls-Royce. This report is published in support of the General Atomics NGNP Preconceptual Design Studies Report, which is the principal summary deliverable for this phase of work.

The Rolls-Royce study had two main objectives. These were:

- Assess the current preferred GA/OKBM concept for Power Conversion System (PCS) and suggest improvements.
- Develop the concept into improved 'Rolls-Royce' PCS concept design.

Both of these objectives have been achieved, by an integrated team pulled together from all four parts of Rolls-Royce (Civil Aerospace, Defence Aerospace, Marine, and Energy).

In the assessment of the GA/OKBM reference PCS design, Rolls-Royce concluded that the concept was elegant and achievable, but carried significant key risks. These were identified as follows:

- Recuperator life and cost considered very high risk.
- Active electro-magnetic bearing/catcher bearing requirements are outside of current world experience - high risk.
- Cost of power electronics required for 4400rpm/286MW generator anticipated to be large (~$50M) - high risk commercially.

An alternative recuperator design has been proposed by Rolls-Royce which would be much more compact and less expensive. It is a cross-corrugated design that is made by the diffusion bonding of corrugated stainless-steel plates. Our analysis shows that moving to this alternative design would meet the required performance of this component for the cycle. Even with this alternative design, the life of the recuperator for the GT-MHR concept should still be considered a moderately high risk.

An alternative concept has been worked up, that addresses some of the key risks identified in the reference concept. This is a combined cycle, consisting of a 66MWt gas turbine generator with the remainder of the power taken by a conventional steam cycle. The key features of this concept are:

- The recuperator is no longer required. A steam generator would be required, but this is considered much lower risk.
- EM bearing risks are reduced by reducing generator weight from 35t to around 10t, and turbomachinery shaft weight from 32t to around 10t.
- Power electronics costs are reduced (since generator is reduced from 300MW to 66MW in gas turbine part).

- Plant efficiency is increased, compared with the GT-MHR Brayton cycle.

- Steam turbines and steam cycle electrical generators are commercial off-the-shelf items - low cost and low risk.

The combined cycle alternative could be expected to have lower plant costs because much of the steam machinery is commercial off the shelf, but this saving would be offset by requiring a bigger containment building and the extra maintenance burden of the steam cycle parts.

In conclusion, both the GT-MHR reference design PCS and an alternative combined cycle PCS have been worked up in the pre-concept study. They both have advantages as described below:

<u>Combined Cycle Advantages</u>

- Reduced EM bearing risk.

- No recuperator.

- Steam equipment (excluding steam generator) would be commercial off-the-shelf.

- Total equipment costs should be lower.

- Steam generator can exploit e.g. AGR experience.

- Flexibility to have process steam instead of electricity from steam plant.

<u>OKBM/GA Reference Cycle Advantages</u>

- More compact - smaller equipment footprint, both inside reactor building and outside.

- More elegant, simpler cycle - less complexity.

It is recommended that further work be undertaken, in the next phase, to decide which cycle should be selected for the NGNP application.

During the PCS preconcept study, other areas have emerged as requiring study in any follow on programme. Some of the more significant are:

- Transient performance. The start-up and transient behaviour of the PCS needs to be better understood so that the transient requirements for the components can be properly assessed. This will require a transient performance model to be constructed for the cycles and is a significant undertaking.

- A study of the control system for the PCS needs to be made. This system's behaviour is intimately bound up with the PCS's transient requirements.

- Further refinement to turbine designs to increase confidence of achieving 60 000hours creep life with uncooled turbine blades at 850ºC for both the reference cycle and the combined cycle.

- Further exploration of the implications of 950°C operation, particularly cost/complexity/performance trade-offs (including blade cooling and thermal barrier coatings).

- A more thorough investigation into EM bearing capabilities and alternative technologies is required because the EM bearings are such a key feature of both concepts.

Contents

List of Figures

List of Tables

1 Introduction

1.1 Project Background

The Energy Policy Act of 2005 required the Secretary of the US Department of Energy (DOE) to establish the Next Generation Nuclear Plant (NGNP) Project. In accordance with the Energy Policy Act, the NGNP Project consists of the research, development, design, construction, and operation of a prototype plant that (1) includes a nuclear reactor based on the research and development activities supported by the Generation IV Nuclear Energy Systems initiative, and (2) shall be used to generate electricity, to produce hydrogen, or to both generate electricity and produce hydrogen. The NGNP Project supports both the national need to develop safe, clean, economical nuclear energy and the National Hydrogen Fuel Initiative, which has the goal of establishing greenhouse-gas-free technologies for the production of hydrogen. The DOE has selected the helium-cooled Very High Temperature Reactor (VHTR) as the reactor concept to be used for the NGNP because it is the only near-term Generation IV concept that has the capability to provide process heat at high-enough temperatures for highly efficient production of hydrogen. The DOE has also selected the Idaho National Laboratory (INL), the DOE's lead national laboratory for nuclear energy research, to lead the development of the NGNP under the direction of the DOE.

As defined in the NGNP Preliminary Project Management Plan (PPMP, Reference 1), the NGNP Project objectives that support the NGNP mission and DOE's vision are as follows:

- Develop and implement the technologies important to achieving the functional performance and design requirements determined through close collaboration with commercial industry end-users

- Demonstrate the basis for commercialization of the nuclear system, the hydrogen production facility, and the power conversion concept. An essential part of the prototype operations will be demonstrating that the requisite reliability and capacity factor can be achieved over an extended period of operation.

- Establish the basis for licensing the commercial version of the NGNP by the Nuclear Regulatory Commission. This will be achieved in major part through licensing of the prototype by NRC, and by initiating the process for certification of the nuclear system design.

- Foster rebuilding of the US nuclear industrial infrastructure and contributing to making the US industry self-sufficient for its nuclear energy production needs

As part of the initial design phase of the NGNP Project, the Battelle Energy Alliance, operator of the INL, contracted with three modular helium reactor (MHR) technology development teams, including a team led by General Atomics (GA), to provide preconceptual engineering services. The GA team consists of Washington Group International, Toshiba Corporation, Fuji Electric Systems, the Korean Atomic Energy Research Institute, OKB Mechanical Design (OKBM) and Rolls-Royce.

1.2 Scope of Rolls-Royce Study

Rolls-Royce has conducted a pre-concept study of the GT-MHR power conversion system (PCS) for NGNP. The Rolls-Royce study had two main objectives. These were:

- Assess the current preferred GA/OKBM concept for PCS and suggest improvements.

- Develop the concept into an improved 'Rolls-Royce' PCS concept design.

This report is published in support of the GA NGNP Preconceptual Design Studies Report (Reference 2), which is the principal summary deliverable for this phase of work.

Both of these objectives have been achieved, by an integrated team pulled together from all four parts of Rolls-Royce (Civil Aerospace, Defence Aerospace, Marine and Energy). The integrated team is shown in Figure 1.

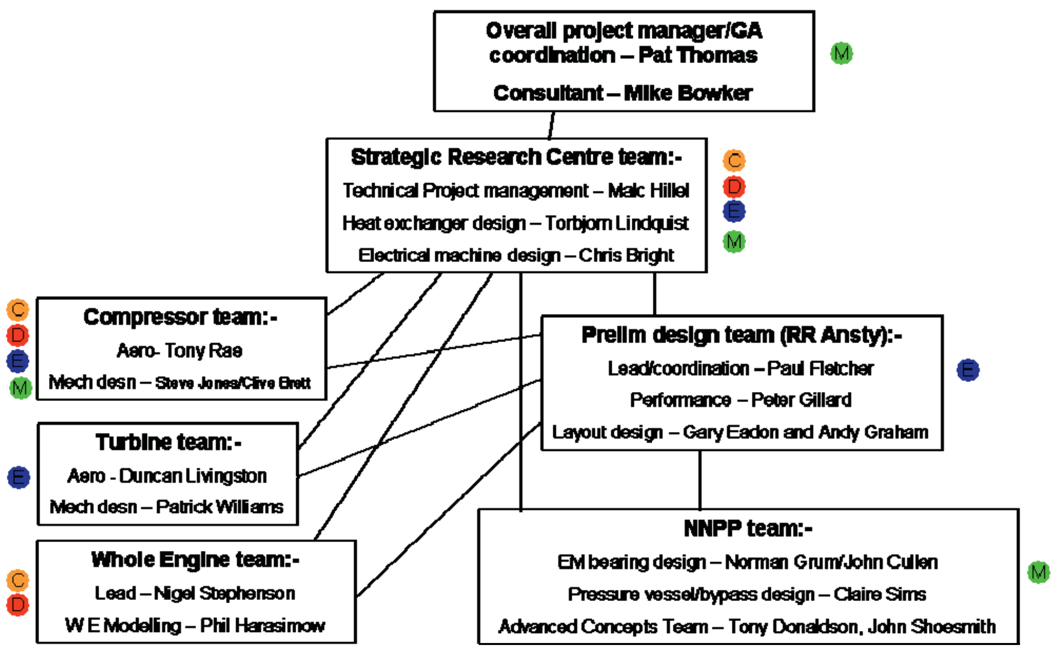


Figure 1 Rolls-Royce Integrated Team to Work on Power Conversion System

In the assessment of the GA/OKBM reference PCS design, Rolls-Royce concluded that the concept was elegant and achievable, but carried significant key risks. These were identified as follows:

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- Electromagnetic (EM) and catcher bearing requirements are outside of current world experience - high risk.

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- Power electronics costs are reduced (since generator is reduced from 300MW to 66MW in gas turbine part).

- Plant efficiency is increased, compared with the GT-MHR Brayton cycle.

- Steam turbines and steam cycle electrical generators are commercial off-the-shelf (COTS) items - low cost and low risk.

The combined cycle alternative could be expected to have lower plant costs because much of the steam machinery is commercial off the shelf, but this saving would be offset by requiring a bigger containment building and the extra maintenance burden of the steam cycle parts.

1.3 Overview of Reactor System

1.3.1 *Overall NGNP Plant Design*

Figure 2 shows a schematic of the NGNP preconceptual plant design. The nuclear heat source for the NGNP consists of a single 600MWt prismatic-block MHR module with two primary coolant loops for transport of the high-temperature helium exiting the reactor core to a direct cycle power conversion system (PCS) and to an intermediate heat exchanger (IHX). The reactor design is essentially the same as for the General Atomics Gas Turbine Modular Helium Reactor (GT-MHR), shown in Figure 3, but includes the additional primary coolant loop to transport heat to the IHX and other modifications to allow operation with a reactor outlet helium temperature of 950°C (vs. 850°C for the GT-MHR). The IHX transfers a nominal 65MW of thermal energy to the secondary heat transport loop, which transports the heat energy to both a SI-based hydrogen production facility (60MWt) and an HTE-based hydrogen production facility (~4MWt). The GT-MHR power conversion system (PCS) is retained with a full rating of 600MWt, to enable electricity-only operation as well as combined hydrogen production and electricity generation.

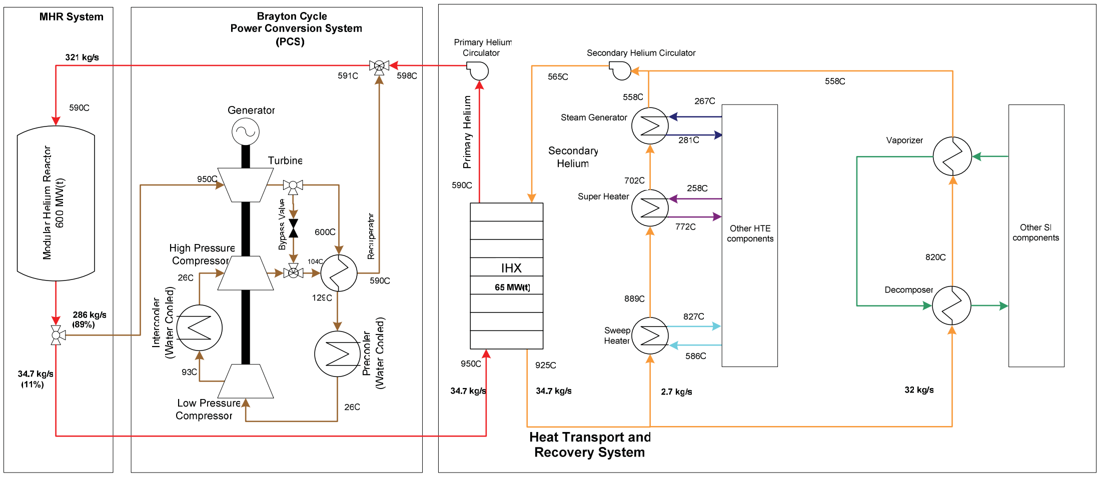


Figure 2 Schematic of NGNP Plant

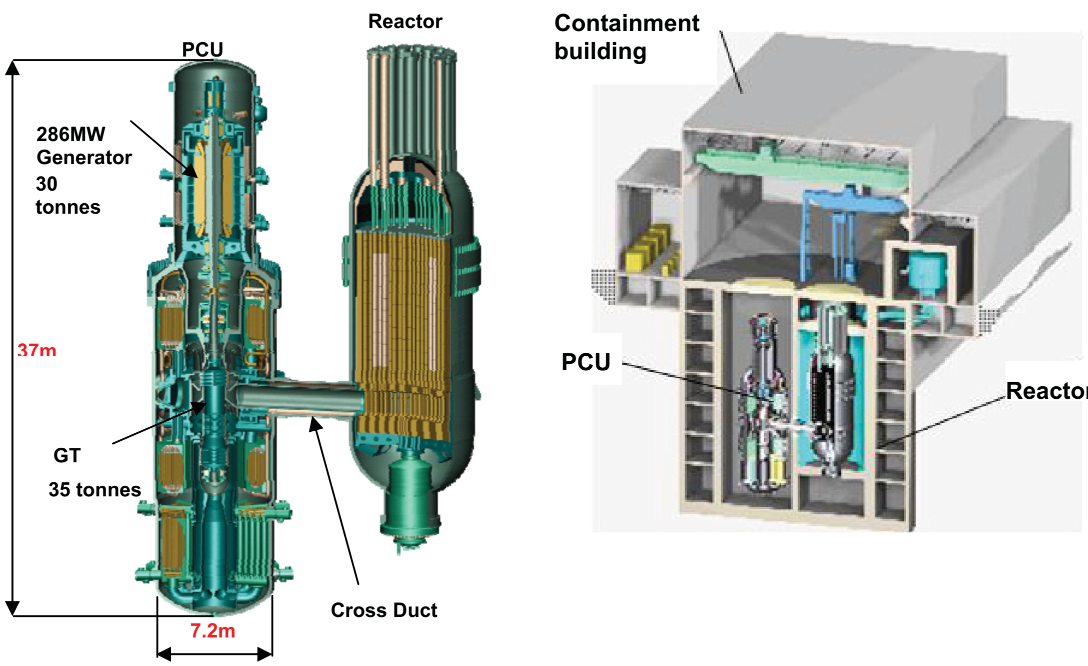


Figure 3 General GT-MHR Layout

1.3.2 *Reactor System*

The NGNP nuclear heat source will be a single GT-MHR reactor module with some modifications to permit operation with a reactor outlet coolant temperature of up to 950ºC (vs. 850ºC for the GT-

MHR). The NGNP nuclear systems include the Reactor System, Cross Vessel and Hot Duct Assembly, Reactor Pressure Vessel (RPV), Shutdown Cooling System, and Reactor Cavity Cooling System. Figure 4 shows a cross-sectional view of the Reactor System.

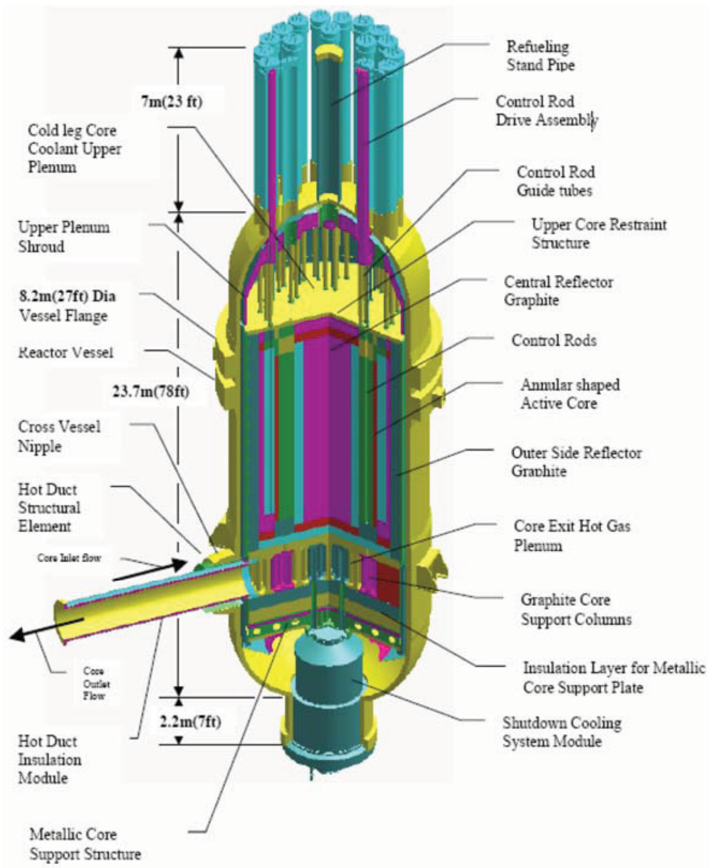


Figure 4 Reactor System

The GT-MHR core consists of UCO TRISO fuel (<20% enriched), in fuel compacts loaded into hexagonal graphite blocks. These blocks are loaded in 102 fuel columns in three annular rings with 10 fuel elements per fuel column, for a total of 1020 fuel elements in the active core. For the equilibrium fuel cycle, one-half of the core is reloaded every 425 full-power days.

Passive safety features of the MHR include (1) ceramic, coated-particle fuel that maintains its integrity at high temperatures during normal operation and loss of coolant accidents (LOCAs); (2) an annular graphite core with high heat capacity that limits the temperature rise during a LOCA; (3) a relatively low power density that helps to maintain acceptable temperatures during normal operation and accidents; (4) an inert helium coolant, which reduces circulating and plateout activity; and (5) a negative temperature coefficient of reactivity that ensures control of the reactor for all credible reactivity insertion events. The fuel, the graphite, the primary coolant pressure boundary, and the low-pressure vented containment building provide multiple barriers to the release of fission products. Taken together, the core has a high degree of inherent safety, and is designed such that there should be no significant fuel failure, even for the most severe accident events.

1.3.3 *Power Conversion System*

The GT-MHR power conversion cycle is an intercooled and recuperated closed Brayton cycle with helium as the working fluid. Helium as a working fluid has many advantages. It does not become radiologically active and is chemically inert. It also has excellent heat transfer properties making it an effective reactor coolant and allowing the heat exchangers in the cycle to be efficient and compact. One disadvantage is that to achieve a given pressure ratio across a compressor or turbine requires many more stages than an air-breathing engine.

Figure 5 is a schematic of the helium coolant cycle within the PCS. High-pressure helium from the reactor core outlet plenum flows through the hot duct inside the PCS cross vessel to the turbine where it expands. The mechanical energy generated in the turbine is used to drive the generator and the low and high-pressure compressors, which are all arranged on a common shaft. Downstream of the turbine, the helium flows through the low-pressure side of the recuperator where heat is transferred to the helium flowing back to the reactor through the high-pressure side of the recuperator. Upon exiting the low-pressure side of the recuperator, the helium passes through the precooler, where it is cooled to about 25°C, before passing through the low-pressure compressor (LPC). Following the LPC, the helium passes through the intercooler where it is again cooled to about 25°C before passing through the high-pressure compressor. The helium then flows through the recuperator high-pressure side, where it is heated up to the reactor inlet temperature and flows back to the reactor through the annular gap between the PCS cross vessel and the hot duct.

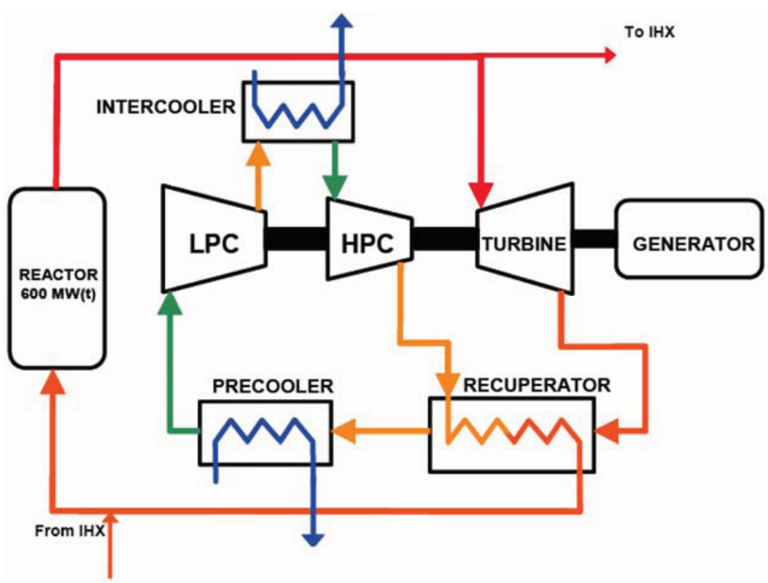


Figure 5 PCS Flow Diagram

The attractive features of this PCS design include:

- A direct Brayton cycle that improves efficiency and economics.
- A vertical shaft that minimizes blade/stator clearances to reduce bypass flows, reduces plant footprint and associated capital costs, allows vertical lifts for maintenance, and the use of gravity to offset turbine thrust.
- Electromagnetic bearings that reduce energy losses and eliminate the possibility of lubricant ingress into the primary circuit.
- A single stage of intercooling that improves thermal efficiency by about 2%.
- A submerged generator that eliminates a rotating seal in the primary pressure boundary and reduces leakage of primary helium coolant.

In 1994, the GT-MHR was selected as the basis for a joint effort by the US and Russia to design a MHR to be used for disposition of weapons-grade plutonium. OKBM in Nizhny Novgorod was given responsibility for the GT-MHR design development and is the Chief Designer of the reactor plant. In support of this arrangement, DOE also negotiated a contract with OKBM to perform R&D work. OKBM has further developed the original GA PCS design through preliminary design and has made several design improvements. OKBM supported the Rolls-Royce study both through the provision of reports, drawings and information, as well as a visit by OKBM staff to Rolls-Royce in February 2007. In particular, this review has drawn upon the OKBM Technology Development Programme (TDP, Reference 3).

Further, reference has been made to INL's own studies, one such is the Independent Technology Review Group (ITRG) report on design features and technologies uncertainties for NGNP (Reference 4). Rolls-Royce comments on the ITRG technical risk assessment are included in Annex A.

1.4 Overview of Power Conversion Unit

1.4.1 *System Description*

The existing design is a closed loop helium cycle system consisting of a nuclear reactor vessel connected by a cross duct to the Power Conversion Unit (PCU).

The PCU vessel contains the vertically mounted gas turbine (GT), which is surrounded by the recuperator, precooler and intercooler systems. The GT, which rotates at a speed of 4400rpm, is connected to a 286MW generator by means of a flexible coupling. Power electronics are used to convert the generator output to be synchronous with the local electricity grid[1].

1.4.2 *Gas Flows*

The GT-MHR PCS design was originally designed for 850°C delivery temperature from the reactor. This design study for the NGNP PCS has initially worked with 850°C reactor delivery temperature, but has also considered the technology requirements for operation at 950°C.

[1] A synchronous version was considered but rejected due to the massive increase in weight of the GT rotor, which was prohibitive on cost and was far outside the experience of bearing suppliers.

A schematic of the PCU gas flows is shown in Figure 6.

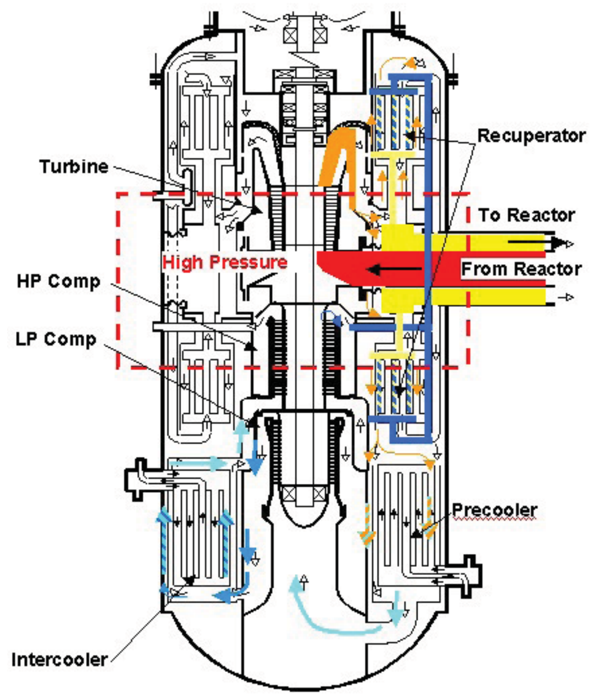


Figure 6 Schematic of PCU Gas Flows

Helium is heated in the reactor and passes through the centre of the hot duct and cross vessel and into the High Pressure Module (HPM) of the GT where it is then directed into the 9 stage turbine. Flow continues from the turbine through the hot side of the recuperator into the precooler and into the 10 stage LP compressor, from here the flow is then directed through the intercooler and back into the 13 stage HP compressor, from here the flow goes through the cold side of the recuperator and is then directed along the outside of the cross duct back to the reactor.

The precooler and intercooler are both water cooled with the water circuit being of a lower pressure than the helium circuit, thus preventing ingress of water into the helium circuit in the event of a leak.

The GT is a single shaft design supported on electro-magnetic bearings (EMBs) with additional catcher bearings (CB) providing back-up should power to the EMBs be lost. The electrical generator features the same type of bearings. The thrust bearings for the GT and generator are positioned either side of the flexible coupling in order to minimise the thermal movements, and thus minimise induced stresses in the coupling. Conventional hydrodynamic bearings are not possible in a submerged high temperature helium environment and cannot be placed outside the PCU without adding the complexity of rotating shaft seals (and the resulting helium leakage).

The need to deliver helium flow to and from the GT (to and from the recuperator, precooler and intercooler) necessitates the use of sealing features to prevent leakage. This is achieved by fitting 7 seal rings ranging from Ø1m at the bottom to Ø3.2m at the top of the GT. When the GT is lowered vertically into position these 7 seals slide into housings in the main vessel and complete the sealing.

The turbine features conventional shrouded blades with fir tree root fixings and a solid bolted disc construction.

Both compressors feature conventional blades with circumferential root fixings and a hollow bolted disc construction.

The existing design layout is very good in terms of achieving an overall small plant footprint but at the expense of some complexity in certain features. These features will be discussed in the following sections

1.4.3 *Turbomachinery*

The GT rotor, which weighs ~35 tonnes, is to be supported vertically using EMBs with catcher bearings acting as back-up in the event of EMB failure. Currently, worldwide there is no similar application of EMBs acting at this combination of weight, speed and dynamic response/stiffness.

Analysis has shown that the rotor shaft is subject to 4 critical speeds during start-up and shut-down which will be controlled by varying the EMB stiffness. However with failure of the electrical supply to these bearings the rotor will see the full force of these critical speeds during rotor run down which could lead to secondary extensive blade failures. Whilst sufficient redundancy can be built into the EMB control system in order to reduce the probability of failure, there will always be the chance that loss of power supply to the control system will occur. Should this happen the catcher bearings will be needed to support the GT rotor during run down.

Although EMBs are physically scaleable to this application, a significant development programme is needed (and is included in the OKBM TDP) for the GT-MHR. The catcher bearings are also challenging, as when the EMBs fail they undergo significant crushing loads and frictional heat build-up. CBs rely on ball/roller or journal bearings to support the rotor but since no oil is allowed in this environment, heat build up may rapidly lead to over-temperature, failure of the CBs and subsequent secondary GT failure. A method for braking the shaft could be employed which would assist the CBs and prevent over-temperature.

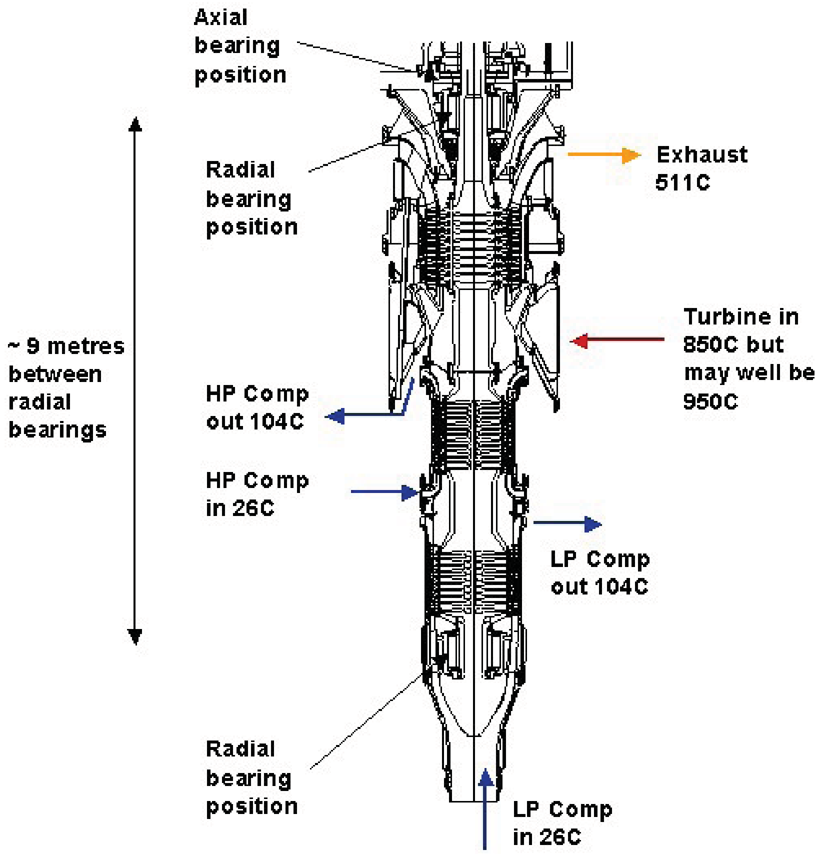


Figure 7 GT-MHR Turbomachinery

1.4.4 *Internal High Pressure Module*

The Internal High Pressure Module (IHPM) is a complex component that has to perform a number of functions, which include:

- Delivering hot, high pressure helium from the reactor to the turbine.
- Collecting recuperator return helium and delivering back to the reactor.
- Collecting turbine exhaust helium and delivering it to the low pressure side of the recuperator.
- Collecting HP compressor helium and delivering to the high pressure side of the recuperator.
- Providing support to the GT via seal housings.

It consists of two thick walled concentric rings with cut-outs for the reactor feed/return pipe and accompanying opposing pressure balance piston. The pipe and piston are connected to the module via bellows, which should accommodate any differential thermal movements, particularly during initial start-up. These bellows also connect the IHPM to the inner wall of the PCU vessel. Various pipes are welded to the IHPM to assist in transporting the helium.

The IHPM is ~ Ø6.5 m x 4.0m long with 100mm wall thickness operating at 70bar and 850 ºC (initially, with potential to increase to 950°C).

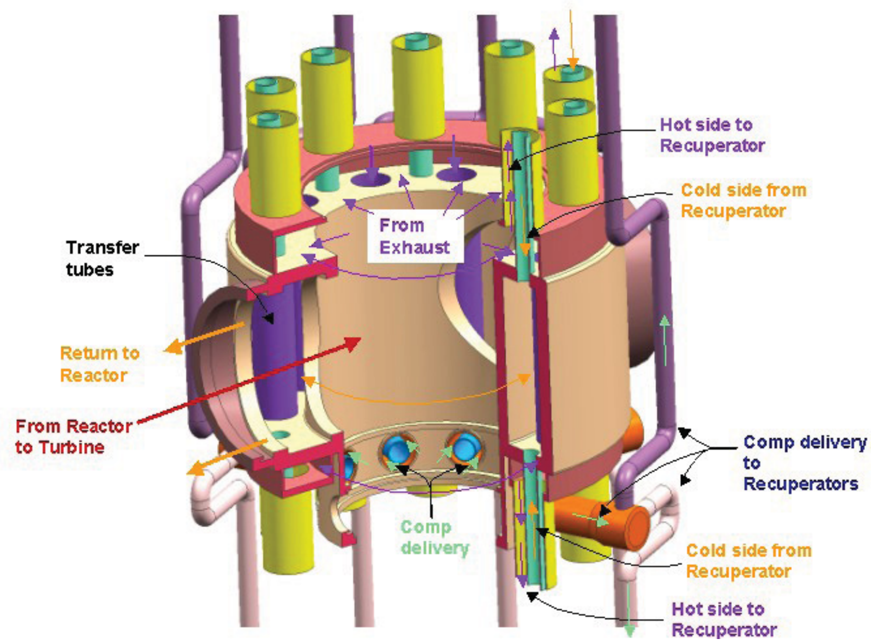


Figure 8 Internal High Pressure Module

1.4.5 *Support and Seal Arrangement*

The GT is lowered vertically into position in the PCU vessel and is bolted at its outer end. Seal rings along the GT length are needed to seal where helium is taken to and from the recuperator, precooler, intercooler and return to reactor. These seals are fitted blind when the GT is lowered into position. Although there appears to be adequate 'lead in' for the seals which should help during fitting, the GT rotor weighs ~35 tonnes and therefore it is considered that this weight will force the seals into position if they are misaligned. This could lead to seal breakage and possible loss of performance. Borescope inspection of the seals after assembly may help but the seal is buried and thus it may not be possible to verify the seal condition. It should be possible during the development programme to prove a method for fitting the GT but a method for confirming seal integrity should also be considered.

Another aspect of the seal rings is that they must assist in supporting the GT along its length. By their nature seal rings have some clearance so it is not entirely clear how they will support the GT.

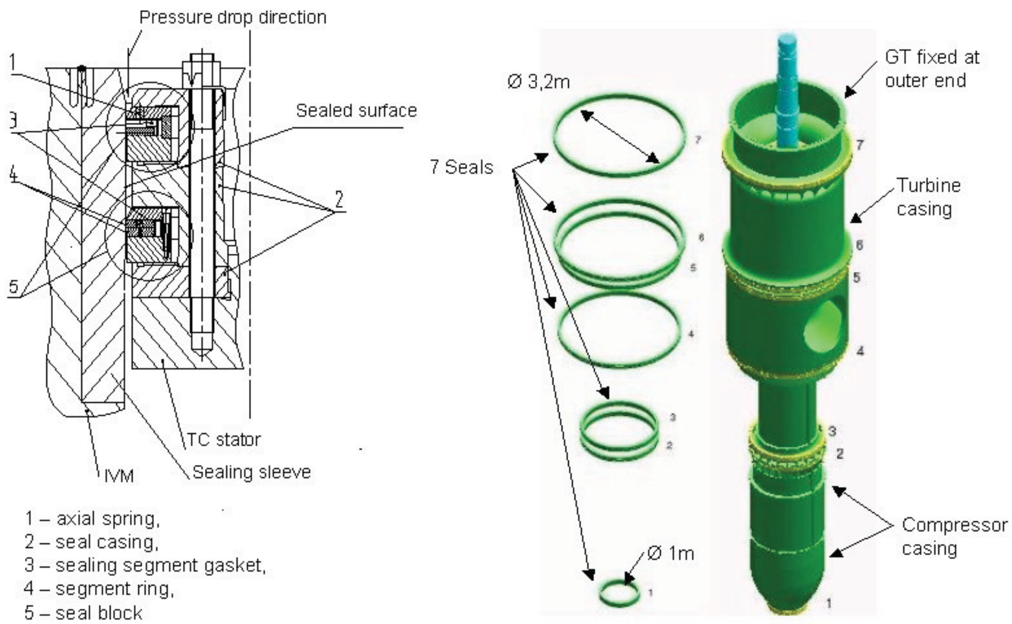


Figure 9 Support and Seal Arrangement

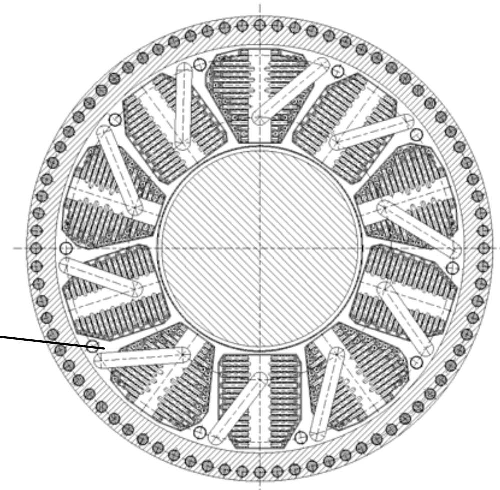
1.4.6 *Recuperator*

The recuperator consists of 20 modules spaced circumferentially between the GT and the PCU vessel inner wall. Each module contains 200 tubes Ø106mm x 2.2m length, and inside each tube is a matrix of folded and welded sheet which separates the LP and HP flows.

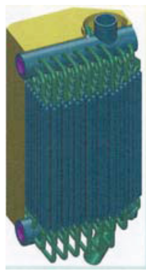
The manufacture of the recuperator is extremely labour intensive. It is estimated that to manufacture a single recuperator assembly about 350 tonnes of material is required, along with some 796 000 weld operations and some 50km of total weld length. Whilst it is accepted that automation will be used extensively to assist both manufacture and inspection it will still be a very costly item.

This arrangement seems very heavy and is located within the PCU thus making maintenance difficult, particularly in the event of a failed tube.

Weight	**350 tonnes**
No of welds	**796 000**
Total weld	**50km**



20 off similar modules containing a total of 4000 tubes



Very heavy, very costly, labour intensive and is located within PCU making maintenance difficult

Figure 10 Recuperator

2 Assessment of the GT-MHR PCS Design for NGNP

This section assesses the current GA/OKBM PCS design for the GT-MHR, as proposed for NGNP. The following subsections address key subsystems, as well as overviews of the performance cycle and whole engine performance. Each subsection describes the associated system, appraises it and makes suggestions for improvement and further study.

2.1 Performance Cycle

2.1.1 *System Description*

The GT-MHR power conversion cycle is an intercooled and recuperated closed Brayton cycle with helium as the working fluid. The minimum helium pressure in the cycle is 25 times atmospheric, giving a proportionate increase in power density. The overall pressure ratio of the cycle is around 2.8, which is very low compared to conventional air breathing gas turbine engines. The cycle is efficient, offering net electrical efficiency approaching 50% and is sufficiently compact that all major components can be contained within a single pressure vessel. The cycle is shown in Figure 11.

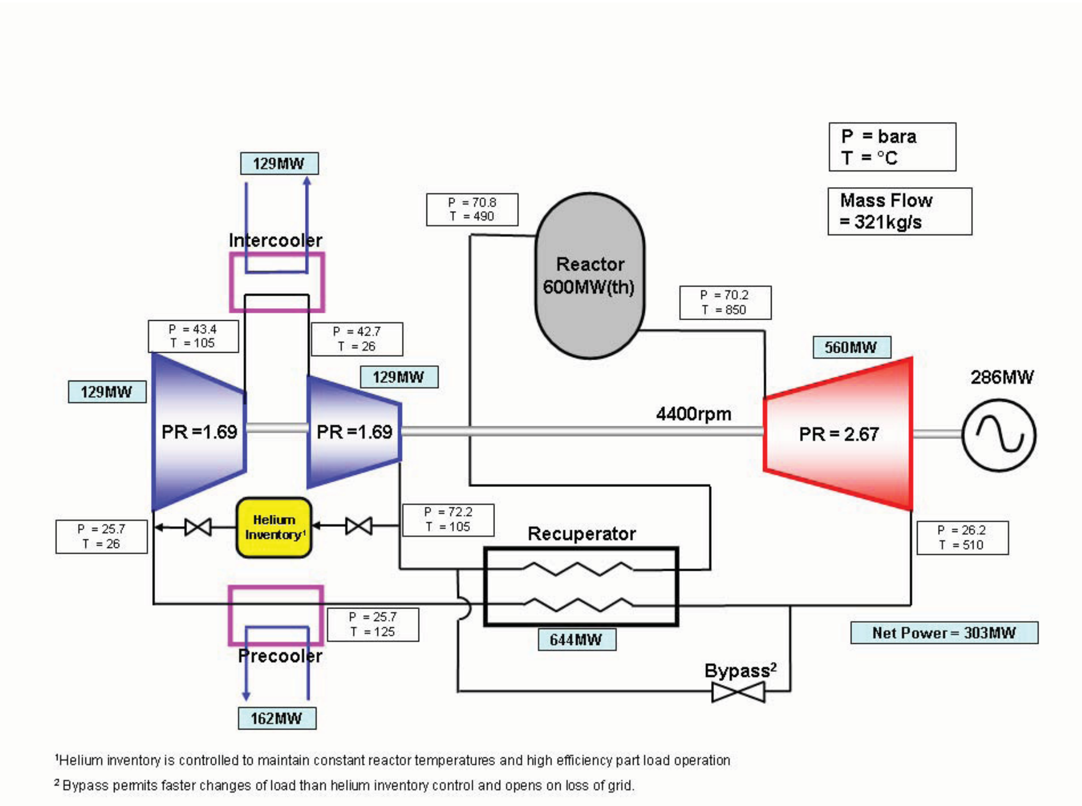


Figure 11 GT-MHR Cycle

Helium has advantageous properties both as a coolant for high temperature nuclear reactors and as a working fluid in a power conversion cycle. As a nuclear coolant it does not become radiologically active and it is chemically inert, so it does not degrade reactor materials at high temperature. Furthermore, it has a specific heat capacity (C_p) five times greater than that of air and a high thermal conductivity (k), making it a very effective medium for reactor cooling.

In the power conversion unit the excellent heat transfer properties are again an advantage allowing the recuperator, precooler and intercooler to be relatively compact and to have low pressure losses. Although the high C_p of helium results in each kg of working fluid doing more work, the high specific heat ratio (γ) means that many compressor and turbine stages are required to achieve a given pressure ratio.

In simple cycle gas turbines high efficiency is favoured by high pressure ratio, as the high temperature drop across the turbine reduces exhaust temperature and hence the energy 'wasted' to the environment. At lower pressure ratios cycle efficiency can be greatly enhanced by recuperation. Here much of the heat remaining in the gas at turbine exit can be 'recycled' back into the combustor/heater delivery stream, reducing the amount of heat input required to achieve a combustor/heater outlet temperature and much enhancing the cycle efficiency. An intercooler further enhances the efficiency by reducing the average work of compression in the cycle. Although this takes heat out of the combustor/heater delivery stream, this debit can be recovered in full from the recuperator, further reducing the exhaust temperature.

For an air breathing gas turbine operating at a combustor/heater outlet temperature of 850˚C, the optimum pressure ratio for an intercooled and recuperated cycle is around 7 (Reference 5). Because of the high C_p and γ, the optimum pressure ratio for a recuperated and intercooled cycle operating with helium is much lower at around 2.5 (see Section 2.1.2). This is close to the value at which the GT-MHR cycle operates.

Since the cycle is closed a precooler is required downstream of the 'hot side' of the recuperator to remove low grade heat from the cycle and reduce the average temperature of compression.

Closed cycle operation has a number of advantages. First, as noted above, the power density rises in proportion to the pressure in the circuit and hence increasing the pressure greatly reduces the size of the machine. Second, controlling the pressure allows the power density to be controlled without changing any of the non-dimensional parameters that govern gas turbine behaviour. Part load operation is therefore very efficient as the thermodynamics of the cycle are (to the first order) unchanged.

From the above introduction to the GT-MHR cycle the following is concluded:

- That helium is an appropriate working fluid for the power conversion cycle.
- That an intercooled and recuperated cycle is appropriate for the cycle pressure ratio and gives high efficiency.
- Use of a high-pressure closed cycle has considerable advantages for both compactness and part load performance of the machine.

2.1.2 *Assessment of Design*

A design point model of the GT-MHR cycle has been built in the commercial code 'Thermoflex 16'. The various flows, temperatures, pressures, efficiencies and pressure drops etc around the cycle were taken directly or inferred from material supplied by General Atomics and OKBM. The details are given in Table 1.

Table 1 Assumptions in Thermoflex Model of GT-MHR Cycle

Parameter	Value	Reference
Reactor Power	600MW	GA-A23952 (Reference 6)
Recuperator Effectiveness	95%	Meeting with OKBM in Derby 12/3/07
Cycle Pressures and temperatures	Various	GA-A23952 (Ref. 6)
LPC efficiency	89% poly (87.8% isen)	Gives correct ΔT at ΔP
HPC efficiency	89% poly (87.8% isen)	Gives correct ΔT at ΔP
Turbine efficiency	91.8% poly (93.2% isen)	Gives correct ΔT at ΔP
LPC PR	1.69	GA-A23952 cycle
LPC PR	1.69	GA-A23952 cycle
LPC PR	Floats to 2.68	Matches GA-A23952 cycle
Mass Flow	321kg/s	GA-A23952 (320kg/s) OKBM meeting 12/3/07 (322kg/s)
Recuperator ΔP/P	2% on both hot and cold sides	From GA-A23952 cycle pressures
Reactor ΔP/P	0.85%	From GA-A23952 cycle pressures
Pre-cooler ΔP/P	0.13%	From GA-A23952 cycle
Pre-cooler outlet temperature	299.2K	From GA-A23952 cycle
Intercooler ΔP/P	1.6%	From GA-A23952 cycle
Intercooler outlet temperature	299.2K	From GA-A23952 cycle
Generator efficiency	95%	Gives correct power output of 288MW. p75 of Reference 3 states efficiency = 97.7%
Mechanical losses	Not modelled	N/A
Bleeds and seal leaks	Not modelled	Leaks total 0.4% of total flow (Reference 7)

The model readily recreated all parameters quoted in the various General Atomics and OKBM references. It should be noted that not all loss mechanisms were included in the Thermoflex model. No attempt was made to include mechanical losses on the shaft, helium leaks, turbine disk cooling bleeds, windage in the generator or similar. However, in order to match the 48% quoted cycle efficiency, it was necessary to assume a generator efficiency of 95%, somewhat lower than the 97% to 98% expected for a generator with frequency conversion power electronics.

It is judged that the degradation in generator efficiency, which had to be applied to match the quoted cycle efficiency, is sufficient to allow for these unaccounted losses. Furthermore, more detailed work done on the turbines, compressors and heat exchangers (reported elsewhere in this document) has shown that the individual efficiency, effectiveness and pressure drop assumptions of the various components are also reasonable.

It is therefore concluded that the quoted cycle efficiency of around 48% is realistic. Further work supporting this conclusion is presented in the following sub-sections.

Sensitivities to Component Efficiencies

A number of studies were undertaken to explore the sensitivity of the overall cycle efficiency to the performance of individual components. This work can be summarised as follows:

- 1 percentage point HP compressor efficiency is worth ~0.25 percentage points cycle efficiency.
- 1 percentage point LP compressor efficiency is worth ~0.25 percentage points cycle efficiency.
- 1 percentage point turbine efficiency is worth ~0.4 percentage points cycle efficiency.
- 1 percentage point recuperator effectiveness is worth ~0.5 percentage points cycle efficiency.
- 1 percentage point pressure loss ($\Delta P/P$) applied simultaneously to both sides of the recuperator is worth ~0.8 percentage points cycle efficiency.

Sensitivity of Cycle Efficiency to Leakage Flows

There are six annular sliding seals, which limit helium leaks between the turbo-compressor external cavities. Design leak flows for these six seals were presented in Reference 7. The design leak flows (totalling 0.4% of the total mass flow) were incorporated into the Thermoflex model. It was found that the cycle efficiency fell by 0.15 percentage points. This result supports the conclusion that the numerous small losses not accounted in the modelling can be absorbed within the 2 percentage points by which the generator efficiency was degraded to match the 48% efficiency quoted for the GT-MHR cycle.

Sensitivity to Overall Pressure Ratio and Reactor Outlet Temperature

The current GT-MHR cycle has been designed for a reactor outlet temperature of 850˚C to reduce the materials risk, particular in the turbine. The intention with the NGNP application is to develop the baseline reactor up to a core outlet temperature of 950˚C. Studies have been performed with the Thermoflex model in order to understand the efficiency that might be gained from operating at temperatures in excess of 850˚C, and to understand how a cycle optimised at these conditions would differ.

The current GT-MHR cycle is designed so the reactor inlet temperature is 490˚C. It is understood that a limit of 500˚C was assumed when designing the cycle to limit the operating temperature of the reactor pressure vessel (the reactor inlet gas passes around the outside of the core and strongly influences the pressure vessel temperature). Figure 12 shows that the requirement to hold the reactor inlet temperature below 500˚C has compromised the cycle. At 850˚C reactor outlet temperature, the efficiency peaks at a pressure ratio of ~2.35 (and an inlet temperature of ~545˚C). Not only does the higher pressure ration reduce the cycle efficiency by around 0.5 percentage points, it also requires more compressor and turbine stages to achieve the increased pressure ratio, making the GT shaft longer and heavier.

Increasing the reactor outlet temperature to 950˚C is an NGNP objective and it is also understood that the design of the reactor and/or pressure vessel would be altered in order to permit reactor inlet temperatures of up to 590˚C (Reference 8). Figure 12 shows that there would be a considerable benefit if this were possible. First, the increased reactor outlet temperature improves the 'Carnot' efficiency of the cycle and is worth around 3 percentage points at constant cycle pressure ratio. Second, if a reactor inlet temperature of 590˚C were attainable the cycle could operate very close to the optimum pressure ratio (~2.4). This not only increases cycle efficiency by a further 0.5 percentage points, it also reduces the pressure ratio of the cycle with the benefit of reduced turbomachinery stage counts.

Clearly however, the increased technical risks with pursuing higher reactor outlet temperature are considerable. The turbine blades would need to be cooled which would incur an efficiency penalty not accounted for in the calculations presented here. Furthermore, the recuperator, turbine structure and transfer ducting would all be required to operate at temperatures 100˚C greater than the current design.

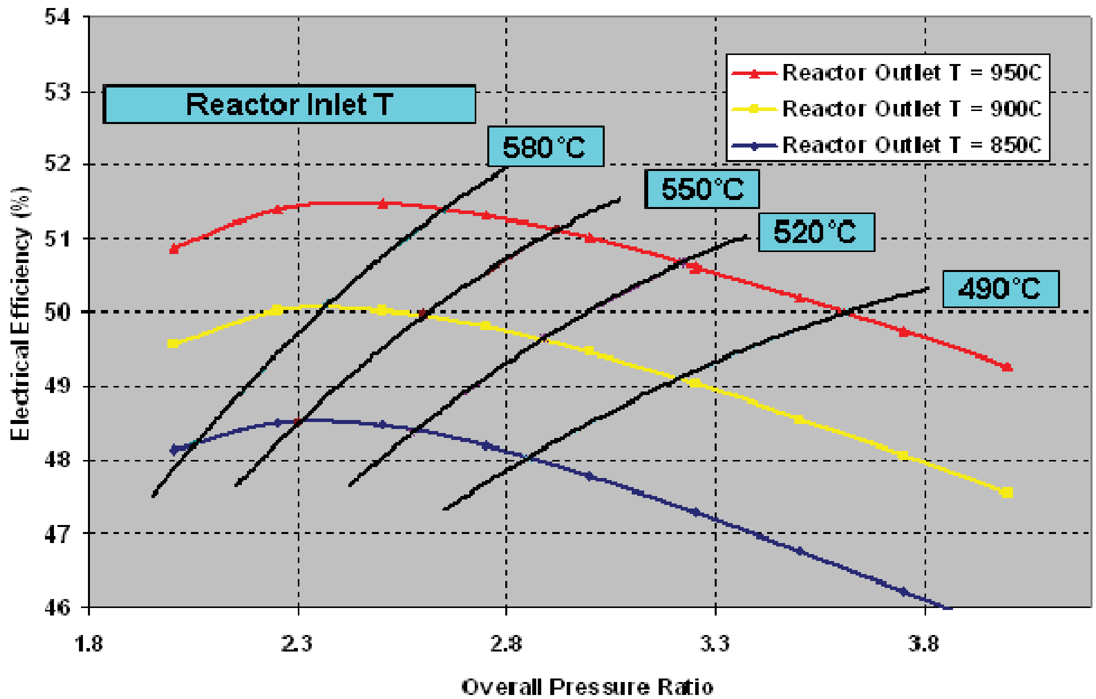


Figure 12 Variation of Cycle Efficiency with Pressure Ratio, Reactor Inlet Temperature and Reactor Outlet Temperature

Limits on the minimum reactor inlet temperatures also apply. At a given reactor power, reducing the inlet temperature at a fixed outlet temperature necessitates a reduction in coolant mass flow. If the inlet temperature is too low, heat transfer from the fuel to the coolant is impaired and fuel temperature becomes excessive. At 850˚C the minimum reactor inlet temperature is 400˚C, at 950˚C the minimum inlet temperature is 510˚C.

2.1.3 *Off-design Performance Modelling*

The Thermoflex model is only capable of design point studies. In order to allow the modelling of off-design and, if required in future, transient performance, the GT-MHR cycle has also been modelled using the Rolls-Royce corporate gas turbine performance method, RRAP.

Off-design Performance – Hot Day / Cold Day

The off-design model has been used to assess the sensitivity of the GT-MHR cycle to the temperature of the heat sink. The design point for the cycle assumes that the precooler and intercooler both cool the helium to 26˚C. Clearly, the achievable helium temperature depends upon the heat sink available at the plant location. If the location were on a coast, a low temperature heat sink would be available with little variation in sink temperature as ambient conditions vary. If a cooling tower is assumed then the achievable helium temperature will vary considerably as ambient temperatures rise and fall. It should also be noted that if the heat sink is the atmosphere, then there are two exchanges of heat to be considered: from helium to water and then from water to the air. It is suggested that a 26˚C helium temperature might only be achievable at conditions cooler than ISA (i.e. 15˚C). It is therefore of interest to model the response of the cycle to variation of the minimum helium temperature.

In performing the modelling the following has been assumed:

- That the reactor outlet temperature is not permitted to rise above the design value (850˚C).
- That the reactor power is not permitted to rise above the design value (600MW).
- That the helium pressure at the reactor is not permitted to rise above the design value (70.8bara).

It has been found that, at minimum helium temperatures below the design value, helium pressure in the GT cycle has had to be reduced to avoid reactor power increasing beyond the limit at constant reactor outlet temperature. Conversely, at minimum helium temperatures above the design value, helium pressure in the GT cycle has been increased until the pressure at the reactor returns to the design value. This has been done to minimise the loss of electrical power.

The response of the GT-MHR cycle to changing minimum helium temperature is shown in Table 2 below.

Table 2 Response of GT-MHR Cycle to Helium Temperature after Heat Sinks

Minimum Helium Temperature	Net electrical power	Reactor Power	Net electrical efficiency
16˚C	294MW	600MW	49.2%
26˚C (design condition)	288MW	600MW	48.0%
36˚C	265MW	567MW	46.7%
46˚C	240MW	533MW	45.1%

The GT-MHR cycle is clearly sensitive to the minimum achievable helium temperature. If this should rise by 20˚C the achievable electrical power falls by almost 50MW. This happens for two reasons. First, the electrical efficiency of the cycle falls as the temperature at which heat is rejected rises. This loss of 'Carnot' efficiency results in the loss of 17MW of electrical power. The second and larger effect is a result of a necessity of reducing the reactor power.

This happens because, as the inlet temperature to a compressor constrained to a fixed rotational speed rises, the pressure ratio across it falls (Reference 5). In the GT-MHR cycle one of the effects of falling pressure ratio is a reduced temperature drop across the turbine. At constant reactor outlet temperature the temperature at turbine outlet must rise. The increased heat in the turbine exhaust is recycled back into the reactor inlet flow through the recuperator increasing the reactor inlet temperature. Because the cycle pressure at the reactor has been returned to the design value (by increasing the minimum pressure in the circuit) the mass flow at the reactor remains approximately constant. With constant reactor mass flow, an elevated reactor inlet temperature and a reactor outlet temperature that is not permitted to rise, the reactor power must be reduced.

There are two possible mitigations for this degraded performance at high helium inlet temperatures. First, if the circuit pressure could be permitted to rise beyond the design value, some of the power loss could be recovered. This would not improve the efficiency but would allow the reactor power to be increased. Note, however, that the reactor power would be directly proportional to the circuit pressure in this scenario and hence returning the reactor to 600MW at 46˚C minimum helium temperature would require a helium pressure at the reactor of almost 80bara.

The second possible mitigation would be to increase the GT shaft speed. This would have the effect of restoring the pressure ratio and dropping the reactor inlet temperature. However, given the mechanical constraints on the design of the turbomachinery, it is considered unlikely that this would be viable in practice.

It is concluded that the GT-MHR cycle is sensitive to increased minimum helium temperatures, which may arise from poor heat sink availability at the installation site or high ambient temperatures. No effective mitigation for this has been identified.

<u>Off-Design Performance – Cycle Designed for 950˚C Reactor Outlet Temperature Operating at 850˚C</u>

The off-design model has also been used to investigate a second off-design condition: if the cycle were designed for operation at 950˚C reactor outlet temperature, what would be the performance at 850˚C?

The results are shown in Table 3 below.

Table 3 Off-Design Performance with Reactor Outlet Temperature Varied

Plant Design /Operating Condition	Net electrical power	Reactor Power	Net electrical efficiency*
Plant designed for 950˚C operating at 950˚C	306MW	600MW	51.1%
Plant designed for 850˚C operating at 850˚C	285MW	600MW	47.9%
Plant designed for 950˚C operating at 850˚C	260MW	551MW	47.4%

* Note that the RRAP model predicts efficiencies very slightly different to the Thermoflex results presented in Figure 12

There are two points of interest in these results. First, the efficiency degradation is relatively small. A cycle designed for operation at 950˚C is nearly as efficient when operated at 850˚C as is a cycle designed for operation at 850˚C. However, in common with the ‘hot day’ analysis (and for similar reasons) significant electrical power output is lost when reactor outlet temperature is reduced.

At fixed shaft speed, reduced reactor outlet temperature reduces the pressure drop across the turbine and hence the cycle pressure ratio. The power lost can be partially recovered by increasing the cycle helium inventory so that the pressure at the reactor is restored to the design value. However, the reduced pressure ratio results in an elevated turbine outlet temperature, increased reactor inlet temperature (due to the recuperator) and a reduced temperature rise in the reactor.

Because the reactor outlet temperature is lower, the turbine can accommodate more flow and hence the cycle mass flow rises. However, this rise is insufficient to offset the reduction in reactor temperature rise and hence the reactor output power must fall. The most obvious solution to this would be to increase the shaft speed but this would almost certainly be precluded by mechanical considerations.

It is concluded that if the GT-MHR cycle were designed to operate at 950˚C and were then operated at 850˚C the resultant efficiency drop would be tolerable. However, since there is no control over the reactor inlet temperature (which is a function of the recuperator effectiveness and the turbine outlet temperature) there is an unavoidable loss in electrical output due to a reduction in temperature rise across the reactor.

Off-Design Performance – Bypass Flow

The GT-MHR design includes a cross duct which, when the controlling valves are opened, permits a fraction of the flow to leave downstream of the HP compressor, bypass the reactor and turbine and return upstream of the hot side of the recuperator. Opening the bypass valves provides controllability by reducing the pressure ratio across the turbine and hence the electrical power developed on the shaft.

One of the main roles of the bypass is to allow the shaft power (and hence the accelerating torque) to be rapidly reduced in the event of loss of grid load, thus preventing shaft overspeed. It also plays a role in the more benign transients of normal operation. Short-term load changes are achieved by controlling the bypass flow. For example, during a rapid load reduction the bypass valve would be opened to reduce the power developed on the shaft. The helium inventory would then be steadily reduced until the bypass valves have fully closed once again. The bypass valves are also to control the speed of the shaft during synchronisation when the generator/motor is neither importing power to, nor exporting power from the shaft.

The action of the bypass valve has been modelled in RRAP. Making the assumption that reactor outlet temperature is held constant as the valve opens, it was found that 45% of the cycle flow would need to pass down the cross duct to reduce the net shaft power to zero. This was a surprising result as bleed flows in conventional gas turbines are much lower. The reason for this is thought to be that the turbine power is very high relative to the compressor power (due to the low pressure ratio and intercooling) and thus the net power is high relative to the turbine power. This means that a much higher fraction of the turbine power must be lost to achieve zero net shaft power than is the case for a conventional gas turbine.

This result has two main implications. First, the cross duct and associated valves will need to be large and also need to be capable of coping with the large changes of fluid momentum when the valves are opened. Second, because the bypass flow is so large there is a considerable effect on the HP compressor. The RRAP model predicts that the pressure ratio across the HP compressor falls almost to unity at 45% bypass flow. This dramatic increase in capacity at the back end of the compressor at high flow could result in aerodynamic instability as the compressor is operating a long way from its design point. It is recommended that this be investigated further.

Off-Design Performance – Starting Procedure

It is understood that the start-up procedure for the GT-MHR cycle is as follows:

- Establish precooler and intercooler flows.
- Raise helium inventory to ~7% (33psia).
- Motor turbomachinery up to full speed (4400rpm).
- Start reactor and use nuclear power to zero out generator motoring power whilst increasing pressure to ~147psia.
- Disconnect generator and control shaft speed with bypass valves.
- Use bypass valve to control speed to synchronise.
- Engage automatic control:
    - Reactor power controls core outlet temperature.
    - Bypass valves and helium inventory control electrical output. Output is varied over short timescales by adjusting the bypass valve. Over longer timescales the helium inventory is varied to eliminate the bypass flow.

As the start up occurs over a relatively long period it can be modelled as a series of steady states. The purpose of the modelling is to confirm that the proposed sequence is feasible and that component stability is not threatened. Table 4 summarises the steady state cases that have been run with the RRAP model.

Case 1 models the point at which the helium inventory has been raised to 7% (33psia) and the shaft is being motored at full speed. It has not been possible to replicate the state where the reactor is fully shutdown with negligible decay heat, because a small amount of reactor heat input has been found to be necessary to stabilise the model. Hence the situation modelled is that where the reactor has been started and is operating at low power, yet considerable power (8MW) is still being supplied to the shaft from the motor.

Case 2 is the point at which the reactor power has been raised sufficiently that there is no net power on the shaft and the motor can be disconnected.

Cases 3 and 4 show that the minimum cycle pressure can be raised in order to increase the reactor power output whilst maintaining zero net power and constant speed on the shaft. In this phase the speed of the shaft will be controlled by the bypass flow such that the generator can be synchronised.

Case 5 shows that when synchronisation has been achieved the bypass valve can be closed and power can be exported from the machine. The net power exported for this case is around 15% of the design value.

Table 4 Steady-State Modelling of Start-up Conditions

Case	Shaft Speed (rpm)	Net shaft Power (MW)	Reactor Power (MW)	Min Cycle Pressure (psia)	Bypass Flow (% of main flow)	Reactor Outlet Temperature (°C)
1	4400	-8	6.6	33	25	200
2	4400	0	25	33	25	540
3	4400	0	65	85	25	540
4	4400	0	113	147	25	540
5	4400	46	139	147	0	540
6	4400	144	290	186	0	850
7	4400	286	600	373	0	850

Case 6 is at 50% power, which confirms that the output power of the machine can be varied by changing the helium inventory alone, with no change in the thermodynamics of the cycle. Case 7 is the full power condition.

Though this analysis has not been exhaustive, it has been sufficient to demonstrate the following:

- That the available control parameters (reactor power, bypass flow and helium inventory) provide sufficient flexibility to start the machine and hence that the proposed start up sequence does appear to be feasible;

- That there should be no issues with compressor stability (all of the cases above are operating further from the compressor surge lines than the full power case).

2.1.4 *Areas of Uncertainty / Issues for Further Study*

The areas of uncertainty where more work is required are off-design and transient performance. The limited off-design work done so far has highlighted issues with the performance of the cycle at 'hot day' and reduced reactor outlet temperature conditions. At these conditions the pressure ratio falls and, as a result of the recuperator, the reactor inlet temperature rises. The size of the turbomachinery limits the flow that can be passed through the reactor and hence the reactor power output must fall. It would be of interest to do more studies at various off-design conditions to understand whether this behaviour has other implications.

It would also be of interest to model the start and shut down sequences (as a series of steady states) in order to understand whether the components remain stable and to confirm that the proposed start sequence is possible.

Transient modelling would also be of great interest. It is expected that normal manoeuvres will be relatively benign due to the slow rates at which they occur. However, the performance of the cycle under fault and accident conditions will be very important for the preparation of safety cases for the nuclear plant. Of particular interest is the loss of grid event and the management of it using the bypass flow.

2.2 Compressor Aero/Mechanical Design

2.2.1 *System Description*

Rotating Assembly

The compressor shown in Figure 13 is a two-spool arrangement, a 10-stage LP section and a 13-stage HP section joined by an intershaft. At the front of the LPC is a location disc to which the forward radial magnetic bearing rotor is attached. At the rear of the HPC is a shaft section to a flanged joint which mates with a rotating seal disc and another intershaft attached to the turbine module.

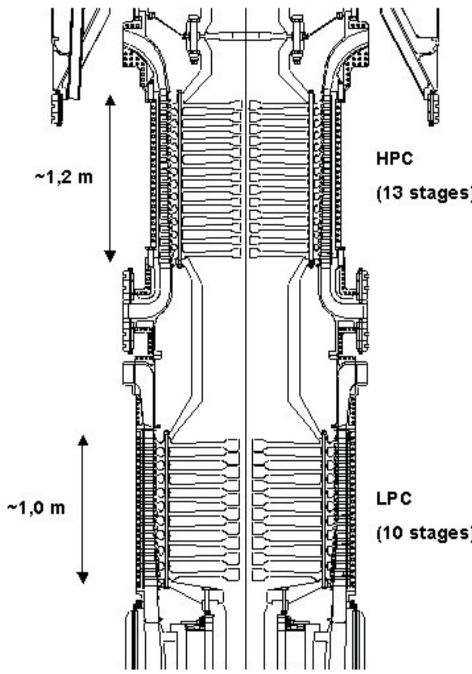


Figure 13 Compressor General Arrangement

Based upon PCU general view drawing ref. PHAT.501379.005BO (Reference 9)

The compressor system thus comprises the magnetic bearing rotor, location disc, ten LPC discs, intershaft, thirteen HPC discs, and the HPC rear shaft. Each of these parts is located relative to adjacent parts by a spigot joint. The whole stack of LPC discs and adjacent parts, and HPC discs and adjacent parts are secured together by long tie bolts. The length of these tie bolts is approximately 1.2 meters for the LPC and 1.38 meters for the HPC.

Rotor blades are located in circumferential root features in the disc rims in both the LPC and HPC sections. Blade numbers in each stage is not known. Axial spacing between blades and stator vanes appears small compared with blade axial chords.

Casing Assemblies

All casing sections surrounding the rotating assembly are assumed to be manufactured from steel and are split casings to enable build of the compressor sections. The split sections are for the LP compressor, the LPC exit duct, the HPC inlet duct, the HP compressor, and HPC exit duct. Casing sections appear to be located relative to each other by dowel at each section external bolted joint. There is an internal bolted joint between the LPC exit duct and HPC inlet duct that will require securing prior to assembly of split sections around the rotating assembly. The HPC inlet duct section carries sliding seal units that maintain separation between LPC exit and HPC inlet flows. Forward of the LPC casing is the magnetic bearing support casing and forward of that, the intake section which carries a sliding seal at its forward face to mate with in-vessel structure.

Location features for the stator vanes are incorporated in each of the compressor casing sections. The vanes inner fixings provide sealing against disc rim seal features.

2.2.2 *Aerodynamic Assessment of Design*

There are two compressors in the GT-MHR reference concept, a low pressure 10 stage machine blowing 1.69 pressure ratio and a high pressure 13 stage machine also blowing 1.69 pressure ratio. These compressors have been analysed with the Rolls-Royce's preliminary aerodynamic design tools (modified to incorporate helium as the working fluid) and claimed efficiencies appear sensible. No major issues with aerodynamically designing for helium operation are envisaged at this stage. Rolls-Royce has designed compressors in this low Mach number design space before (albeit for air as the working fluid).

Predicted polytropic efficiencies for these compressors are 91.9% and 91.1% for LP and HP respectively. The surge margins for the compressors are also predicted to be acceptable. With the pressure ratio split between the compressors altered very slightly to 1.6 (for LP) and 1.8 (for HP) the predicted achievable surge margins are 20% for both compressors. Until the transient requirements for these components are better understood, this seems a sensible level of surge margin to have at the design point in the pre-concept design phase.

Aerodynamic test validation for these compressors in the helium environment looks difficult, but two possible ways round this could be:

- Aerodynamically design cautiously, so that the risk of inadequate performance is small.
- Fit instrumentation in the first turbomachinery module so that aerodynamic design can be developed for subsequent modules.

In summary, the compressors in the GT-MHR design are seen aerodynamically as low risk. No major concerns have been identified at this stage.

2.2.3 *Aerodynamic Assessment of Design*

Rotating Assembly

Comparison was made with an existing civil aero/industrial engine produced by Rolls-Royce. Discs from this Rolls-Royce engine were scaled on radius and rotational speed and compared with the disc profiles seen in the PCU general view drawing (Reference 9). Rim loads using blade masses deduced from the Rolls-Royce aero model of the GT-MHR were also considered in the comparison. Based on this comparison, the disc assembly is within Rolls-Royce experience and does not present any particular mechanical concerns.

There is considerable scope for weight reduction and therefore alleviation of the duty requirements for the magnetic bearing, catcher bearing and its support structure. Moving to an electron beam welded titanium assembly will significantly reduce the weight of the discs removing the need for extra disc material around the tie bolt holes. Additionally the use of bladed discs (blisks) could save an extra 20% of weight relative to a bladed, welded disc. On aero engines a disadvantage of blisks is that foreign object damage to blades can cause the whole blisk to be replaced even though only one blade is damaged. On the GT-MHR, there should be no sources of foreign object damage making blisks particularly well suited to this application.

Containment of blades and discs needs careful consideration, particularly on the HPC due to its proximity to the hot gas duct into the reactor. Full FMECA analysis is needed to ensure that all the risks are adequately mitigated.

Casing Assembly

Split casings are machined as matched pairs to ensure circularity and concentricity of the rotor paths and stator vane location features. The GT-MHR design of compressor casings would be produced from steel and the static application (not weight restricted like an aero engine) will allow the casing to be thickened to ensure sufficient stiffness. The pressure differential across the casings causing compressive loading will also be of benefit to stability of the casing circularity.

The inlet and exit casings are necessarily split to enable them to be fitted to a balanced rotor assembly. The duct section will incorporate spokes between the outer and inner walls and presumably a spoke will be at the split line position to minimise helium leakage. Care needs to be taken to ensure asymmetry in the casing geometry does not induce any distortion into the compressor casing sections thus affecting tip clearances. At the split lines, particularly between zones of large pressure differential, measures may be required to minimise leakage of helium. Bolt spacing and face flatness may need to be controlled at all joint positions to reduce leakage. Axial joints will need to incorporate location features to maintain concentricity through the casing stack.

The stator vanes are located in features machined on the inside of the casing. Normal aero engine practice is to incorporate sacrificial stainless steel liners at these location positions to prevent undue wear on casing features. Stator vane numbers are not known, and the general view drawing does not clearly show the seal features between vane inner fixing and rotor discs.

The bearing support casing and intake casing are secured to the front flange of the LPC casing and are required to withstand loading resulting from the magnetic bearing under normal operation and those generated in the catcher bearing in a failed magnetic bearing scenario. The sliding seal/spigot location at the front of the intake casing will require minimal clearance and its effect needs to be considered as part of the WEM analysis.

Radial Clearances - Electromagnetic Bearing Clearances

Electromagnetic bearings of the size needed for the rotating system are not currently available and manufacturers cannot provide definitive estimates of the magnetic bearing or catcher bearing air gap requirements. For smaller bearings currently in use for 1.5 ton machines, the air gap between the rotor and the catcher bearing is 0.25mm. For this assessment, a catcher bearing air gap of 0.75mm (0.030) has been assumed reflecting the much bigger size of the rotor assembly (see Figure 14 below). When static, the rotating assembly movement is limited by the catcher bearing air gap (0.75mm). Under normal operating conditions, it is assumed that the magnetic bearing will control the rotating assembly radial movement within +/- 0.075mm (0.003). During failure conditions (surge/ EMB failure) the rotating assembly can move and contact the catcher bearing taking up the 0.75mm air gap clearance and an undetermined amount of catcher bearing compliance designed to absorb shock loading.

In determining the effect that the radial freedom in the EMB will have on radial tip clearances, the assumptions for failure conditions and run on requirements are critical. For the purposes of this assessment, it is assumed that any failure which causes the rotor to touch the catcher bearing and which uses up some of the catcher bearing compliance would be a 'major' failure and it would not be expected to continue operating the gas turbine after this e.g. EMB failure. For more minor failures (e.g. surges) it is assumed that these would potentially touch the catcher bearing but not use up any compliance in it and, for these failures, the compressor would be expected to continue running unaffected.

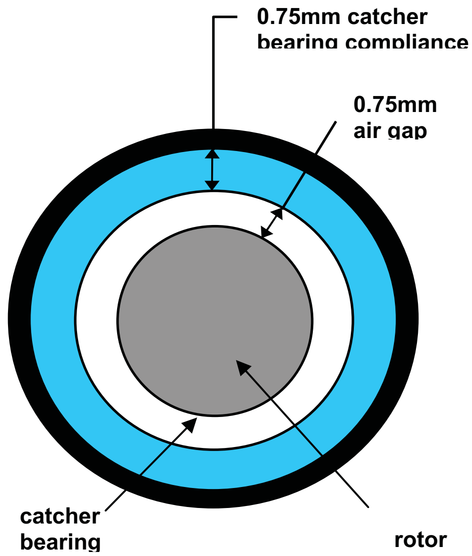


Figure 14 Compressor Radial Clearances - EM Bearing Assumptions

<u>Blade Tip Clearance</u>

Radial clearance between the rotor blade tips and the casing, and stator vane inner seal features and rotor drum, need to be closely controlled to ensure compressor performance. Typically this equates to a 1.5%rms value of clearance divided by blade height (equivalent to 1.5mm tip clearance for a blade of 100mm in length).

Figure 2.3 below shows the assumed cold build clearances, cold running clearances (for start up) and hot running clearances. In assessing tip clearances it is assumed that no rub with the casing will occur as this minimises the risks associated with blade tip rub induced failures and with radioactive contamination of abradable lining material. It is assumed that the radial tip clearances need to accommodate a minor failure (e.g. surge) so a 'failure allowance' of 0.75mm should be included in the clearances, equal to the assumed EMB air gap size (see previous section). The CF growth and manufacturing tolerances estimates are based on current engine design data and whirling effects have been estimated using engineering judgement.

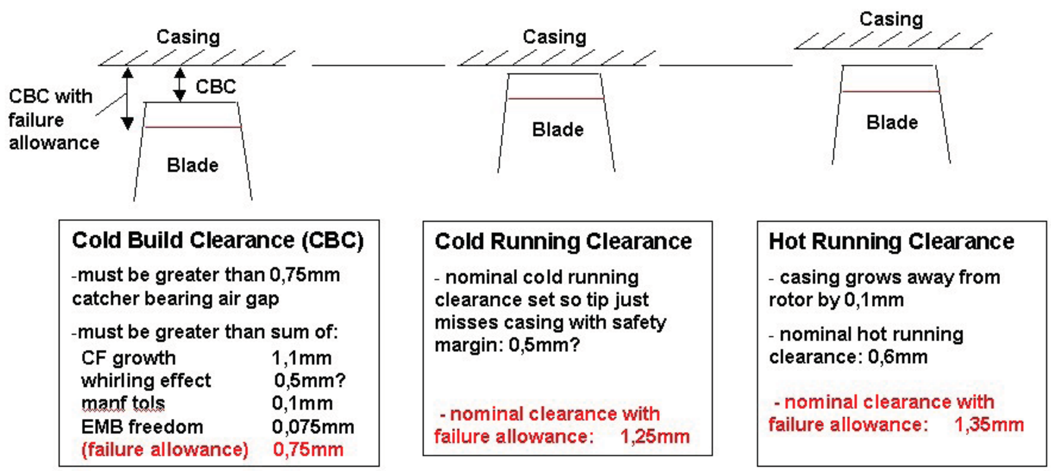


Figure 15 Compressor Radial Clearances

For the blade heights being considered the 1.35mm tip clearance shown above is close to the 1.5%rms target and is considered acceptable given the pre-concept stage of the assessment and the unknowns relating to the EMB clearances. In the event that detail analysis shows that the proposed tip clearance to be inadequate, then linings may be needed. However, it would still be intended that there would be no rub under normal operating conditions, and that only under extreme conditions, such as a surge, would a rub occur.

Other means of controlling tip clearances under normal operating conditions may be available depending upon the final compressor configuration. For instance, if here is sufficient hade in the outer annulus line, advantage can be taken of the differential axial growth between casings and bladed rotor, i.e. if the casing has a 3 degree hade then 10.0mm axial forward movement of the casing will result in a 0.5mm reduction in tip clearance.

The stator vane inner seal radial clearance has not been specifically reviewed but radial movement from the EBM in a failure case could generate heat in the seal fins causing cracking. Careful consideration therefore needs to be given to determining acceptable clearances for these seals.

<u>Blade Number</u>

Rolls-Royce created an aero model of the GT-MHR for analysis purposes, which, on the single iteration that was run, showed high blade numbers were required. The number of blades required by the model was not able to fit in the space available on the disc rim however further iterations of the model could probably have reduced the blade numbers required on each stage. The blade number on the proposed GT-MHR number is not known for each stage but presumably it has been properly assessed and the blades will fit properly in the disc rim without having to resort to skewing of platforms, which, on circumferential blades, can lead to chocking of blades under operating conditions.

<u>Axial Clearances</u>

Axial spacing between blades and vanes is controlled by aerodynamic requirements, aerofoil vibration criteria, surge allowance, and the amount of differential movement between the rotating assembly and static parts (vanes). The axial location for the whole rotating system is the plane of the axial magnetic bearing which is 1.7 meters rearward of the rear radial magnetic bearing and

10.5 meters from the front face of the LPC. While there is possibly a small amount of differential axial adjustment within the axial magnetic bearing, the thermal effects dominate, amounting to the casing growing approximately 10.0mm forward of the rotor system over the compressor sections. Hence the axial gap between blades needs to accommodate the 10.0mm growth, tolerances, min. LE/TE gaps, aerofoil LE/TE blend radius, and a surge allowance amounting to approx. 25.0mm total. The spacing shown on the general view drawing may not meet this requirement particularly in the HPC section. A more detailed analysis may require some increase in rotor and casing length above those estimated depending upon accurate thermal growths and better understanding of surge behaviour.

<u>Balancing Procedure</u>

Rotating assembly construction using spigot fits with tie bolts is widely used especially in the power generation industry. For commercial aero engines where rotor dynamics are a critical consideration, the use of a multi-spigoted long rotor stack is thought to be a potential risk to achieving an acceptable level of dynamic balance. Very careful measurement and recording of swash and run-out at each spigot feature is required, then appropriate positioning and assembly of each joint. Special hydraulic tooling may be needed to hold the assembly in place during the sequential build, and the torque tightening procedure of the tie bolts will have to be precisely controlled. Such precision through a stack of twenty-seven components - fifty-two spigot features, is considered challenging. In the worst case, high speed balancing may be required which, for such a large bladed rotating assembly, will dictate provision of a dedicated balance facility.

Maintainability Review report ref. 1001231 produced by EPRI refers to existing technology that may be utilised to dynamically balance the 'turbo machinery' without physical access to the rotating shaft – ref. page 3-13. This technology basically employs balance ring assemblies mounted permanently at balance planes on the shaft. A smart controller senses vibration and determines optimum balance correction. The balance ring then moves the weighted rotors to the proper positions electromagnetically, minimising vibration in seconds – ref. page C-1. Rolls-Royce has no experience of such a system but has in the past, when investigating automatic balance correction options, discounted them on the grounds of weight, complexity, and reliability.

<u>Dry Film Lubricants</u>

An oxide layer forms on the surface of titanium at room temperatures and this layer can cause wear and frettage between the blade root and disc circumferential groove mating surfaces. As such, it is normal practice to apply a dry film lubricant to the mating surface of the blade root fixing. The lubricant usually has a graphite or molybdenum base and its application is by controlled process specification.

The risk of self-welding of the blades to the disc if parent metal is exposed in a helium environment was also considered. The use of Dry Film Lubricants (DFLs) would be an obvious mitigation to this risk however there was some concern that the DFL could wear away, enter the gas stream and become a source of radioactive contamination. The use of DFL and this risk would be eliminated by the incorporation of blisks.

2.2.4 *Development Programme*

Technology development programmes of work in support of the mechanical design of the compressor would be limited, with only potential studies identified at the moment dependent on further, more detailed risk assessments:

- Helium embrittlement of titanium and steel.
- Self welding of materials in helium environment.

The OKBM TDP generally covers the mechanical requirements for the GT-MHR compressor. However, materials development should not be completely limited to the turbine materials as there is inadequate information for the titanium and steels which will be used in the compressor regarding prolonged exposure to a helium environment (including contaminants).

2.2.5 *Materials, Lifecycle, Maintenance and Operability*

Materials for the GT-MHR design have not been specified in detail but for the rotating assembly titanium should be used to minimise weight and EMB requirements.

Whilst analysis of specific components has not been done, maintenance interval and life requirements should be achievable, being comparable to those of industrial gas turbines. Maintenance or inspection while the compressor is in situ is not considered practical due to access and contamination issues and the compressor should be designed to be robust enough to fit and forget until the next shop visit.

Levels of radioactive contamination need to be considered by specialists to evaluate any limitations during overhaul.

2.2.6 *Technical Risk Assessment*

In summary, no major issues are envisaged for the compressors. The key risks are as follows:

- Maintaining acceptable radial clearances (especially following on from a catcher bearing deployment).
- Understanding the effect the high pressure helium environment has on compressor materials, particularly titanium.
- Issues with contamination. First, the contamination of the helium cycle with compressor released materials such as abradable materials in the tip and hub seals and dry film lubricants commonly used in blade roots. Second, the contamination of the compressor with radioactive substances – silver in particular can have a detrimental effect on titanium.

2.2.7 *Areas of Uncertainty / Issues for Further Study*

Improvement of Mechanical Design

By getting a better understanding of the optimum stage loading/stage number design could be reached by:

- Acquiring a better understanding of surge margin requirements and mechanical constraints.
- Performing 2D designs of the compressor to understand radial loading and to what extent diameter can be reduced.

Validation of Helium Compressor

Design codes and design criteria can be applied to this design having installed helium gas properties. The doubt though will still remain, does helium behave very differently to air is an axial compressor?

Two approaches can be applied:

- Either be cautious on the aerodynamic design,
- Or except that the first engine is have enough instrumentation to develop the compressor.

It is highly unlikely you could make a rig that would run at the correct inlet pressures and temperatures and on helium.

Helium Embrittlement

Hydrogen embrittlement of steel is a well-known problem, which produces a reduction in elasticity and possible sudden failure. Hydrogen is also highly soluble in titanium, causing hydrides and possible crack initiation sites.

Helium also has a high diffusion constant but the effect on diffusion into steel is not thought to be a problem. Information for diffusion into titanium is scarce. Impurities in helium, such as hydrogen, may cause increased diffusion depending upon quantity. Helium, as an inert gas, is used for shielding in plasma arc welding and weld material analysis shows no helium contamination however it is considered that material testing in a helium environment (including exposure time, temperature, and appropriate levels of impurities) is required to assess the possibility and extent of embrittlement and effect on material properties.

Radial Clearance in EMBs.

To finalise estimates for tip clearance for normal operating conditions and the possible need for incorporation of sacrificial rotor path linings for the EMB failure scenario, definitive EMB and catcher bearing air gap information is required. The tip clearance estimates are based on an assumed value of 0.75mm, which requires qualification. The failure modes and run on requirements are also critical to the tip clearance estimates and must be defined.

2.3 Turbine Aero/Mechanical Design

2.3.1 *System Description*

It is understood that the concept design of the OKBM turbine was undertaken by the Kuznetsov Scientific and Technical Complex at Samara. The turbine is a conventional axial design with nine stages of high hub-tip ratio, high aspect ratio blading. Pressure ratio is 2.68:1.

One notable feature of the design is the constant blade height (constant inner and outer hade lines). A design of type does not represent an optimum aerodynamic performance and it is an unusual choice. We are not aware of the reason for this feature, but we may suppose that there is a perceived advantage in assembly, manufacturing or cost. The implications of this design choice have been assessed below.

The rotor blades are shrouded and feature large axial gaps between stators and rotors. Both of these design features are consistent with the large axial movements that might be expected on a shaft of this length.

Overall the construction of the turbine (as with the other parts of the rotating machinery) is very robust and likely to be heavy compared to modern gas turbine design.

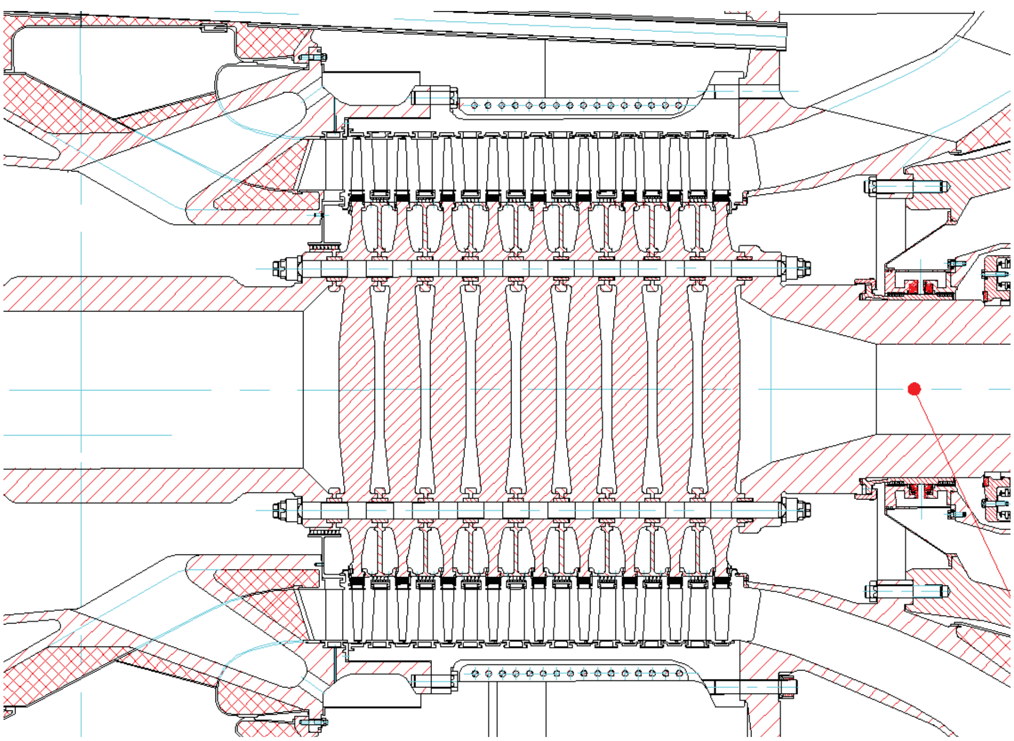


Figure 16 Baseline GT-MHR Turbine Design

2.3.2 *Aerodynamic Assessment of Design*

The assessment is based on a General Arrangement (GA) of the gas turbine (Reference 9) and the published cycle parameters (below). No information was available on the aerodynamic design of the turbine, or even the number of blades in each stage.

Table 5 OKBM/Samara Cycle Turbine Parameters

Parameter	Value
Mass Flow	$320 kgs^{-1}$
Shaft Speed	4400rpm
Inlet Total Pressure	7070kPa
Inlet Total Temperature	848°C
Outlet Total Pressure	2640kPa
Outlet Total Temperature	488°C

Hence the assessment of the design is in two parts. First, an initial view has been made on the turbine as a whole against the duty that it is intended to perform. Second a series of stage-by-stage turbine designs have been produced that match the geometric data available (number of stages, blade chord, blade height). Within these constraints a number of different 'styles' of aerodynamic design have been assessed.

The following conclusions have been made based on an initial view of the turbine as a whole:

- There are nine turbine stages, with an overall mean stage loading of 1.763. This indicates that the turbine is roughly the correct size and has the correct number of stages. Indeed, this stage loading may be considered to be conservative – seven or eight stages may be sufficient.

- The first stage inlet Mach number is very low (0.067) and the last stage exit axial Mach number is quite low (0.264). This is a consequence of the material properties of helium and suggests that high efficiency should be possible.

- The first stage flow coefficient is 0.387 and the last stage flow coefficient is 0.704. These numbers are both feasible (if low) for a helium turbine. However, the difference between the numbers suggests that the aerodynamic design has been compromised to fit the constant hade lines of the turbine. In other words, as the cross-sectional area of the turbine is fixed the helium is accelerating as it expands.

- Nozzle guide vane (NGV) and rotor Reynolds numbers are high ($\sim 1 \times 10^6$). This is outside of normal gas turbine experience but does not cause particular challenges.

Overall therefore the initial overview seems reasonable. However, the stage-by-stage design process has identified a number of areas of concern.

A series of different 'styles' of design have been considered. For simplicity, only two are discussed here:

- A 50% reaction design would typically be the most efficient as it minimises Mach number and losses throughout the turbine.

- An 'axial' design is a high impulse, low reaction design where the flow is turned parallel to the axis of the turbine after each stage. This style of blading is more common in steam turbine design.

The conclusions of this detailed design exercise are summarised below:

- The blade count is very high. For the 50% reaction design 147 rotor blades are required on the fourth stage. For the axial design this rises to 294. The high blade count increases part count and cost significantly. In addition the packaging of the blade roots is likely to be very difficult.

- The blade stress is likely to be very high and material selection for the blade very difficult.

- The rotor turning angles are low (<83 degrees), which is likely to lead to flat, thin rotors. This may result in low stiffness blades with unsuitable dynamic response, which would be compounded by the requirement to carry a shroud.

- The turbine is far from being choked with Mach numbers throughout the turbine less than 0.4. This is important, as it is standard practice to choke the flow within the turbine to control the flow and the rotor speed. It is therefore a feature of the OKBM design that the rotor speed must be actively controlled by the generator/starter.

- The axial chord and height data extracted from the GA is very similar between stages. Put together with the other aerodynamic data from this study there is a strong suspicion that the turbine has not been designed beyond an initial sizing study and that the GA is only an indicative sketch.

- Despite the concerns highlighted above a polytropic efficiency of 93% should be possible from this design if a modern aerodynamic blading design is used.

The analysis has produced a good understanding of the important features of the design and this allows us to make recommendations for how the OKBM design may be improved:

- The number of stages should be reduced to around 7. This increases the stage loading of the remaining stages, which improves the blade shape, and reduces weight.

- The axial chord of each stage should be increased to fill the axial gap created by removing stages. This reduces part count and improves the blade root and disk rim design.

- A curved inner and outer hade line should be introduced to match the expansion of the helium through the turbine. This allows the optimum flow coefficient to be selected for each stage.

- Using a radial turbine to replace two or three axial stages would reduce cost and improve the robustness of the design considerably. However, this concept has been rejected as the radial turbine would have to be very large (~3.5m diameter) and would be creep limited. It also does not fit within the current layout.

To illustrate these points a concept aerodynamic design has been generated that meets the aerodynamic requirements for the GT-MHR and fits within the existing design layout with minimal changes. This design is shown diagrammatically in Figure 17 (blue superimposed on OKBM design) and it features:

- Seven stages.
- First stage tip radius of 0.591m.
- Last stage tip radius of 0.926m.
- Reduced blade root stress (170MPa for first stage).
- Optimised work split and stage loading across all stages.

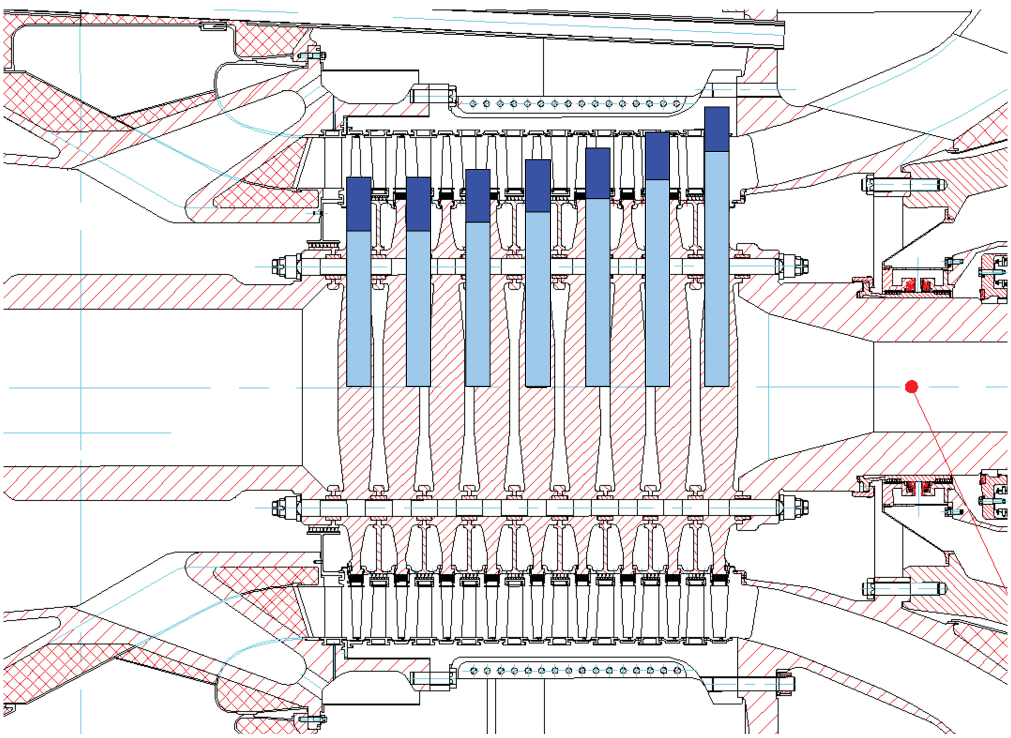


Figure 17 Revised Turbine Design

2.3.3 *Mechanical Assessment of Design*

The GT-MHR reference design has a 9 stage turbine operating at 850°C on a constant hubline. This design has been assessed mechanically, based on the assumed aerodynamic design discussed above. The conclusion is that the design solution is possible with refinement.

Creep life is considered to be the life limiting factor in the design. The materials investigated are typically those used in high pressure turbines in modern aero engines and in the power generation sector. The conclusion drawn is that to achieve 60 000 hours life with an entry temperature of 850ºC, the disks need to be cooled. HPC delivery gas requirement of approximately 0.4%/stage for this design layout and all stages of disk would be cooled. With refinement of the design, uncooled blades should achieve the required creep life, but the design may have to be shroudless for this to be possible – further work is needed to resolve this. The study has concluded that 850ºC is at the extreme limit of acceptability for an uncooled turbine blade solution to satisfy a creep life requirement of 60 000 hours.

For operation at up to 950ºC blade cooling and/or thermal barrier coatings will certainly be required because the limiting stress values decrease so much in going from 850ºC to 950ºC. It should be possible to design cooling for the turbine blades, but some design methods development will be required to account for the very high heat transfer rates in helium as compared to air. Development work will also be required to see whether aero-engine derived thermal barrier coatings are suitable for the helium environment.

Potential for weight reduction compared with the GT-MHR design exists by refining the design. This has to consider the optimised turbine solution in terms of speed, stages and blade count against material choice, blade profile, material selection and the benefits and practicality of hollow blade design.

Un-shrouded blades have considerable weight advantage, but require more conceptual design to achieve the required performance characteristics with tip clearance, turbine movement and control of the helium working gas.

The turbine mechanical design has been considered in two stages:

- Assessment of the current GA/OKBM datum.
- Assessment of an improved design based upon the GA/OKBM concept

<u>Assessment of the GA/OKBM datum design</u>

Two layouts have been identified:

- Using a straight hub line defined from initial concept.
- Using a conical hub line defined by turbine aero based on constant AN2 concept.

Assessment and design work completed to define potential blade layouts using aero/thermodynamic data and blade data.

Table 6 provides information on the materials considered for the turbine based on Rolls-Royce information. Although these materials well recognised in the Aero Propulsion sector, CMSX4 (advanced single-crystal Ni-based) turbine blade materials are also widely recognised within the Power Generation turbine sector, and hence have been applied as the most appropriate solutions related to this type of technology.

Table 6 Commonly Used Disc & Blade Materials (RR Database)

Application	MSRR number	Alpha Code	Common name
Disc	MSRR7090	QXZ	Inco 718
Disc	MSRR7250	RCZ	Waspalloy - non EB Weld
Disc	MSRR7220	QSQ	Waspalloy
Disc	MSRR7090	QFP	Inco 718
Blade	MSRR7248	RCY	CMSX4
Blade	MSRR7257	RFA	CMSX4
Blade	MSRR7215	QNF	Inco 738
Blade	MSRR7048	QDC	713 LC
Blade	MSRR7150	QPD	MAR-M002

The initial datum straight hub line design has been assessed as having the following basic layout covering the blade numbers, profile and comparisons of the based as shown in the various charts in Figure 18.

Straight hub line, Temperature 850°C, 4400rpm, 9 Stage Turbine, Aerofoil layout

50% Reaction

	Number off aerofoils Stator	Number off aerofoils Rotor	For PPD Hub translation (Pitching) Stator	For PPD Hub translation (Pitching) Rotor
Stage 1	91	132	45.57	31.42
Stage 2	120	138	34.56	30.05
Stage 3	126	145	32.91	28.60
Stage 4	134	147	30.95	28.21
Stage 5	127	144	32.65	28.80
Stage 6	135	144	30.72	28.80
Stage 7	130	135	31.90	30.72
Stage 8	138	141	[illegible]	29.41
Stage 9	147	121	28.21	31.27

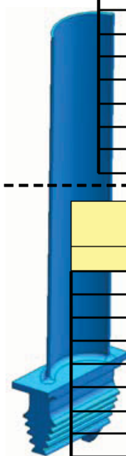
STAGE 4
50% REACTION BLADE

STAGE 1
CONICAL ANNULUS BLADE

Axial

Largest
reaction

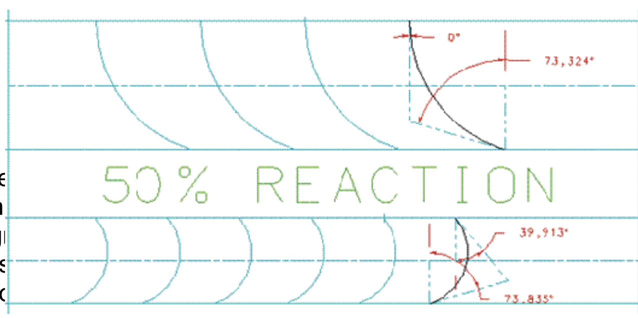
Fillet radius shown 3mm



176 off

Fillet radius shown 2.5mm
LE = 0.6mm
TE = 1.1mm (nominal)
Platform width = 18.3mm
Using modified existing firtree for comparison only

	Number off aerofoils Stator	Number off aerofoils Rotor	For PPD Hub translation (Pitching) Stator	For PPD Hub translation (Pitching) Rotor
Stage 1	59	272	78.24	15.25
Stage 2	70	282	59.24	14.71
Stage 3	74	293	56.04	14.15
Stage 4	79	294	52.49	14.11
Stage 5	76	284	54.58	14.60
Stage 6	81	278	51.20	14.92
Stage 7	78	257	53.17	16.14
Stage 8	84	263	49.37	15.77
Stage 9	89	220	46.59	18.85



Design

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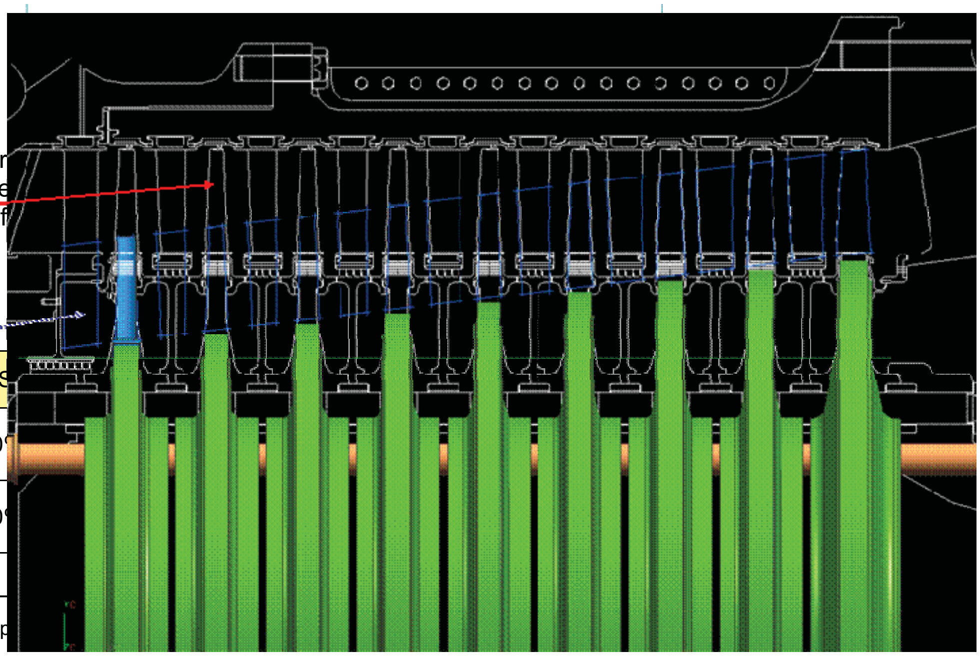


The initial str
at 100% spee
a 0.1% proof

Straight
Hub line

Conical
Hub line

S
100
120

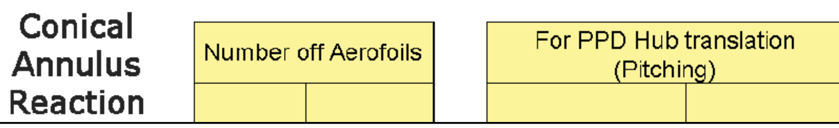
140% Speed	Assumed max speed before disc burst	339MPa

Note that the disc post region would typically have a similar level of stress to the blade shank through the mean cross sectional area. The disc post additionally carries the CF load of the blade root section and the top half of the disc post itself.

The mean stresses from the above table show that a typical proof RF for the disc post would be 1.8 at the design condition and 1.3 at the overspeed condition.

Comparisons of the blades within this arrangement are shown in Table 8 and the turbine blade assembly in Figure 20.

Table 8 Datum Concept, Conical Hub Line Turbine Blade Arrangement



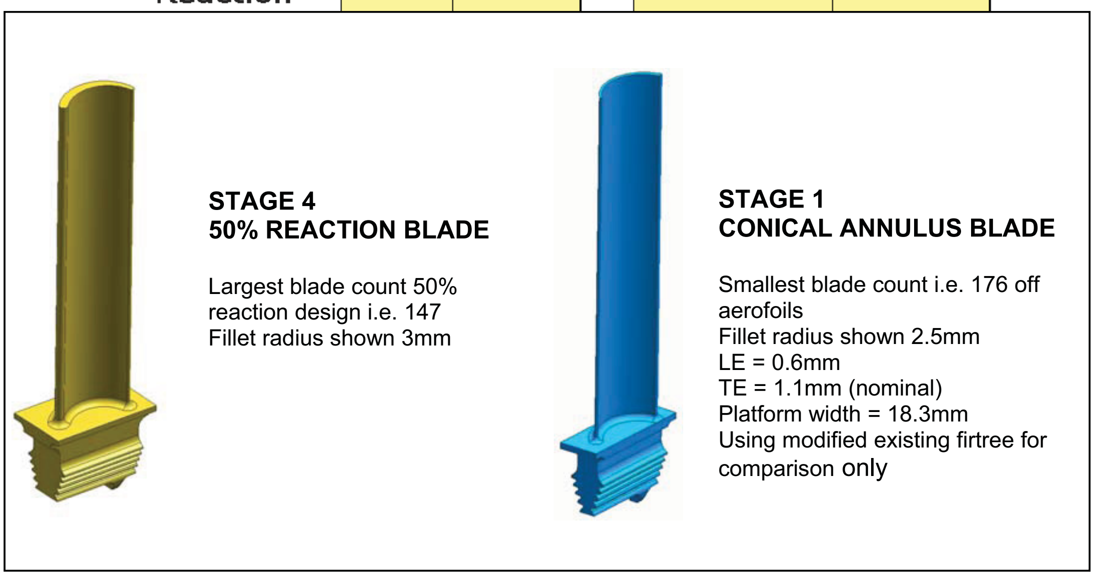


Figure 20 Turbine Blade Comparison

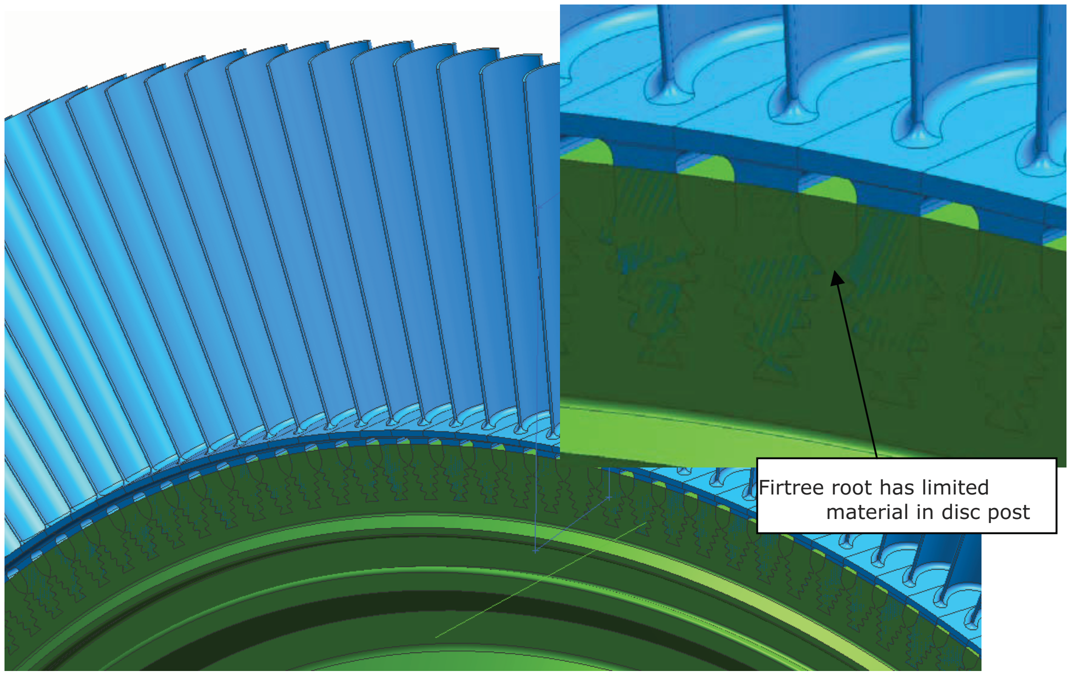


Figure 21 Stage 1 Turbine Blade Assembly

Both the straight hub line and conical turbine layouts immediately generate concerns associated with the blade mounting. Using a typical fir tree arrangement the material available at the platform on the outer disc would not be acceptable under the loads and temperatures envisaged under these operating conditions (Figure 21).

Ease of removal of these blades during any repair and overhaul, or replacement, process has to be considered in relation to the procedures that may need to be applied for any radiation-contaminated components.

Operating with helium as the working gas generates a range of specific design characteristics for the turbine that have been identified in the system description for the turbine. These material requirements will be common for the datum GA/OKBM design or any derivatives that may be offered on this concept.

From the mechanical context the selection of the materials is a critical element of any of the turbine designs. The materials requirements for turbine require high temperature alloys, to accept combined requirements of CF loads, creep life at the thermal and RPM conditions proposed.

Preferred materials:

- Disc - Inco 718 (QFP), Waspalloy (RCZ) also a candidate, option of FV 535 Steel based disc considered but discounted due to the temperature and load constraints relative to the requirements.

- Blades - CMSX4 single crystal (RFA/RCY).

- Blades at 850ºC may require cooling, and thermal barrier coatings, depending on the design options for the turbine, and the scope in the concept for increases in temperature to achieve higher levels of efficiency.

- Coating would be preferable with cooled blades to achieve creep and efficiency requirements.

Consideration of the Datum design and these derivatives has to be given against the proposed creep life. It is proposed that a creep life of 60 000 hours will be used. Under a continuous 24 hour/7 day week operation this equates to 6.8 years. For the materials identified for the disc and blades the following relationships as shown in Figure 22.

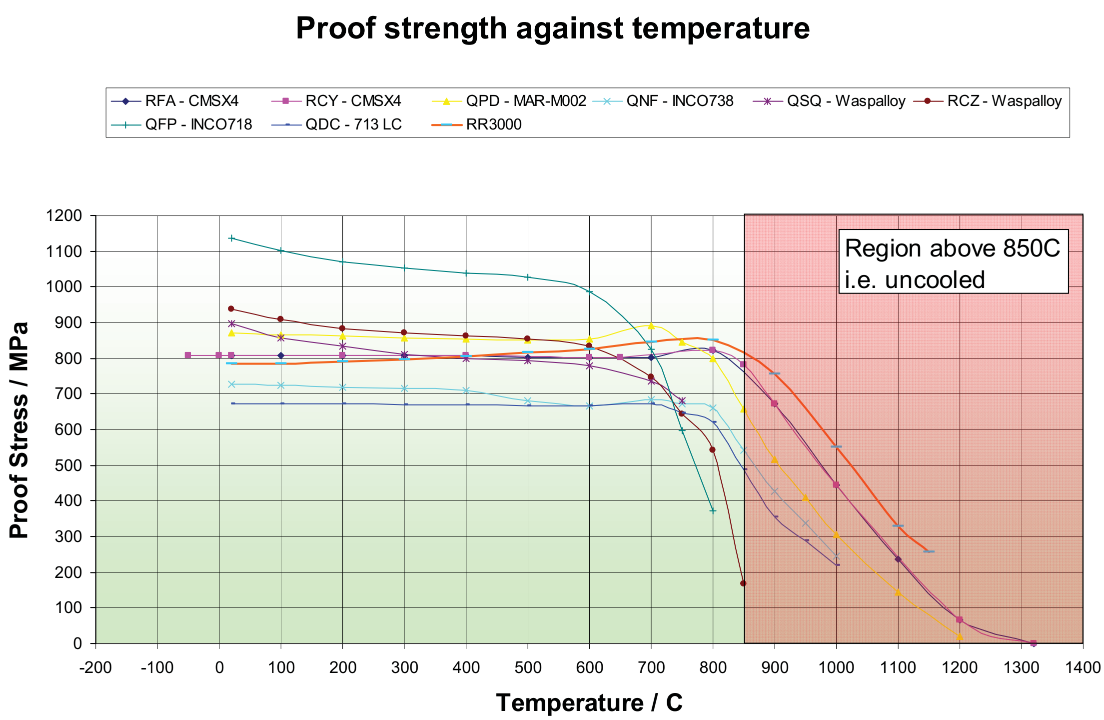


Figure 22 Turbine Disc and Blade Material Proof Stress Properties

From the characteristics shown in Figure 23, at 850°C blade temperature a stress of 200MPa is the limit to be set as a design constraint if a creep life of 60 000 hours is to be achievable.

A preliminary thermal model of the bladed disc assembly with only disc cooling, shows that with a minimum amount of cooling fluid circulated through the turbine bore, the majority of the disc sections can be maintained within 5 degrees of the coolant temperature. The aerofoil and platforms have such a large area in contact with the fluid that unless they are cooled with internal passages their temperature will be within 5 degrees of the gas path temperature. The disc post and blade root regions form a conduction path along which the temperature varies between these two extremes.

The resulting disc post temperature at the outer circumference of the disc should have a temperature of between 550°C and 650°C. Figure 19 below shows the allowable stresses to satisfy the creep life of 60 000 hours, which for Inco 718 (QFP) is 520MPa down to 120MPa for this range of temperatures. The equivalent stress range for using the Waspalloy (RGZ) material is 500MPa down to 220MPa. In both cases a temperature of around 550°C should be the design target temperature.

Figure 23 also clearly shows that an uncooled disc design, which would have a temperature everywhere almost equal to the gas path temperature of 850°C, is not an option due to failure via the creep mechanism.

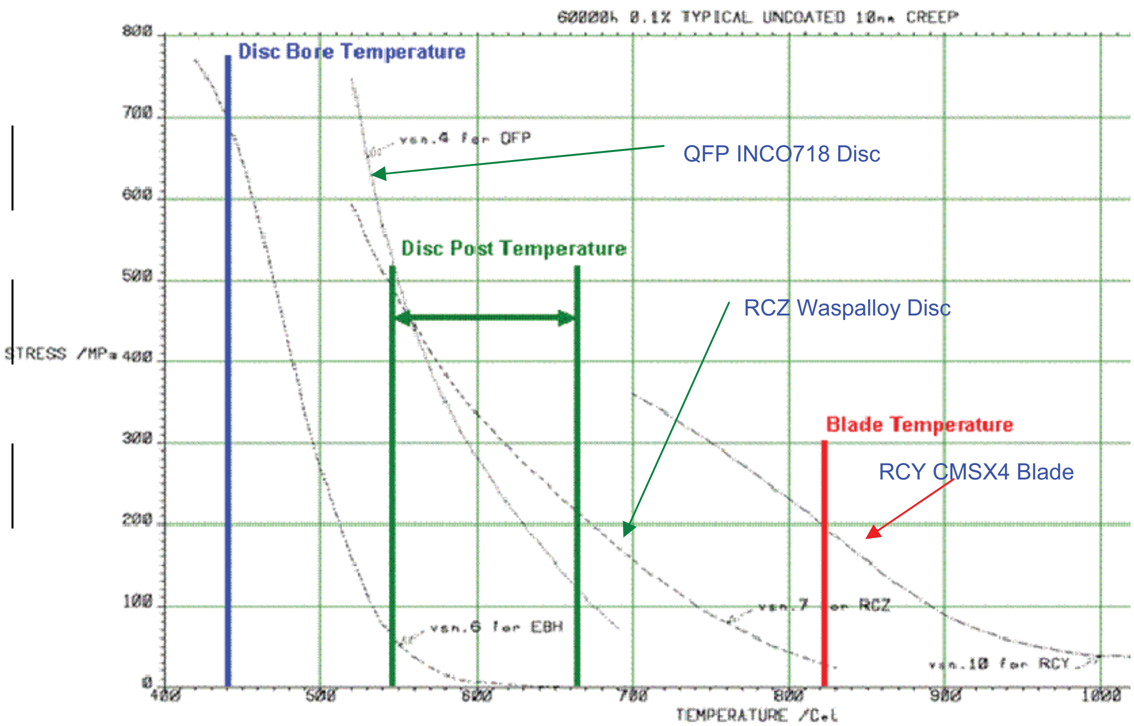


Figure 23 Creep Characteristics Against Temperature

Relating these generic design considerations to the datum GA/OKBM design as received, show that in an uncooled state of 850°C, disc bore stresses in the region of 380MPa at 100% RPM, and peak in-plane stresses of 440MPa, see Figure 24.

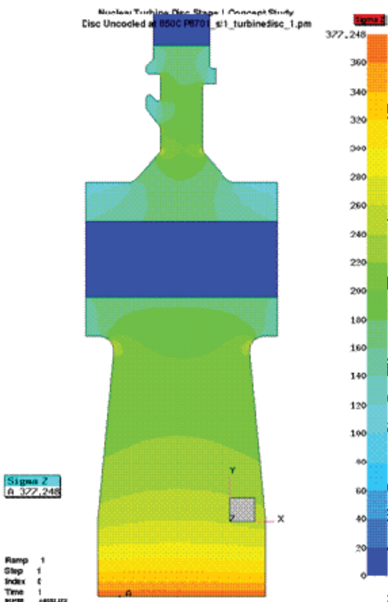
The in-plane stresses are in localised areas and could easily be reduced by the optimisation of the design. The disc is overly massive for the blade CF requirement, with the disc itself contributing excessively to the high stress levels.

However the overall situation is clear when assessed against Figure 23 that a cooled disc is recommended to achieve the required life.

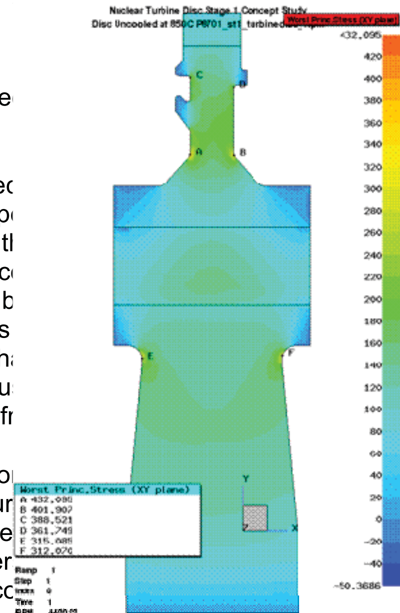
100% Speed Condition – Uncooled @850C

Figure 24: Preliminary Stress Calculations for Datum Arrangement Turbine Disc

Hoop Stress



Worst In Plane (XY) Stress, Swxy



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have to be considered. Efficiency may be affected very marginally by the bleeding off of HP compressor helium, although with the use of cooled thermal barrier coated blades this in theory can be recovered by increasing the temperature from the reactor slightly. The result of taking a higher technology solution in overall terms though is an increase in complexity of the design, and an associated increase in cost.

An alternative option would be a solution that had a reduced life, or blade and disc loads and hence allowed uncooled blades and reduced cost.

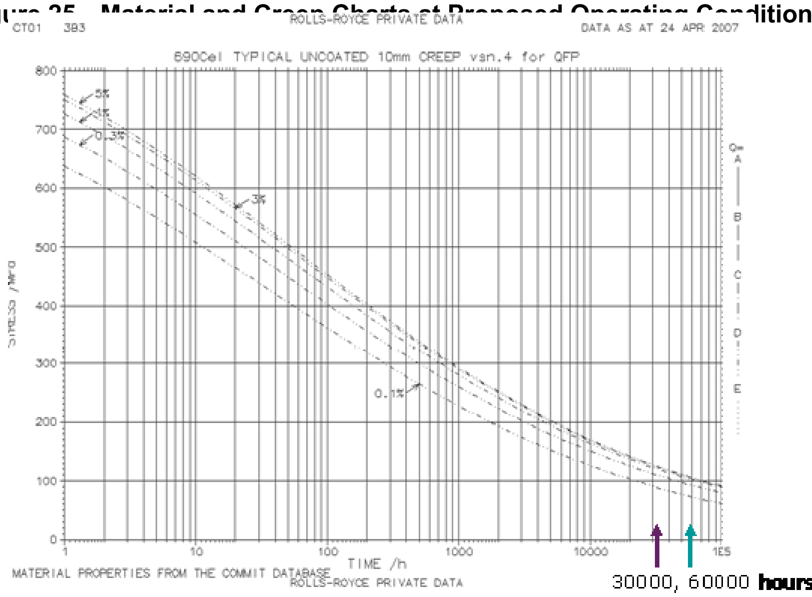
Consideration therefore should be given to a reduced creep life. An option exists to consider the removal of the GT Power Conversion Unit during a reactor refuelling shutdown. Although Figure 25 shows the stresses allowed do not vary greatly it does allow some margin, and critically removes the need for more complex design solutions and the associated development and production costs if this option were to be pursued.

At the temperatures proposed here, the trade off required between uncooled and cooled blades, shrouded and un-shrouded turbine blades are significant considerations in achieving the target turbine creep life. It should also be recognised that, in order to achieve the creep life of 60 000 hours, a margin of safety should be applied.

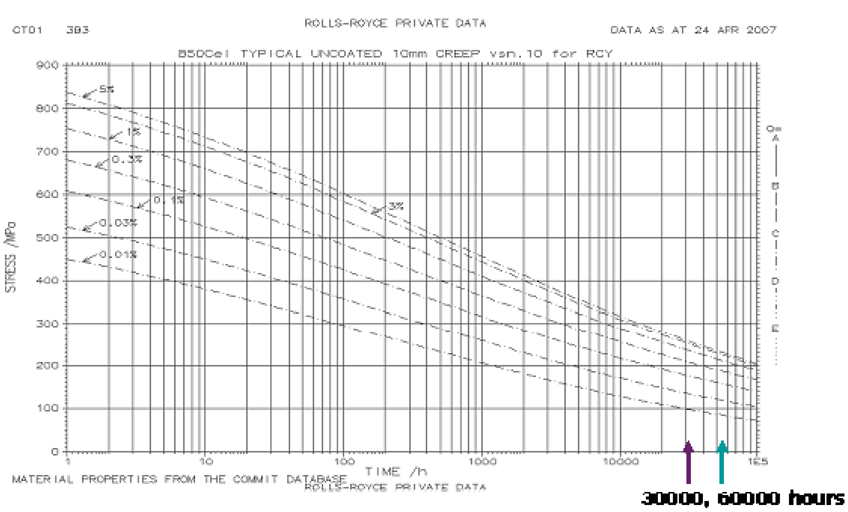
To achieve the creep life requires the stresses at the blade shank, which is typically the area of greatest concentration, to be minimised to <200MPa. This requires consideration of the shaft speed, blade mass, and the diameter of the turbine itself. These have to be optimised to achieve the required aerodynamic and thermodynamic performance to achieve the overall efficiency for the PCU, combined with the required mechanical performance.

QFP Inco 718 Turbine Disc material, uncooled, 690ºC.

Figure 25 – Material and Creep Charts at Proposed Operating Conditions



RCY CMSX4 Turbine Blade material, uncooled, 850ºC.



Conclusions Against Mechanical Concept for Turbine

The overall OKBM design solution is judged to be possible, especially if refined.

Creep life is considered achievable with blade and disc cooling, and blade thermal barrier coating, but requires optimisation through the follow-on programme phases. HPC delivery gas requirement is approximately 0.4%/stage for this design layout and all stages would be cooled.

There is potential for weight reduction as part of the turbine design optimisation. This has to consider the optimised turbine solution in terms of speed, stages and blade count against material choice, blade profile, material selection and the benefits and practicality of hollow blade design.

Unshrouded blades have considerable weight advantage, but require more conceptual design work to achieve the required performance characteristics with tip clearance, turbine movement and control of the helium working gas.

An uncooled blade unshrouded offers a more simple mechanical solution, however from the assessments here of a seven stage turbine it will only achieve anything approaching the required life at 850°C with lower shaft speeds e.g. 440rpm or 360rpm. Hence this requires further work to create an optimised solution between shaft speed, turbine stages and layout that approaches the life with the typical expected margin, or a cooled turbine approach will need to be considered as part of an overall trade off process as the design is developed. Creep life is considered achievable with blade and disc cooling, and blade thermal barrier coating at 5000rpm with the five stage turbine concept, but requires optimisation through the follow on programme phases.

Option for operation at up to 950ºC may be possible with further design and development work to optimise cycle and turbine design with the cooled blade and thermal barrier coated solution.

2.3.4 *Development Programme*

A development programme to verify any models of the turbine in helium to confirm the efficiency at this temperature would be appropriate, and verify the characteristics of helium, in relation to the design, in order to optimise the design for this environment. This would require an iteration of modelling design, development and physical test rigs work.

Consideration of a rig programme that verifies the turbine performance in He and provides some stress/thermal load and initial life usage information should be included, at the operating temperature. This should be completed without using a nuclear plant as the heat source, in order to allow refinement of the design, and physical assessment of the material condition and life usage without radiation issues.

The issue of the radioactivity and means of managing the gas turbine, PCU and the turbine replacement should be considered further with more information on the severity of the radiation, the radioactive decay, and safe working practices (see Section 2.9). Applicable techniques and technology for handling such hardware exists in the nuclear industry, hence a technology transfer as opposed to new technology programme is envisaged. An outline procedure for the removal of the turbomachinery as a radioactive component has been outlined by GA, and further study of this and the subsequent maintenance/replacement/reinstallation operations is needed. This study will also need to consider the safe working practices, shielding and the management and disposal of the PCU and its components, and the associated equipment needed, which may include automation and robotics.

2.3.5 *Materials, Lifecycle, Maintenance and Operability*

Material choice is critical if the turbine life target of 60 000 hours is to be achieved. High-grade steels are likely to be at the edge of their operating limits, hence typically an alloy such as Inco 718 or Waspalloy, with cooling would be required for the disc. Other non-Aerospace materials can also be considered.

Based on knowledge of Aerospace materials the requirement for the blades fits on the boundary between a cooled option and an uncooled blade. An uncooled solution simplifies the overall solution and associated cost, but will have to be developed as part of the effort to optimise the turbine performance with the number of stages, blade count, rotational speed and the blade and disc layout. The use of single crystal type materials and Aerospace alloys such as CMSX4 would allow this option to be developed.

Any move towards 950ºC as the operating temperature would require a more complex solution with cooling and possibly a thermal barrier coating on top of the single crystal material, assuming the 60 000 hour turbine life remains a target, with the materials identified. This obviously has implications on design and development costs as a result of the increased complexity of design and materials, the verification needed to ensure the concept is reliable under the helium based operating conditions and temperatures, but also will have implications on the production unit costs.

Consideration is needed also on the benefits of turbine coatings. Thermal barrier coatings (TBCs) have been identified as a possible means of controlling turbine blade temperatures when used with cooled blades. Sulzer-Metco 204 has been assessed for this, but the bond coatings that support the TBC have other benefits in protecting the alloys against corrosion or materials that may attack the physical structure of the blade, which will need consideration.

2.3.6 *Technical Risk Assessment*

A risk assessment has been completed considering the turbine in normal and abnormal operating conditions and under maintenance repair and overhaul. The summary is shown in Table 9.

Table 9 Turbine Technical Risk Summary

Normal Operation Risk	Mitigation
Can a design be achieved with the temperature and specified life requirements for 60 000 hours, initially at 850°C (with the potential for 950°C later), with helium, and efficiencies required	1. Adjust cycle temperature or life requirement. 2. Design for requirements, manage thermal and CF loads within stage temperatures
Aerodynamic design may not be optimised. *Comment: Helium has a high specific heat capacity, gas constant and ratio of specific heats compared with air but, as an ideal gas, traditional design methods should be easily applied. Obtaining reliable efficiency numbers due to the high Reynolds number, high gamma etc may result in a sub-optimal aerodynamic design solution.*	Low risk, mitigate by: 1. Survey of existing data for working fluids in relevant regimes (e.g. steam and ammonia systems). 2. Develop a helium test programme to gather relevant efficiency data.

Helium is likely to have an increased risk of generating embrittlement of blades, and silver can affect nickel-based alloys. *Comment: Further assessment is needed on the effect of helium on corrosion, and the effect of silver (Ag110) in the quantities likely, and cobalt and tellurium on the alloy components, and the need for coating.*	1. Consideration with TBC and yttrium-based bonding coat to protect against helium. 2. Single crystal blade should reduce impact of silver on crystal boundaries. 3. Assessment of the turbine materials in high temperature helium.
Abnormal Event Risk	Mitigation
Over-speed, by other than failure of shafts *Comment: The interaction between the mechanical design and the control system is an essential feature in the development programme, combined with mechanical brakes, and a means of dumping He from the turbine, to stop turbine overspeed in such an event. The turbine design will also need some margin to handle these events, but this will need to be optimised with the other mass and load constraints.*	1. Not considered a major risk, assuming control system is effective. 2. Emergency mechanical brake and control system to reduce speed by dump of helium on this event to a bypass area. 3. Design turbine up to 120% as survivable one off occurrence limit
Over-speed by shaft failure, (turbine to generator shaft failure worst expected case). Over-speed due to dramatic reduction of load.	1. Increase shaft strength so event cannot occur. 2. Emergency mechanical brake and control system to reduce speed by dump of helium on this event to a bypass area. 3. Design turbine up to 120% as survivable one off occurrence limit.
EM Bearing failure	Bearing failure will almost certainly cause unbalance, and failure of the turbine. Design for containment as far as possible. Likely that damage to extend to all areas of turbo-machinery.
Over-temperature	Considered unlikely on the basis that the nuclear reactor will have extensive temperature monitoring and control.
Blade failure, from manufacture defect, creep or possibly HCF. LCF is not considered a risk in this usage scenario.	1. Design of blade to allow safety margin within overall thermal and mechanical design of turbine. 2. Design of disc post to allow single blade off without unzipping the turbine. Note that position of hot gas entry from nuclear reactor for hot gas, could mean uncontained blade could pass to reactor
Maintenance, Repair and Overhaul Risk	Mitigation
Radiation and turbine overhaul or component replacement activities (as defined in Section 2.9).	A manageable risk, given suitable safeguards and engineering.

2.3.7 *Costs*

Costs associated with turbine will be defined by the complexity of the solution to achieve the life and efficiency requirements. If the solution can be achieved without cooling this will dramatically reduce the cost associated with both the development process and the parts production cost, whilst retaining efficiency as a result of not ducting He for cooling.

If the solution moves to higher temperatures, then the need for turbine cooling increases and hence the associated cost, however this option has to be considered against the benefits of higher operating temperatures and efficiencies.

A further option that should be considered is the cost benefits of having a lower life turbine. An option could exist of GT removal when the nuclear fuel is changed. This may allow lower cost materials to be used, and would be a more ‘disposable ‘ solution.

2.3.8 *Areas of Uncertainty / Issues for Further Study*

See comments in previous sections.

To achieve the required life at 850ºC requires blade stresses of <200MPa as the design criteria. This should be achievable.

Opportunities for weight saving through reducing the number of stages (see above).

Management of the radioactivity through processes for gas turbine removal within the plant, and storage, decontamination processes for safe turbine replacement.

2.4 Electrical Generator

2.4.1 *System Description*

A closed-cycle helium gas turbine cycle produces shaft power to drive an electrical generator. The turbo-generator rotates at a speed of about 4400rpm to match the mechanical characteristics of the turbine. The proposed generator is a two-pole synchronous generator with brushless excitation. The rotor will carry a DC winding embedded in slots in the generator rotor. The DC winding will be supplied by a brushless excitation system comprising an AC exciter generator and a rotating rectifier, both on the same shaft as the generator rotor.

The generator output will be 20kV 10kA output at 73.3Hz, corresponding to 4400rpm. A generator transformer will raise the voltage to that required by the utility that the GT-MHR will supply, and a power electronic frequency converter will convert the generator output frequency to the frequency required by the utility system.

The PCU turbo-machinery should be designed for 60 000 hours (about 7 years) between overhauls, whilst most of the rest of the PCU has a target design life of 60 years of operation.

To minimise penetrations of the reactor pressure boundary, the whole of the helium gas turbine and also the generator will be housed in the reactor pressure vessel, with electrical connections made through the RPV. The gas turbine and generator are housed in the power conversion unit (PCU) in the RPV.

2.4.2 *Assessment of Design*

The main challenges are the high speed of the turbo-generator and the location of the generator in a high-temperature high-pressure helium atmosphere. The 4400rpm speed is much higher than most designs of large generators, which are typically rated at 3600rpm (60Hz output). The helium atmosphere, at 2.6MPa, is much denser than the hydrogen atmosphere, typically 0.3MPa, used in most large generators. The GT-MHR design favours a two-pole synchronous generator with brushless excitation similar in many ways to existing power station generators. A power electronic frequency converter will convert the output frequency (73.3Hz) to the 50 or 60Hz frequency of the utility that the generator will supply.

Key aspects of the proposed design are discussed in the following subsections.

2.4.3 *Ionising Radiation*

Although the reactor core and fuel elements are designed to reduce the risk of radioactive contamination of the helium gas, some contamination may occur. This presents three main electrical risks:

- Ionisation of insulation (helium and the solid insulation of the generator windings). This will encourage electrical breakdown.

- Degradation of the solid insulation of the generator windings.

- 'Metal plating' of electrical insulation. Radioactive daughter products may escape from the fuel as gases (generally radioactive, such as silver 110) that may deposit on insulating surfaces aided by the electric field in the vicinity. Significant deposition could form conductive paths leading to catastrophic failure of the insulation.

The OKBM design manages this risk by locating the generator within a radiologically clean pressurised enclosure that is isolated from the primary helium coolant by a buffer seal. The helium used to cool the generator is fed from the helium clean-up circuit and is therefore essentially uncontaminated. A slight overpressure ensures that the gas flow across the buffer seal is from the generator cavity into the turbomachinery section.

2.4.4 *High Temperatures Decomposing the Electrical Insulation*

Most common insulating materials used to wind electrical machines degrade significantly if the temperature rises above 150ºC. Therefore, the electrical insulation is at risk from the high temperatures of the helium working fluid.

The OKBM design will manage this risk by forced helium cooling which rejects waste heat to an intermediate water cooling loop. The generator will be in helium cooled to 40ºC and the design temperature of the generator winding will be between 110-115ºC. The generator helium cooler is located in the generator cavity, in the annular space between the PCU vessel and the generator stator.

The electrical insulation will be a monolithic design manufactured by Electrosila in St. Petersburg, Russia. However, the partners in the GT-MHR project will need to confirm that satisfactory quality control is applied to the materials and manufacturing processes used by Electrosila. The insulation is not expected to release volatile chemicals that will attack other components, and helium, being an inert gas, is not expected to attack other components either. The risk of any attack by helium on the generator components being aggravated by ionising radiation is believed to be negligible. This is because helium is an inert gas, radiation will be kept low, and temperatures in the generator will not be unduly high.

The generator/turbomachinery buffer seal will offer some protection against the high helium temperatures in the rest of the PCU. Also, the turbine drive shaft is hollow and there will be a thermal barrier in the coupling. These design features will reduce heat being conducted along the shaft to the generator.

GA and OKBM have proposed building an experimental generator to study helium cooling.

2.4.5 *The Electrical Insulation of Helium*

OKBM has experience of helium cooled generators in the range 500-800MW. Therefore, the OKBM TDP will design the generator according to previous experience with helium rather than adapting an air-cooled or hydrogen-cooled design and scaling the design according to the properties of helium.

However, it is necessary to consider reduced helium pressure. There are two main effects. One is the reduced electrical insulating properties of helium, which must be considered in the generator design. The other is the consequence of rapid depressurisation (the 'bends'). During normal operation at working pressure, helium will diffuse into the electrical insulation and other components. Any subsequent sudden rapid depressurisation would cause the helium to expand, possibly causing damage to the electrical insulation. The GT-MHR generator operates within the PCU vessel at a nominal pressure of 2.6MPa. However, this may rise to 11.6MPa during pressure testing. During starting, the pressure will be about 0.3MPa. The rated voltage of the generator is 20kV. If the helium pressure falls below 0.3MPa, the working voltage will be reduced to 3kV to allow for the reduced electrical breakdown strength at that pressure.

The OKBM TDP will manage the risks presented by helium by carrying out laboratory tests on samples of the proposed insulation. These tests would compare the electrical breakdown stress in helium before and after helium depressurisation.

2.4.6 *Generator and Exciter Current Lead Outs*

<u>Pressure withstand and helium leak tightness</u>

The proposed electrical rating of the generator lead-outs is 20kV and 10kA. The general construction of each lead-out will be co-axial with a circular penetration through the PCU vessel by a cylindrical ceramic insulator around a central conductor.

OKBM have experience with ceramic insulation on electrical pumping equipment and this technology is tried and tested. Strong ceramic resists pressure up to 20MPa (well above the pressures during normal operation, 2.6MPa, and pressure testing, 11.6MPa). The helium cooling of the generator will also help the cooling of the current lead-outs. OKBM are therefore confident that this technology will maintain satisfactory helium leak tightness and differential thermal expansion. The OKBM TDP will carry out laboratory tests on samples of the proposed insulation. These tests will simulate nominal conditions of voltage, current, temperature and pressure (including sudden depressurisation).

OKBM do not see any advantages of alternative technologies such as glass or elastomeric insulation.

Eddy current and magnetic hysterisis losses

The proposed electrical rating is 20kV and 10kA. The magnetic induction produced by such electric currents could present a significant risk of undue electromagnetic losses.

The main risk is magnetic hysterisis losses and this risk will be managed by non-magnetic concentric inserts around the lead-outs. Losses could be reduced still further by installing the lead-outs in concentric or trefoil installation.

2.4.7 *High Speed Operation*

There are four main challenges presented by the high speed design: centrifugal forces, windage losses, flow-induced vibration and frequency conversion.

Centrifugal Forces

The most serious challenge is the strong centrifugal forces, which increase with the square of the speed. Already this presents a challenge to existing designs of power station generators, which run at 3000rpm (3600rpm on 60Hz systems). This will require the generator rotor to be made from high tensile material. Usually steel is used, partly because of its magnetic properties, but principally due to its mechanical properties. Typical steels are alloys of nickel, chromium, vanadium and molybdenum.

Windage Losses

Previous studies have identified windage losses as significant (Reference 10). The windage losses are estimated from extrapolation of data from conventional synchronous generator designs. In a 500MW 3000rpm power station generator cooled by hydrogen at about 0.3MPa, the mechanical losses are as follows: bearings, 750kW; hydrogen seal, 150kW; and windage, 190kW. The windage losses are about 0.04% of the generator rating.

In the case of the GT-MHR, windage losses are aggravated by the conditions in the PCU as follows:

- Windage losses are proportional to gas pressure since pressure increases the density of the gas. The pressure of the helium gas is 2.6MPa, 8.67 times the pressure of hydrogen used in the 500MW generator previously described. Therefore the windage losses increase by 8.67 times due to pressure alone.

- Helium is twice the density of hydrogen at the same temperature and pressure. This doubles the windage losses since windage losses are proportional to the density of the gas.

- The higher angular velocity of the generator. Windage losses are proportional to the cube of the peripheral speed and so an increase in speed from 3000rpm to 4400rpm increases the losses accordingly. However, the radius of the generator rotor will have to be reduced inversely in proportion to the square of the speed in order to manage the centrifugal forces. The combination of increased angular velocity peripheral but reduced rotor diameter means that the peripheral speed varies inversely with angular velocity and so the windage losses will be vary inversely with the cube of the angular speed. Therefore, the losses will reduce by 0.317 due to the increase in speed.

Therefore, the windage losses in the proposed generator are estimated to be:

0.04% x 8.67 x 2 x 0.317 = 0.22% of the generator output

This is not an unduly large loss but it does represent a loss of output, and therefore revenue. Also, these losses will require helium coolers that are larger than hydrogen coolers of conventional hydrogen cooled generators of similar ratings.

Flow-Induced Vibration

The density of the helium flow is much greater than that of hydrogen in conventional generators. Previous studies (Reference 10) have identified this as another serious challenge facing the generator design. This may require theoretical studies and experimental tests on critical components to determine risk presented by flow-induced vibration.

2.4.8 *Frequency Conversion*

The Need for Frequency Conversion

A power electronic frequency converter will be needed to convert the output of the generator, 73.3Hz (corresponding to 4400rpm) to 50 or 60Hz system frequency. The power electronic converter is regarded as a low risk item as it will follow existing design principles. For example, converters of similar rating to that proposed are used in sub-sea interconnections (e.g. the England-France DC link), where converters of 500MW are used: well above the 286MW rating proposed for the GT-MHR. The cost will be about $50M and the converter, plus switchgear and filtering, and the converter will occupy about 2500 m^2 of land.

Typical converter efficiencies are 98% or better (Reference 11). However, losses in the converter reduce the overall efficiency of the power plant and this must be included in any analysis of otherwise higher cycle efficiencies that might result from higher speed operation.

The OKBM TDP does not address frequency conversion and power transformation. This is probably because frequency conversion and power transformation lie outside the scope of study. Although the OKBM TDP does not state this explicitly, it is reasonable to suppose this to be so since frequency converters and power transformers are not integral parts of the GT-MHR and can be supplied separately by other manufacturers.

Previous studies (Reference 10) have identified the following risks: harmonics, current rating and fault contribution. These risks may be managed as follows:

Harmonics

These can be controlled by existing technology. Advanced electronic switching can reduce the generation of harmonics and existing technology is well able to control harmonics.

Current and Voltage Rating

The generator will use a generator transformer and the frequency converter will be connected to the high voltage winding of that transformer. This is more economic than a frequency converter connected between the generator and the generator transformer since this design would involve much higher currents. However, the generator transformer will have to be designed for 73.3Hz instead of 50 or 60Hz as used in utility systems. A 73.3Hz design is regarded as low risk since 73.3Hz is only 22% higher than 60Hz and existing 60Hz design principles can be used with minor modifications.

<u>Low Fault Current Contribution</u>

Most converters often have a current limiting feature. Ironically, this can result in not enough fault current being available to operate overcurrent electrical protection reliably, if at all. The solution is to use alternative designs of protection such as voltage controlled overcurrent protection, distance protection or current comparison protection (Reference 12).

<u>Motor Generator Set</u>

Motor generator sets offer an alternative to frequency conversion. However, efficiencies are lower than power electronic converters and reliability is poorer. A typical motor generator rated at 286MW would have an efficiency of about 97% and the losses would therefore be about 1% more than that of an electronic frequency converter. However, the capital costs would be much less: about $12M.

Motor-generator sets for frequency changing work best if there is a simple ratio between the frequencies. Therefore, the turbine generator speed should be raised to 4500rpm to produce an output of 75Hz, which is in the ratio of 5:4 for 60Hz and 3:2 for 50Hz.

2.4.9 *Generator Starting*

The generator will be used as a starter motor to rotate the turbine compressor to pressurise the helium and circulate it through the reactor. The frequency converter normally used to convert the generator output to the utility frequency will be used in reverse to supply power to the generator.

Utility converters used in DC transmission are often designed for bi-directional power flow and operated accordingly, and so the frequency converter proposed for the GT-MHR could be designed similarly. The risk presented by this method of starting is low, however a bi-directional power converter would be more expensive than a unidirectional frequency converter. An alternative to using a bi-directional power converter is to install an additional frequency converter for starting only. This method of starting is similar to that already used in natural gas fired CCGT power stations.

2.4.10 *Vertical Arrangement*

The shaft of the turbo-generator will be vertical rather than the horizontal arrangement used in most power plants. This does not present undue problems to the generator as such, since many hydro-generators are installed in a similar manner.

Existing generator designs wedge the generator rotor and stator windings tightly in place and this will offer the necessary support to the windings despite the generator being in a vertical position. The OKBM TDP will seek advice from experienced generator manufacturers to manage the risks presented by the vertical arrangement.

The estimated weight of the generator rotor is 15 tonnes. This assumes that the rotor diameter will be less than that of 3000 and 3600rpm designs in order to manage the centrifugal forces.

2.4.11 *Magnetic Bearings*

The magnetic bearings are a significant risk to the project. This risk would be mitigated by reducing the weight of the turbo-generator so that the forces on the magnetic bearings are less. One way of reducing the weight is to design the turbo-generator to operate at a much higher speed than the 4400rpm presently proposed. This will raise the power density and achieve the required reduction in weight. A higher speed design of generator is considered later in this report. The magnetic bearings are described in Section 2.6 in this report.

2.4.12 *Overall Assessment of the Generator Design*

It should be possible to design and build an electrical generator for an NGNP plant by 2018, and the OKBM TDP target efficiency of the generator of 97.7% should be achievable. The cost is estimated to be $6M for the generator.

The electrical generator is a comparatively low risk item of plant compared to other components in the GT-MHR reactor. Essentially, the generator is a development of existing generator technology, even allowing for the relative novelty of high speed operation and operation in a high pressure high temperature helium environment with the possibility of radioactive contamination. The risks therefore may be divided into two main types: risks due to the helium environment and risk due to high speed operation.

The main risk due to the high pressure high temperature helium environment is windage losses which are calculated to be around 0.22% of the output of the generator depending on the final design. This will affect the overall efficiency of the plant.

Radioactive contamination and high temperatures will be managed by enclosing the generator in a separate compartment at a pressure slightly above the rest of the PCU and with cooling to avoid subjecting the generator to undue temperatures.

The main risk associated with high-speed operation is centrifugal forces exerted on the rotor. These forces are believed to be manageable and it should be possible to build a synchronous generator to operate at these speeds. If this is not possible, alternative generator designs that have a simpler rotor may be used instead. These are described in the following subsections.

Another risk is the static frequency converter necessary to convert the generator output to the 50Hz or 60Hz utility system that the GT-MHR will supply. This risk is judged to be small as the static converter would be a modest development of existing HVDC utility converters.

2.4.13 *Alternative Higher Speed Designs of Generator*

Reducing the size and weight of the turbine generator assembly would reduce the weight upon the magnetic bearings and therefore reduces the risk presented by this component. The size and weight of the turbine generator assembly may be reduced by operating at higher speeds. Other advantages of a smaller size of turbine generator is a smaller PCU, saving costs, and a smaller mass of PCU and turbine generator at risk from radioactive contamination. These savings in risk and cost favour operation higher than 4400rpm, however this increases the centrifugal forces suffered by the generator, requiring a simpler design of generator rotor.

At present the generator proposed for the GT-MHR is a synchronous generator. The rotor will carry a DC winding supplied by an excitation system comprising an AC exciter generator and a rotating rectifier both on the same shaft as the generator rotor. This is a mechanically complex arrangement and is vulnerable to high centrifugal forces.

An alternative design is the induction generator. An induction generator has a mechanically simpler winding and therefore is more robust and better able to withstand centrifugal forces. However, an induction generator of 286MW is well above the ratings of commercially available induction machines and would therefore require considerable development. Also induction machines are about 1% less efficient than synchronous generators, and the rating of the frequency converter would have to be increased by about 15% to supply the reactive demand of the induction generator.

2.4.14 *Development Programme*

The development programme proposed by OKBM is reasonable and addresses the issues identified in this section.

2.4.15 *Materials, Lifecycle, Maintenance and Operability*

No maintenance risks are foreseen. The turbine compressor is designed for a maintenance interval 60 000 hrs, approximately 7 years, which would suit electrical generators since they are sufficiently reliable that a breakdown would not be expected during this period.

2.4.16 *Costs*

- 286MW electrical generator for the helium turbine: $3M.
- Power electronic frequency converter: $50M.

2.4.17 *Areas of Uncertainty / Issues for Further Study*

- Centrifugal forces: whether a synchronous machine with brushless excitation is feasible.
- Windage losses in high-pressure helium.
- Electrical generator current lead-outs: cooling, helium leak tightness, dielectric strength, and the ability to withstand differential thermal expansion.
- The gradual diffusion of helium into electrical insulation (and other components), followed by a sudden depressurisation due to an accident or otherwise.
- Flow-induced vibration due to high velocity high-pressure helium.

2.5 Heat Exchangers

2.5.1 *Overview*

Helium as a working medium in the heat exchangers is different from e.g. air in many respects. Helium does not become radioactive in the reactor. It is chemically inert, so it does not degrade reactor materials or the heat exchanger materials at high temperatures. The specific heat capacity (C_p) of helium is very high; 5.2kJ/kg.K compared to 1.02 kJ/kg.K for air. The specific heat ratio is also very high; 1.63 compared to 1.4 for air. The thermal conductivity for helium is high; ~200W/m.K compared to 40W/m.K for air [at 500K], which enables excellent heat transfer properties in the heat exchangers within the cycle. The Mach numbers are very low as speed of sound is very high for helium.

The GT-MHR has three principal heat exchangers, the recuperator, pre-cooler and intercooler, rated as follows:

- Recuperator - 644MW
- Intercooler - 129MW
- Precooler - 162MW
- The working fluids are pure helium and water.

This review has focussed on these, although there are a number of minor heat exchangers, such as those used to cool the generator cavity and as part of the helium clean-up system.

The pre-cooler and intercooler designs are relatively conventional and the designs seem reasonable. There is a minor system risk from the potential for leaks of the cooling water used, but this is minimised by using a lower pressure than the helium circuit. Thus, provided the reactor is pressurised, any leaks will be from the helium side to the water side. It is understood that a leak detection system is planned for this eventuality.

The recuperator is, however, more novel. Experience shows that recuperators can be difficult to design for long life and the overall GT-MHR efficiency is particularly sensitive to recuperator performance. The following subsections therefore provide a discussion of this subsystem, its design and performance.

2.5.2 *Recuperator System Description*

The primary function of the recuperator is to recover most of the heat from turbine exhaust flow. The recovered heat is used to preheat the compressor discharge flow before entering the reactor. The key requirements of the recuperator are to achieve:

- A recuperator effectiveness of 95%.
- Total pressure loss on the low-pressure side of around 1.1% (0.03MPa).
- Total pressure loss on the high-pressure side of 1.0% (0.07MPa).
- A 60-year service life.
- High mechanical integrity (very high delta pressure across the heat transfer plates).
- Low-cost automated manufacturing technology.
- Low maintenance.

Figures 27-30 illustrate the OKBM recuperator design.

The recuperator consists of twenty gas-to-gas heat exchanger modules within the PCU vessel. Hot helium flow from turbine exhaust is distributed evenly among the 20 modules on the low-pressure side of the heat exchanger elements and the cold helium flow from compressor discharge is distributed evenly among the 20 modules on the high-pressure side of the heat exchanger elements. Heat is transferred from the low-pressure side to the high-pressure side. The modules are arranged in two rings around the turbomachinery, one above and one below the vessel cross duct connection, as shown in Figure 26. Each module contains around 200 heat exchanger elements, hence in total around 4000 individual elements. Each heat transfer element is a tube with a total heat transfer length of 2175mm, where the main heat transfer process takes place, and with an outer diameter of 106mm and an inner diameter of 99mm as shown in Figure 27. The primary heat transfer surface inside each element is of plate and fin type and it is divided into three similar segments in the heat transfer element. Folding a 0.35mm thick and 2200mm long sheet of metal in a triangular-like zigzag shape makes up each segment in the tube. Each segment consists of 32 cells. In between every second folded plate, 0.25mm thick offset strip sheet metal fins are inserted on the low-pressure side. The fins prevents the low-pressure side from collapsing, as there is a delta pressure of around 45 bar across the plate when operating, and they also enhance the heat transfer on the low-pressure side. There are no fins on the high-pressure side. The length between 2 sheets of metal on the high-pressure side is 0.53mm and on the low-pressure side it is 1.65mm. All three folded plate segments within a heat transfer element have to be welded together along the sides (~2200mm) and at the top between the metal sheets forming the low-pressure cells to seal off the high-pressure side from the low-pressure side. The three heat transfer segments are inserted into the tube and welded to it along the upper and lower circumference. Each element has one inlet and one outlet nozzle at each end of the tube for the high-pressure helium to enter and exit. The heat transfer elements are joined to a header in groups of two to three. The low-pressure helium enters through several horizontal slots in the heat transfer element tube at the top and exit trough similar slots at the bottom of the tube.

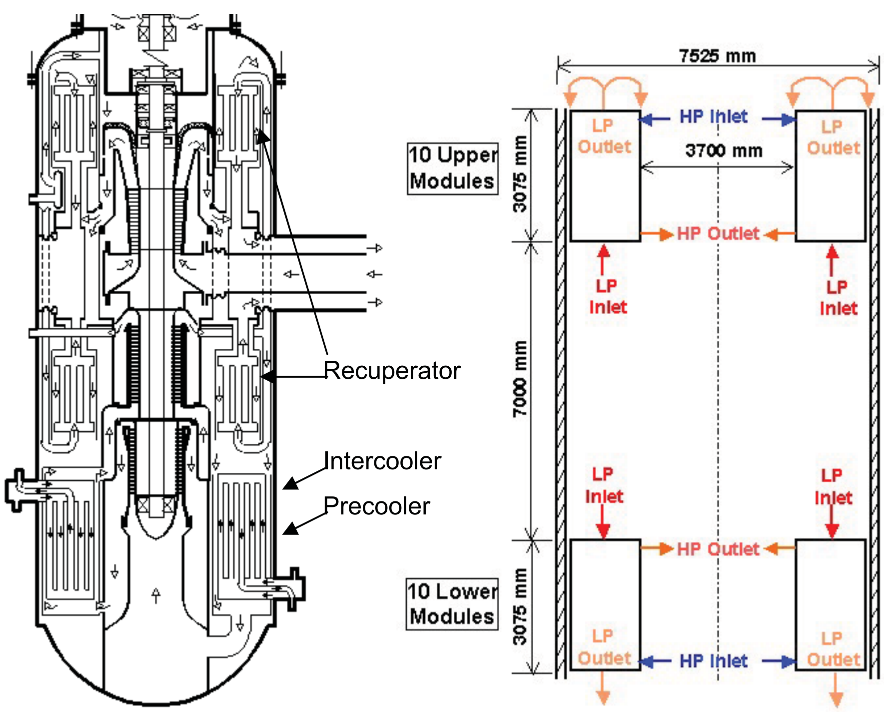


Figure 26 Schematic of GT-MHR Coolers and Recuperator

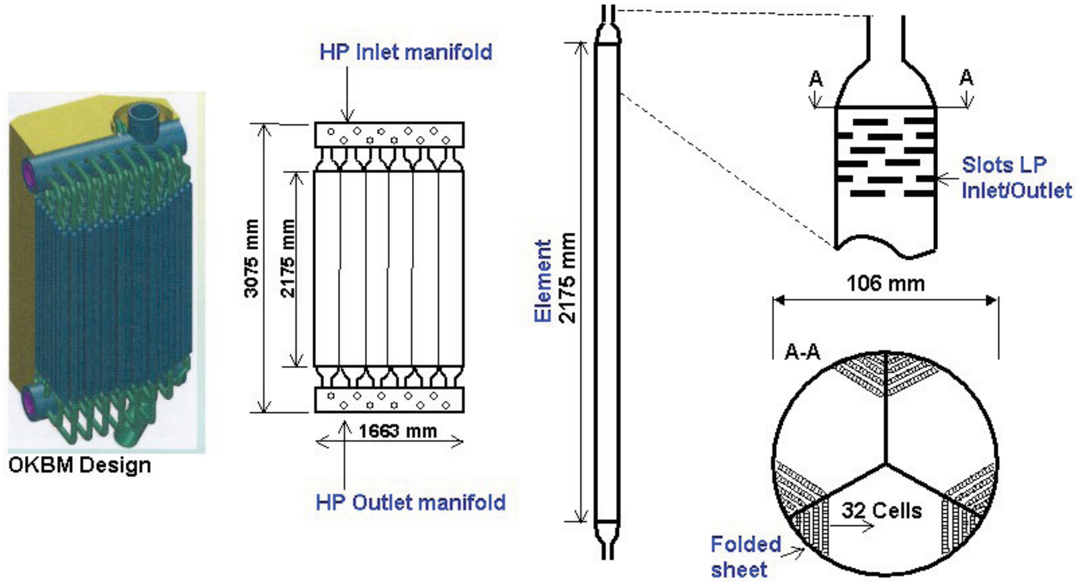


Figure 27 GT-MHR Recuperator Elements

Figure 28 GT-MHR Heat Transfer Element.

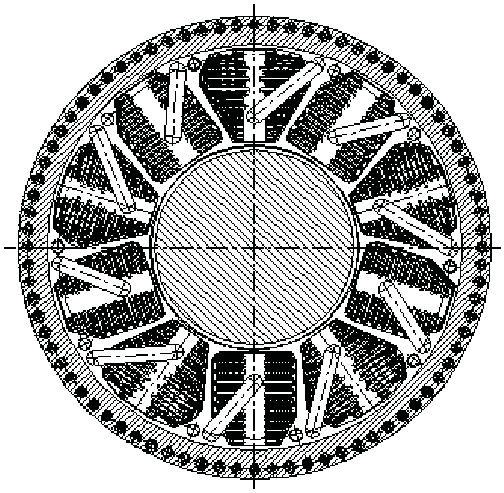


Figure 29 Recuperator Modules - Top View.

The recuperator is designed to transfer 644MWt of heat at rated plant operation with a thermal effectiveness of 95% and with maximum pressure losses of 0.03MPa (1.1%) on the low-pressure side and 0.07MPa (1%) on the high-pressure side. The inlet helium mass-flow rate on both low-pressure and high-pressure side is $321 kgs^{-1}$. The hot low-pressure recuperator inlet temperature is 510°C and the cold high-pressure inlet temperature is 105°C. The helium recuperator inlet pressure on the high-pressure side is 72.2 bar and the helium recuperator inlet pressure on the low-pressure side is 26.2 bar, hence the delta pressure across the heat transfer plates is 46.0 bar. Table 10 summarises the recuperator physical characteristics.

Table 10 GT-MHR Recuperator basic data

Parameter		Baseline Actual
Recuperator weight	[tonnes]	350
Number of modules	[-]	20
Number of elements	[-]	~4000
Total recuperator volume	[m^3]	172
Total elements volume	[m^3]	122
Total heat transfer volume	[m^3]	67
Total frontal area	[m^2]	56
Total frontal area elements	[m^2]	31
Heat transfer element length	[mm]	2175
Heat transfer element outer diameter	[mm]	106
Heat transfer element inner diameter	[mm]	99

2.5.3 *Assessment of Recuperator Design*

The assessment of the recuperator is split into two areas; thermal performance and manufacturability. Each of those areas is described below.

Thermal Performance

To perform a thermal performance analysis of the proposed GT-MHR recuperator, the detailed design of the fins were estimated and correlations for Nu numbers and friction factors were established. The Nu and f correlations for the proposed plate and fin geometries have been obtained from Manglik and Bergles (1995) and are shown in Figure 30. The method of designing the heat exchanger is according to Kays and London (1984).

The proposed heat exchanger is of a very compact design, where the hydraulic diameter on the high-pressure side is 1.1mm and on the low-pressure side it is 1.6mm. The unit cell length of 2.88mm includes 2 sheets of metal each 0.35mm thick, high-pressure flow path of 0.53mm and low-pressure flow path of 1.65mm. The offset strip-fins on the low-pressure side are spaced by 2mm and the longitudinal length of the fins is 6mm. The longitudinal length of the fins is subject to optimisation and it has been claimed by OKBM that 6mm is the optimal length when optimising thermal performance and hydraulic losses.

The physical dimensions, i.e. total frontal area, length of heat transfer surface and volume of all elements, have been constrained when calculating the thermal and hydraulic performance of the proposed GT-MHR recuperator.

It has been shown that, when using the correlations described above and assuming the same frontal area while modelling the thermal performance as well as the physical dimensions, the length required of the heat transfer surface achieving the required heat transfer is 1708mm. The physical length of the effective heat transfer area is 1935mm, which is 227mm longer than the required length of 1708mm. However, the theoretical model does not take into account the inlet and outlet non-uniform flow distribution (because little information is known about these effects) that will especially appear where the low-pressure flow enters/exits the element through the slots in the tube at the top and bottom. The offset strip fins can be designed to facilitate the evenly distribution of low-pressure entry and exit flows, but it is unavoidable that there will be areas where the high-pressure and low-pressure flows will be cross-flow to each other rather than fully counter-flow. There is also a global non-uniform flow distribution, as the upper and lower recuperator modules have different helium flow path length and it is technically challenging to balance all flow resistances to achieve the same flow-rates to all modules and elements, hence some elements will be more loaded than others. The total pressure losses calculated in the heat transfer matrix only are 0.25%: 0.08% on the high-pressure side and 0.17% on the low-pressure side. The pressure losses in pipes connecting the elements with the headers and to get the helium flow to and from the compressor/turbine down to the pre-cooler have not been estimated but, compared with the OKBM figures, there is still 0.92% on the high-pressure and 0.87% on the low-pressure side to account for. This is sufficient as the Mach numbers are generally low.

The weight of the heat transfer area only, i.e. the metal sheets and fins, have been estimated at 205 tonnes and the balance of the heat exchanger, tubes, connection pipes and headers, account for another 119 tonnes. The total weight of the recuperator is therefore estimated at 324 tonnes, which is close to the quoted GT-MHR recuperator weight of 350 tonnes.

In summary, the proposed GT-MHR recuperator appears to have been designed to achieve the required thermal and hydraulic performance targets that were set out in the project specifications. The main uncertainty with the current GT-MHR design with respect to thermal performance is how non-uniform the helium flows are, both globally to each module and locally within each element. However, having flow restrictors in the flow paths does introduce additional parasitic pressure losses and optimising the fins in the elements to distribute the flow evenly in the heat transfer matrix can mitigate the non-uniform flow distribution uncertainty.

Table 11 Modelling of GT-MHR and R-R Proposed Recuperator

Parameter		GT-MHR Actual Design	GT-MHR Model Results	R-R Proposed Design
Total frontal area	[m^2]	30.8	30.6	19.9
Total effective length of heat transfer area	[mm]	1935	1708	1349
Total heat transfer volume	[m^3]	59.6	52.2	26.9
Total pressure drop HP/LP	[%]	n/a	0.08/0.17	0.11/0.14
Reynolds number HP	[-]	n/a	1940	2977
Reynolds number LP	[-]	n/a	1255	2911
Heat transfer coefficient HP	[$W/m^2.K$]	n/a	1744	4182
Heat transfer coefficient LP	[$W/m^2.K$]	n/a	1132	2345
Weight estimates of Recuperator				
Heat transfer area	[tonnes]	n/a	205.0	57.6
Element tubes	[tonnes]	n/a	78.7	16.0
Connecting pipes	[tonnes]	n/a	16.8	10.0
Main feed pipes	[tonnes]	n/a	23.5	n/a
	Total:	**n/a**	**324.0**	**83.6**

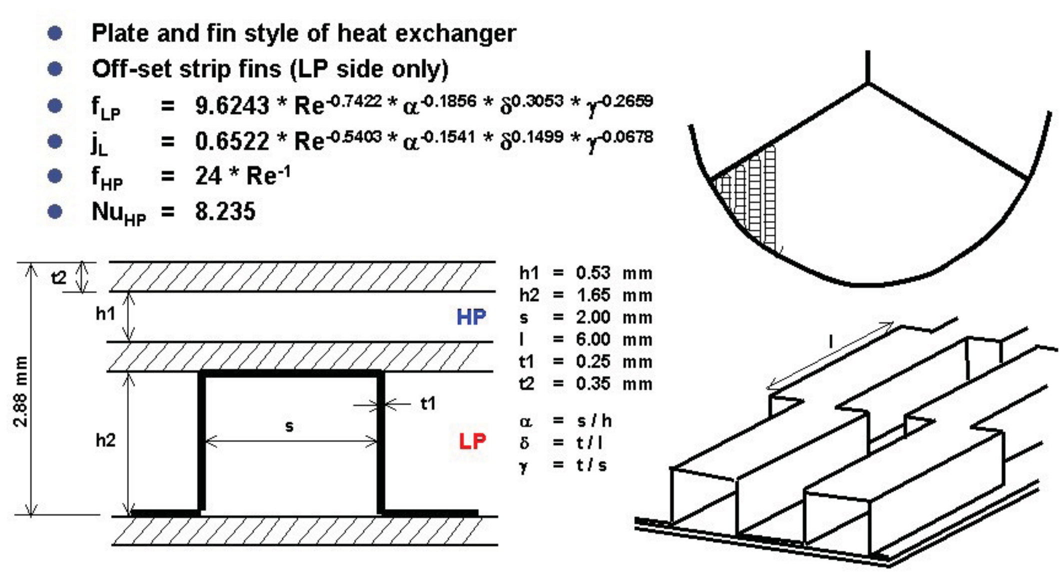


Figure 30 Recuperator Plate and Fin Geometry

<u>Manufacturability</u>

The GT-MHR recuperator consists of around 4000 heat transfer elements arranged in 20 modules with individual helium feeding pipes. The technical challenges are to manufacture a large number of leak tight and high integrity heat transfer elements and to arrange and connect them to the network of pipes and ducts feeding high-pressure and low-pressure helium to and from the recuperator modules and elements.

In each heat transfer element, the heat transfer surface has to be large enough to minimise the number of welds. In the proposed design, a large sheet of metal is folded to create the flowpaths of the heat transfer matrix. This manufacturing technique requires a large number of defect free sheets of metal to be manufactured and techniques have to be developed to bend the sheets into the right shape. It also requires a large number of off-set strip fins, with individual widths, to be inserted in between every second plate. Techniques also have to be developed to join the ends and sides of the folded plates by welding to seal off the high-pressure side from the low-pressure side. The estimated length of all 796 000 welding operations to manufacture just the heat transfer elements is around 50 km.

It will be very technically challenging to set up a manufacturing line for the manufacture of leak-tight and high integrity heat transfer elements when considering the number of welding, bending and cutting operations required. The quality control of the manufacturing processes will be technically challenging and the individual testing of each element prior to installation will have to be comprehensive to ensure that all elements are leak-tight. A leakage in one or several elements will not lead to any loss of helium as the elements are all contained within the main PCU pressure vessel, but the effect on overall PCU thermal performance, electrical efficiency and power output, will be detrimental.

Due to the amount of manual and automated operations required, when manufacturing all heat transfer elements, it will also be very challenging to keep costs at an acceptable level. The quantities of raw materials, in total around 350 tonnes and most of it is required in thin metal sheets, will add to the challenge to keep the costs at a low level.

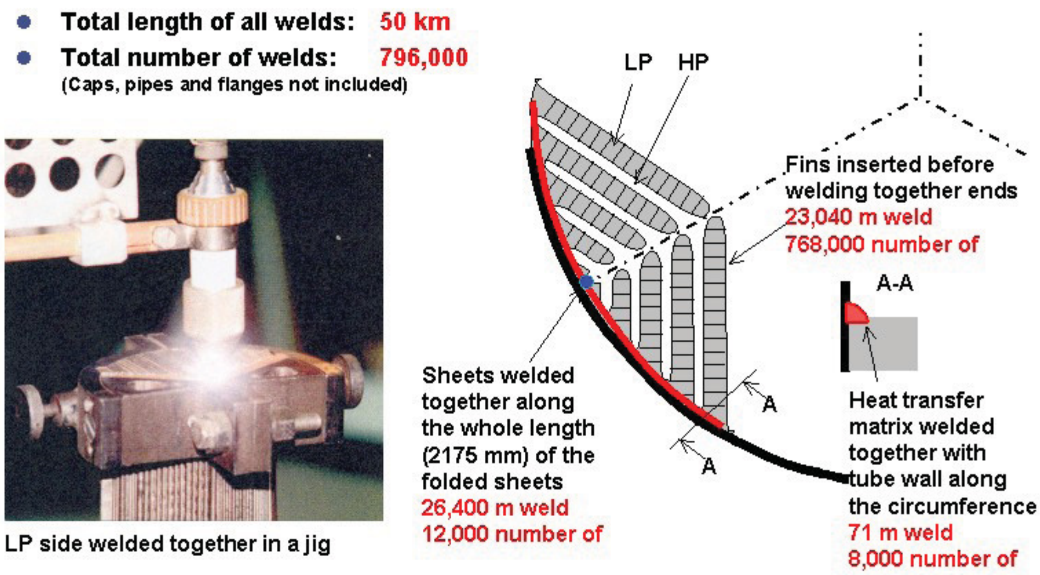


Figure 31 Recuperator Manufacturing

<u>Alternative Recuperator Designs</u>

OKBM have proposed some fallback commercially available recuperator options in case the proposed baseline design, for some reason, will not be feasible. The first option is a commercially available Ingersoll Rand recuperator design such as shown in Figure 32. However, this design currently operates in some applications with a much lower delta pressure across the heat transfer plate, hence it may well have to be changed to be used in the proposed GT-MHR application. Other options include using commercially available heat exchanger designs that use braising or molecular bonding techniques instead of welding.

Figure 32 Ingersoll Rand Recuperator

Rolls-Royce have considered alternatives to the GT-MHR recuperator design that could potentially offer a more compact and lower weight solution to the current proposed design whilst achieving the same hydraulic losses, such as a fusion-bonded cross-corrugated plate heat exchanger design. Kleeman investigated this style of heat exchangers for high temperature helium gas applications in 1978. It was concluded then that this style of heat exchangers features a highly compact installation and volume whilst achieving very low-pressure losses.

The Roll-Royce proposed alternative recuperator is shown in Figure 33 below. This is based upon a design produced by Alfa Laval, and utilises the experience gained from a Rolls-Royce-funded research programme at Oxford University.

This type of heat exchanger consists of cross-corrugated stamped stainless steel plates, frame plates, pressure plates and connections. The manufacturing process, i.e. the fusion bonding technique, was developed a few years ago. No welding is required, except for manifold connections/pipes between heat exchanger modules. The current material in these heat exchangers is stainless steel type 316 and it is quoted that temperatures up to 550°C can be achieved. The hydraulic diameters in the cross-corrugated heat exchanger are higher than those in the GT-MHR design Figure 34:

- High-pressure side: 1.5mm for cross-corrugated and 1.1mm for plate and fin.
- Low-pressure side: 2.7mm for cross-corrugated and 1.6mm for plate and fin design.
- The compactness of the cross-corrugated design is 773 m^2m^{-3} and for the plate and fin design it is 2273 m^2m^{-3} on the high-pressure side and 1511 m^2m^{-3} on the low-pressure side.

It has been shown that, even though the cross-corrugated design has larger hydraulic diameters and is less compact (in terms of heat transfer area) than the plate and fin design, the cross-corrugated design would be much smaller and lighter than the plate and fin design for the same hydraulic losses. The reason for this is that the overall heat transfer coefficient in the cross-corrugated design is superior to the ones in the plate and fin design. For the same requirement, 12 heat exchanger modules of the size shown in Figure 35, arranged e.g. as per Figure 36 would suffice. The volume of all 12 cross-corrugated heat exchanger modules is about half of the volume of the GT-MHR plate and fin design and the weight is about ¼ of the weight of the GT-MHR design as well (Table 12).

In summary, it can be concluded that there are alternatives to the GT-MHR recuperator design that could potentially offer a more compact, lower weight and lower risk solution to the current proposed design whilst achieving same hydraulic losses. Moreover, the cross-corrugated style of recuperator could also be a cheaper option as the design does not incorporate welding and requires less material. The estimated cost for this alternative design of recuperator is around $12M for GT-MHR (i.e. ~$1M/module). However, the Alfa Laval recuperator design is not off-the-shelf technology as of yet on this scale, as these heat exchangers are currently manufactured in smaller unit-sizes than what would be required for the GT-MHR application. It is judged that this is not a major shortfall and the GT-MHR requirement could be met without stretching the current design technology too much.

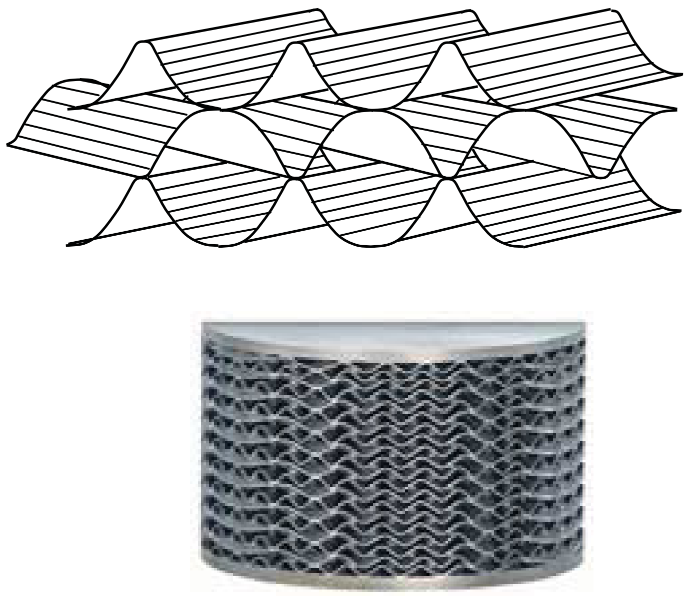


Figure 33 Cross-Corrugated Heat Exchanger

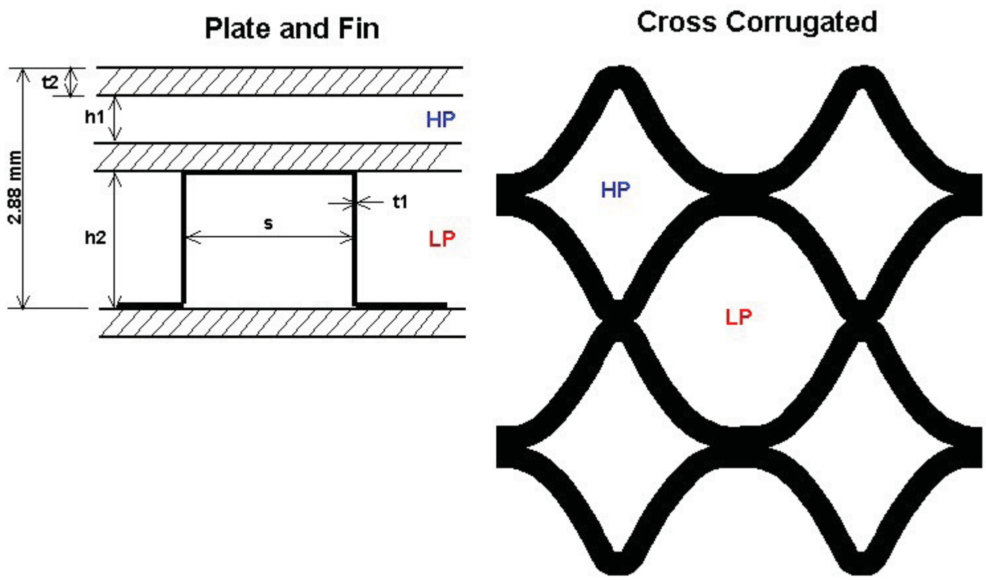


Figure 34 Comparison of Recuperator Heat Transfer Surface Areas

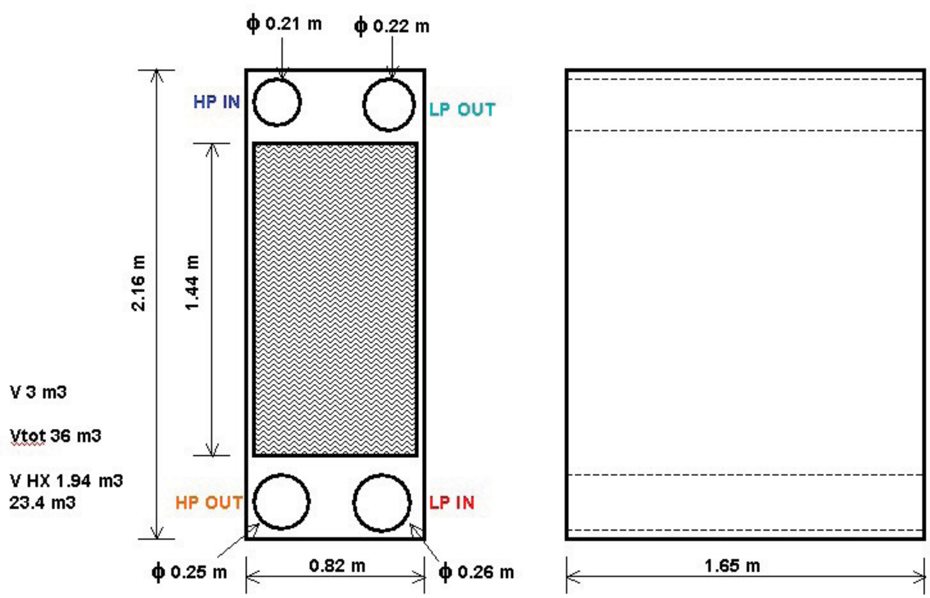


Figure 35 Proposed Cross-Corrugated Recuperator Module (1 of 12 in total)

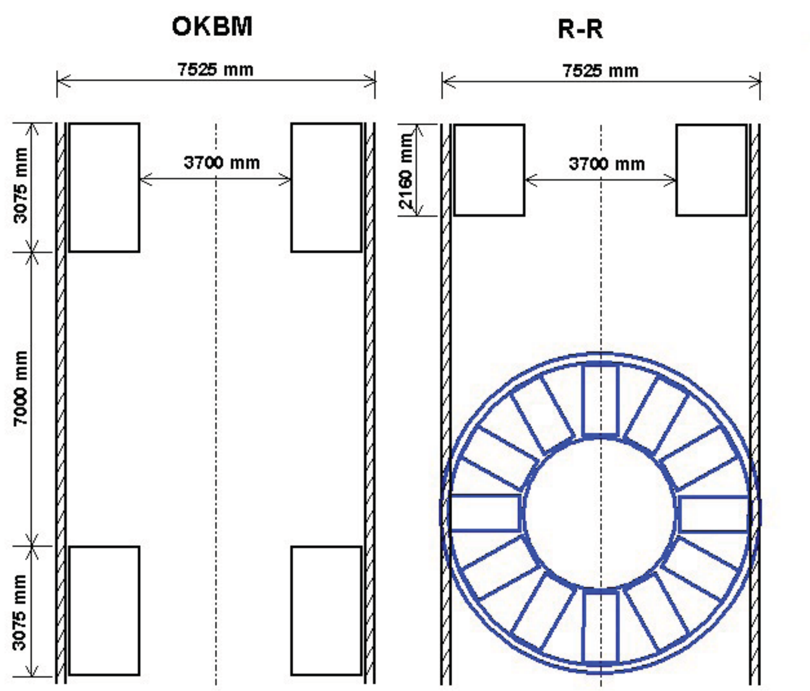


Figure 36 Comparison of Recuperator Baseline and Revised Designs

Table 12 Modelling of Baseline and Revised Recuperator

Parameter		Baseline Actual	Baseline Model	Revised
Total FA	[m^2]	30.8	30.4 (99%)	14.8
Total L	[m]	1.935	1.644 (85%)	1.440
Total Vol	[m^3]	59.6	49.8 (84%)	21.4
dp HP/LP	[%]	n/a	0.07/0.16	0.06/0.09
Weight HX	[te]	n/a	203.4	45.7
ReHP	[-]	n/a	2029	4159
Re_{LP}	[-]	n/a	1310	4056
h_{HP}	[$W/m^2.K$]	n/a	1833	5292
h_{LP}	[$W/m^2.K$]	n/a	1187	2985
Weight Estimates				
Heat transfer area	[te]		203.4	45.7
Element tubes	[te]		78.6	16.0
Connecting pipes	[te]		16.8	10.0
Main feed pipes	[te]		23.5	n/a
	Total:		**322**	**71.7 (22%)**

Conclusions

- The OKBM proposed recuperator design will most likely transfer the required heat from the low-pressure turbine exhaust helium to the high-pressure compressor discharge helium whilst achieving target pressure losses. However, uncertainties in how non-uniform the helium flow will be in operation will impact the thermal performance of the proposed design. A lower recuperator effectiveness than 95% will result in lower electrical efficiency and less power output.

- It will be very technically challenging to set up a manufacturing line to manufacture a large amount of leak-tight and high mechanically integrity heat transfer elements when considering the amount of welding operations (800 000 operations and 50 km of weld), bending operations and cutting operations required.

- Based on a competitive $/kW plant price, the OKBM recuperator design is likely to be hard to manufacture competitively.

- There are alternatives to the GT-MHR recuperator design that could potentially offer a more compact and lower weight solution whilst achieving the same hydraulic losses. We have proposed a fusion-bonded cross-corrugated plate heat exchanger design. The volume of this is about half of the volume of the GT-MHR plate and fin design and the weight is about ¼ of the weight of the GT-MHR design. Moreover, the cross-corrugated style of recuperator could also be a cheaper option as the design does not incorporate welding and requires less material. However, it is not off-the-shelf technology as of yet on this scale, but it could potentially be without stretching their current design technology too much.

2.5.4 *Development Programme*

The recuperator development programme proposed by OKBM is reasonable and addresses the issues identified in this section.

2.5.5 *Materials, Lifecycle, Maintenance and Operability*

The proposed materials are as to expected for the recuperator, pre-cooler and intercooler in the GT-MHR application.

The recuperator is likely to capture a significant fraction of any radioactive elements released from the fuel. This will have an impact on any repair and maintenance activities, which are therefore likely to need to be handled remotely. In mitigation, the modular design aids removal/replacement of recuperator modules, both for the OKBM and cross-corrugated designs (especially the latter).

Experience would indicate that recuperator reliability is likely to be a problem. Both the OKBM and any proposed alternative designs should be extensively tested and/or have a robust replacement strategy.

2.5.6 *Technical Risk Assessment*

Key risks:

- **The proposed recuperator may not achieve the required effectiveness of 95% and total hydraulic losses of 2%** - This will result in a lower plant efficiency and lower power output. This can be mitigated by enlarging the recuperator modules or improving the flow distribution to each heat transfer element if the flow is non-uniform. It can also be mitigated by having a different recuperator design.

- **The recuperator may not achieve the desired lifetime** - Whilst the temperatures experienced in the recuperator are not too challenging (and don't require special high temperature capable materials), the pressure differences between the two sides are quite large. Making a delicate structure to survive a 60-year lifetime in this environment is very challenging. Both the OKBM reference design and the alternative cross-corrugated design should be considered to have a high risk of not achieving the required life.

- **The recuperator is not leak tight** - This will results in lower plant efficiency and lower power output. This can be mitigated by developing repair and replacement processes.

- **Fabrication of the proposed GT-MHR recuperator is too expensive** - This will result in a non-competitive plant solution. This can be mitigated by improving the manufacturing technologies or changing the recuperator design. The proposed cross-corrugated design should be significantly cheaper, but carries a minor risk associated with scaling up to the desired GT-MHR size.

2.5.7 *Costs*

The estimated manufacturing cost for a GT-MHR recuperator element is $20k, hence the total costs for all elements is estimated at $80M.

The estimated manufacturing cost for the alternative cross-corrugated recuperator module is approximately $1M, hence total costs for all modules are estimated to $12M.

2.5.8 *Areas of Uncertainty / Issues for Further Study*

Further work is required in the following areas:

- Manufacturing processes for proposed GT-MHR recuperator design,
- Design and off-design performance of proposed GT-MHR recuperator design,
- Understanding helium non-uniform flow distributions, both globally and locally,
- Develop a process to detect leaking recuperator modules,
- Develop a process to repair and replace faulty recuperator modules, and
- Research alternative heat exchanger alternatives, such as the cross-corrugated design.

2.6 Electromagnetic and Catcher Bearings

2.6.1 *System Description*

There are 2 axial EMBs on either side of coupling and 4 sets of radial bearings i.e. either side of generator and either side of turbomachine. The GT-MHR design requires support of 35 tonnes in the generator rotor and 32 tonnes in the turbomachinery rotor, at a speed of 4400rpm. The EMBs are theoretically feasible and the technology is scaleable. Furthermore, physically, the bearings will fit into the available space.

2.6.2 *Assessment of Design*

Electromagnetic Bearings

EMBs are good at supporting static loads. Problems arise with dynamic load capability since the frequency response could be as low as a couple of hertz, i.e. performance in a seismic environment could be compromised. Thus, transient load forces are important. A further problem is that there are iron losses in rotor components that have to be dissipated. However, reducing loads does derisk the EMBs somewhat as described below.

EMB electromagnets typically use organic insulants as these offer long life (measured in decades) when operated at up to ~180°C. Above this temperature, ceramic insulation systems are needed. These are little used in the market at present, although they are technically feasible.

EMB power consumption is typically ~10% of that expected in an equivalent oil-cooled bearing. It is also relatively easy to use EMBs to accurately position the rotating shaft (<<1mm), potentially allowing very tight control of turbine blade tip clearances.

Rotor dynamic design is required, as the rotating body needs to be matched to the capabilities of the bearing and control system. In particular the designer needs to know the critical speeds on both the EMBs and the backup bearings – these critical speeds will be different. Both bearings will need to be suitably stiff.

Catcher Bearings

The OKBM design is at the limit of catcher bearing possibilities: 2 million DN (diameter x rpm) is the limit of catcher bearing burst capability.

Rolls-Royce has some experience in this field. For example, having developed bearings for a 1.25 tonnes mass rotating at 8000rpm. This design underwent 12G shock test and was design to provide a multi-use (up to 10 times) run-through capability, i.e. the machinery continued operation through the shock transient.

The design of catcher bearing for the GT-MHR needs careful optimisation of materials selection and design. In particular, the bearing loads and the requirement for numbers of reuses before replacement will be the key variables.

There may be the need for more than one type of catcher bearing, with different stiffnesses.

2.6.3 *Relevant Experience of EM Bearing Manufacturers*

The thrust bearings with their large loads are perceived as particularly challenging and thus four leading EMB manufacturers were approached to gain their views on the viability of such bearings for the PCU. Two generator/turbomachinery rotor weights and speeds were specified:

- 35 tonne at 4400rpm.
- 10 to 12 tonne at 5000rpm (this is to cover the turbomachinery design covered in Section 4).

In addition the manufacturers were advised that the thrust bearings would have to cope with seismic loads.

The manufacturers' responses can be summarised as follows. The order of the responses is from those Rolls-Royce would have least confidence in to those in which greater confidence could be placed:

Synchrony

Synchrony have not designed thrust bearings for such large loads before, not even at the 10 tonne level. However, given sufficient engineering resources and some engineering development, they believe that the EMBs could, in principle, be made. However, the seismic loads would be a major concern for Synchrony.

Synchrony's current interests in packaging and miniaturisation are such that they were keen to direct Rolls-Royce to their competitors for the present work.

Waukesha (formally Glacier)

Waukesha are currently manufacturing the 24th of a batch of EMBs for horizontal synchronous motor/compressor units. These units are fully speed variable in the range from 600rpm up to 6300rpm. The weights of the rotors of the motor and compressor are 10 tonne and 1.5 tonne respectively. The EMB thrust bearing is rated at 7.5 tonne. Thus, in terms of load and speed, Waukesha have experience similar to that needed for the present 10 tonne rotor. Indeed, they say they would be completely confident of success if they undertook such an EMB project.

The 35 tonne rotor is outside Waukesha's experience. However, they have undertaken a number of design studies, the most relevant of which relates to EMBs and back-up bearings for the 60 tonne rotor of a 230MW vertical generator in a Pebble Bed Modular Reactor operating in a seismic environment.

Approximately a decade ago Rolls-Royce was twice involved on EMB projects with this company when it was known as Glacier. Both projects were unsuccessful from the perspective of Rolls-Royce. However, Waukesha have since had some success in the marketplace, although their key technical staff have not changed significantly. Thus Waukesha have clearly learnt something from their experiences. However, Rolls-Royce would be very wary of working with them again.

SKF (formerly known as Revolve)

SKF have build experience of EMB thrust bearings with loads up to 100kN (about 10 tonne) operating at 16 000rpm. Thus, SKF claim experience similar to that required for the present ~10 tonne rotor. However, it is not clear whether their designs are suited to a seismic environment.

The 35 tonne rotor is outside SKF experience but not, they say, outside the capability of EMBs. Their approach would be to undertake an engineering design study at the outset in order to:

- Identify areas of risk such as, they believe, the back-up bearing system.
- Find viable solutions.
- Estimate costs.
- Estimate the added value advantage that an EMB system would have over a conventional bearing solution.

S2M

S2M are generally considered to be the world's most successful EMB supplier. Most of their designs are for shaft diameters in the range 50mm to 400mm but the firm has experience at larger sizes too and also at high rotational speeds (up to 120 000rpm). Thousands of their products are installed on turbo-molecular pumps in the semiconductor industry and an increasing number are being installed on turbo-machinery used in the natural gas/oil industry.

Specific S2M experience relevant to the current project includes provision of thrust EMBs (with 1200mm diameter thrust disks) for several vertical shaft water turbines of 35 tonne load. These units have been installed in Japan, a renowned earthquake zone, where they have been operating successfully.

S2M have also undertaken an outline design study for a thrust EMB in a reactor system where the load is about 110 tonne.

For the current requirements at 4400rpm and 5000rpm S2M estimate that the thrust disks could have maximum diameters of 1300mm and 1000mm which they claim suit the 35 tonne load and, of course, the ~10 tonne load.

2.6.4 *Summary and Conclusions on Thrust EMBs*

Synchrony does not have the experience or the interest to supply thrust EMBs for this project. Waukesha have experience in the lower load range (~10 tonne) but Rolls-Royce has had poor experiences working with them. SKF too has experience only in the lower load range, although it has undertaken studies at the higher load level of 35 tonne. Only S2M has experience of 35 tonne vertical load thrust bearings in a seismic environment. However, their build experience, as opposed to studies they have undertaken, does not extend to the 100 tonne load level.

This review has shown that most of the EMB industry can cope with thrust loads at the 10 tonne level but that only S2M has proven experience in a seismic environment at the 35 tonne level. None of the companies contacted has claimed experience (as opposed to studies) at significantly higher load levels.

The above suggests that at the ~10 tonne load level Rolls-Royce could work with SKF, S2M and, perhaps, with Waukesha. However, at the 35 tonne level, S2M is the only possible collaborator. At significantly higher loads there are no known firms to partner.

2.6.5 *Development Programme*

Generally-speaking, the OKBM TDP recognises the technology challenges and risks inherent in the EM and catcher bearings for this design. Given, however, the world experience available, Rolls-Royce would recommend that the bearing development programme attempts to gain better engagement from industry partners with relevant experience.

2.6.6 *Materials, Lifecycle, Maintenance and Operability*

EMB materials are well established, although the pedigree of ceramic windings for higher temperature applications is more limited. Reliability is anticipated to be high for the components installed within the PCU vessel. Redundancy can also be built into the components and associated control electronics and software to allow graceful degradation in the event of single component failure.

Catcher bearing materials are more of a challenge and the OKBM TDP recognises this. More work is need to demonstrate the rotor loads can be supported over several engagements of the bearing system.

Maintenance of both EM and catcher bearings will be difficult but, provided catcher bearing life can be proven, is unlikely to be significant problem. It is anticipated that any required maintenance will be carried out during turbomachinery maintenance cycles (i.e. when the PCU vessel is opened for rotor extraction).

2.6.7 *Technical Risk Assessment*

See discussion above.

2.6.8 *Areas of Uncertainty / Issues for Further Study*

See discussion above.

2.7 Pressure Vessel and Bypass System

This section provides a high-level evaluation of the Power Conversion Unit (PCU) vessel and cross duct design.

The overall scope of the study being carried out by Rolls-Royce is given in Reference 13. This memo represents Phase 1 of the study for the PCU vessel and cross duct, which, from Reference 13, is: 'evaluation of the OKBM/GA preferred layout and identification of improvements'.

OKBM Report 08.03-006.01 (Reference 14) gives the proposed design that is being evaluated.

As agreed in the Programme Meeting on 6 March 2007, an evaluation of the structural integrity of the PCU In-Vessel Metalwork (IVM) is to be included with the vessel and cross duct evaluation. Functional evaluation of the IVM is to be covered by the aerodynamic evaluation being carried out for the turbomachinery.

2.7.1 *System Description*

At present the layout and major dimensions for the OKBM/GA preferred layout have been defined, as given on the Power Conversion Unit General View Drawing (Reference 9).

The material/materials for the PCU vessel have not yet been defined. However it is understood from the meeting with OKBM in Derby on 13 March 2007 that an American Society of Mechanical Engineers (ASME) steel would be used.

The material/materials for the IVM components have not yet been defined, however some possible alloys were proposed in a presentation entitled 'GT-MHR International Project, PCU' by OKBM in 2007. The presentation also shows that it is envisaged that the IVM will be fabricated from a mixture of plate, sheet and tube.

At present, the designers have not carried out a detailed structural assessment of the PCU. It is understood that stress modelling, fatigue and fracture assessments would be carried out at a later stage in the design process.

2.7.2 *Assessment of Design*

<u>Materials Issues</u>

Since materials have not yet been defined, this section contains general comments assuming that typical pressure vessel steels are used.

Corrosion control should not be a particularly difficult issue for this design, subject to usual nuclear build cleanliness and atmospheric control requirements. Water is used in the pre-cooler and intercooler within the design; if a water leak occurred this could cause a potential corrosion issue on nearby pressure vessel surfaces. During operation, the helium will be at a higher pressure than the cooling water, hence any defects occurring in water pipes would lead to loss of helium rather than leakage of the water. However during maintenance, when the plant is depressurised, it is reasonably foreseeable that some leakage of water could occur. Therefore this scenario should be considered further as the design is developed, in terms of factors such as cooling water chemistry, leak size/rate probability, leak detectability and materials corrosion resistance.

The yield strength of a material gives a guide to its suitability for use at a given temperature (although other measures of strength may be used in assessments). The yield strength of the pressure vessel material will reduce with increasing temperature. This is illustrated in Figure 37, which shows some typical strength against temperature curves for ASME ferrous forging materials. Values have been taken from Section II Part D of the ASME Boiler and Pressure Vessel Code (Reference 15). If variations to the temperature cycle are considered as the design is developed, the amount by which temperatures could be increased will be limited by material yield strength (since there are practical limits to compensation that can be made by adjusting wall thickness).

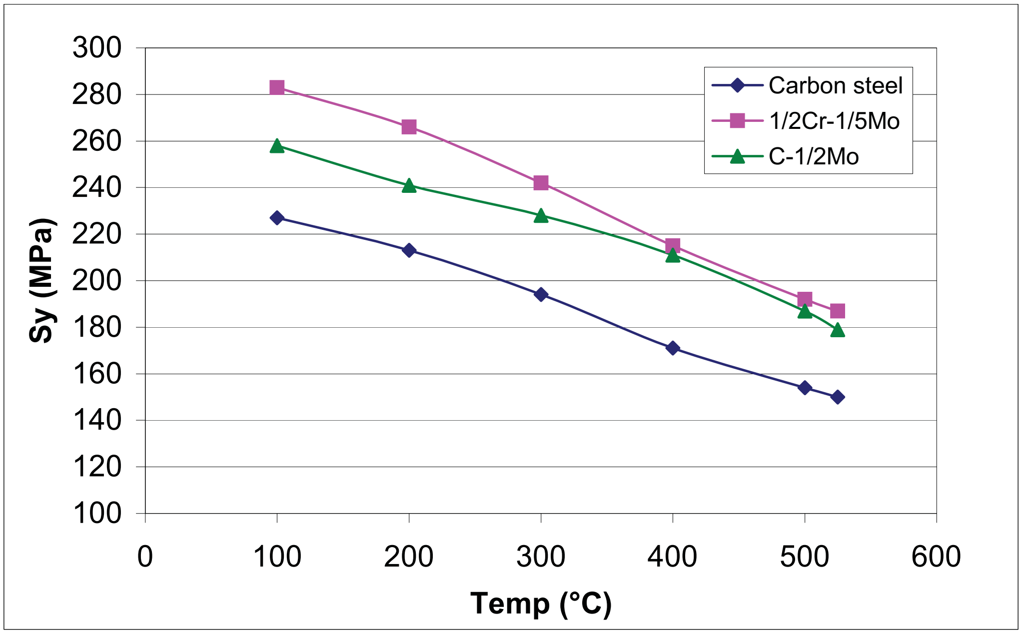


Figure 37 Yield Strength of Some ASME Ferrous Forging Materials

Creep may be an issue due to the proposed high temperature operation of this plant. The potential for creep to occur within the vessel, cross duct and IVM components should be investigated, together with an assessment of consequences of creep for areas that may be affected. From Reference 16, 'it is essential to be able to define the 'high' temperatures at which creep and creep fracture generally become important. With pure metals, high temperatures can be defined as above about $0.4T_m$, where T_m is the absolute melting point. This temperature is critical because, at around 0.4 T_m and above, atomic rearrangements in the crystal lattice can occur by diffusion.' It should be noted that:

- This statement is for pure metals, not alloys. Alloys may be more creep-resistant, but the level of resistance depends on the material microstructure. Therefore, it is sensible to use this as a point above which creep may be an issue. However, in practice, creep may not be an issue until much higher temperatures are reached.

- This refers to absolute temperature - i.e. in Kelvin (K).

A typical pressure vessel steel T_m may be taken as being about 1640K, with the range in T_m being approximately 1470K to 1670K. Therefore the temperature above which creep may potentially become an issue is 588K (i.e. 0.4 x lower T_m), but more typically temperatures of concern will be above 656K (i.e. 0.4 x typical Tm).

The highest proposed temperature within the plant is 1121K at the reactor outlet. This temperature is seen by the inside tube of the cross duct and by the IVM at the ducting to the turbine inlet. Although these do not form part of the pressure boundary, the safety and functional implications of creep at high temperature regions such as this should be assessed as part of the design development. The highest helium temperature that would be experienced by the pressure boundary in the current design is 764K, which is the high pressure recuperator outlet temperature experienced by the PCU vessel outlet and the outer co-axial tube of the cross duct. This is above the temperature at which creep may be an issue. It is also a sensitive region with regard to thermal stresses, fatigue and geometry effects, as will be discussed in the section on Bypass Design.

Section II Part A of the ASME Boiler and Pressure Vessel Code (Reference 15) gives ferrous materials specifications and contains guidance on which materials are suitable for high-temperature applications. Section II Part D gives material properties. Table 5 of this memorandum, which shows some typical ferrous steel forging properties, illustrates the need to carefully select an appropriate material for high temperature operation. Further information received from GA has indicated that, further to the OKBM design, they have given more attention to material selection for high temperature operation and related this to allowable temperature parameters.

There has been some discussion as to whether helium at high temperature and pressure could affect material properties. Helium is inert and hence does not chemically react with metals. Due to its atomic size, however, it is highly penetrating and could have some effect on the strength/elasticity of the pressure vessel material. A literature search on helium embrittlement has been carried out by Rolls-Royce. The results from this search revealed many published technical papers on the subject of helium embrittlement. These have not been reviewed in detail, however it appears that most were produced to support nuclear fusion work, where helium can occur within materials due to transmutation. Therefore some further investigation is required to establish whether or not helium embrittlement may be an issue in helium-cooled fission reactors.

Some transition welds are proposed in the current design. Transition welds generally need to designed and manufactured with extra care, due to issues such as thermal mismatch, weld pool dilution and corrosion behaviour.

<u>Fabrication</u>

The OKBM design is forged rings for the main shell of the PCU pressure vessel. The height of the rings, and hence the number of sections, will be limited by the capability of forgemasters of large pressure vessels.

Within all pressure vessels, especially those for nuclear applications, the number of welded joints should be minimised wherever possible. This is because even the best quality welds are still likely to contain some defects, which can act as fracture or crack growth initiation sites.

The current OKBM design appears to have adopted the principle of minimising welds on the PCU pressure vessel whilst allowing for achievable forging size sections.

There has since been a suggestion by GA that rolled plate sections, with a longitudinal seam welds could be used instead of forged rings. From the safety point of view, this would be a less favourable option (unless it can be demonstrated that the inherent reactor safety mitigates a lower integrity pressure vessel design). A longitudinal weld in a pressure vessel is acted upon by the hoop stress (sh), which is greater than the longitudinal stress (sl) that acts on circumferential welds (sh=PD/2t and sl»PD/4t; where P is pressure, D is internal diameter and t is wall thickness). Using a longitudinal seam weld may also increase the overall amount of structural welding in the vessel, hence increasing the number of defects likely. A design with a longitudinal seam weld would also increase the cost of any in-service inspections.

The fabrication of the IVM has not been reviewed, as details are not known beyond the statement that it will be fabricated from a mixture of plate, sheet and tube.

Loading and stress considerations

The design would/may be subject to the loads and stresses shown in Table 13.

Table 13 Loads and Stresses Potentially Affecting the Design

Load/stress type	Description/Comments	Applicable to PCU Vessel?	Applicable to IVM?	Applicable to Cross Duct?
Internal pressure		Yes	No	Yes
Pressure difference in each section		Yes	Yes	Yes
Weight	Stress due to reactions to weight at mountings	Yes	Yes (component weight is significant)	Negligible
Thermal gradients across sections		Yes	Yes	Yes
Thermal transients	Stresses due to warm-up, cool-down, etc	Yes	Yes	Yes
Weld residual stresses	This could be reduced by post-weld heat treatment (PWHT), if it is to be carried out	Yes	Yes	Yes
Bolt-up loads	Stresses around bolted regions due to bolt tension	Yes	Yes	No
Internal loads due to expansion of dissimilar IVM materials	Note that the current design of the IVM does incorporate thermal expansion compensators in certain areas	Yes	Yes	No
Thermal stresses around the pre-cooler and intercooler		Yes	Yes	No
Vibration	Small loads: may be below fatigue threshold. However if they are not, the high frequency may cause fatigue.	Yes	Yes	Yes
Pressure fluctuations from turbine/compressor operation		Yes	Yes	Yes

Load/stress type	Description/Comments	Applicable to PCU Vessel?	Applicable to IVM?	Applicable to Cross Duct?
Loads transmitted by external mountings: i) fit-up ii) thermal expansion	These loads could be corrected in the mounting arrangement design and by environment control	Yes	No	No
Seismic		Yes	Yes	Yes
Shock loads due to external hazards		Yes	Yes	Yes
Bearing failure loads		Yes	Yes	Yes
Other accident loads		Yes	Yes	Yes

Based on the current understanding of the PCU design, it has been assumed that the PCU and IVM would not need to be designed to withstand internal shock loads due to blade loss. Such failures are assumed to be completely contained within their immediate casing.

PCU Design

It is not clear why there is a change in PCU vessel diameter just above the pre-cooler water outlet. A simple cylindrical vessel shell would be inherently stronger than one with a section change, as well as being easier to manufacture. However, if this is an important design feature, for example due to geometry constraints, then attention should be paid to the design of this area, with the blend made as smooth as possible. At present the change in section appears to be rather too sharp.

There are several regions of the PCU vessel where there appear to be quite sharp changes in wall thickness, which would lead to stress concentrations. An example of this is around the vessel mounting, as shown in Figure 38. The vessel mounting is also a highly loaded area; therefore the design of this region should be reviewed.

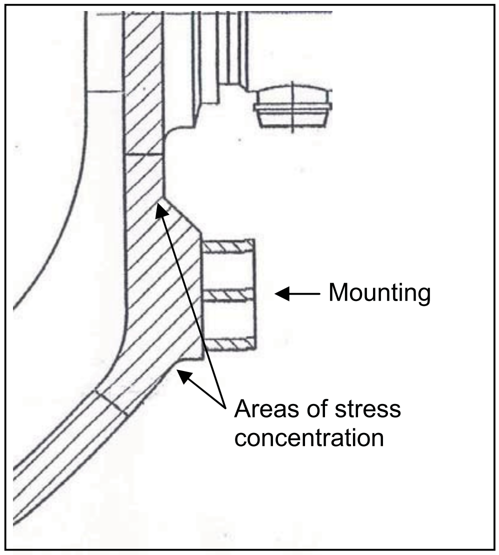


Figure 38 Current Design of PCU Vessel Mounting Region

The current design of PCU vessel is to be sealed using a welded toroid seal. This is a well-established method of sealing nuclear pressure vessels. Although the nuclear industry has suffered some problems in the past with toroid seals, there is now much published experience that can be utilised in producing the detailed design.

The PCU design features a main bolted flange at the top of the cylindrical shell section, plus other smaller bolted connections, such as handholes. Bolted connections are widely used in nuclear pressure vessels, so no new problems are envisaged. However this is an area of the design that will need particular focus as the design is developed, to ensure appropriate bolt sizes, thread length, flange depth, flange width and ligament width are used.

<u>In-Vessel Metalwork Design</u>

Details of the IVM design are likely to be subject to changes from this concept design, driven by improvements to component design and layout. As the IVM design is developed, the advice given elsewhere in this section regarding materials, welds, geometry effects on stress concentration and bolted joints should be considered.

<u>Cross Duct Design</u>

The OKBM proposed cross duct design is very similar to that designed for the Chinese HTR-10 Generation IV nuclear power system, details of which are given in Reference 17. The Chinese design is also a horizontal coaxial double-tube duct carrying helium between the reactor and another vessel. The Chinese design has thermal insulation material packed into a space between an inner tube and a liner tube, to resist heat transfer between the hot helium and the cooler helium. It is not yet known what insulation arrangements will be used for the GT-MHR design.

The Chinese design has undergone detailed assessments and full-scale thermal performance tests in a helium test loop. The conclusion was that all analyses and experimental data indicate that the duct was well designed and reliably tested.

This experience gives good confidence in the basic design of the GT-MHR cross duct.

However the GT-MHR cross duct design has a higher operating temperature, different loadings and long design life (60 years), hence a more specific read-across is not possible.

Fatigue is an issue that would have to be given high significance in the assessment of the cross duct. The blend radii between each of the coaxial tubes and the adjoining vessels are areas that will be particularly susceptible to fatigue, due to geometry effects. Also the cross duct will experience more severe thermal transients than other areas of the PCU vessel.

As highlighted in the section on materials, creep is also potentially a concern for this region.

Conclusions

This memorandum presents a summary of the issues potentially affecting the PCU vessel, cross duct and In-Vessel Metalwork, which may be used for reference and guidance as the design is developed. No detailed assessment of these components has been carried out as part of this Phase 1 study.

A few areas of potential concern have been raised. In particular these are:

- Material strength at high temperature - This must be a key factor in material selection.

- Fatigue - This could be a serious challenge to the proposed design life of 60 years. It is thought that the blend radii between each of the cross duct coaxial tubes and the adjoining vessels are areas that will be particularly susceptible to fatigue.

- Creep - The cross duct will be the most susceptible region within the scope of this memorandum, as it sees the highest temperature.

All of the above issues have the potential to limit plant operation. Whether or not this is the case would have to be determined by further analysis once a more detailed design was available to assess.

Suggestions for PCU vessel design improvements have been made, such as attention to the geometry of certain areas.

No immediate problems have been found with the PCU vessel, cross duct or In-Vessel Metalwork design that should impede development of the current design.

2.7.3 *Technical Risk Assessment*

No major risks.

2.7.4 *Costs*

No independent estimates have been made at this point - utilise existing GA and OKBM data.

2.7.5 *Areas of Uncertainty / Issues for Further Study*

The loads listed in Table 13 should be examined on more detail.

2.8 Whole Engine Mechanical Modelling

2.8.1 *System Description*

The gas turbine rotor system is very large and very heavy. The distance between the bearings is ~10 metres and the weight of the rotor is ~ 33 tonnes. The rotor is connected to the electrical generator through a flexible connection, so the two rotor systems may be considered to be separate systems. The weight of the electrical generator is approximately 35 tonnes. The rotors are mounted vertically and are supported by electromagnetic bearings. There are two axial EMBs one just above the gas turbine rotor and one just below the electrical generator system. There are 4 radial EMBs two at each end of the GT and two at each of the electrical rotor system. To support a weight of ~35 tonnes the EMB would need to be bigger than any EMBs yet produced in the world. Being vertically mounted the function of the radial bearings is to react any sideways loads as a result of rotor unbalance or to react to rotor movements due to other causes (e.g. seismic events).

2.8.2 *Prediction of the Critical Speeds of the GT Rotor System*

The rotordynamic assessment considered the critical speeds of the rotor system and tried to identify associated risks. The original OKBM design was assessed and also the shorter gas-turbine proposed for the combined cycle application (see Section 4). This preliminary assessment did not model the stiffness of any of the surrounding structure and did not take into account the stiffness or damping effects of the electromagnetic bearings (EMB).

The Rolls Royce proprietary code Rotordyn was used to do a preliminary analysis of the GT rotor system. The geometry of the rotor system was estimated from data provided by OKBM and estimates of blade weights and numbers were made. It was assumed that steel or Inco would be used for the rotor. For the bearing stiffnesses two conditions were assumed:

- Rigidly connected to earth (using a spring stiffness of 1 x 10^8 N/mm).
- A flexible bearing support was also considered, as it was thought that there maybe considerable flexibility in the facility casings. This was very difficult to estimate so a value of 1 x 10^5 N/mm was used, a value that is has been used in preliminary studies for aero applications.

The critical speeds calculated by Rotordyn for these two boundary conditions are given in Table 14 and Table 15 and shown graphically in Figure 39 and Figure 40.

Table 14 Critical Speeds Calculated with Rigid Bearing Support (K= 1 x 10^8 N/mm)

Mode	RPM	%N	Direction
1	689.8		backward
2	**699.2**	15.9%	forward
3	2485.3		backward
4	**2545.4**	57.9%	forward
5	4034.3		backward
6	**4241.6**	96.4%	forward
7	5444.2		backward

Table 15 Critical Speeds Calculated with Flexible Bearing Support (K = 1 x 10^5 N/mm)

Mode	RPM	%N	Direction
1	520.2		backward
2	**522.3**	11.9%	forward
3	1177.3		backward
4	**1198.0**	27.2%	forward
5	1897.6		backward
6	**1979.7**	45.0%	forward
7	3293.6		backward
8	**3596.0**	81.7%	forward
9	6083.9		backward

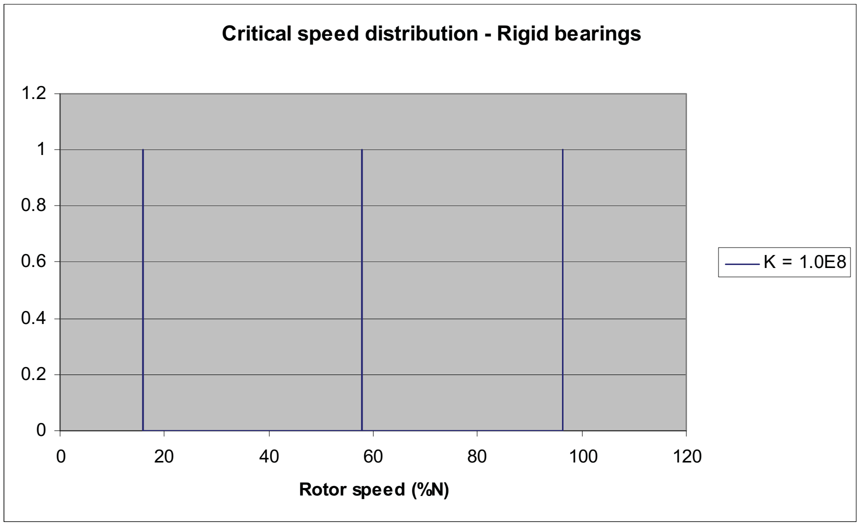


Figure 39 Critical Speeds, Rigid Bearings

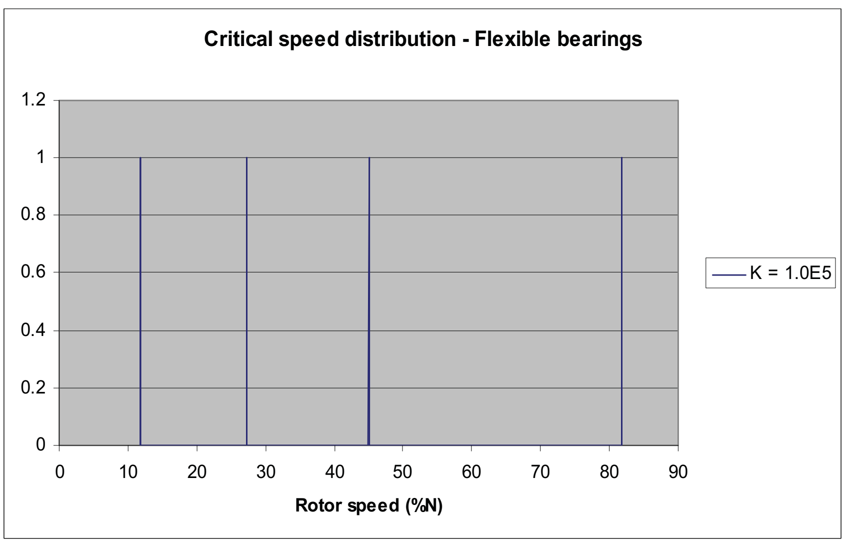


Figure 40 Critical Speeds, Flexible Bearings

Figures 37 to 40 show the mode shapes of the 4 forward critical speeds predicted for the flexible bearing configuration (K = 1 x 10^5 N/mm).

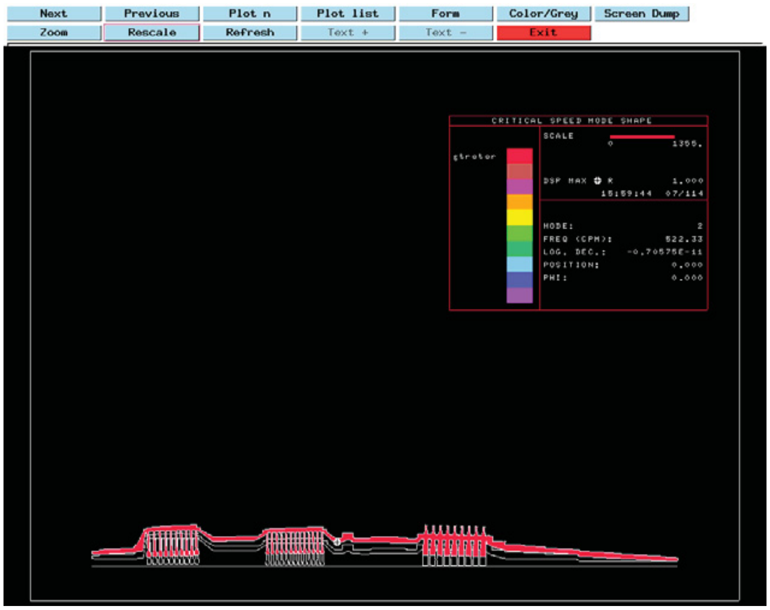


Figure 41 Flexible Bearings - First Forward Critical Speed 522rpm (11.9%N)

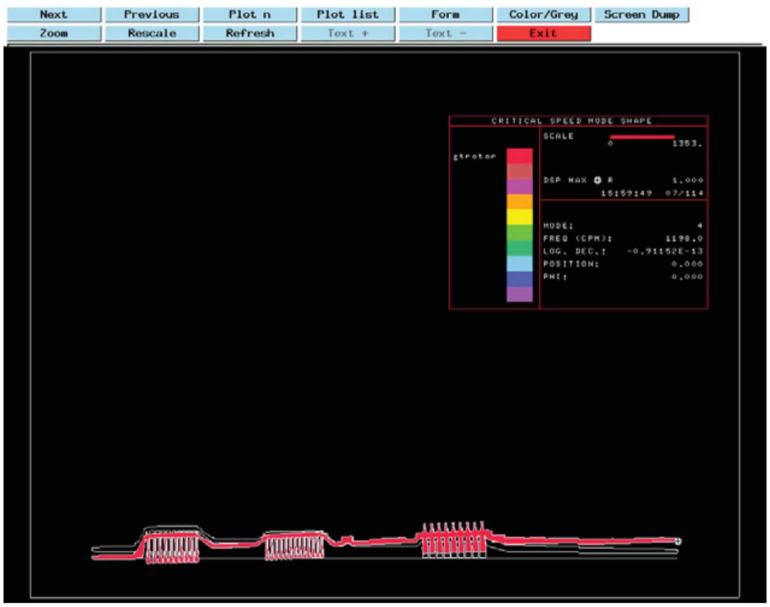


Figure 42 Flexible Bearings - Second Forward Critical Speed 1198rpm (27.2%N)

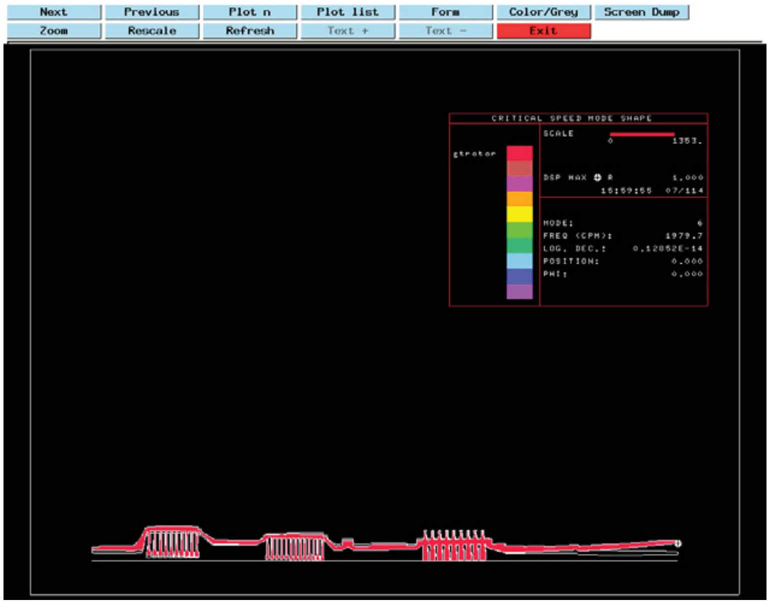


Figure 43 Flexible Bearings - Third Critical Speed 1980rpm (45%N)

Figure 44 Flexible Bearings - Fourth Critical Speed 3596rpm (81.7%N)

The GT will run at a constant (maximum) speed of 4400rpm. It was estimated that it would take around 15 minutes to accelerate the rotor system up to this speed. The EMB control system will be designed to modify the bearing support stiffness (and damping) so that the rotor may pass through each critical speed, keeping its response to any imbalance within limits.

For the rigid bearing condition Rotordyn calculated 3 critical speeds within the running range. The highest, at 96%N, would be a problem so close to the running speed. The rigid bearing configuration is much stiffer arrangement than expected in practice, but it is useful to note the response of the rotor for the stiffest possible support configuration.

The flexible bearing configuration gave 4 critical speeds in the rotor running range, the highest at 82%N.

The key issue is designing the EMB to control the response of the rotor when passing through these critical speeds. A properly designed control system with filters at the correct frequencies to modify the EMB response will be essential. In addition to the EMB there are secondary or back up bearings, which are conventional bearings. These are set at a smaller gap than the normal working gap within the EMB, such that the catcher bearings are engaged before the rotor comes into contact with the EMB magnets. These bearings support the rotor if the EMBs fail (e.g. if the electrical supply fails) or if there is some shock to the system that the EMBs cannot cope with, as EMBs are very poor in handling transient behaviour. It will be necessary, therefore, to analyse the rotor system without the EMBs functioning to see how the system will rundown. Note that it is assumed that the reactor scrams and the turbine bypass valve opens on a loss of the EMBs.

2.8.3 *Steady-State Forced Response to Out of Balance*

A steady state forced response was run in Rotordyn to see how the rotor would respond to a nominal out of balance of 10 oz.in (7200 g.mm). This represents the out of balance that would expect from blade failure or partial blade failure. The maximum response in the rotor to this out of balance was 0.007mm. This is a low value and suggests that out of balance due to blade loss would not be a problem.

Balancing however, may be quite a significant problem because the rotor system is very heavy and because it will require dynamic balancing, as it will be operating above several critical speeds. Rotor balancing is not within the scope of this assessment but it will be important to determine, at some stage, how well the system may be balanced and what residual imbalance could remain after balancing. This will be very important for the radial EMB design.

The response of the rotor systems to seismic activity has not been covered in this assessment. Shock loading is likely to cause sufficient displacements for the back-up bearings to be contacted. Whether there is a need for a run-through capability, whereby the turbomachinery continues operation through a transient engagement of the catcher bearings, has yet to be established.

2.8.4 *Conclusions*

- For the OKBM GT design 4 critical speeds of the rotor were calculated in the running range. This is assuming a bearing stiffness of 1.0 x 10^5 N/mm.

- The response of the rotor when passing through these critical speeds will be controlled by the radial electromagnetic bearings.

- It is essential that the control system of the EMBs is corrected tuned to the critical speeds of the GT rotor. Filters will be required within the control system at the critical speed frequencies to respond correctly when the rotor is at these speeds.

- It is important to know how the system will respond if the EMBs fail (say due to loss of power). In this situation the loads will be taken by the conventional back-up bearings. It will be important to know how the system will rundown if the rotor only has the support of the back-up bearings.

- Blade failure is not seen as a problem from a rotordynamic point of view because of the large total weight of the complete rotor system.

- Rotor balancing could be difficult due to the large weight and the requirement for dynamic balancing, as the rotor will be operating above several critical speeds of the system.

- The back-up bearings could see skidding of the rotor because the rotor system is mounted vertically and thus may be loaded relatively lightly in the radial direction.

- Rotor over-speed conditions and control system responses need to be established.

- Disc failure need to be considered for containment and impact considerations.

- Response of the rotor system to seismic activity is required.

2.8.5 *Areas of Uncertainty / Issues for Further Study*

Need EM bearing stiffness.

2.9 Implications of Radioactivity for Repair and Overhaul

Radioactivity issues exist with a range of isotopes that are by products from the fuel used in the nuclear reaction process. Of main consideration for the GT-MHR are silver Ag110, caesium Cs134 and Cs137. These have beta and gamma radiation. Table 16 shows the radioactivity issues associated with these isotopes.

Table 16 Radioactive Characteristics of Isotopes Likely to be Released from TRISO Fuel

Isotope	Half-life	Radiation
Caesium 134	2.1 years	Beta and gamma
Caesium 137	30 years	Beta and gamma
Iodine 131	8.1 days	Beta and gamma
Tellurium 132	78 hours	Beta, gamma and X-rays
Silver 110m	270 days	Gamma

The available information on decay of these isotopes, shows that activity levels and likely decay after nearly seven years of operation, and the decay over four years would be as shown in Table 17.

Table 17 Relative Radioactive Decay of Caesium and Silver Isotopes

Isotope	% of gamma dose rate at 7 years of operation	Activity level from decay of initial dose rate in one year	Activity level from decay of initial dose rate in 3 years	Activity level from decay of initial dose rate in 4 years
Ag110	50%	36%	5%	2%
Cs134	25%	71%	36%	26%
Cs137	25%	98%	93%	91%
Weighted mean of initial dose rate	100%	60%	35%	30%

Consequently a storage period of 3-5 years is likely prior to any form of turbine replacement activity. However this has to be assessed further because it is thought that the concentration of Ag110 could between 1-10 part per billion and Cs and Te 10-100 parts per billion. In addition Te does not go well with Nickel based alloys, which are common on high temperature gas turbine systems.

Decontamination studies have been undertaken on the German HHT programme, which has indicated it is possible to undertake decontamination processes with Cs. This may reduce the issues with caesium, but this needs further evaluation.

The safe turbine replacement process would have to be developed with health physicists, and appropriate safety measures and equipment developed or acquired depending on the policy defined for GT overhaul. Removal and replacement of PCS components. Depending on the analysis of the impact of radiation after a period of storage combined with the hardness of the gamma radiation which will define the thickness of screens etc, and the amount of contamination (in parts per million) from the nuclear reaction and the overall health and safety evaluations it will be possible to define an appropriate Repair and Overhaul policy for the whole GT and the turbine. Gamma rays from Ag110 are harder than those from Cs, and this will define the level of shielding.

From the available information there are clearly a range of risks that are present, and the mitigation of these will be a combination of:

- Safe working processes and containment of any radiation during the GT removal process.
- Containerisation of the GT for a period of 3-5 years for decay of the radioactivity to more acceptable levels.
- Possibly decontamination tasks for Cs.
- Strip and build processes either using automated equipment, or remote handling equipment or if the radioactive contamination is at a safe level the use of humans possibly in radiation suits and gloves.

To enable this, a design philosophy for the plant may be to have a crane installation that allows the GT to be removed on a contained and cleanable/filtered environment and places the contaminated

GT in a sealed screened container in the facility, and then installs a replacement GT that has been delivered to the facility.

Depending on the anticipated reliability, and the impact of external objects on any other failure modes consideration may be needed for up to three GT units being available per reactor plant. One installed, one on standby, and one in radioactive storage decontamination and overhaul. This would guarantee high utilisation of the plant for power generation, but would come at a cost. If these plants are clustered then it would be possible to have a single GT as a spare for a number of them, with others in operation or radioactive storage. This would have to be defined on the basis of probability for failure to define any spares requirements.

From the work so far this is clearly not conclusive, and hence analysis needs to be part of the next phase of the study.

Within the study, consideration is also needed with the design for maintainability. Assessment would be needed of the design needed to support automation or remote equipment, which would impact the selection of nuts/bolts/fasteners/locking techniques and the associated flanges. But also the types of tooling used and the compatibility with any other equipment, spacing between components assuming gloves may be used, 'observability' and access to components if camera or masks are to be used, and the process for manoeuvring the GT to gain access and remove the turbine within an automated or remote process in relation to the types of robotic or remote equipment required.

The acceptable safe radiation exposure limits in the UK are as follows: and are defined against the types of gamma radiation and severity defined in rem units, where 100rem = 1Sv. The notional maximum doses are <50mSv per year, but a dose over 5 years of <100mSv. These values vary for specific circumstances of 'radiation worker'.

3 Alternative PCS Designs for NGNP

3.1 Layout Design Objectives

In the assessment of the GA/OKBM reference PCS design, Rolls-Royce concluded that the concept was elegant and achievable, but carried significant key risks. These were identified as follows:

- Recuperator life and cost considered very high risk.
- Active electro-magnetic bearing/catcher bearing requirements are outside of current world experience - high risk.
- Cost of power electronics required for 4400rpm/286MW generator anticipated to be large (~$50M) - high risk commercially.

Several design options have been investigated in an attempt to best satisfy the following brief list of perceived layout requirements. Each feature introduced attempts to simplify, cost reduce or de-risk the equipment.

3.1.1 *General Requirements*

<u>Simplest solution is best</u>

Rationale: Easier to analyse – less to go wrong.

The general approach to creating an alternative layout was to develop and simplify the current GT-MHR proposal but given more time some significant assumptions would have been re-examined, i.e.:

- The maximum environmental temperature of the magnetic bearings could be increased, especially with reference to the Japanese design. This would give greater freedom for the design layout.
- The impact of casing seal clearances on rotor stability, and the need to understand its dynamic response during shut down with failed electromagnetic bearings, may lead to the currently proposed assembly method being unacceptable.
- Catcher bearing technology acquisition, especially foil bearings, and appropriate damping mechanisms
- Seal technology. Should the cylindrical seals between casing and vessel, and bellows between casing and vessel be replaced by fixed casings on the gas turbine and piston seals between the GT and vessel?

Small footprint for equipment exposed to possible contamination

Rationale: Reduced land usage, lower building costs.

- Retain vertical orientation.
- Minimise vessel diameter.

Reduce risk of heat exchanger failure

Rationale: Current GT-MHR recuperator appears to be a leap in technology, could use a proven steam generator design using known construction methods.

3.1.2 *Vessels*

Two smaller cylindrical pressure vessels are more feasible than one large one

Rationale: Large size may limit the number of possible suppliers and make transport to site difficult.

Simple scaling at constant wall stress suggests two small vessels should not increase material volume significantly.

Maintain low temperature near pressure vessel walls

Rationale: Maintain material strength, especially with respect to creep. This means the relatively cold steam generator exit flow should be used to bathe steam generator and PCU vessel walls.

There are reduced internal stresses due to differential thermal expansion if temperature excursions are minimised.

Use concentric ducts between pressure vessels

Rationale: Maintain lower temperature duct wall by using cooler outer flow to shield hot inner flow.

Reduces number of possible overboard leak paths.

Use three vessels arranged 'in a line'

Rationale: Reduces bending stresses on interconnecting tubes.

External sliding supports for GT-PCU vessel co-planar with reactor interconnecting tube

Rationale: Minimises bending stresses in interconnecting tube.

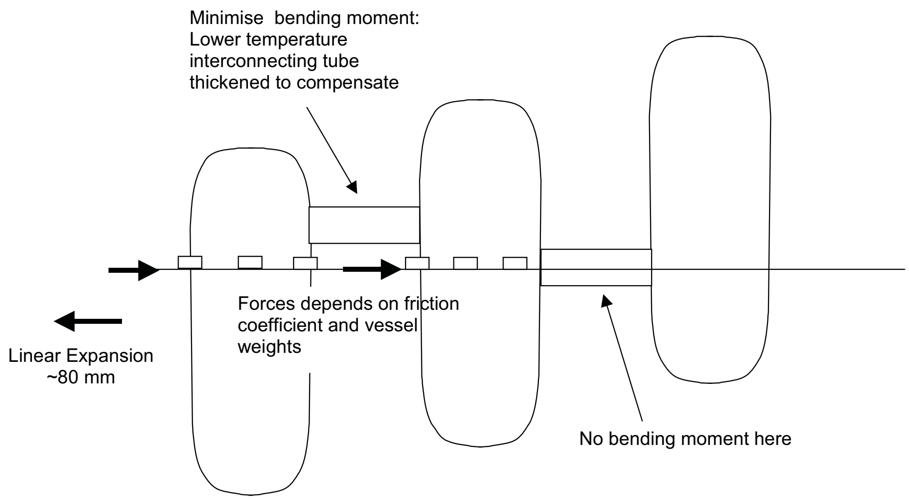


Figure 45 Concept of Combined Cycle Vessel Mounting

3.1.3 *Shaft Orientation and Main Bearings*

<u>Minimise static force on journal bearings</u>

Rationale: Electromagnetic bearings must maintain a reserve capacity to allow for the correction of dynamic forces. A vertical orientation means the journal bearings have little static load thus most of the capacity can be allocated to dynamic load capacity.

During acceleration to full speed the rotor system must pass through several potential resonant speeds. To minimise lateral excursions the bearing stiffness must be adjusted as a function of rotor speed. This is achieved by increasing the ‘pulling force’ on each opposing magnetic pairs. Thus to allow maximum capability to vary the stiffness the static load should be minimised

Note: When running hydrodynamic bearings have a large static load capability and a much larger dynamic load capacity before contact occurs. This is because it takes some time for the fluid to escape as bearing faces are forced together. Unfortunately potential contamination by lubricating liquid lubricant is unacceptable and gas bearings have not yet achieved an acceptable state of development.

<u>Reduce rotor weight</u>

Rationale: Ease catcher bearing design, Reduce the amount of electromagnetic bearing development. Combined cycle leads to significant reduction in gas turbine generator weight because power from GT generator is greatly reduced.

3.1.4 *Catcher Bearings*

Rolling element journal bearings are assumed on the gas turbine at compressor inlet and turbine exit and at each end of the generator

<u>Rolling elements must not contact rotating shaft during normal operation</u>

A method of attracting the rollers to the fixed outer race is required, perhaps by magnetic attraction.

<u>Must adequately load journal catcher bearing to reduce 'skidding'</u>

Rationale: Catcher bearing design is the limiting factor to allow the rotor system to survive EMB failure. The loss of control of bearing stiffness combined with the inevitably long rundown time increases the chance of damage to compressor and turbine blade tips. On detection of EMB failure a method of loading the roller bearing is required to ensure that the elements do not skid thus creating possibly large transient forces and significant local heating.

<u>Foil bearings may provide a suitable journal catcher bearing function</u>

The claimed benefits of foil bearings are

- No external gas supply is required – should continue to operate if external supplies fail
- The bearings are non-contacting in normal use
- High surface speeds are desirable, unlike the normal D-N limitation for rolling element types
- Good transient capability. Lower technical risk than highly loaded journal bearings. Good load capability

Rolls-Royce has limited experience in the use of foil bearings hence a significant technology acquisition programme

<u>Active magnet thrust bearings may present a lower technical risk than journal bearings</u>

It appears feasible to simple sliding surface thrust bearing to allow the rotor system to safely coast to rest.

In the worst case the all the rotational energy in the rotor would be dissipated in the catcher thrust bearing. There is mitigation as some energy will be dissipated by pumping helium through the by-pass valve and some may be absorbed by the generator, but a significant proportion will inevitably end up as heat in the catcher bearing.

3.2 Alternative Layout Designs and Configurations

Several alternative configurations were studied. These are summarised below:

3.2.1 *Twin Shaft - in Parallel*

It is possible to reduce rotor weight by dividing the power into multiple units i.e. gas turbines in parallel. However the helium flow through each compressor will be difficult to predict and control because one shaft may have a slightly better turbine and worse compressor which could lead to gross power imbalances between the shafts. Compressors operating in parallel are notoriously unstable as it is extremely unlikely that their performance characteristics will match at all operating conditions. This option has therefore been rejected.

3.2.2 *Twin Shaft - in Series*

Another arrangement considered was having two shafts in series, as shown schematically in Figure 46. This would prevent the control issue of the twin parallel shafts arrangement. The design chosen was to have a high pressure turbine and low pressure turbine in series within single PCS vessel of current size. This looks feasible and with shorter shafts and half-sized generators it would reduce bearing loads and shaft dynamics issues. When looked at in more detail, however, the complexity of the interconnecting pipework makes this option look very difficult to design and maintain. Figures 46 to 51 give an indication of how complicated the installation in one pressure vessel would be.

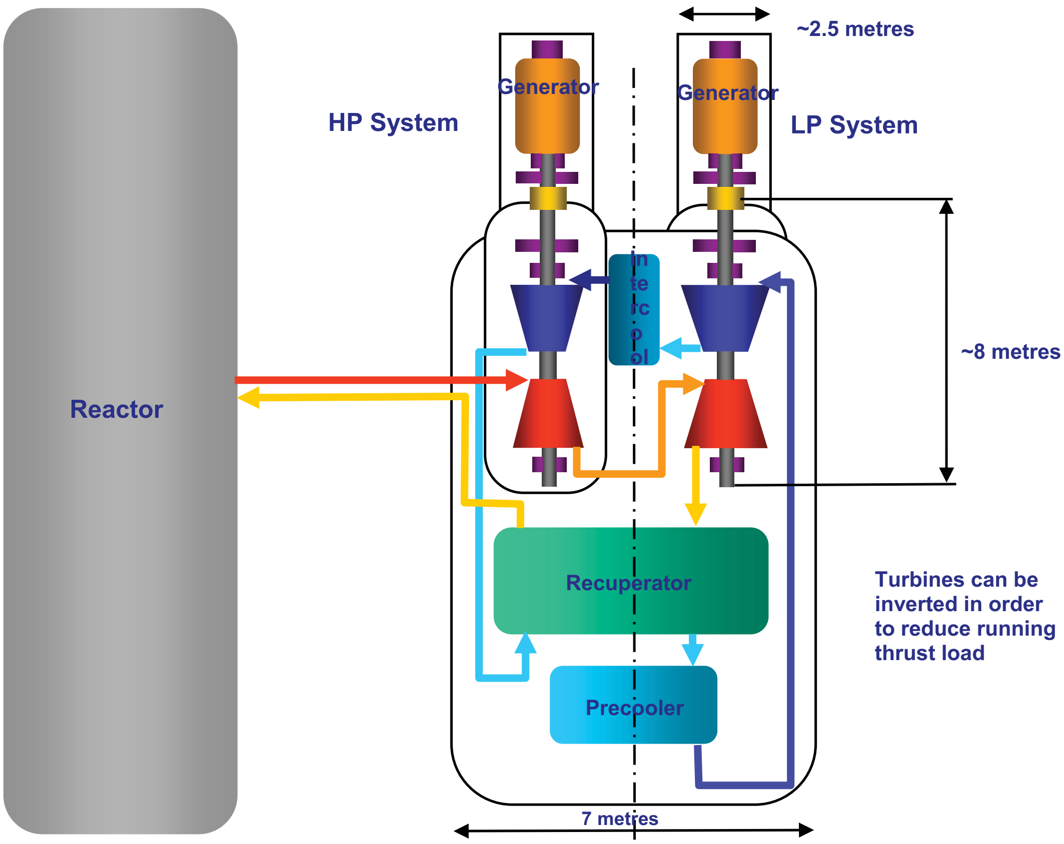


Figure 46 Schematic of In-Series Twin Shaft Design

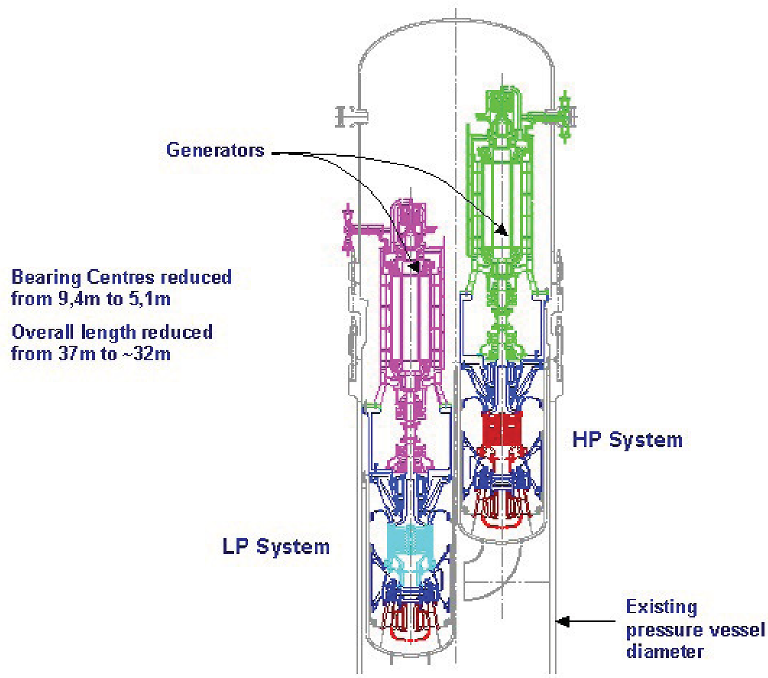


Figure 47 Overview of Twin Shaft Concept

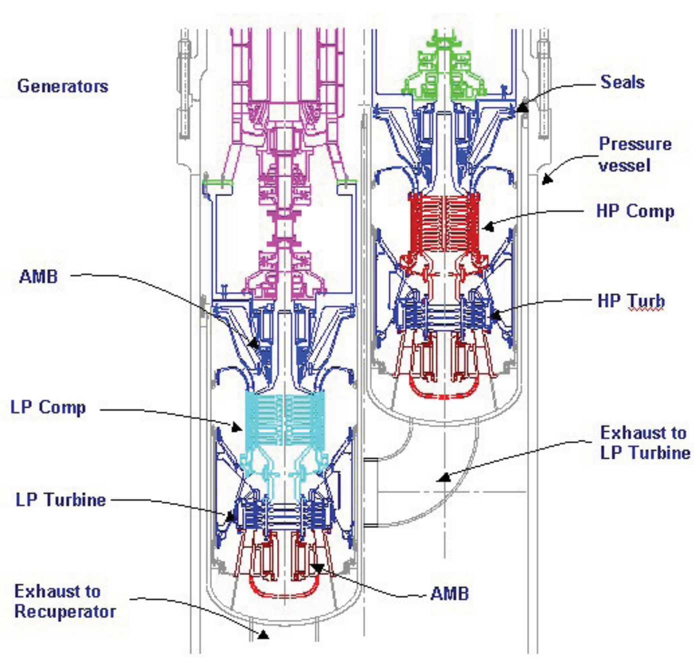


Figure 48 Twin Shaft - General Layout

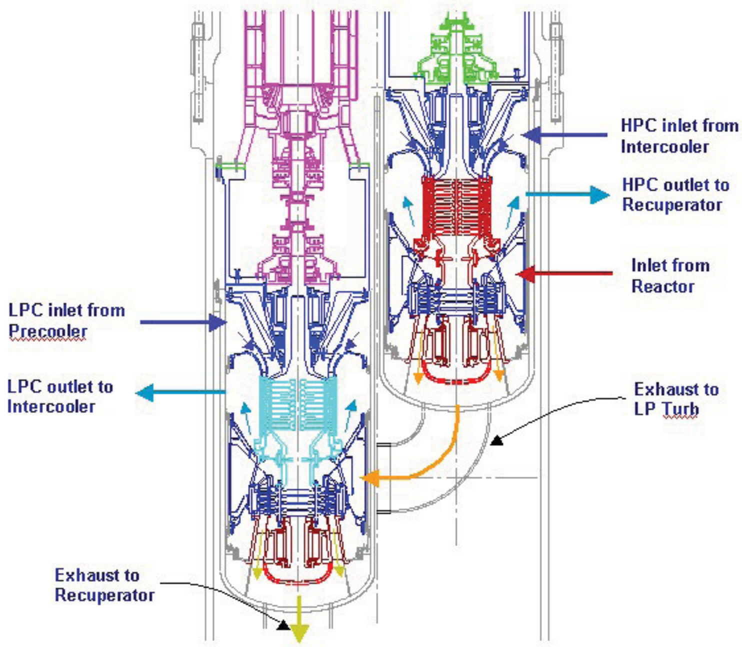


Figure 49   Twin Shaft - Gas Flows Through Turbomachinery

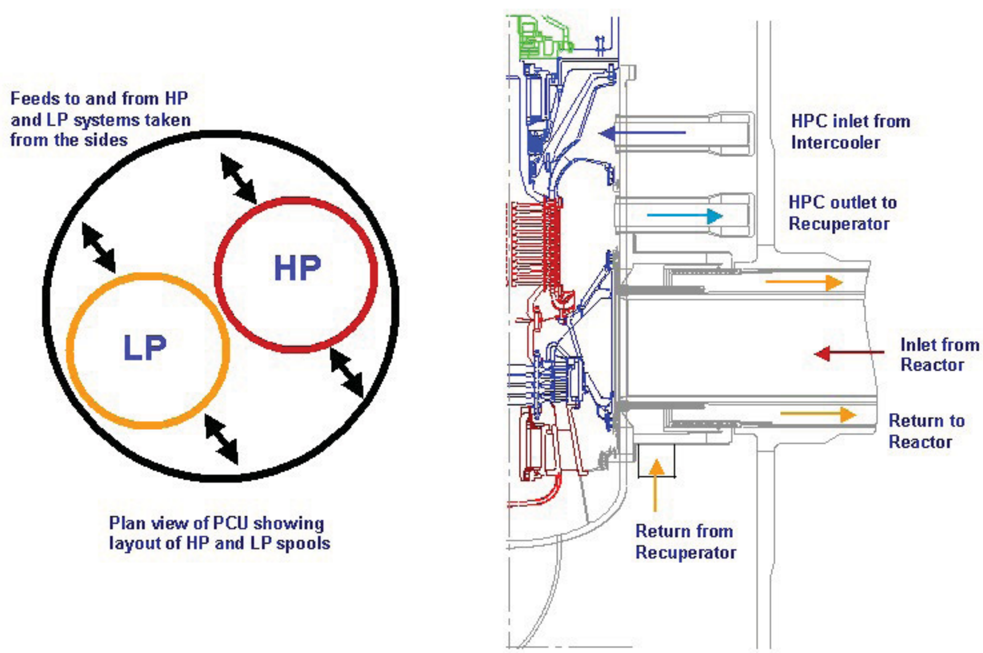


Figure 50   Twin Shaft - Internal Pipework

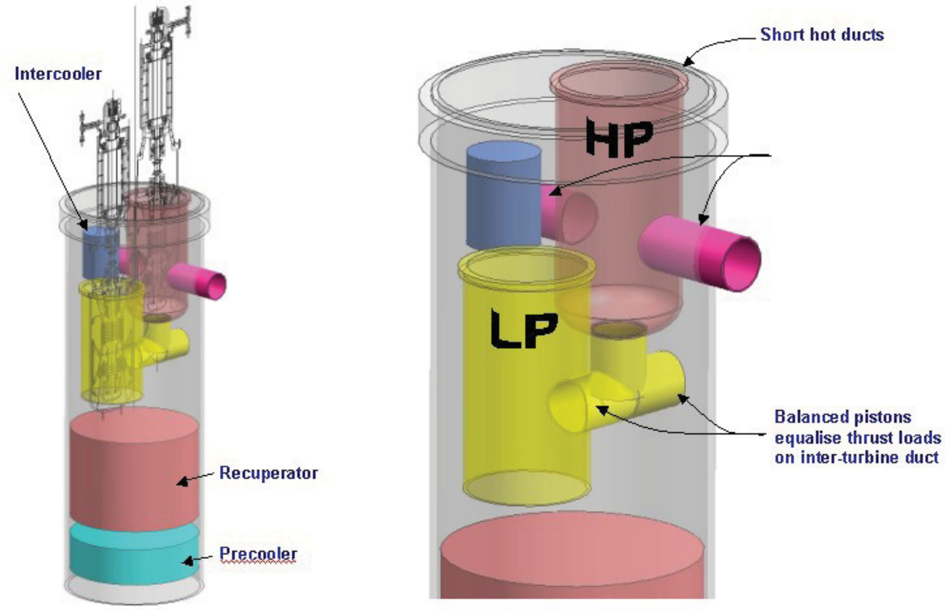


Figure 51 Twin Shaft - Vessel Loading

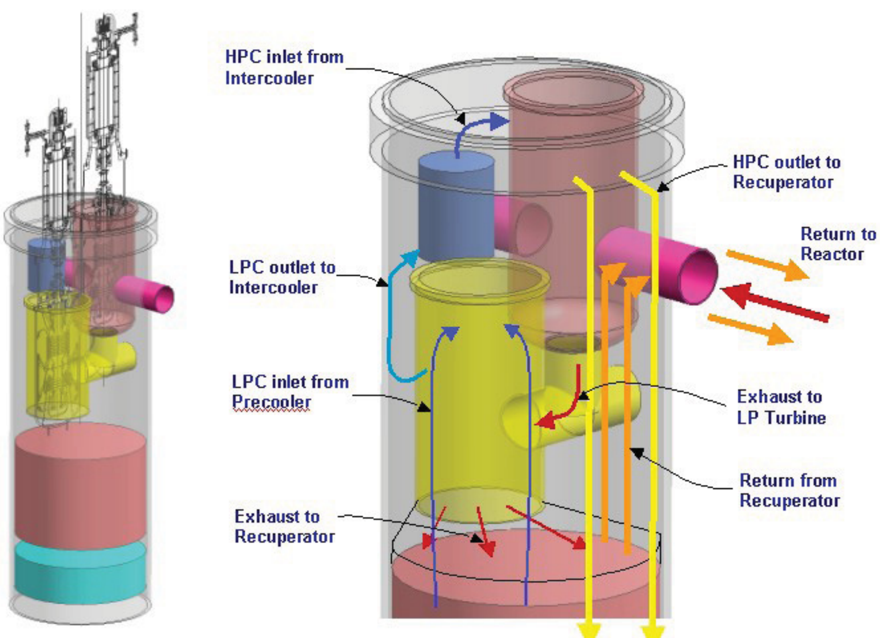


Figure 52 Twin Shaft - Gas Flows To/From Turbomachinery

Strengths:

- One pressure vessel
- Shorter rotors allow possibility of higher speeds, hence smaller rotating machinery
- Bearing Centres reduced from 9.4m to 5.1m
- Overall length reduced from 37m to ~32m
- Contained within existing 7m vessel diameter
- Simpler recuperator layout
- Simpler precooler layout
- Smaller generator

Weaknesses:

- Heat soak into intercooler during shutdown
- More complicated vessel
- Complex pressure-balanced high pressure duct work
- Potentially hotter EM bearings

<u>Conclusions</u>

Twin spool version is feasible:

- Short shafts may ease shaft dynamics issues
- Should have reduced bearing loads

But:

- Excessive complexity of high pressure ducting
- More expensive than OKBM version
- Turbine bearings are in a hot region
- Recuperator access still difficult

Although perhaps theoretically feasible the added complication of the proposed ductwork seemed to be problematic and expensive. This option was rejected as a result.

3.2.3 *Free Power Turbines*

The use of a free power turbine in single vessel or in two vessels had two aims: lower bearing load and retention of synchronous generation (to save cost in the power electronics).

The concept illustrated in Figure 53 has two rotors, each of relatively low weight. The gas generator would be compact and could be mounted horizontally. The power turbine would need to be pressure balanced. The most convenient way to achieve this would be a double-ended turbine design in which hot helium enters near in the middle then flows out in both directions along the shaft. Clearly this doubles the number of turbine stages and doubles component costs. Also a separate starter motor would be required because there is no direct connection between the generator and compressor – this would add extra cost and complexity.

All the presented multi-rotor systems require turbines in series for stability reasons. This means hot, high-pressure inter-turbine ducts of complex shape. Reliability would be problematic.

Maintenance might be cheaper for this layout due to the need only change the HP shaft after 60 000 hours while the power turbine may last considerably longer.

In the layouts considered so far there appears to be the need for at least two magnetic bearings to operate in a high temperature environment.

This option has been deemed unattractive. No detailed layout design has been created.

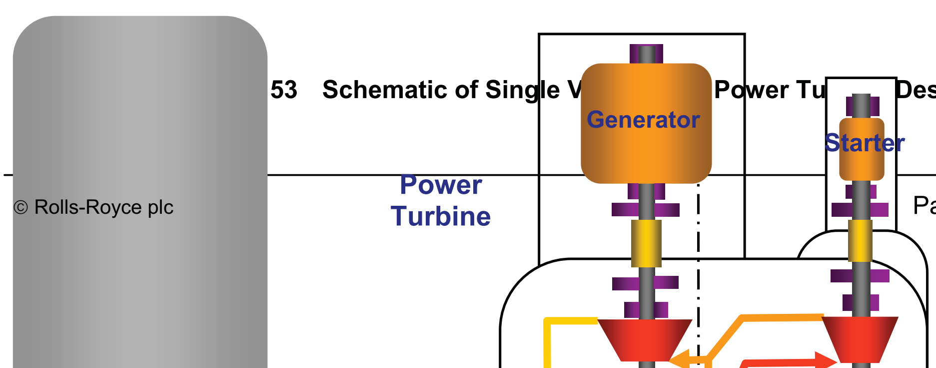
<u>Free Power Turbine - Single Vessel</u>

Strengths:

- Shorter rotors
- Contained within existing 7m vessel diameter
- Small gas generator could be mounted in small separate pressure vessel
- Lower speed generator & free power turbine longer life, less frequent service intervals
- Synchronous generator – higher efficiency

Weaknesses:

- More complicated vessel
- More complicated ducts
- More bearings
- May need two generators in series to lower thrust load



53 Schematic of Single V Power Tu Design

Free Power Turbine - Twin Vessel

Putting the power turbine in a separate pressure vessel was also considered.

Strengths:

- Shorter rotors
- Contained within existing 7m vessel diameter
- Small gas generator could be mounted in small separate pressure vessel
- Lower speed generator & free power turbine longer life, less frequent service intervals
- Synchronous generator – higher efficiency

Weaknesses:

- Extra pressure vessel
- More complicated ducts
- More bearings
- May need two synchronous generators in series to lower thrust load

3.3 Performance Cycles

Rolls-Royce has looked at several alternative PCS concepts that mitigate some of the key risks identified in the reference GT-MHR design. The most promising alternative concept appears to be a combined cycle, described in Section 3.3.4. This has been worked up to a pre-concept level to understand feasibility and make comparisons with the reference cycle. The conclusion of this work is that the combined cycle option looks feasible and may be slightly more efficient than the reference cycle. Its costs should be similar and the concept goes some way to mitigating the key risks identified with the GT-MHR design.

3.3.1 *Simple Cycle*

A simple cycle was modelled by removing the recuperator from the existing cycle. Heat was rejected from the cycle through a precooler prior to compression and an intercooler midway through the compression. As expected, this cycle performs poorly. The cycle efficiency was in the low 30%s and even this could only be achieved with a pressure ratio exceeding 10 (which would require a very large number of compressor stages). This cycle was not pursued further.

3.3.2 *Reheated Cycle*

In a reheated cycle the partially expanded helium is returned to the reactor for reheating to the reactor outlet temperature. The expansion is then completed in a second turbine. The arrangement of recuperators, intercoolers and precoolers was the same as that in the GT-MHR cycle. It was found that the cycle efficiency improved by a couple of percentage points relative to the current cycle at constant reactor outlet temperature. However, the addition of heat part way through the expansion greatly increases turbine outlet temperature and hence (following heat transfer across the recuperator) greatly increases reactor inlet temperature. Given also that reheat would require a redesign of the reactor, this cycle was not considered further.

3.3.3 *Double Intercooled Cycle*

In the current cycle heat is rejected from an intercooler approximately midway through compression. This improves the efficiency of the cycle by reducing the work required to compress the helium to reactor inlet pressure. The heat the intercooler rejects from the cycle is recovered in the recuperator and hence the intercooler does not result in a reduction of reactor inlet temperature (and hence a need for a higher reactor power to achieve the reactor outlet temperature at a given mass flow). Adding a second intercooler is an obvious means of improving the thermodynamic performance of the cycle. The total work of compression is further reduced and the reactor inlet temperature can still be maintained. The recuperator is required to transfer 10% more heat as the pressurised cold side inlet (down stream of the compressors) is reduced in temperature. It was found that a double intercooled cycle improved the cycle efficiency by ~1 percentage point.

There are considerable disadvantages, however. Introducing an extra intercooler would require an extra compressor with associated offtakes and returns (and a lengthened shaft) and, as noted above, the recuperator would need to be 10% larger. It is considered unlikely that the relatively small improvement in cycle efficiency would be sufficient to offset the extra complication and length/weight of PCU that such a cycle would require.

3.3.4 *Combined Cycle*

The GT-MHR cycle recycles the heat remaining in the turbine exhaust back into the reactor inlet stream and therefore makes good use of the lower grade heat in the cycle. An alternative means of using this low-grade heat is in a combined cycle, where the recuperator, precooler and intercooler are effectively replaced by a steam generator (downstream of the gas turbine). The steam produced is expanded in a conventional steam turbine located on a separate shaft with its own generator. The gas turbine is retained in the primary reactor coolant circuit where it performs three functions:

- To act as a coolant 'circulator' to drive the helium through the reactor.
- To extract mechanical work and produce some fraction of the whole cycle electrical output.
- To reduce the temperature of the helium gas stream to levels that can be tolerated in the steam generator.

The efficiency of a combined cycle is maximised when the steam turbine inlet temperature is maximised. Metallurgical limits tend to limit steam temperatures to between 550˚C and 600˚C and therefore a GT-MHR combined cycle would be optimised into this range. At these conditions the bottoming cycle has notable similarities with the steam cycles employed in UK nuclear plant, in particular the Advanced Gas Reactor (AGR) stations.

<u>Similarity of the Bottoming Cycle to Existing UK Nuclear Plant</u>

In AGR power stations, the reactors are cooled by carbon dioxide pressurised to around 40bar. The carbon dioxide is driven around the primary circuit by four gas circulators per reactor. The coolant is forced upwards through the reactor and then passes down through steam generators.

There are twelve of these per reactor and they are located around the periphery of the core within the concrete pressure vessel.

The AGRs were designed to use standard (at the time) fossil fired steam turbine plant. The steam temperatures are therefore well in excess of those in the previous generation 'Magnox' gas cooled plants and PWR plants. The thermal efficiency of AGR power stations is therefore high; values of 42% are typically quoted. The steam at turbine inlet is at around 170bar and 540˚C. After expansion to 40bar the steam is reheated in the steam generator to improve the power density of the cycle and improve the quality of the steam at the exhaust.

Reference 18 gives an excellent description of the steam generators in the AGR plants at Torness and Heysham 2, the last station pair to be commissioned in the late 1980's. This reference details the pressures and temperatures in both the gas and steam sides of the steam generators as well as giving information on the materials used and measured plant performance. The operating parameters are summarised in Figure 54 below.

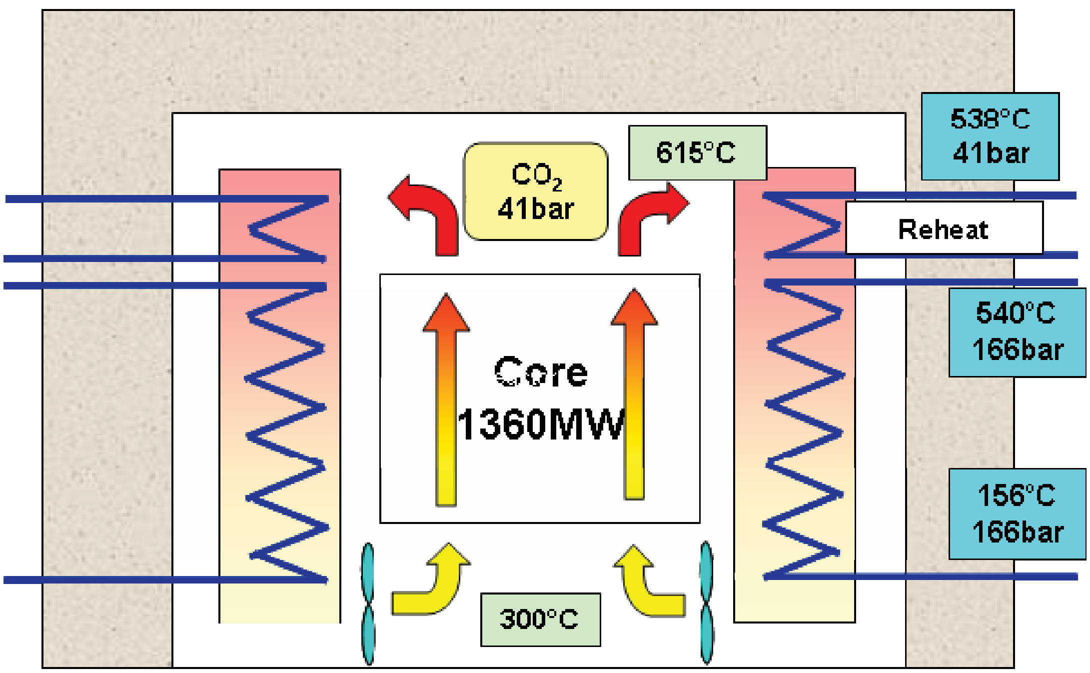


Figure 54 UK Advanced Gas Reactor (AGR) Plant

The similarities with the AGR cycle lend credibility to the choice of a combined cycle because of the generally successful performance of this plant over the last 30 years[2]. Most importantly, it

[2] The AGR steam generators have not given completely trouble free service and 2 of the plants are currently operating at reduced capacity due to concerns about weld integrity in the steam generator headers. Nevertheless these concerns have arisen after around 200 000 hours of service and the improved designs in the later plants have suffered far fewer problems. The operating experience gained with AGR plant would be invaluable should steam generators be designed for a GT-MHR combined cycle plant.

demonstrates that transferring heat from a pressurised coolant gas to a steam cycle in a nuclear environment, and operating such plant commercially, is readily achievable.

Thermoflex 16 was used to model a combined cycle in which the bottoming cycle matched the AGR cycle as closely as possible. At a reactor outlet temperature of 850˚C a GT pressure ratio of 1.95 was required to achieve a steam generator inlet temperature of 615˚C. A 75˚C minimum pinch between the steam generator inlet gas and reheat/superheat was assumed, giving steam conditions at turbine inlet of 540˚C and 166bar. It was found that the predicted cycle efficiency was approximately 49% offering a 1 percentage point advantage relative to the existing GT-MHR cycle[3].

Description of Proposed Combined Cycle Solution

The cycle proposed has deviated slightly from exactly replicating the AGR cycle as the bottoming cycle. The efficiency of the cycle has been improved by increasing the temperature of the steam at turbine inlet from 540˚C to 580˚C. This increase is justified because:

- Steam turbine technology has advanced in the interim with 600˚C inlet temperatures now a common maximum. A brief search has shown that plant for reheated steam cycles for use in combined cycles is available ‘off the shelf’, in the correct power range, with the capability of accepting steam at 177bar and 600˚C[4].

- The far superior heat transfer properties of helium relative to carbon dioxide permit a much smaller pinch between the gas inlet and steam outlet temperatures to be assumed. The pinch assumed is 40˚C.

- There is only a 5˚C increase in the steam generator inlet gas temperature (i.e. the hottest temperature that any of the components will need to withstand). This is judged to be insignificant.

The efficiency of the cycle has been enhanced by feedheating, i.e. using steam bled from the latter stages of the steam turbine to warm the returning condensate. A deaerator has also been modelled. The cycle is shown at a high level in Figure 55, more details of the fluid streams modelled in Thermoflex are shown in Figure 56.

[3] It should also be noted that if the reactor outlet temperature is reduced to 615˚C and the GT pressure ratio is set to 1, the cycle becomes an AGR cycle. When modelled in Thermoflex the efficiency predicted for this scenario was 42%, which is the value quoted for AGR stations. This gives confidence that the modelling assumptions in the Thermoflex steam cycle model are valid.

[4] The Siemens SST-5000 available at power outputs between 120MW and 500MW is an example of steam turbine plant with these capabilities.

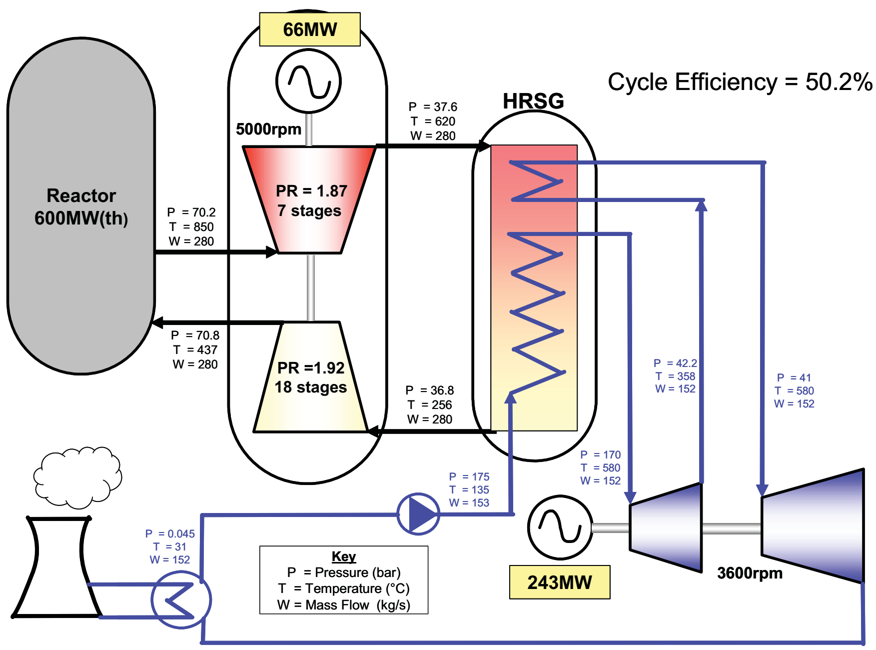


Figure 55 Proposed Combined Cycle for GT-MHR

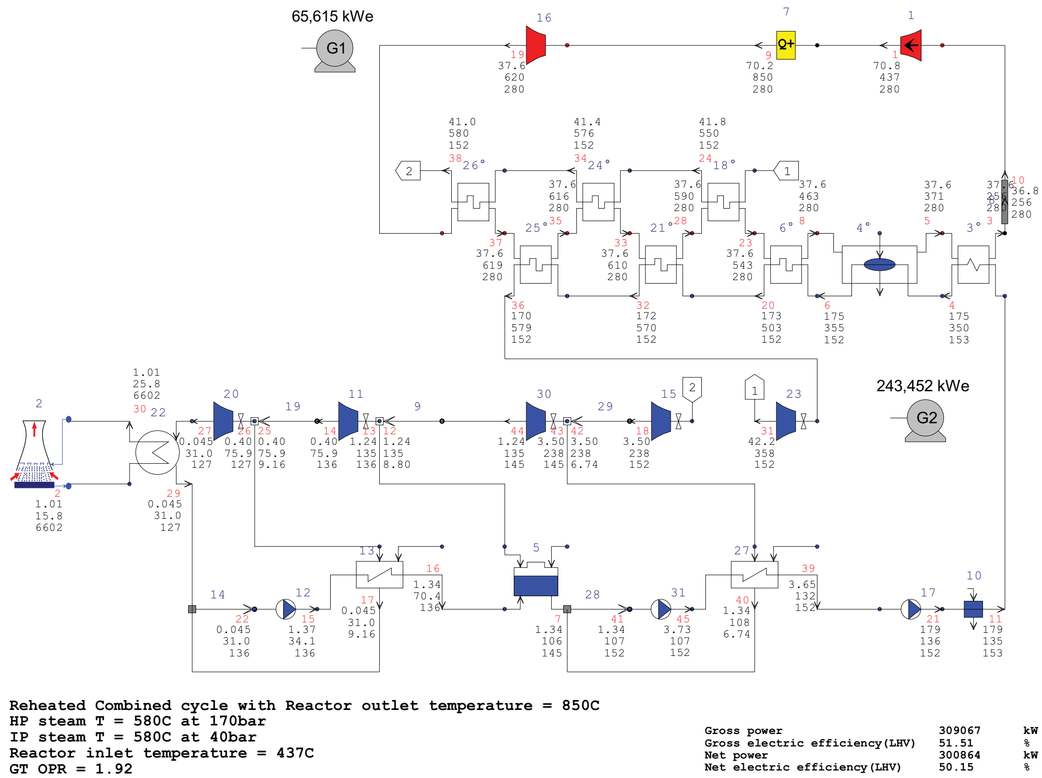


Figure 56 Proposed Combined Cycle for GT-MHR in Detail

Note: (number triplets for each stream are pressure [bara], temperature [˚C], flow [kg/s])

Figure 56 shows the superheater and reheater components 'interwoven' three times to achieve very similar superheat and reheat temperatures. This complex arrangement was set up for modelling convenience and would not need to be replicated in practice.

The following features of the cycle are worthy of note:

- A condenser vacuum of 45mbar has been assumed. This is consistent with sea water cooling at a coastal location and would be optimistic if a cooling tower is required (depending on the climate at the plant location). Nevertheless, the saturation temperature at 45mbar is 31˚C and it is judged that the cooling performance required to achieve this would be roughly equivalent to that required to achieve a minimum helium temperature of 26˚C in the GT-MHR cycle. Therefore, it is considered that 45mbar condenser pressure is a fair assumption to use when comparing the two cycles.

- Only 66MW of electrical power is generated on the GT shaft compared to 243MW in the steam cycle. This has three main advantages. First, the generator on the GT shaft will be much smaller and lighter reducing the weight of the GT shaft. Second, the steam turbine will run at synchronous speed and therefore only 66MW of power will need to be frequency converted by power electronics. This reduces the cost of the power electronics installation and also the inherent power losses. The third advantage is that the torque transmitted on the GT shaft is lower and hence the construction of the shaft can be less substantial and lighter.

- Given that both the pressure ratio and the power output are lower it might be expected that the GT turbomachinery would be substantially smaller for the combined cycle than for the GT-MHR cycle. The turbomachinery sizes scale with the quasi-dimensionless group W√T/P where W is flow (kg/s), T is temperature (K) and P is pressure (bar). A comparison of the values of this group for the GT-MHR and combined cycle turbomachinery is shown in □ below. Whilst it is clear that the turbomachinery is generally smaller for the combined cycle it is also clear that the reduction in size is nowhere near in proportion to the reduction of work done by the shaft.

- The net electrical efficiency of the cycle is a little over 50%. This is around 2 percentage points higher than the GT-MHR cycle.

Table 18 Comparison of GT Component Sizes for GT-MHR and Combined Cycles

Component/Stream	GT-MHR Cycle W√T/P	Combined Cycle W√T/P
LP compressor inlet	213	-
LP compressor output	144	-
HP compressor inlet	130	175
HP compressor outlet	86	105
Turbine inlet	153	134
Turbine outlet	343	223

<u>Proposed Combined Cycle Solution – Modelling Assumptions</u>

Care has been taken when modelling the combined cycle to make assumptions at a similar level of optimism/pessimism to those in the model of the GT-MHR cycle. Where components are common, for example the reactor, compressors and turbine, identical assumptions on pressure drops and efficiencies have been made. For components unique to the combined cycle, performance assumptions are based either on calculations (for example the pressure drop in the steam generator) or on readily available published data (for example the efficiency of the synchronous generator). The full set of assumptions and a comparison with those assumed in modelling the GT-MHR cycle are shown in Table 19 below.

Table 19 Modelling Assumptions GT-MHR and Combined Cycles

Parameter	GT-MHR cycle	Combined Cycle
Reactor power	600MW	600MW
Recuperator effectiveness	95%	N/A
LPC efficiency	89% poly (87.8% isen)	89% poly
HPC efficiency	89% poly (87.8% isen)	89% poly
Turbine efficiency	91.8% poly (93.2% isen)	91.8% poly
Recuperator $\Delta P/P$	2% on both hot and cold sides	N/A
Reactor $\Delta P/P$	0.85%	0.85%
Pre-cooler $\Delta P/P$	0.13%	N/A
Pre-cooler outlet temperature	299.2K	N/A
Intercooler $\Delta P/P$	1.6%	N/A
Intercooler outlet temperature	299.2K	N/A
GT generator efficiency	95%	96%
GT mechanical efficiency	100%	99% (based on turbine power)
Steam turbine efficiency	N/A	91.8% (poly)
Steam turbine mech efficiency	N/A	99%
Steam turbine gen efficiency (assumed synchronous)	N/A	98.5%
Condensate pump efficiency	N/A	85% (isen)
Reheater steam side $\Delta P/P$	N/A	3%
Steam generator water side $\Delta P/P$	N/A	5.0%
Steam generator + Reheat He Side $\Delta P/P$	N/A	2.2%
Steam generator Superheater & Reheater minimum pinch	N/A	40K
Condenser pressure	N/A	45mbar

One difference between the two sets of assumptions is the generator efficiency for the GT shaft. An efficiency of 95% is assumed for the GT-MHR cycle but 96% is assumed for the combined cycle. However, the GT-MHR cycle assumes that there are no mechanical losses on the GT shaft (or, rather, that the mechanical losses are included in the difference between the assumed 95% generator efficiency and the 97 to 97.5% efficiency actually expected for a generator with power electronics). The combined cycle assumes a mechanical loss of 1% on the GT shaft. Since the power debit due to shaft efficiency is calculated as a fraction of the turbine power (which is much greater than the net shaft power) the combined cycle treatment is the more pessimistic.

Combined Cycle at 950˚C Reactor Outlet Temperature

If reactor outlet temperature were raised to 950˚C the GT pressure ratio would need to rise to 2.4 so that the steam generator inlet temperature would stay constant and hence the thermodynamics of the steam cycle would be unchanged from the cycle at 850˚C. In contrast to the GT-MHR cycle, where increasing the reactor outlet temperature reduces the cycle pressure ratio (see Section 2.1.2), the increase in pressure ratio would require more compressor and turbine stages with the inherent increases in GT shaft length and weight.

The electrical power generated on the GT and steam turbine shafts would be 87MW and 227MW respectively. The net electrical efficiency is predicted to be 52.4%. Figure 57 below compares the GT-MHR and combined cycle efficiencies at reactor outlet temperatures of 850˚C and 950˚C.

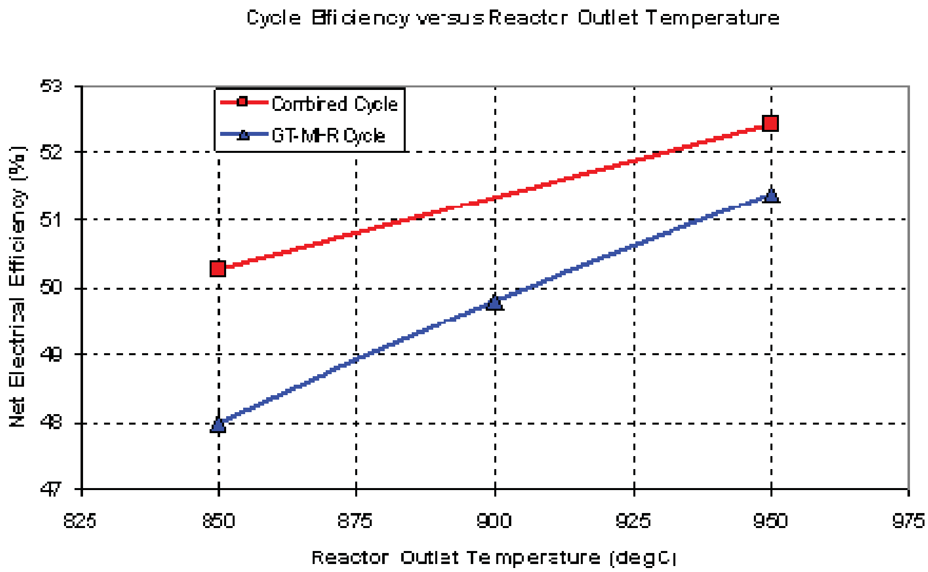


Figure 57 Comparison of GT-MHR and Combined Cycle Net Efficiencies

3.3.5 *Combined Cycle – Off-Design Performance Modelling*

A model of the combined cycle has been built in the Rolls-Royce corporate performance method RRAP in order to model the performance of the cycle off-design. The model does not currently model the steam cycle other than to model an extraction of heat from the GT cycle at the appropriate place. Heat is extracted such that the steam generator gas outlet temperature remains constant. Although this assumption is crude, it does mirror AGR practice where the steam flow is controlled to maintain constant steam generator outlet temperature.

<u>Combined Cycle at Off-design Conditions – Cycle Designed for 950˚C Reactor Outlet Temperature Operated at 850˚C</u>

This situation was considered for the existing GT-MHR cycle in Section 2.1.3 where it was found that the GT-MHR cycle lost almost 50MW of power when the reactor outlet temperature is reduced from 950˚C to 850˚C. It will be recalled that the reason for this is that pressure ratio falls and reactor inlet temperature rises, but reactor mass flow remains essentially constant. The consequence is that reactor power has to be reduced to hold the reactor outlet temperature down.

Analysis has shown that the combined cycle performs better in this situation. The most successful strategy was found to be:

- To slow down the GT shaft from 5000rpm to 4680rpm. This reduces the pressure ratio across the GT and matches the steam generator inlet temperature to the optimum (620˚C).

This has the effect of driving the compressor towards choke (i.e. away from surge and hence there is no threat to stability)

- To increase the helium inventory to match the helium pressure at the reactor to the design condition.

- To adjust steam flow so that the steam generator gas outlet temperature matches that at the design condition.

Following this strategy results in the flow at the reactor rising (by around 3%) which partially compensates for the reduced reactor outlet temperature and means that the reactor power need not be reduced as much as is the case for the GT-MHR cycle. The GT-MHR reactor power needs to fall from 600MW to 551MW, whereas the combined cycle only needs to fall to 573MW. Furthermore, by slowing the shaft to control the pressure ratio the thermodynamics of the steam cycle are unchanged. The overall result is that the combined cycle loses 27MW when derated from 950˚C to 850˚C but the GT-MHR cycle loses 46MW.

The mechanisms of this difference are complex but the reason for the difference is essentially that the combined cycle has more parameters that can be controlled, in particular the steam flow. This additional control allows the cycle to respond more flexibly to changing conditions and it is considered that this is likely to be an advantage in other off-design/part load scenarios.

Combined Cycle at Off-design Conditions – Hot Day

Analysis of this scenario has not yet been performed. However, it is expected that the combined cycle will again have an advantage over the GT-MHR cycle at increased ambient temperatures. It was shown in Section 2.1.3 that the GT-MHR reactor power has to fall on a hot day because GT mass flow cannot be increased, but reactor inlet temperature rises due to the reduced pressure ratio.

For the combined cycle there will be no increase in reactor inlet temperature because the extra flexibility of control afforded by the steam cycle (control of water mass flow and feedheating) allows the compressor inlet temperature to be held constant. The GT cycle will therefore be ‘unaware’ of the increased ambient temperature and therefore no reduction in reactor output power will be required. Clearly, there will still be a loss of power output due to the loss of efficiency of the steam cycle but the fact that the reactor output power can be held constant means that the loss of power will be far less than for the GT-MHR cycle.

3.3.6 *Areas of Uncertainty / Issues for Further Study*

In common with the analysis of the GT-MHR cycle, the areas of uncertainty where more work is required are off-design and transient performance. Although it is believed that the extra flexibility of control of the combined cycle should allow the performance off-design to be good, this remains to be proven. Also, at the time of writing the starting and part load strategies for this cycle have yet to be defined. Transient modelling of the cycle and, in particular, transient interactions between the steam and helium halves of the cycle, is of great interest and needs to be explored further.

4 Proposed Alternative PCS Design for NGNP Application

4.1 Overview

4.1.1 *System Description*

Rolls-Royce has chosen to present a directly heated helium circuit in conjunction with a combined cycle as the preferred solution. The proposed layout consists of two pressure vessels: one containing the GT and generator and another contains the steam generator. The rest of the plant is considered to be low risk 'commercial-off-the-shelf' equipment and as such has not been analysed in any detail at this stage. Some extra refinements may be called for, such as oversized condensers to allow the steam turbine to be bypassed or perhaps bigger HP steam vents to allow heat to be rejected quickly.

4.1.2 *Assessment of Design*

Because of the limited time available the proposed design is a simple development of the existing GT-MHR concept. The concept layout is shown in Figure 58.

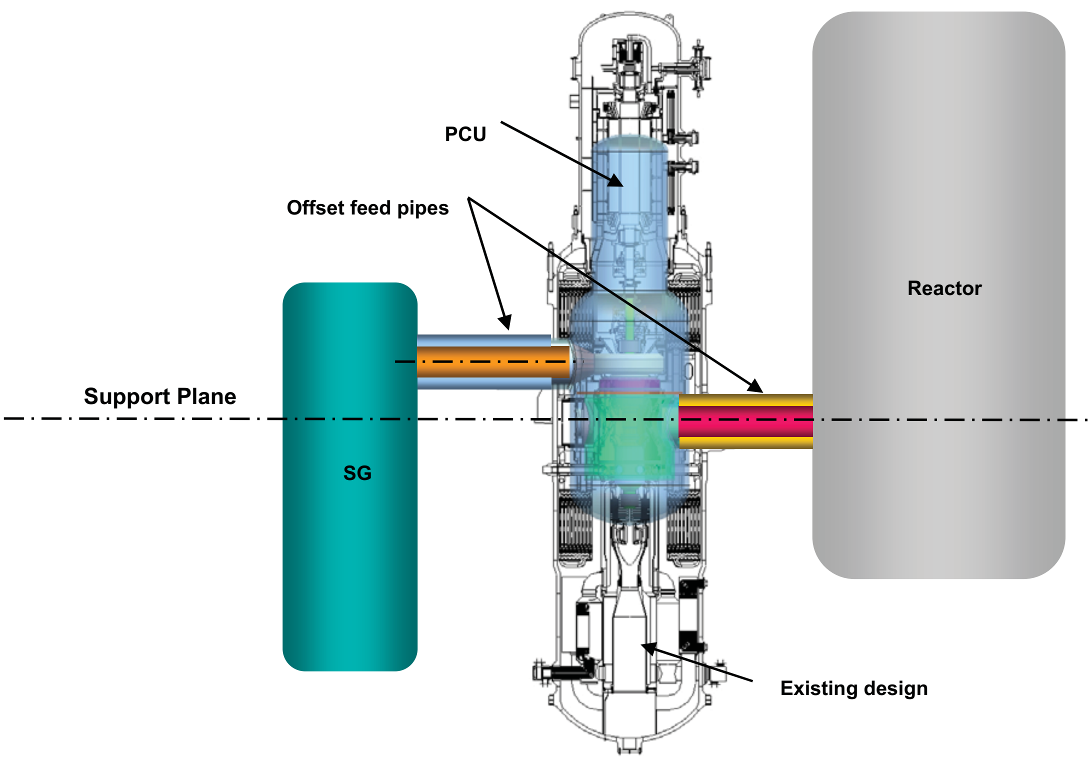


Figure 58 General Layout of Combined Cycle for GT-MHR

In the proposed high efficiency combined cycle the gas turbine inlet temperature is high. This means the size of the turbo-machinery is not significantly reduced even though the power output is less than one third of the previous design. The generator weight, however, is reduced in proportion with the power.

There is now only one compressor as there is no need to route helium to and from the intercooler as in the existing design. This results in a shorter overall compressor length and so reduces the bearing span from ~9.5m to ~7.6m. This should improve its lateral stiffness, but shaft speed has been increased from 4400rpm to ~5000rpm hence there will still be a need to vary the bearing stiffness during the acceleration and deceleration.

The vertically-oriented GT shaft is still preferred because this method leaves maximum capability to vary this stiffness; however the difficulty in creating a reliable catcher bearing solution should not be underestimated.

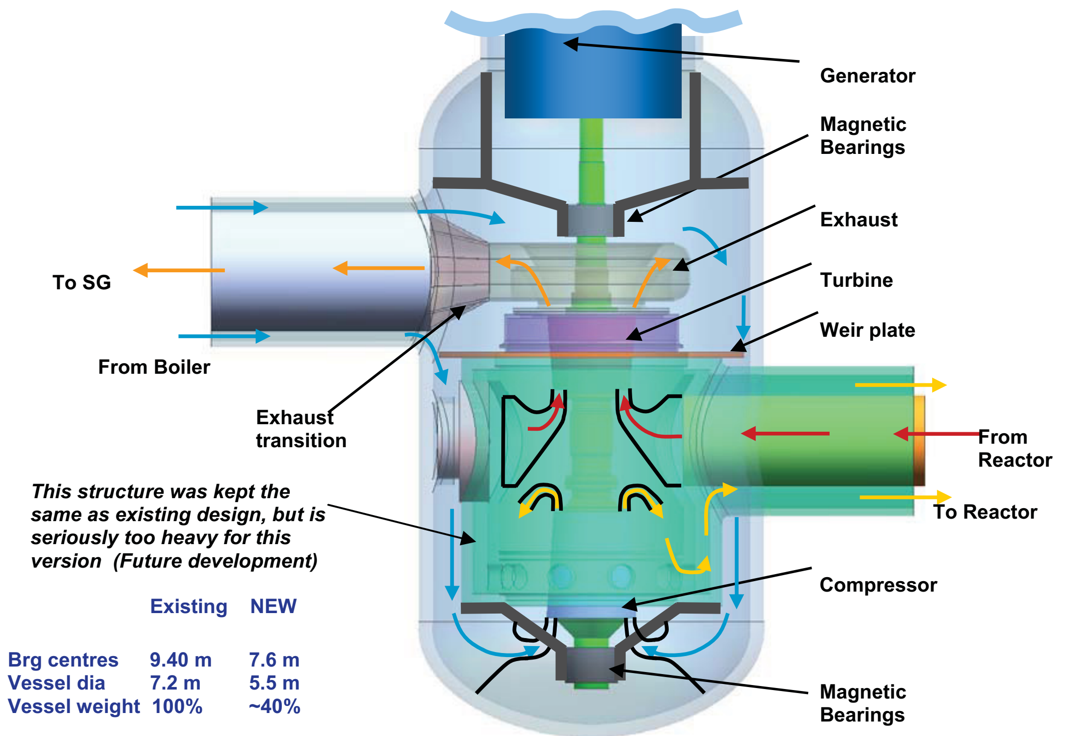


Figure 59 General Layout of PCU for GT-MHR

The flow path in the PCU is now much simpler than the existing GT-MHR design due to the elimination of the recuperator, precooler, intercooler and attendant pipework.

Helium from the reactor flows through the turbine and then via a simplified exhaust out of the PCU to the steam generator to produce steam. Return from the steam generator is allowed to flow around the inside wall of the PCU vessel before entering the compressor inlet - a weir plate is used to ensure even distribution of flow.

Flow through the compressor is then directed back to the reactor to complete the cycle. Compressor bypass and process off-take procedures are at the moment the same as the existing design, but could be reviewed at a later stage.

The PCU vessel has been reduced in both length and overall diameter to produce a weight saving of ~60% compared to the existing design. Reducing the diameter from 7.2m to 5.5m should help in terms of manufacturing and transport to site.

The chosen combined cycle arrangement now features 3 vessels (reactor, PCU and steam generator) connected by two similar feed/return pipes. It is assumed that the reactor will be grounded with the thermal movements of the pipes pushing against the PCU and the Steam generator. These pipes are not located in the same plane, which could introduce bending moments into the pipes and/or vessels. Careful design of the vessel mounting system should ensure that these bending moments are kept to a minimum and thus have little affect on vessel/pipe stress.

An alternative arrangement with in-plane feed/return pipes was considered which would have reduced any induced bending moments in the vessels. It was rejected, however, due to the complex exhaust and balance piston geometry, see Figure 60.

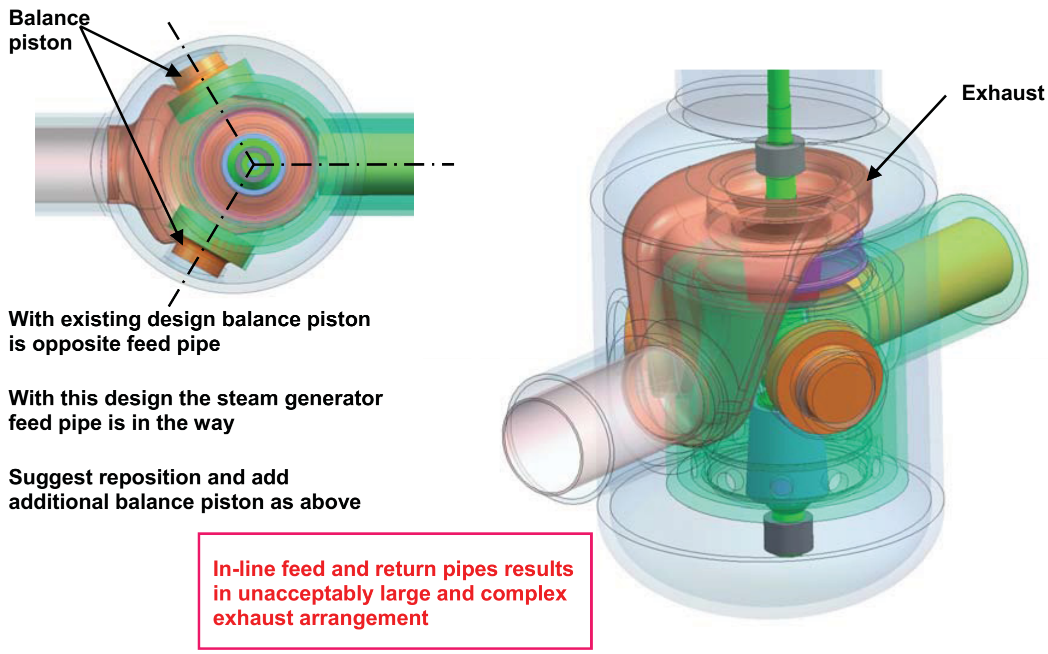


Figure 60 Alternative Inline Feed/Return Pipe Layout

Advantages of the combined cycle:

- Simpler PCU layout removes the need for recuperator, precooler and intercooler.
  - Eliminates recuperator risk.
- Reduced EMB bearing risk.
  - Generator weight reduced from ~35 tonnes to ~12 tonnes.
  - Turbomachine weight reduced from ~32 tonnes to 12 tonnes (due to shorter shaft).
- Steam equipment (excluding steam generator) would be commercial off-the-shelf. Total equipment costs should be lower.
- Flexibility to have process steam instead of electricity from steam plant.

Disadvantages of the combined cycle:

- Increased equipment footprint, both inside reactor building and outside
- Increased complexity (but mainly COTS steam plant)
- More vessels and interconnects

4.1.3 *Future Work*

For the combined cycle option to progress further, there are many things requiring further study. These include the following:

- Because there is no intercooler, the number of times helium must be taken into and out of the gas generator is reduced. It may be possible to reduce costs and simplify the sealing arrangement between the semi-structural internal high-pressure toroidal pressure ring and the interconnecting duct to the reactor.
- Any helium leaking from the high pressure section of the gas turbine or exhaust will combine with flow entering the compressor and could leave hot streaks. It may be beneficial to move the economiser section of the steam generator close to the GT inlet to mitigate this issue.
- The extensive surfaces of gas turbine and steam generator outer casing and the inner surface of the pressure boundary will need to be lagged in some way. The integrity and reliability of the fixing system needs to be demonstrated due to the potential for damage the compressor caused by debris ingestion.
- Power leads and instrumentation connections to the gas generator could be taken out to another interconnection at the bottom of the pressure vessel easing the problem of running such connections through the high pressure/temperature section. However this may lead to issues where such leads pass through the pressure vessel.
- The ambient temperature of the electromagnetic bearings may become an issue if a suitable thermal insulation and cooling strategy cannot be adequately demonstrated. The plate out of silver could also cause reliability issues for the EM bearing coils.

4.1.4 *Development Programme*

The combined cycle is considered to be lower technical risk while offering higher cycle efficiency. The concept builds on existing technologies. Combined cycle machines match the high temperature capability of the gas turbine with steam cycle adding the high efficiency. The steam plant is known technology and although complex should offer a low risk especially for plants operating in the near future. Indeed, the plant may be lower cost due to a substantial reduction in the cost of power electronics and the need to develop and maintain a viable recuperator. The steam generator may seem a difficult component but the later British AGR steam generators have a similar duty and have demonstrated high reliability.

Because of the relatively high inlet temperature the gas turbine remains a large component albeit simpler due to not being intercooled. Total compressor stage count is lower than the GT-MHR design but the component is large. The design is believed to be a low technical risk but the extension to operation in high pressure helium makes testing expensive. The technology development plan for the GT-MHR suggests a low cost approach by the use of scaled models in combination with particular full scale rig tests attempting to validate particular features where computation is uncertain. This leads to some risks remaining at the plant commissioning stage, but a large high pressure helium rig capable of demonstrating the gas turbine rotor at full scale could be prohibitively expensive.

The key risk area is the development of the gas turbine and steam generator since all other elements are virtually standard commercial equipment. The steam generator would inherit design details from the successful British AGRs and hence is thought to be low risk. Clearly, modern pipe manipulation and welding techniques will need to be assessed and incorporated where possible.

To an extent, the steam generator decouples the steam from the gas turbine commissioning process i.e. steam can bypass the steam turbines and go straight to the condenser or even be vented. This means that the gas turbine and reactor need not see the full extent of a load rejection albeit within the confines of the capacity of the chosen heat rejection system.

The gas turbine is now a much simpler machine but the power density when operating at full helium pressure remains considerable. The development plan for the GT-MHR design maps out details of component development. The plan starts with small component rigs and culminates in a full rotor test at full temperature but reduced pressure. This approach is endorsed and equipment should be simpler for the proposed combined cycle design due to the absence of the intercooler. In theory, and at the cost of complexity of the test facility power control system, power could be recycled from the generator into the simulated reactor but a detailed assessment of the part load performance and prediction of the effect of the bypass valve would need to be undertaken before meaningful predictions could be made.

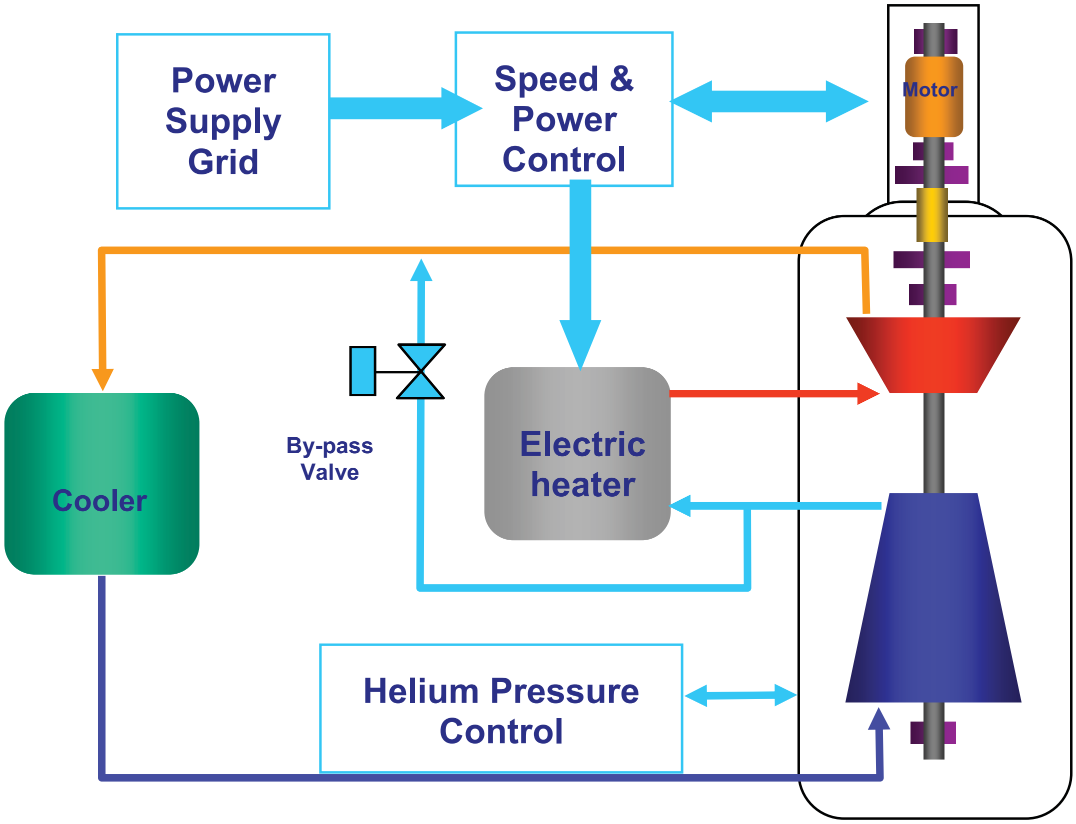


Figure 61 Proposed Gas Turbine Test Facility

4.1.5 *Materials, Lifecycle, Maintenance and Operability*

The requirements of materials within the combined cycle proposal are very similar to the requirements of the GT-MHR design, although the amount of high specification material will be reduced due to the smaller vessel volume. Clearly the specification of the steam generator pipe-work can be optimised. It is also noted that the quality of welding may be easier to assess in the steam generator components compared to features near the ends of the proposed recuperator tubes.

Helium temperature near the pressure vessel surface is now greater than the recuperator outlet temperature of the GT-MHR design but is still generally low. It is not believed that this will have a significant cost effect.

The In-Vessel-Metalwork of the proposed design is considerably simpler than the GT-MHR design and the number of large diameter seals that must engage when the gas turbine is lowered into place has reduced from 7 to 5. The inter cooler and pre-cooler are now replaced by the steam generator and thus do not complicate the main pressure vessel.

It is believed that the inner surface of the pressure vessel may need to be lagged and, although this is a simple structure requiring virtually no maintenance, it is credible that it could detach and present a risk to the inlet blades of the gas turbine. A cost benefit analysis should be undertaken. Note there will be a requirement for lagging on the external surface of the exhaust hood and transition pipe between gas turbine and steam generator so the risk of foreign object damage to the gas turbine or steam generator cannot be reduced to zero.

It is felt that the high pressure seals between the gas turbine and the high pressure exhaust duct could be optimised differently. Perhaps a mechanism with seals and bellows mounted on the gas turbine could be created, especially if the seal was held in place using process gas pressure, rather than bolts, to ease remote assembly/disassembly. This type of modification could significantly reduce the level of possible exposure to contamination during maintenance.

For the turbomachinery blades, our study has focused on typical modern turbine materials such as titanium alloys in the compressors and single crystal nickel alloys in the turbine. Creep life for the turbine blades is the life limiting issue, which forces the requirement for blade cooling for operation above 850°C. In future follow on studies, more exotic materials, such as silicon-carbide or carbon-carbon are worth investigating. These have potentially excellent creep properties at high temperatures without the need for blade cooling. They are, however, a long way away from being mature technology.

The operation of the gas turbine will be similar to the GT-MHR design; overall power will be controlled by the reactor heat output. Generator power electronics (frequency converter) will be used to control rotor speed and the compressor outlet bypass valve can still be used for emergency shutdowns.

The combined cycle offers extra flexibility in that short-term load changes can be buffered by dumping steam direct to the condensers, bypassing the steam turbines. In the extreme, steam could be vented.

4.1.6 *Technical Risk Assessment*

The general approach to new designs is to manage requirements and reduce risks. At this early stage it has not been possible to explicitly define a meaningful set of criteria so subjective judgment has been used to make assessments. The areas identified so far are described below.

<u>Bearings</u>

The area of greatest concern is the development of a viable magnetic bearing system including catcher bearings and adequate stiffness control. It is worth noting, however, that the combined cycle turbine and generator lie with the range of speeds and weights already provided by some EMB manufacturers.

There is concern that the catcher bearings will not be able to demonstrate an adequate number of demands, which may make the system expensive to maintain. This could be because they release debris to contaminate the helium circuit. It is noted that the catcher bearings will have little ability to control the rotor response as it transitions critical speeds during rundown.

<u>Turbine & Compressor</u>

It is believed that a turbine using un-cooled blades can be created. Currently, Rolls-Royce has little experience in the use of materials in a high temperature helium environment, especially in the presence of possible contamination by silver or other fission products. It is feared that the silver could adversely affect the properties of titanium or nickel-based alloys. The design also has an unusual operating regime i.e. a very low number of cycles and an extremely long operating time. Proving that design codes are adequate in this mode may be difficult.

Design codes are built on many years experience of operation in air. The effect of changing to helium has not been quantified. It will be necessary to validate most of the characteristics by analysis using CFD and rigs. The aeroelastic properties of helium must be verified, especially in order to adequately predict blade vibrations.

The effect of possibly contaminated helium on known abradable coatings and the effect of their debris on the reactor is also of concern.

The containment of debris from a credible gas turbine failure is always a concern due to the very high energies involved, especially noting the proximity of the turbine to the interconnection with the reactor.

While the extent of radioactive contamination of serviceable turbomachinery components is unclear it is not possible to assess the cost of maintenance. This cost could vary between simply holding the rotor for a period in a safe area up to the need to develop a fully remote robotic system.

Dry film lubricants are used extensively in the assembly of current aero engines. Their use in long running helium environments must be adequately assessed.

Off-design operation, especially through start-up, must be understood by the gas turbine designer. The interaction with control measures such as the operation of the bypass valve could lead to unstable behaviour of the compressor and magnetic bearings. It is thought that sudden changes in the bypass valve could affect the stability of the axial magnetic thrust bearing control.

Although the GT-MHR design is axi-symmetric, blade tip clearances especially near compressor outlet may case instabilities. A detailed thermal and structural understanding of the design under all conditions will be required before long term operation can be allowed. This understanding must cover assembly, start-up and emergency shutdown.

Detailed assessment of seal systems has already been recognised within the OKBM TDP. Credible leaks could seriously distort compressor inlet temperature distribution which could lead to instabilities and surge during normal operation. It is not clear how the integrity of these seals can be assessed after assembly. Indeed the seals form part of the support structure to the electromagnetic bearings and such any uncertainties could be compounded by the complex requirements to control bearing stiffness. It appears necessary to route power and instrumentation cables for the compressor end bearings through the turbine and other very hot sections of the gas turbine casings which could lead to unreliability. The routing will also be close to these seals.

While the risk of water contamination in helium in the reference is low the proposed combined cycle design increases the risk due to the increased wetted surface area (number of steam generator tubes) and high water pressure (>150 bar)

The very large diameter of the pressure vessels in the GT-MHR design introduces large programme risks due to transport and possible on-site fabrication issues.

The design of the recuperator components, specifically the number of welds on folded sheets in the 4000 tubular elements and the complexity of the high pressure piping, are considered very high risk. Essentially this unit has to last for the life of the PCU and is extremely difficult to access for maintenance. It is not clear that a viable economic design can be created

Key risks for the generator include the cost and reliability of the power electronics. The operation of the machine in high pressure helium will require a better understanding of the effect on rotor losses and conductor vibration and thermal management.

Areas where the proposed Rolls-Royce design has mitigated some risks are highlighted in green in the following table. In one instance, the risk has got worse. This is highlighted in red.

Table 20 Comparison of GT-MHR and Combined Cycle Technical Risks

Risk	GT-MHR		Combined Cycle		Mitigation
	Likelihood	Impact	Likelihood	Impact	
Electromagnetic Bearings					
Cannot adequately control the rotor	L	H	L	H	Detailed rotor dynamic modelling/bearing control early in concept design
Cannot achieve bearing stiffness at acceptable clearances	H	M	M	M	PCU shaft is lighter, Steam plant on conventional bearings
Reliability of system not adequate to achieve required life	M	H	L	H	Less exposed to electronic system failures
Catcher Bearings					
Can't achieve adequate number of demands	H	H	M	H	Reduced demand on development of capture bearing design
Adequate stiffness and damping can't be achieved during deployment	H	H	M	H	Development of suitable backup bearing system
Contamination risk to reactor system during deployment (due to bearing wear)	H	L	H	L	Ball bearings/material choice
Catcher bearings require shaft brake	M	M	M	M	
Turbine and Compressor					
Inability to design turbine for adequate life due to temperature of environment	L	M	L	M	
Inability to design for adequate life due to helium in environment	L	H	L	H	

Risk	GT-MHR		Combined Cycle		Mitigation
	Likelihood	Impact	Likelihood	Impact	
Inability to design turbine for adequate life due to silver contamination in environment (material choice issues)	?	H	?	H	
At 850°C we require blade cooling and that impacts cycle efficiency unacceptably	L	M	L	M	
Inability to use abradable linings (or dry film lubricants in compressor) because of cycle contamination risk leads to performance shortfall	L	H	L	H	
Turbine is not efficient enough because the aerodynamic design does not account for helium properties properly	M	M	M	M	
Achievable clearances are not sufficiently tight - impacts performance	M	M	L	M	Combined cycle less sensitive
Designing for EMB failure impacts turbine efficiency and/or requires turbine removal	L	H	L	H	
No acceptable root fixing design to achieve design life	L	M	L	M	
Disk cooling system is inadequate to achieve design life because not much temperature difference between reactor inlet and exit	M	H	M	H	
Blade off impacts reactor	L	H	L	H	
Sudden removal of load causes unacceptable over-speed - disk burst impacts reactor	L	H	L	H	
Shaft failure causes turbine over-speed - disk burst which impacts reactor	L	H	L	H	
Radioactive contamination (with long half lives) makes maintenance impractical	M	H	M	H	
Surge/system failure causes excessive radial/axial movements because of EMBs response which leads to blade failures through rubbing or high disk stresses	L	H	L	H	
Designing for surge tolerance impacts radial and axial clearances and gives poor efficiency and reduced surge margin	M	H	M	H	

Risk	GT-MHR		Combined Cycle		Mitigation
	Likelihood	Impact	Likelihood	Impact	
Inadequate surge recovery leads to blade failures	L	H	L	H	
Silver contamination causes life issues with titanium	L	H	L	H	
Inability to use dry film lubricant because of cycle contamination leads to self welding through excessive frettage	L	H	L	H	
Radioactive contamination (with long half lives) makes maintenance impractical	L	H	L	H	
Helium damping of blades is inadequate to suppress blade vibration	L	M	L	M	
Surge margin not adequate for starting procedure (because starting procedure is not well understood)	?	H	?	H	
Unknown air system issues lead to asymmetries in casings - causes unacceptable clearances - poor efficiency and surge margin	L	H	L	H	
Unknown air system issues lead to leakage paths in and out of the compressor reducing surge margin and efficiency	L	M	L	M	
Higher surge loads (long duration) because of large downstream volume causes blade failures	L	H	L	H	
Heat Exchangers					
Recuperator design life is inadequate - starts leaking and fails	H	H	n/a	n/a	Cross-corrugated design minimises GT-MHR risk
Recuperator is too expensive for plant economics	H	H	n/a	n/a	Cross-corrugated design minimises GT-MHR risk
Steam generator design life is inadequate	n/a	n/a	L	H	Proven AGR type design
Helium Cycle Generator (Steam Turbine System is COTS)					
Cost of power electronics is prohibitive	H	H	L	M	Anticipate power electronics getting cheaper with time.

Risk	GT-MHR		Combined Cycle		Mitigation
	Likelihood	Impact	Likelihood	Impact	
Poor reliability of power electronics cause failure of turbo-machinery	L	H	L	H	Power electronics getting more reliable with time.
Poor reliability of power electronics disconnects generator from system	L	L	L	L	
Windage losses are higher then anticipated	L	M	L	M	
Generator becomes contaminated with radioactivity - cause generator failure	L	H	L	H	
Generator cooling inadequate - causes generator failure	L	H	L	M	Smaller generator - easier to cool
CF forces too big causing inadequate life	L	H	L	H	
Helium flow-induced vibration of generator windings, etc - causes generator failure	M	H	M	H	Experimental test and theoretical studies
Whole System Risks					
Inexperience with helium causes unforeseen issues (aerodynamic/material life/leakages)	M	M	M	M	
System validation looks impossible before building a prototype - prototype is inefficient	M	M	M	M	
Infrastructure costs to support development and maintenance programme are prohibitively expensive	M	L	M	L	
Relative thermal/seismic movements of reactor/PCU/(and steam generator) give unacceptable stresses and fails pipe-work	L	L	M	L	Plant occupies more space, probably more vulnerable
Bellows seals will not be adequately maintainable and will fail causing major leaks in the cycle	L	M	L	M	
Piston ring seals will not adequately support the turbo-machinery casings - this will cause unpredictable effects on the EMB bearing stiffness and cause a failure	M	H	M	H	Develop alternative mounting/sealing concepts
Cabling running through hot part of the cycle to the EMBs will fail	L	M	L	M	

Risk	GT-MHR		Combined Cycle		Mitigation
	Likelihood	Impact	Likelihood	Impact	
Risk of breaking ring seals on assembly of PCU. Breakage could be undetected	M	M	M	M	
Insulation detaches and causes FOD leading to failure	L	H	L	H	
Complex helium flows cause asymmetry in compressor inlet conditions causing surge.	L	M	L	M	
Water contamination of helium cycle causes issues for reactor	L	H	L	H	Fort St Vrain experience Theoretical probability higher with combined cycle SG, but true impact unproven at this stage
EMB wiring insulation can't be adequately cooled - life not achieved	L	H	L	H	
Pressure vessel has inadequate integrity because of being welded up on-site	M	H	L	H	Smaller vessels

4.1.7 *Costs*

At this stage there has been insufficient time to assess costs with any degree of accuracy. It is currently believed that the cost of the power electronics associated with a non-synchronous generator of around 300MW will more than offset the increased cost of combined cycle equipment for the suggested smaller ~66MW generator and associated steam plant. While this cost advantage is likely to apply to the first unit produced it is accepted that development of technologies (especially power electronics) may reverse the situation for the Nth plant. It should also be noted that the combined cycle plant requires a larger reactor containment building and an additional building for the steam turbine.

4.2 Compressor Aero/Mechanical Design

4.2.1 *Aerodynamic System Description*

The aerodynamic issues for the combined cycle compressor are very similar to those for the reference design. Only one compressor is required for the combined cycle requirement – no intercooling is required for this alternative. A single 18 stage design blowing 1.93 pressure ratio has been analysed with the Rolls-Royce preliminary design tools, and has been predicted to give 89.5% polytropic efficiency and 27% surge margin at the design point. The design appears acceptable with no major aerodynamic risks.

Some optimisation work was carried out to this design, and it showed that increasing aerodynamic loading could be expected to increase efficiency (by around 0.8%). The aerodynamic loading could be increased by reducing the compressor blade radius, reducing the blade and vane numbers, reducing the number of stages, or reducing shaft speed. All these options would reduce rotor shaft weight and help reduce the risks associated with the EM bearings and catcher bearings. Increasing the loading would reduce the design point surge margin (to around 20%) – but without a better understanding of the transient requirements of the system, we do not know what level of design point surge margin would be acceptable. This should be an output of a follow on transient performance study in the next phase.

4.2.2 *Mechanical System Description*

Rotating Assembly

The LPC and HPC of the initial GT-MHR based proposal is now replaced by a single eighteen-stage compressor of increased tip diameter. The diameter of discs for this unit is outside R-R current experience but at the proposed running speed of 5000rpm, is not considered to be impractical. Again it is proposed to utilise a welded titanium drum construction but due to the overall increase in length (now approx. 1.75 meters), a bolted joint may need to be incorporated within the drum construction for manufacturing purposes (similar to that used on HPC drums where the material of the rear stages changes from titanium to a nickel based material). Helium temperatures across this compressor design are now increased to 256°C at inlet and 437°C at outlet. Although blisks of these proportions are outside of Rolls-Royce current experience, it is thought that they would be feasible and they are therefore specified. The blade numbers derived in the Rolls-Royce aerodynamic model for the combined cycle are reduced from those of the GT-MHR and adequately fit the disc rim proportions.

The overall length of the compressor is significantly reduced from the GT-MHR design (see Figure 62 and Table 21 below)

Table 21 Comparison of Compressor Axial Lengths mm (ins)

	GT-MHR	GT-MHR(rr)	Combined Cycle
LP - 10 stages	1025 (40.35)	1060 (41.75)	
HP - 13 stages	1190 (46.85)	1148 (45.2)	1753 (69.0) [18 stages]
Intershaft	1385 (54.5)	1385 (54.5)	
Total length	3600 (141.7)	3593 (141.5)	1753 (69.0)

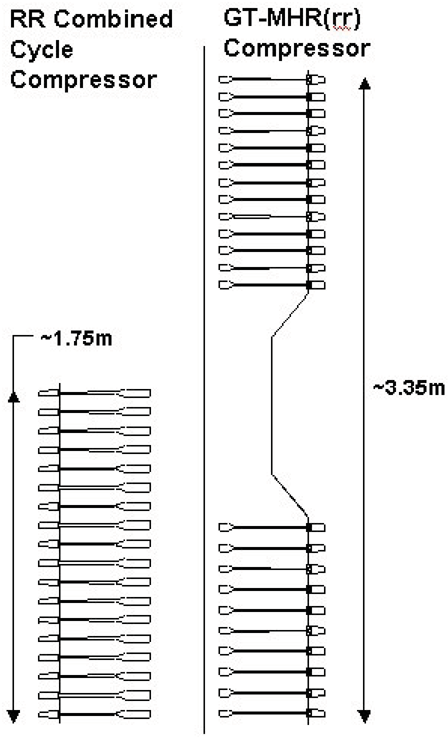


Figure 62 Comparison of GT-MHR and Combined Cycle Compressors

Casing Assembly

The casings are again produced in steel and are of split configuration to enable assembly around a bladed rotor assembly. There is now no requirement for exit and inlet-casing sections with their spoked curved duct profiles, and so are significantly simplified. Careful design and disposition of material is required to minimise distortion and maintain circularity. The increased length however may require the casing to be split into fore and aft sections with will need suitable joint location features. The casings will again incorporate location features for the vane outer fixings and stainless steel liners.

4.2.3 *Assessment of Mechanical Design*

Rotating Assembly

As with the assessment of the GT-MHR design, discs from an existing civil aero/industrial engine produced by Rolls-Royce were scaled on radius and rotational speed. Rim loads using blade masses deduced from the Rolls-Royce aero model of the combined cycle were also considered. Due to the increased radius on the combined cycle engine, the resulting blisk cross section that was produced was significantly bigger than the originally scaled engine and therefore only a first approximation. Although the blisk is outside of Rolls-Royce experience in terms of its size and proportions, it does not present any particular mechanical concerns. Manufacturing would need to review the implications of a blisk of these proportions in detail however initial assessment suggests it would not cause significant problems.

Although the temperature across the compressor in now increased and at exit is 437°C, Rolls-Royce investigation of recent engine configurations has indicated that titanium has acceptable creep properties at this elevated temperature.

The reduction in compressor length combined with that of the turbine produces a shorter, stiffer and lighter rotating assembly. The span between radial magnetic bearings is reduced from 9.4 meters to 7.6 meters changing the rotor dynamics and reducing the duty on the magnetic bearing system.

Containment of blades and discs again needs careful consideration, particularly in relation to the proximity of the compressor to the hot gas duct that leads into the reactor. Full FMECA analysis is needed to ensure that all the risks are adequately mitigated.

Casing Assembly

The casing assembly will again require machining as a complete assembly (either a split whole length casing or split two section) to maintain circularity and concentricity. The reduction in casing sections will reduce leakage, which can particularly occur across the inlet and exit sections due to the high differential in pressure. Again the baseline configuration is for no rotor path linings to be incorporated – see radial clearance discussion in Section 2.2.

Radial Clearances

The combined cycle design has the same EMB and blade tip clearance issues as discussed for the GT-MHR in Section 2.2.

Axial Clearances

Due to the unchanged position of the axial EMB relative to the turbine and compressor, the combined cycle design has similar axial clearance issues as the GT-MHR as discussed in Section 2.2. Although it is shorter than the GT-MHR design, there is still a significant distance between the axial bearing and the compressor, which needs to be properly evaluated.

Balancing

The reduced distance between the radial magnetic bearings improves the balancing of the combined cycle design relative to the GT-MHR. The generic issues discussed for the GT-MHR in Section 2.2 still apply however the welded construction should significantly improve the ability to balance this assembly.

Weight

Figure 63 shows a comparison of typical disc and blisk cross-sections for the following engines:

- GT-MHR (OKBM): This is the GT-MHR design as taken from the PCU general view drawing (Reference 9).
- GT-MHR (RR): This is an approximated bladed disc section scaled on speed and radius to the GT-MHR conditions from an existing Rolls-Royce civil aero/industrial engine. It assumes that the disc will form part of a welded assembly

- RR Combined Cycle: This is an approximated blisk section scaled on speed and radius to the RR Combined Cycle conditions from an existing Rolls-Royce civil aero/industrial engine. It assumes that the disc will form part of a welded assembly.

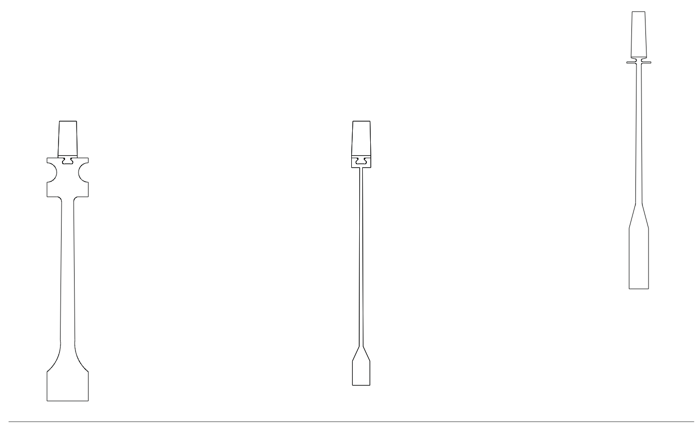


Figure 63 Comparison of Compressor Disc and Blisk Cross-Sections

NB: Cross sections shown are Rough Order of Magnitude (ROM). Disk designs have had no stress modelling and proportions could change.

Assuming similar materials (titanium alloys) are used in all three discs/blisks, visually it can be seen that the GT-MHR(OKBM) design is significantly heavier than the GT-MHR(RR) design and the Rolls-Royce Combined Cycle blisk. The GT-MHR(RR) bladed disc design saves weight because it is intended to form part of a welded assembly and hence does not need the extra material around a tie bolt hole. However, irrespective of this weight saving, the GT-MHR(RR) origins as a scaled aero derivative industrial engine make it apparently much more optimised than the GT-MHR(OKBM) design (in the order of 50% lighter). Note that additional weight savings in the order of 20% could be made if blisk designs had been used on the GT-MHR(RR).

The Rolls-Royce Combined Cycle blisk and blade weight is over twice the GT-MHR (RR) weight because of the increased radius and speed, which outweighs the benefits of going to a blisk design. Because there are 23 stages (10 LPC/13HPC) on the GT-MHR(RR) engine vs 18 stages on the Rolls-Royce Combined Cycle engine, the rotor assembly weight will only increase by 70%.

The GT-MHR(RR) and combined cycle discs/blisks have been scaled on speed and radius only and therefore have not been increased in size to account for the increased torque/power that the helium cycles demand. However discs are not limited by torque stresses and it is thought that this doesn't account for the discrepancies in size relative to the GT-MHR(OKBM) design. It is recommended that further, more detailed assessments of the disc/blisk cross sections be carried out so that more confident and precise weight conclusions can be made.

4.2.4 *Conclusions*

The mechanical compressor issues for the proposed combined cycle alternative are very similar to those for the reference GT-MHR concept. There is now only a single compressor (of 14 stages) rather than two compressors with intercooling in between. One difference is that the compressor inlet temperature is higher for the combined cycle (256°C). Titanium will still have acceptable properties at these increased temperatures.

4.2.5 *Development Programme*

There are no differences, relative to the GT-MHR, in the technology development programmes of work for the combined cycle engine design.

4.2.6 *Materials, Lifecycle, Maintenance and Operability*

There are no differences, relative to the GT-MHR, in the materials, lifecycle, maintenance or operability issues for the combined cycle engine design.

4.2.7 *Technical Risk Assessment*

There are no differences, relative to the GT-MHR, in the risk assessment for the combined cycle engine design.

4.2.8 *Costs*

Preliminary (ROM) costs based on industrial Rolls-Royce compressor scaled for size and weight (material cost) gives compressor system costs in the order of $3M - $3.5M at present day costs. The GT-MHR(RR) option uses less material but incorporates complex inlet and exit split casings, whereas the Combined Cycle is simpler design but is larger in diameter using more material. No account has been included for 1 off (prototype) escalation for materials or manufacture.

4.2.9 *Areas of Uncertainty / Issues for Further Study*

The Rolls-Royce combined cycle engine has the same areas of uncertainty as with the GT-MHR discussed in section 2.3.7 (helium embrittlement and radial clearances in EMBs). Additionally, there is the following area of uncertainty relating to silver contamination.

<u>Silver Contamination</u>

Silver in small quantities may be produced as part of the reaction process. Silver reacts with Ti above 300ºC, causing corrosion and possible crack initiation sites that can be difficult to detect. Therefore the effect of Ag on Ti could have been a significant potential issue for combined cycle option. However the severity of corrosion is dependent upon the quantity of silver present and preliminary information suggests silver contamination levels are extremely low and are not considered to be sufficient to cause a problem. Additionally, the small amount of silver that may be produced by the reactor has to progress a long way through the turbine and pre-cooler before it reaches the compressor and it is considered unlikely that it will travel so far.

4.3 Turbine Aero/Mechanical Design

4.3.1 *System Description*

To assess the viability of a helium turbine for the combined cycle power conversion unit, a concept turbine has been designed. The overall length of the turbine has been chosen to be the same as the OKBM/Samara design. All other geometric parameters have been derived from aerodynamic and mechanical constraints.

The cycle parameters for the turbine have been reproduced here for clarity.

Table 22 Combined Cycle Turbine Parameters

Parameter	Value
Thermal Power Rating	66MW
Mass Flow	$280kgs^{-1}$
Shaft Speed	5000rpm
Inlet Total Pressure	7020 kPa
Inlet Total Temperature	850°C
Outlet Total Pressure	3760kPa
Outlet Total Temperature	620°C

4.3.2 *Assessment of Design*

The concept design of a suitable turbine has been undertaken in considerably more depth than the concept design of the OKBM GT-MHR turbine presented in Section 2.3 above. The aerodynamic assessment has included:

- Current understanding of material limitations.
- Turbine efficiency requirements.
- Diffuser integration.
- Mean height blading feasibility designs.
- Blade, shroud and platform mass.
- Blade root feasibility.
- Disk and overall rotating mass.

As a result the aerodynamic design is a good compromise between the competing requirements. This has allowed us to gain a good understanding of the levels of technical risk associated with each aspect of the design. The features of the design are discussed first, followed by a discussion of the technical risks in Section 4.3.5 below.

- Five Stages.
- First stage tip radius of 0.671m.
- Last stage tip radius of 0.773m.
- Shrouded high root-tip ratio blading.
- Optimised work split and reaction across stages.

Compared to the OKBM GT-MHR turbine the combined cycle design is a similar overall size. However, the increased blade axial chord significantly reduces the blade count with a maximum of 67 blades in the final stage (see Figure 64).

The blade tuning is around 107 deg for each stage which results in a good blade shape which should present no problems from a vibration/dynamics standpoint. However, this has resulted in blades that are individually heavy (1.59kg for the final blade). However, the size and shape of the blading is such that it is suitable for hollow casting.

The blade pitch/chord ratios attained are suitable aerodynamically and also allow for reasonable shroud and platform design.

Figure 64 Combined Cycle Last Turbine Blade Profile

Compared to aerospace design practice the disk rim load is low, even though the blade mass is high. An estimation of the overall mass of the rotating components has been made based on solid blading and optimised disk design. The mass of the turbine is estimated at 1200kg.

Based on our current understanding of material limitations, the blades are uncooled but the disks will require cooling. A cooling flow is likely to be required for the disk post and blade root of all stages.

Combined Cycle Gas Turbine - Mechanical design

The alternative approach proposed by Rolls-Royce features a turbine design with five or seven stages.

The analysis completed focuses on the five stage turbine, although some preliminary assessments have been done on the seven stage design to identify the options this creates, for progression in any future phases as part of a greater optimisation of the solution.

The basic outline of the design is as shown in Table 23.

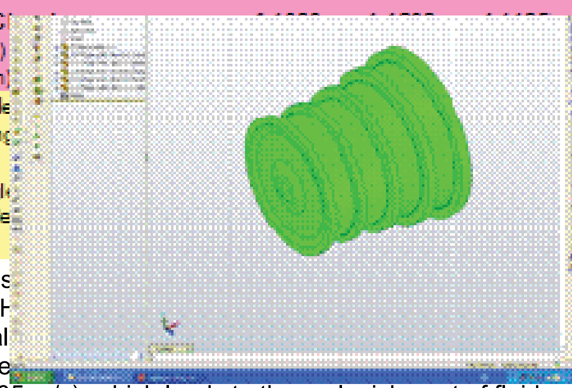
Table 23 Parameters for Combined cycle Gas Turbine – 5 Stage Turbine

	Stage 1	2	3	4	5
Inner Annulur Radius (m)	0.536	0.534	0.560	0.596	0.644
Outer Annulus Radius (m)	0.671	0.673	0.698	0.730	0.773
Inlet Total Temperature (K)	1123.15	1082.57	1041.99	997.94	949.03
Exit Total Temperature (K)	1123.15	1082.57	1041.99	997.94	949.03
Inlet Relative Total Pressure (Pa)	6702138	6032002	5380835	4721030	3919762
Exit Relative Total Pressure (Pa)	6661127	5993698	5342125	4681393	3893235
NGV Ideal Blade Count	43	57	59	49	35
Rotor Ideal Blade Count	59	62	64	51	67
NGV Pitch / Chord	0.8692	0.8174	0.8297	0.8523	1.3026
Rotor Pitch / C[illegible]	[illegible]	[illegible]	[illegible]	1.1331	0.7596
NGV Chord (m)	[illegible]	[illegible]	[illegible]	0.0990	0.0990
Rotor Chord (m)	[illegible]	[illegible]	[illegible]	0.0714	0.0870
Vane Inlet Angle	[illegible]	[illegible]	[illegible]	-42.063	-40.646
Vane Outlet Ang[illegible]	[illegible]	[illegible]	[illegible]	67.280	70.917
Vane Turning	[illegible]	[illegible]	[illegible]	109.343	111.563
Rotor Inlet Angl[illegible]	[illegible]	[illegible]	[illegible]	36.474	52.608
Rotor Exit Angle	[illegible]	[illegible]	[illegible]	-68.396	-62.744
Rotor Turning	[illegible]	[illegible]	[illegible]	104.870	115.352

The ba[illegible] e a shroud[illegible] bly of discs.

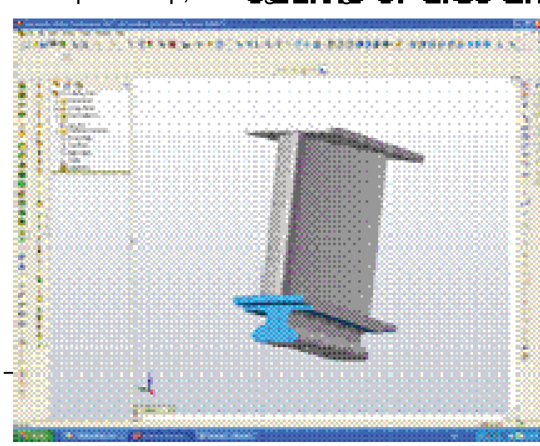
This all[illegible] chieve the life

As men[illegible] conside[illegible] the suitabil[illegible]



The free disc surfaces [illegible] ondition using heat transfer coefficients (H[illegible] atural convection mechanisms. Normal [illegible] buoyancy, is associated with lower levels of he[illegible] discs creates a massive acceleration (order 105 m/s), which leads to the replenishment of fluid as soon as any heat has been picked up, leading to large heat transfer coefficients.

Outline of disc arrangement as a drum



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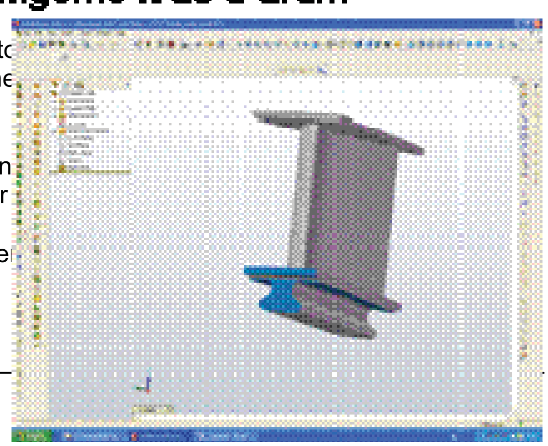
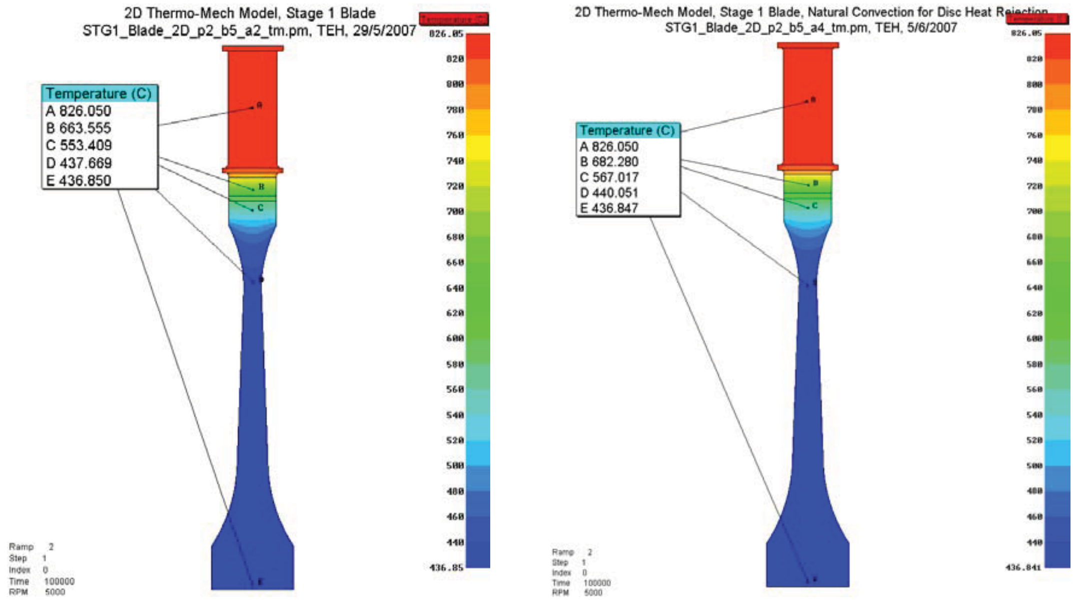


Since the blade root material is much more capable of handling the higher temperatures whilst maintaining a reasonable creep life, the disc post region is of prime interest. The disc post / blade root region is modelled with a merged region property, which would be better modelled with a 3D model. One interpretation of the 2D model, assuming the blade root and disc post geometries and the thermal conductivities of the two materials are similar, is that the temperature quoted reflects the blade to disc interface temperature.

The model however does give an indication of temperatures to expect, 660°C to 680°C.



Results with forced convection as heat rejection mechanism

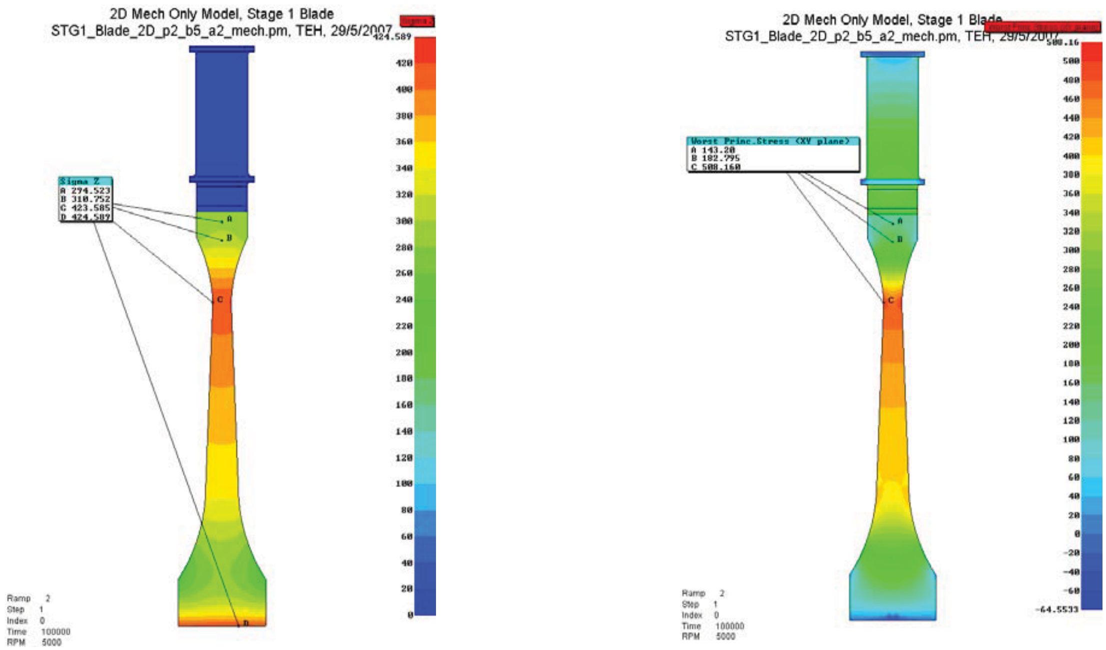
Results with natural convection as heat rejection mechanism

Figure 66 5 Stage Turbine, Stage 1 Thermal Model - Temperatures (°C)

The profile of the disc is scaled from an aero compressor disc to support the proposed blade CF load and produce the same radial displacement at the blade root section. As can be seen the profile is now much less massive:

- Peak Hoop Stress of 425MPa at the disc bore.
- Peak In-Plane Stress of 510MPa at the diaphragm to disc rim transition.

As shown in Figure 67 below, the disc profile is not optimised, clearly the diaphragm section needs to be slightly thicker to reduce the radial stresses below the hoop stresses at the bore. Since the turbine design is only at a concept stage and the aerofoil loadings are changing quite significantly with both rotational speeds and stage numbers, the disc optimisation has not been performed at this stage.



Hoop Stress **Worst Principal In-Plane Stress**

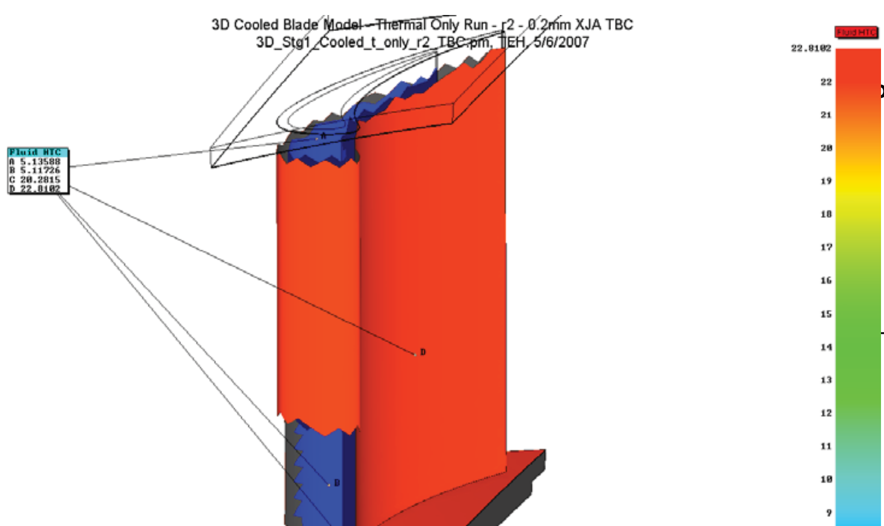
Figure 67 5 Stage Turbine, Stage 1 Turbine Disc Stresses - CF Loading

Cooled Blades

A 3D model of the first stage of a simply cooled turbine blade has been created to evaluate possible cooling using some compressor bleed helium flow. Different cycles have been considered resulting in gas path temperatures of both 850°C and 950°C. The outcome from this analysis is shown in Figures 67 to 70, with the associated commentary.

The external HTCs have been generated from a forced convection correlation, resulting in HTCs of 22 W/m^{2}K at the design condition.

The internal HTCs have been generated from the maximum of forced and natural convection using 2% bleed flow for the first stage. This results in HTCs of 5 W/m^{2}K at the design condition. The smaller flow area through the root generates HTCs of 20 W/m^{2}K



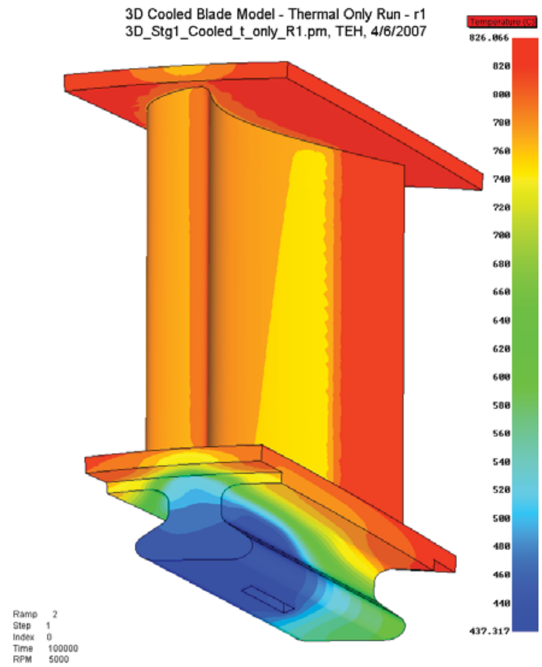


Figure 69 Stage 1 Simply Cooled Turbine Blade Temperatures

The efficiency of helium as a coolant is much greater than that of air – the results in Figure 69 show that there is little radial gradient showing the 2% bleed mass flow rate is excessive. However the conductivity of the material is not sufficient to cool the trailing edge, which would result in large thermal stresses.

With a reduced cooling mass flow rate the natural convection mechanism would dominate the blade cooling.

A second model including a 0.2mm layer of Metco 204NS (Rolls Royce 3-alpha code XJA) thermal barrier coating is shown in Figure 70. The use of the coating is effective in reducing metal temperatures when used in conjunction with effective blade cooling. Possible fringe benefits might exist in shielding the blade from contaminants. However, use of the coating exacerbates the differential thermal gradients and is not recommended. If cooling is to be used, provision of trailing edge cooling is probably going to be necessary, which will also serve to reduce the local stiffness, which is desirable.

In addition consideration is needed to the bonding of the coating, and the impact of failed coatings on the PCU, and the resulting failure of a turbine blade if the coating failure is not detected. These are additional risks that have to be managed within a cooled/thermal barrier coated solution.

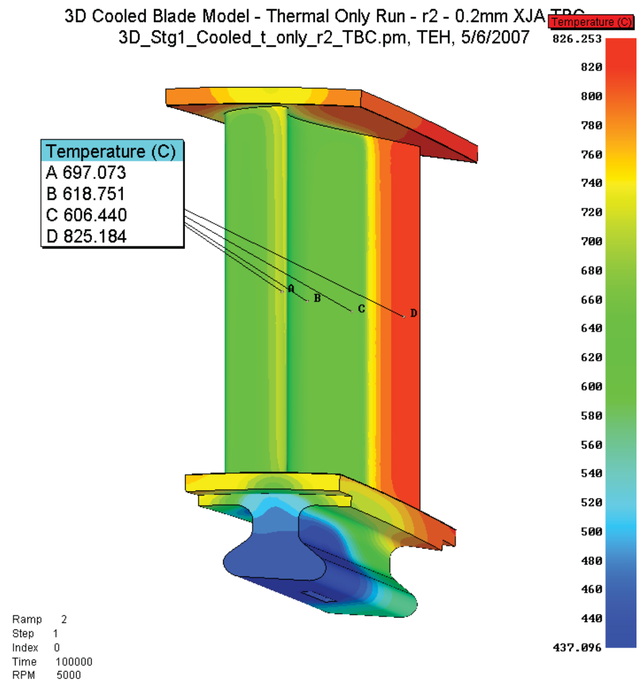


Figure 70 Stage 1 Turbine Blade Temperatures With Metco Thermal Barrier Coating

A third model including the thermal barrier coating, reduced cooling mass flow of 0.4% and a higher gas path temperature of 950°C has been analysed. This is shown in Figure 71.

The temperatures have increased by approximately 130°C from the second model, of which 100°C is due to the higher gas path temperature and 30°C is due to the reduction in cooling flow.

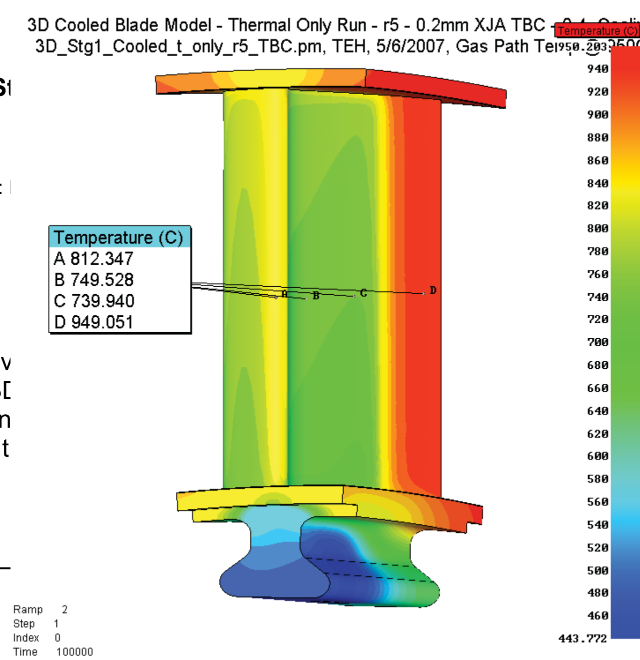


Figure 71 St… th Increased Gas

Including: … w of 0.4%

Uncooled Blades

Uncooled blades hav… reducing the speed to 3600 rpm. Two 3D… en analysed at the three speeds now un… based upon a seven stage turbine within t…

An internal design review commented that the typical aero engine creep envelope was different to that of an industrial or power generation turbine, and that an alternative material might exist with more suitable properties than CSMX4. However, it has been confirmed that CMSX4 is widely used for the first stage turbine of modern power generation turbines, and hence it's continued use in this concept study.

The graph below shows the mean stresses of a shrouded blade at the worst section, generally the blade root shank. Only the 3600rpm design shows satisfactory life at 850°C, and none of the speeds are satisfactory for 950°C with the current technology of materials identified.

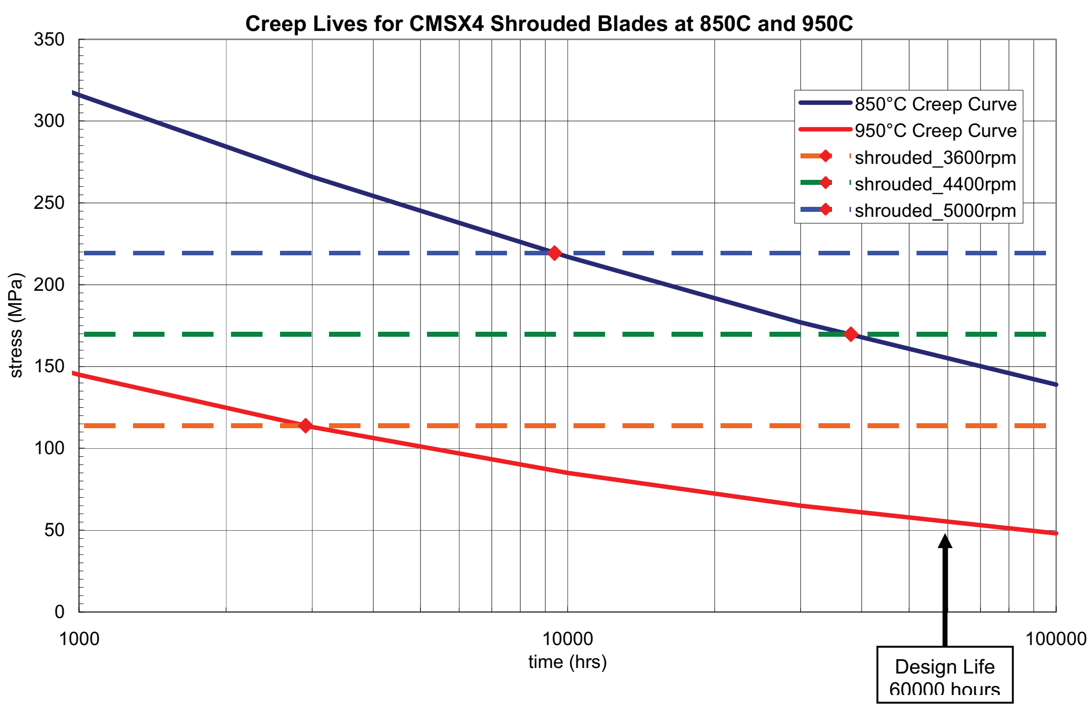


Figure 72 Creep Lives – Shrouded Blades

The graph below shows the mean stresses of a shroudless blade at the worst section, generally the blade root shank. The 3600 and 4400rpm designs shows satisfactory life at 850°C, but still none of the speeds are satisfactory for 950°C.

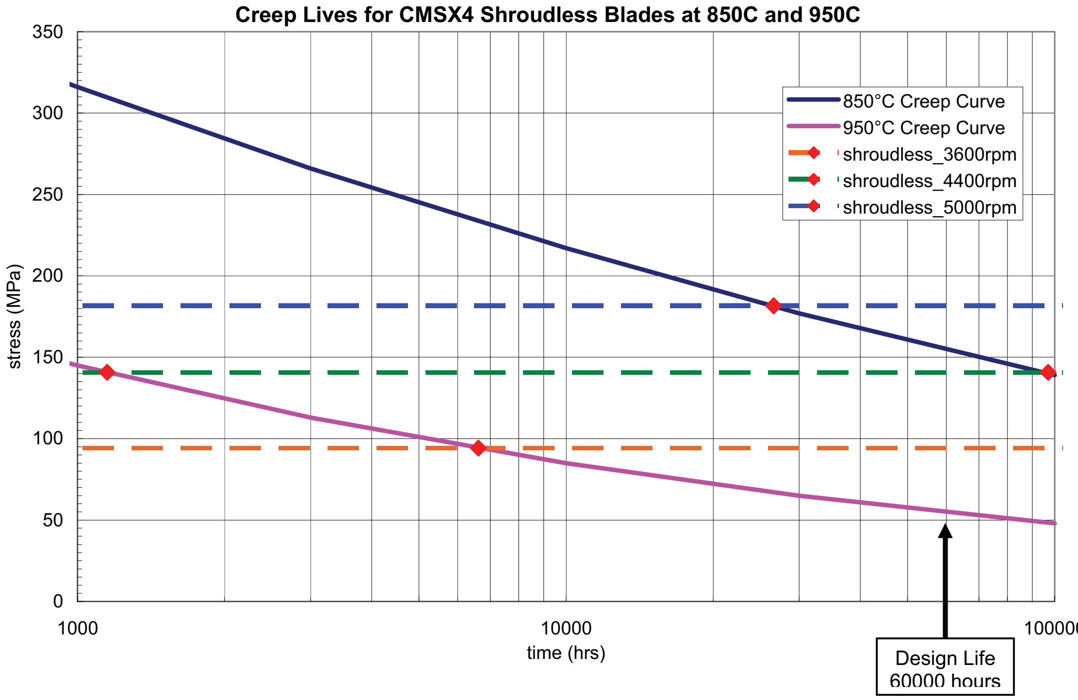


Figure 73 Creep Lives – Shroudless Blades

Conclusions Against Mechanical Concept for Turbine

The design solution is possible with refinement. Concerns exist though in achieving the range of design requirements with the current turbine layout, with an uncooled blade. An uncooled shroudless blade operating at a shaft speed of <4400rpm, is the most likely to achieve the target life. A shaft speed of higher than this would require a reduction in the target life, typically in the 25 000 to 30 000 hour region. A shrouded blade would require a maximum shaft speed of 3600rpm.

Therefore the optimisation process has to be progressed to define the most suitable aerodynamic and thermodynamic solution within the mechanical constraints, which will result in further assessments of the number of stages, shaft speeds and turbine diameter.

At higher shaft speeds creep life is considered achievable with blade and disc cooling, and blade thermal barrier coating, but requires optimisation through the follow on programme phases, and assessment of the thermal barrier coatings.

HPC delivery gas requirement is approximately 0.4%/stage for this design layout and all stages would be cooled.

The option for operation at up to 950ºC may be possible with further design and development work to optimise cycle and turbine design and with a cooled and thermal barrier coated blade solution. This will have to be considered against the risks and complexity of a cooled turbine solution, and the importance of the barrier coating bonding, where failure would almost certainly result in a turbine blade failure and damage to the gas turbine.

Potential for weight reduction exists by turbine design optimisation. This has to consider the optimised turbine solution in terms of speed, stages and blade count against material choice, blade profile, and the benefits of cooling and thermal barrier coatings.

4.3.3 *Development Programme*

The development programme requirements are similar to those defined for the GT-MHR design.

4.3.4 *Materials, Lifecycle, Maintenance and Operability*

The materials, lifecycle, maintenance and operability issues are similar to those defined for the GT-MHR design. However, the smaller and simpler PCU will generally facilitate maintenance and access.

4.3.5 *Technical Risk Assessment*

<u>Aerodynamic Risks</u>

Based on the concept design of the combined cycle turbine, the following technical risks are identified.

- As discussed above, there is a low technical risk that the aerodynamic design of the turbine could be completed successfully. However, there is a medium risk that the efficiency of a turbine designed using existing tools would not represent the optimum efficiency design. This risk should be minimised using a combination of a survey of existing helium turbine experience and a suitable test programme.

- There is a low risk that material selection would limit the aerodynamic design freedom. This would result in increased mass and cost (or reduced efficiency). However, we are confident that an aerodynamic solution would still be achievable.

- There is a high risk that the blade root / disk rim fixing would be unacceptable. Root fixing crushing calculations have shown that the stresses at the fixing are 250% over acceptable limits for the final stage. The mitigations for this are the use of a hollow casting or an increase in blade count (and reduction in blade mass). The design of the root fixing is identified as the highest technical risk for the current design.

Overall, a principal conclusion of the aerodynamic study is that we have a design with sufficient flexibility to adapt to changing material and mechanical requirements. As part of the current study we have considered six and seven stage designs to demonstrate the options that we have for working around potential problems.

<u>Mechanical Risks</u>

Mechanical risks associated with the nuclear application may impact the aerodynamic design; principally material selection and root fixing. The generic risks identified in Section 2.3.6 are also relevant.

<u>Radioactive Contamination Risks</u>

Risks associated with the effect of radiation, which effects the handling of the PCU during removal from the plant, and turbine removal, storage for radioactive decay of specific components and the subsequent repair processes where safety considerations will be needed.

4.3.6 *Costs*

It is anticipated that the combined cycle turbomachinery will offer some cost savings over the GT-MHR, arising from a lower part count, but this has not been assessed in detail at this stage.

4.3.7 *Areas of Uncertainty / Issues for Further Study*

As detailed above.

4.4 Electrical Generator

4.4.1 *System Description*

The proposed design is a combined cycle comprising a 66MW helium turbine generator and a 243MW steam turbine generator.

4.4.2 *Assessment of Design*

This is a less risky design from an electrical point of view since it effectively transfers 220MW of generation from the challenging helium atmosphere of the PCU to steam turbine generation. Steam plant is low-risk commercial off-the-shelf equipment and therefore will not be considered any further in this section.

In some respects, the risks presented by the helium turbine generator are the same kinds as those identified in Section 2.4 of this report, and will not be described again in this section. However, some of the risks are reduced simply because the size, and more importantly the weight, of the generator is reduced.

An exception in the higher speed of operation, 5000rpm, the main effect of which is to increase the centrifugal forces, unless the rotor diameter is reduced accordingly. It is believed, however, that a synchronous generator can be built to operate satisfactorily at such speeds. If not, an induction generator might be considered but this would be a considerable leap in technology as no such machine approaching 66MW has been built.

The generated frequency would be 83.33Hz but this is well within the capability of existing frequency conversion technology.

The effects of the lower rating on risks previous identified in Section 2.4 are briefly assessed in the following subsections.

4.4.3 *Ionising Radiation*

Although the reactor core and fuel elements are designed to reduce the risk of radioactive contamination of the helium gas, some contamination may occur. The three main electrical risks are: ionisation of insulation, degradation of the solid insulation of the generator windings and metal plating of the electrical insulation by gases that decay into metals such as silver 110 or otherwise.

The reduced generator rating does not alter this risk.

4.4.4 *High Temperatures Decomposing the Electrical Insulation*

Most common insulating materials used to wind electrical machines degrade significantly if the temperature rises above 150ºC. Therefore, the electrical insulation is at risk from the high temperatures of the helium working fluid.

The OKBM TDP will manage this risk by forced helium cooling which rejects waste heat to an intermediate water cooling loop. The smaller generator rating eases the cooling and so this risk is reduced.

4.4.5 *The Electrical Insulation of Helium*

A 66MW generator would probably have a rated voltage of 11kV rather than the 20kV proposed for the 286MW generator in the GT-MHR design. This lower rated voltage eases the electrical insulation requirements and so this risk is reduced.

4.4.6 *Generator and Exciter Current Lead Outs*

Pressure withstand and helium leak tightness

The proposed rating of the electrical lead-outs of the 286MW generator is 20kV and 10kA. The corresponding ratings for the 66MW generator would be lower, typically 11kV and 4100A. This considerably eases the design of the generator lead-outs and so this risk is reduced.

4.4.7 *High Speed Operation*

There are four main challenges presented by the high speed design: centrifugal forces, windage losses, flow-induced vibration and frequency conversion.

Centrifugal Forces

The most serious challenge is the strong centrifugal forces, which increase with the square of the speed. A rated speed of 5000rpm produces centrifugal forces that would be almost double that of 3600rpm designs unless the rotor diameter was reduced accordingly. Therefore the risk presented by centrifugal forces is increased.

Windage Losses

Previous studies have identified windage losses as significant (Reference 5). The windage losses are estimated from extrapolation of data from conventional synchronous generator designs. In a 500MW, 3000rpm power station generator cooled by hydrogen at about 0.3MPa, the windage losses are 190kW, that is, 0.04% of the generator rating.

In the case of the proposed 66MW generator rated at 5000rpm, windage losses are as follows:

- Windage losses are proportional to gas pressure since pressure increases the density of the gas. The pressure of the helium gas is 2.6MPa, 8.67 times the pressure of hydrogen used in the 500MW generator previously described. Therefore the windage losses increase by 8.67 times due to pressure alone.

- Helium is twice the density of hydrogen at the same temperature and pressure. This doubles the windage losses since windage losses are proportional to the density of the gas.

- The higher angular velocity of the generator. Windage losses are proportional to the cube of the peripheral speed and so an increase in speed from 3000rpm to 5000rpm increases the losses accordingly. However, the radius of the generator rotor will have to be reduced inversely in proportion to the square of the speed in order to manage the centrifugal forces. The combination of increased angular velocity but reduced rotor diameter means that the peripheral speed varies inversely with angular velocity and so the windage losses will be vary inversely with the cube of the angular speed. Therefore, the losses will be reduced by 0.317 due to the increase in speed.

Therefore, the windage losses in the proposed generator are estimated to be:

0.04% x 8.67 x 2 x 0.216 = 0.15% of the generator output.

The percentage windage losses are in fact less for the higher speed generator than for the 4400rpm generator (0.22%) and so the risk presented by windage loses is lower.

<u>Flow-Induced Vibration</u>

The density of the helium flow is much greater than that of hydrogen in conventional generators. Previous studies (Reference 5) have identified this as another serious challenge facing the generator design. The higher speeds may make flow-induced vibration more serious and therefore the risk is greater.

4.4.8 *Frequency Conversion*

<u>The Need for Frequency Conversion</u>

A power electronic frequency converter will be needed to convert the output of the generator, 83.3Hz (corresponding to 5000rpm) to 50 or 60Hz system frequency. The generator frequency is higher than that of the 286MW 4400rpm generator but well within the capability of frequency conversion technology. The risk presented by the frequency converter is less since its rating will be less than that required by the 286MW generator. The cost will also be less, about $8M.

4.4.9 *Vertical Arrangement*

The challenges presented by the shaft of the helium turbo-generator will be vertical rather than the horizontal arrangement used in most power plants. Using a 66MW helium generator rather than a 286MW generator eases the challenges presented by the vertical arrangement since the weight will be less. The weight of the generator rotor of the helium turbine is estimated to be 4 tonnes. This assumes that the rotor diameter will be less than that of 3000 and 3600rpm designs in order to manage the centrifugal forces.

4.4.10 *Magnetic Bearings*

The magnetic bearings are a significant risk to the project. Using a 66MW helium turbo-generator instead of a 286MW machine reduces this risk. As mentioned in Section 2.6, this design is within the mass/speed experience base of some bearing manufacturers.

4.4.11 *Overall Assessment of the Generator Design*

The combined cycle design has advantages over the GT-MHR design because the combined cycle design uses a much smaller helium turbine generator. The remainder of the generation is low risk steam turbine generation. This reduces the risks presented by the electrical generation and so, it should be possible to design and build electrical generators (helium turbine and steam turbine) for an NGNP plant by 2018, and the OKBM TDP target efficiency for the generator of 97.7% should be achievable.

4.4.12 *Potential for Weight Reduction: Alternative Higher Speed Designs of Generator*

The use of a 66MW helium turbine generator instead of a 286MW generator reduces the weight upon the magnetic bearings and therefore reduces the risk presented by this component. The size and weight of the turbine generator assembly may be reduced further by operating at higher speeds: 5000rpm is proposed and even higher speed may be considered.

As stated in Section 2.4, higher speeds offer several savings but also present several challenges, in particular increased centrifugal forces. At present the generator proposed for the GT-MHR is a synchronous generator, but large centrifugal forces may require an alternative design such as an induction generator (noting that no induction generator at 66MW is commercially available).

4.4.13 *Development Programme*

Many of the challenges presented by the GT-MHR are eased by a combined cycle design. A combined cycle generator could be delivered for an NGNP plant by 2018.

4.4.14 *Materials, Lifecycle, Maintenance and Operability*

See section 2.5. The materials, lifecycle, maintenance and operability of the 66MW generator is similar to the 286MW generator.

4.4.15 *Costs*

The following are ROM cost estimates:

- 66MW helium turbine: generator cost: $2.2M.
- Frequency converter for the above: $8M.
- 243MW steam turbine: generator cost: $5M.

4.4.16 *Areas of Uncertainty / Issues for Further Study*

See Section 2.4.

4.5 Heat Exchangers

The combined cycle offers the following advantages over the inter-cooled recuperated cycle:

- Removes recuperator, intercooler, precooler.
- Reduces weight on shaft.
- No radioactivity in secondary (water) circuit.
- Water side components available ‘off the shelf’.
- Similarities to existing (UK) nuclear plant.

It has, however, the following disadvantages:

- Plant is less compact.
- N'th plant may be more expensive due to higher part count.
- Risk of water entering helium circuit (General Atomics are not concerned about this risk).

This section discusses the concept steam generator module required by the combined cycle approach.

4.5.1 *System Description*

There are 7 Advanced Gas Reactor (AGR) plants in the UK, which have operated successfully for over 20 years. The AGR plant parameters (see Figure 54) are:

- Reactor cooled by CO_2 at 41bar.
- CO_2 transfers heat to a reheated steam cycle.
- The cycle efficiency is around ~42%.

The AGR-based cycle has been improved as follows:

- Steam temperature has been increased from 540˚C to 580˚C (reheated steam turbines for combined cycles are available off-the-shelf with 600˚C capability).
- The pinch in steam generator has been reduced from 75˚C to 40˚C (heat transfer from helium is far superior to CO_2).

The once through heat recovery steam generator has been modelled using correlations and methods described by Kakac and Liu (2002). The following tube arrangements have been considered in the steam generator:

- In-line tube arrangements.
- Staggered tube arrangements.
- Finned tube arrangements.

4.5.2 *Assessment of Design*

<u>Staggered Tube Bundle</u>

To minimise the spacing between tubes, staggered tube arrangements have been investigated. However, it has been concluded that the staggered arrangement as featured in Figure 74 is not attractive for the following reasons:

- Difficult to bend and package the tube rows.
- The module grows too much in 'height' because every tube row has to be offset to be able fit the next layer close to the previous tube row to minimise tube spacing.

- The modules will be difficult to package in a circular pressure vessel.

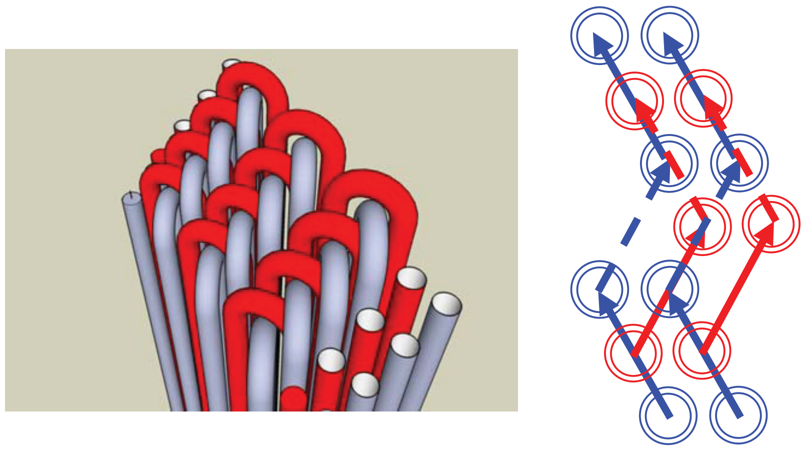


Figure 74 Staggered Steam Generator Tube Bundle

<u>Finned Tube Bundle</u>

A finned tube bundle arrangement could be used in the economiser and evaporator where heat transfer coefficients are significantly different on either side of the tube. By using fins, it is possible to increase the area ratio between 3 and 10 times. However, higher tube pitch is required to fit fins and this has the effect that cross sectional area has to decrease, hence requiring longer tubes and more water/steam entry rows. It will be more difficult to arrange the heat exchanger modules as the economiser and evaporator will have much smaller cross sectional area than the super-heaters and re-heaters.

<u>In-line Tube Bundle</u>

An in-line tube arrangement is the only feasible option. In all heat exchanger modules, except for the reheat modules, the water/steam enters the modules in the first two rows as shown in Figure 75. The steam enters the reheat modules in the first five tube rows. The spacing between every even and odd tube row has been increased to minimise the turning losses on the water/steam side and to avoid excessively tight bend radii. This installation becomes about 30% higher than estimated in Table 24 but this can be reduced again if the spacing in between the tubes is finned, hence the area of the fins corresponds to the number of tubes that can be removed to have the same height as both with and without increased bend radii between every second tube. The tube length in the economiser is 2200mm.

The tube dimensions and geometries are presented in Figure 76. The pitch ratio has to be as low as 1.22 to achieve the right Reynolds numbers and helium velocities between the tubes.

The heat recovery steam generator is divided into an economiser, evaporator, super-heater 1, re-heater 1, super-heater 2, re-heater 2 and super-heater 3. In total the overall height of all heat exchanger modules is around 7.5 m excluding headers underneath the economiser and after super-heater as well as before and after the re-heater. The estimated weight of all tubes is around 230 tonnes, which is of the same order of magnitude as the proposed GT-MHR recuperator.

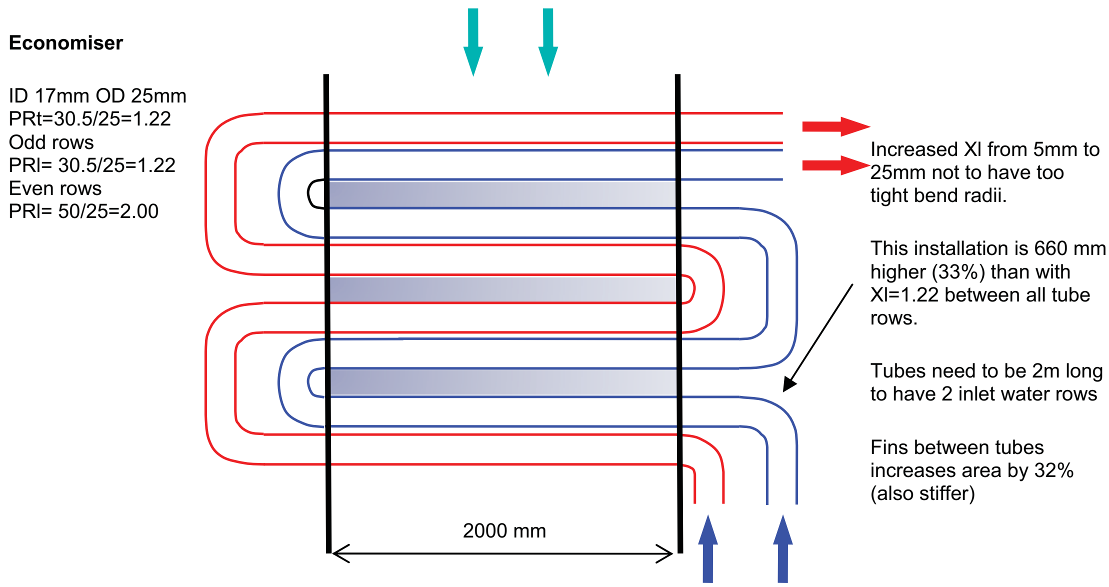


Figure 75 Economiser In-Line Arrangement

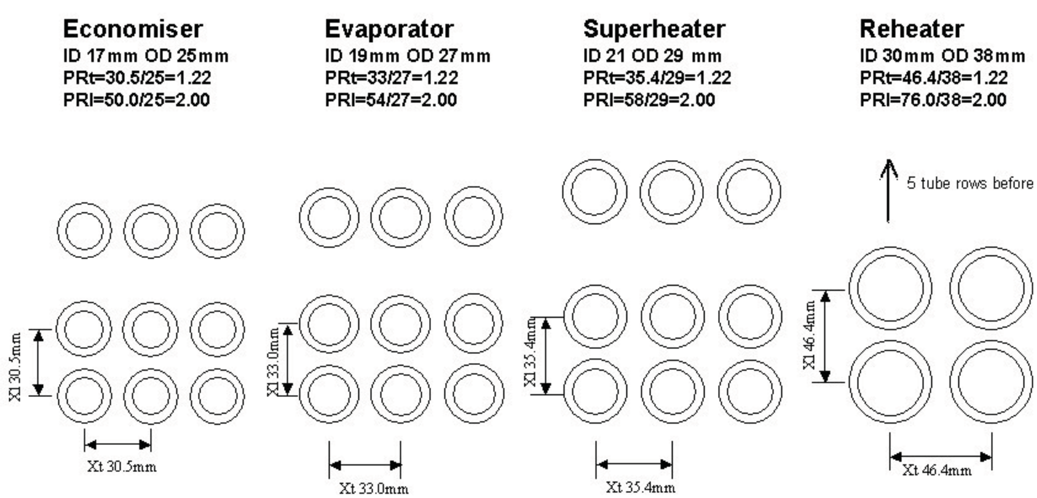


Figure 76 Steam Generator In-Line Tube Arrangements

Table 24 SG Design Data

HRSG		ECO	EV	SH1	RH1	SH2	RH2	SH3	RH3	TOT
Q	MW	167.2	133.8	116.3	68.3	29.1	8.7	4.4	1.5	529.3
LMTD	K	57	53	68	95	60	49	43	56	N/A
Nt*	-	407	407	407	782	407	782	407	782	N/A
CA	m2	12.4	13.4	14.4	14.4	14.4	14.4	14.4	14.4	N/A
OD	mm	25	27	29	38	29	38	29	38	N/A
ID	mm	17	19	21	30	21	30	21	30	N/A
Ho	W/m.K	3200	3160	3100	2940	3290	2990	3320	3000	N/A
Hi	W/m.K	14070	30000	5570	1090	4490	1125	4380	1125	N/A
U	W/m.K	1490	1400	1210	570	1150	585	1150	584	N/A
A	m2	2070	1890	1480	1330	440	320	95	47	7672
NR	–	65 (66)	55 (56)	40	36 (40)	12	9 (10)	3 (4)	1 (0)	228
H	m	1.98	1.81	1.42	1.67	0.42	0.42	0.1	0.04	7.72
Dp	%	0.46	0.39	0.28	0.27	0.09	0.07	0.02	0.01	1.6
W (tubes)	t	61	57	45	41	14	11	4	1	238
W (vessel)	t									250

Q: Heat transfer; LMTD: Log mean temp; Nt: Number of tubes in transverse direction; CA: Cross sectional area; OD Tube outlet diameter; ID: Tube inlet diameter; Ho: Outer heat transfer coefficient; Hi: Inner heat transfer coefficient; U: Overall heat transfer coefficient; A: Heat transfer surface area; NR: Number of tube rows longitudinal direction; H: Height; Dp: Pressure loss; W: Weight

* Number of tubes per row for ECO is 204 (2 entry tube rows), for RH it is 156 (5 entry tubes)

General Steam Generator Installation

There is over 20 years experience from AGR steam generator designs, which are very similar to the proposed steam generator design for the combined cycle GT-MHR. The tubes in the heat exchanger modules have smaller pitch than in the AGR design and because the pinches are lower, it also has more heat transfer area than the AGR design.

The steam generator is fixed below the economiser and it hangs on a centrally-mounted tree and the tubes can expand freely downwards. There are collector pipes before the economiser and after the super-heater as well as before and after the re-heater. Helium enters the pressure vessel at the top to keep the pressure vessel walls at the lowest possible temperature.

The tubes are arranged in diamond shaped modules to optimise the space required to fit them in a circular pressure vessel as shown in Figure 77.

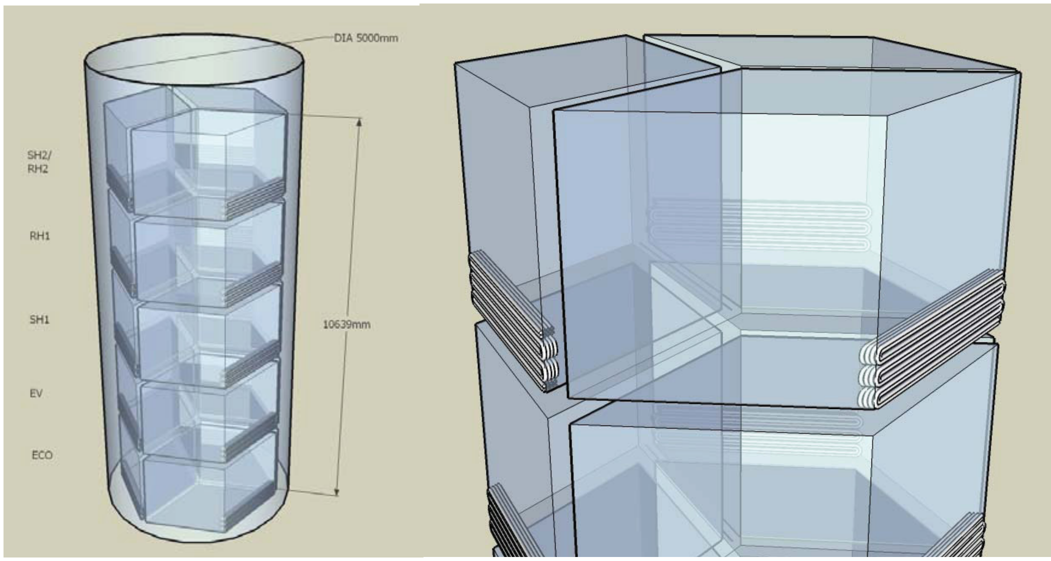


Figure 77 SG Layout

4.5.3 *Materials, Lifecycle, Maintenance and Operability*

The materials, maintenance and operability are similar to the AGR designs and procedures.

4.5.4 *Technical Risk Assessment*

No major technical risks have been identified.

4.5.5 *Costs*

The estimated costs for the steam generator have been estimated at $19M.

4.5.6 *Areas of Uncertainty / Issues for Further Study*

This steam generator should be worked up as a concept design during the next phase of work on the combined cycle option.

4.5.7 *Conclusions*

It has been shown in this preliminary SG study that it is possible to fit the SG in a similar size pressure vessel as the PCU, 5 m diameter and 15 m high.

The in-line tube arrangement is the only possible option.

Installation is similar to the AGR design.

The total tube weight is lower than the evaluated GT-MHR recuperator design.

4.6 Electromagnetic and Catcher Bearings

The EM bearing requirements for the combined cycle concept are easier than for the reference GT-MHR design. The generator's rotor mass is estimated to be around 10 tonnes, and the gas turbine rotor mass is also estimated to be around 10 tonnes. The shaft speed has increased slightly, however, to 5000rpm. Also the length of shaft between radial bearings is reduced. Dynamic analysis of the smaller, lighter gas turbine rotor shaft indicates that the number of resonant frequencies experienced during startup should reduce to 3 (from 4 for the GT-MHR reference design). All in all, it is considered that the combined cycle alternative concept has reduced EM bearing risk when compared with GT-MHR.

Discussions with the world's leading EM bearing suppliers about the combined cycle's bearing requirements have also been conducted. One supplier of EM bearings, SKF, has build experience of EM thrust bearings with loads up to 100kN (about 10 tonne) operating at 16 000rpm. Thus SKF claim experience similar to that required for this application. However, it is not clear whether their designs are suited to a seismic environment.

In consideration of the catcher bearings, the reduced gas turbine shaft output compared to GT-MHR means that the shaft and bearing diameters can be reduced considerably. This, coupled with reduced rotor mass in the generator and gas turbine, will reduce the risks in the catcher bearing design.

4.7 Whole Engine Mechanical Modelling

4.7.1 *System Description*

The details of the OKBM design and the preliminary critical speed analysis of the gas turbine rotor system are presented in Section 2.8. Please note that the calculations are very preliminary and take very little account of the rotor support structure and does not include any effects due to the electromagnetic bearings (EMBs). Section 2.8 shows that for this very simple analysis that we expect 4 critical speeds of the rotor system within the running range (0 – 4400rpm).

The OKBM design was regarded as a very heavy design and would require, at 33te, a design of EM bearing that would be larger than any designed in the world to date. It would, therefore, be very sensible to try and reduce the weight of the system proposed.

The OKBM proposed plan (described briefly in reference 1 and similar to the plan described in the ASME paper, reference 1.2) is a Brayton cycle with primary helium coolant used as the working fluid. The new proposal from Rolls Royce is a combined cycle, and the proposed rotor system for the gas turbine is shorter than the system proposed by OKBM.

4.7.2 *Critical Speeds of the Revised Rotor Design*

No drawings were produced for the new design but the following details were assumed for the combined cycle engine:

- Reduce the total number of compressor stages from 23 to 18
- Remove the HP compressor
- Remove the section of shaft between the LP and HP compressor

- Reducing the shaft length between the bearings from 9.4 m to 7.6 m
- Reduce the number of turbine stages from 9 to 7
- Working speed increased from 4400rpm to 5000rpm.

The Rotordyn model was modified to take into account all these changes.

In additional to the updates listed above a third model was produced which included all these changes plus a titanium compressor. For the initial design steel properties were assumed for the complete rotor system, as it was not know then that weight would be an important design factor. Now it is known that the weight of the rotor system may require a very special EMB it is important to reduce the weight of the rotor system if possible. The original weight of the Rotordyn model of the OKBM design was 32.8 tonnes, which is roughly in agreement with the OKBM values. When modified, as above, the estimated weight reduces to 26.4te but when the compressor material is changed from steel to titanium this reduces to 20.5te.

The critical speed analysis was run for the revised configuration and for the revised configuration with a titanium compressor.

4.7.3 *Predicted Critical Speeds for the Revised Rotor Designs*

Table 25 compares the critical speeds predicted by Rotordyn for the original OKBM design and for the two revised designs. These have been calculated for two bearing stiffnesses: i) rigid bearings (K= 1×10^8 N/mm) and ii) for a flexible bearings (K= 1×10^5 N/mm).

Table 25 Predicted Critical Speeds for the 3 Different Designs - Flexible Bearings

Mode	Frequencies (%N)		
	OKBM	RR All steel	RR Ti compressor
1	522 (11.9%)	635 (12.7%)	689 (13.8%)
2	1198 (27.2%)	1359 (27.2%)	1581 (31.6%)
3	1979 (45.0%)	2332 (46.6%)	2860 (57.2%)
4	3596 (81.7%)	4714 (94.3%)	5209 (104.2%)
Max speed (100%N)	4400rpm	5000rpm	5000rpm

Table 26 gives the critical speeds calculated for the rigid bearings. Although the flexible bearing condition is probably more likely it is useful to see what the rigid bearing configuration gives to envelope the responses.

Table 26 Predicted Critical Speeds for the 3 Different Designs - Rigid Bearings

Mode	Frequencies (%N)		
	OKBM	RR	RR
		All steel	Ti compressor
1	699 (15.9%)	933 (18.7%)	973 (19.5%)
2	2545 (57.8%)	3258 (65.2%)	3542 (70.8%)
3	4242 (96.4%)	4778 (95.6%)	5352 (107.0%)
Max speed (100%N)	4400rpm	5000rpm	5000rpm

4.7.4 *Conclusions*

Stiffening the shaft by making it shorter and reducing the weight of the compressor, by changing the material from steel to titanium, has increase the critical speeds of the rotor system. It may be possible to reduce the number of critical speed in the range to 3. However, this current simple analysis shows the 4th critical speed near the running speed (5000rpm). If this is still the case when analysed more fully, design changes will be possible to put a greater margin between the any critical speed of the rotor and the working speed.

5 Conclusions & Recommendations

Rolls-Royce has conducted a pre-concept study of the GT-MHR PCS for NGNP. The Rolls-Royce study had two main objectives. These were:

- Assess the current preferred GA/OKBM concept for PCS and suggest improvements.
- Develop the concept into improved 'Rolls-Royce' PCS concept design.

Both of these objectives have been achieved, by an integrated team pulled together from all four parts of Rolls-Royce (Civil Aerospace, Defence Aerospace, Marine, and Energy).

In the assessment of the GA/OKBM reference PCS design, Rolls-Royce concluded that the concept was elegant and achievable, but carried significant key risks. These were identified as follows:

- Recuperator life and cost considered very high risk.
- Active electro-magnetic bearing/catcher bearing requirements are outside of current world experience - high risk.
- Cost of power electronics required for 4400rpm/286MW generator anticipated to be large (~$50M) - high risk commercially.

An alternative recuperator design has been proposed by Rolls-Royce which would be much more compact and less expensive. It is a cross-corrugated design that is made by the diffusion bonding of corrugated stainless-steel plates. Our analysis shows that moving to this alternative design would meet the required performance of this component for the cycle. Even with this alternative design, the life of the recuperator for the GT-MHR concept should still be considered a moderately high risk.

An alternative concept has been worked up, that addresses some of the key risks identified in the reference concept. This is a combined cycle, consisting of a 66MWt gas turbine generator with the remainder of the power taken by a conventional steam cycle. The key features of this concept are:

- The recuperator is no longer required. A steam generator would be required, but this is considered much lower risk.
- EM bearing risks are reduced by reducing generator weight from 35t to around 10t, and turbomachinery shaft weight from 32t to around 10t.
- Power electronics costs are reduced (since generator is reduced from 300MW to 66MW in gas turbine part).
- Plant efficiency is increased, compared with the GT-MHR Brayton cycle.
- Steam turbines and steam cycle electrical generators are commercial off-the-shelf items - low cost and low risk.

The combined cycle alternative could be expected to have lower plant costs because much of the steam machinery is commercial off the shelf, but this saving would be offset by requiring a bigger containment building and the extra maintenance burden of the steam cycle parts.

In conclusion, both the GT-MHR reference design PCS and an alternative combined cycle PCS have been worked up in the pre-concept study. They both have advantages as described below:

Combined Cycle Advantages

- Reduced EM bearing risk.
- No recuperator.
- Steam equipment (excluding steam generator) would be commercial off-the-shelf.
- Total equipment costs should be lower.
- Steam generator can exploit e.g. AGR experience.
- Flexibility to have process steam instead of electricity from steam plant.

OKBM/GA Reference Cycle Advantages

- More compact - smaller equipment footprint, both inside reactor building and outside.
- More elegant, simpler cycle - less complexity.

It is recommended that further work be undertaken, in the next phase, to decide which cycle should be selected for the NGNP application.

During the PCS preconcept study, other areas have emerged as requiring study in any follow on programme. Some of the more significant are:

- Transient performance. The start-up and transient behaviour of the PCS needs to be better understood so that the transient requirements for the components can be properly assessed. This will require a transient performance model to be constructed for the cycles and is a significant undertaking.
- A study of the control system for the PCS needs to be made. This system's behaviour is intimately bound up with the PCS's transient requirements.
- Further refinement to turbine designs to increase confidence of achieving 60 000hours creep life with uncooled turbine blades at 850ºC for both the reference cycle and the combined cycle.
- Further exploration of the implications of 950°C operation, particularly cost/complexity/performance trade-offs (including blade cooling and thermal barrier coatings).
- A more thorough investigation into EM bearing capabilities and alternative technologies is required because the EM bearings are such a key feature of both concepts.

6 Abbreviations and Acronyms

AC	Alternating Current
AGR	Advanced Gas-cooled Reactor
ALARP	As Low As Reasonably Practical
ASME	American Society of Mechanical Engineers
CB	Catcher Bearing
CCGT	Combined Cycle Gas Turbine
CF	Centrifugal
CFD	Computational Fluid Dynamics
COTS	Commercial Off-The-Shelf
DC	Direct Current
DFL	Dry Film Lubricant
DOE	Department Of Energy
EB	Electron Beam
EM	Electromagnetic
EMB	Electromagnetic Bearing
FMECA	Failure Modes Effects and Criticality Analysis
FOD	Foreign Object Damage
GA	General Arrangement / General Atomics
GT	Gas Turbine
GT-MHR	Gas Turbine - Modular Helium Reactor
IHPM	Internal High Pressure Module
IHX	Intermediate Heat Exchanger
INL	Idaho National Laboratory
ITRG	Independent Technology Review Group
IVM	In-Vessel Metalwork
HCF	High Cycle Fatigue
HP	High Pressure
HPC	High Pressure Compressor
HPM	High Pressure Module
HTC	Heat Transfer Coefficient
HVDC	High Voltage Direct Current
ISA	International Standard Atmosphere
ITRG	Independent Technology Review Group
JAERI	Japanese Atomic Energy Research Institute
LCF	Low Cycle Fatigue
LE	Leading Edge
LOCA	Loss Of Coolant Accident
LP	Low Pressure
LPC	Low Pressure Compressor
MHR	Modular Helium Reactor

MWe	Megawatts electrical
MWt	Megawatts thermal
NGNP	Next Generation Nuclear Plant
NGV	Nozzle Guide Vane
OKBM	Experimental Design Bureau of Machine Building (trans. from Russian)
PBMR	Pebble-Bed Modular Reactor
PCS	Power Conversion System
PCU	Power Conversion Unit
R&O	Repair & Overhaul
RPV	Reactor Pressure Vessel
SG	Steam Generator
TBC	Thermal Barrier Coating
TDP	Technology Development Programme
TE	Trailing Edge
VHTR	Very High Temperature Reactor

7 References

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Annex A - Comments on ITRG Risk Assessment

Reference - INEEL/EXT-04-01816 Design Features and Technology Uncertainties for the Next Generation Nuclear Plant, Independent Technology Review Group, 30 June 2004

The following is extracted from the above reference, with Rolls-Royce's comments added:

4. POWER CONVERSION SYSTEM

4.1 Introduction

A review of the design, fabrication, and operation of the power conversion systems proposed by the proponents of prismatic gas-cooled reactor concepts and the PBMR concept was conducted with the objective of identifying the major risks associated with these systems. In addition, information was also obtained from background documents prepared by DOE and INEEL for the NGNP.

4.2 Assessment of Risks

In assessing risks, it has been assumed that the ITRG recommendation with respect to reducing the NGNP reactor outlet temperature will be adopted; specifically, that this temperature will be reduced from 1000 to 900 - 950°C. As discussed in the materials section of this report, the metallurgical risks of the 1000°C outlet temperature are unacceptably high.

The reference power conversion cycle for the NGNP is direct; the helium circulating through the reactor core is also circulated through the power conversion system. One of the proponents of a prismatic gas-cooled reactor has proposed a direct-cycle power conversion system, and the designers of the PBMR are in the process of developing and testing hardware for a direct cycle. However, another prismatic reactor proponent has proposed an indirect cycle employing an intermediate heat exchanger to transfer heat to a closed secondary power conversion system.

The proponent of an advanced high-temperature (molten-salt-cooled reactor) proposed a compound indirect cycle: a secondary molten salt loop transfers heat through three salt-to-gas heat exchangers in parallel. The transferred energy is used to power three turbine-compressors in cascade; each heat exchanger reheats the gas working fluid. This relatively complex arrangement is made necessary by the relatively small temperature difference across the primary side of the intermediate heat exchanger. Efficient heat transfer requires relatively small gas temperature rises across the secondary side of this heat exchanger.

The indirect power conversion cycle proponent proposes to use a mixture of nitrogen and helium for a working fluid. The mixture has thermodynamic properties closely resembling air and the combustion gas mixture employed in conventional gas turbine power plants. The single-shaft power conversion assembly is oriented horizontally. It resembles and uses much of the same technology as a conventional combined cycle (a gas turbine topping a steam turbine) power plant. Gas from the turbine exhaust is directed to a steam generator, where it is cooled to a temperature in the 50 - 100°C range before entering the compressor. The steam from the steam generator is used to power a steam turbine, and the steam turbine exhaust is directed to a condenser, the ultimate heat sink for the cycle. Water from the condenser is directed back to the steam generator via condensate and feedwater pumps.

The cycle trades the recuperator, precooler and, for some designs, an intercooler of a Joule-Brayton cycle for the steam generator, condenser, feed pump, and steam turbine. About 80% of the power in this cycle is derived from the steam turbine and 20% from the gas turbine. Because of the highly efficient and fully developed Rankine steam cycle with superheater outlet temperatures of about 500 °C, there is substantially less risk of this cycle failing to meet efficiency targets than with the direct Joule-Brayton cycle. In addition, the low return temperature from the secondary system leads to a low reactor inlet temperature and a high reactor coolant temperature difference. This in turn leads to a low coolant mass flow requirement-hence, a low coolant operating pressure (a high coolant density is not required to transport the energy). The low return temperature allows use of well-developed alloys for pressure bearing parts; the low pressure leads to moderate wall thickness for those parts. On the other side of the ledger, however, the cycle is significantly more complex than a direct Joule-Brayton cycle, as is anycombined-cycle plant. Combining two or more reactor modules with one larger power conversion unit might compensate for this drawback.

Note that the indirect cycle need not be a combined cycle. An IHX will produce, conservatively, a secondary gas outlet temperature 50°C below the reactor outlet temperature. The loss in efficiency of a recuperated Joule-Brayton cycle with a 50°C reduction in turbine inlet temperature is only about 1.5%. However, the primary coolant pumping power required for the indirect cycle will impose an additional efficiency penalty of as much as 1.5%, depending on coolant flow requirements. The indirect-combined cycle is expected to offset most of this penalty.

An important advantage of the direct cycle over the proposed combined cycle is its apparent simplicity. However, for the NGNP, the most significant drawback of the direct cycle is the large number and diversity of the development risks associated with it. These risks are described in Table 27.

An examination of the table shows risks of several kinds:

- Metallurgical. The front end of the turbine is 50°C hotter for the direct cycle. Achieving 6-7 year component life (or better) at the higher temperatures is a development challenge.

- Maintenance/availability. The vertical and compact modular arrangements proposed make for very difficult access for maintenance. If component reliability is not high, lost time for maintenance outages will be large. In addition, the direct cycle is susceptible to contamination (e.g. by fission products or activated silver) of the power conversion unit, which may result in high shutdown radiation levels.

- Design/development. The magnetic bearing systems required to minimize leakage from the direct cycles are in an early stage of development and will require full-scale prototype testing. The auxiliary (catcher) bearings likewise need development. These bearings prevent physical damage to the turbomachinery in the event of a failure of the magnetic bearing system.

- Operational. Control of the power conversion system during normal operations and during upsets involves components and concepts that are in an early stage of development and unfamiliar to utility operators. (In this regard, some aspects of indirect cycle control may also need development.)

Given the large number and formidable nature of the risks associated with the direct cycle and their potential impact on the NGNP schedule, the ITRG concludes that the NGNP should proceed on the basis of a lower-risk indirect cycle (not necessarily the cycle proposed by the proponent of the indirect cycle). The indirect cycles necessitate use of an intermediate heat exchanger-itself a component with significant developmental risks. However, the ITRG considers these risks-described in Section 6 to be more manageable than the aggregate of the risks associated with any of the direct cycle concepts.

The recommended approach is not intended to foreclose the future application of direct cycles to Generation IV power plants. The potential advantages of plant simplicity afforded by a direct cycle are great. Accordingly, the ITRG also recommends that the DOE actively participate in the several direct cycle developments currently under way: the PBMR power conversion system, the US International Team power conversion system, and the Gas Turbine High Temperature Reactor (GTHTR-300) (JAERI) power conversion system. Further, the NGNP indirect secondary circuit can be used as a test platform for direct cycle components.

Finally, as noted above, some of the control challenges of direct-cycle power conversion systems are also present in indirect-cycle power conversion systems. The ITRG considers that the selection of a control system design approach and the evaluation of its effectiveness would benefit from operating utility participation at an early stage. This will provide greater assurance that the control system design approach ultimately adopted will be operationally acceptable.

Table 27 Risk Summary for Direct-Cycle Power Conversion Systems

<table>
<tr><th>Component/ Subassembly</th><th>Risk Issue</th><th>Potential Consequence of Risk Issue</th></tr>
<tr><td>High-temperature end of turbine</td><td>Life of components, replacement cost and outage time requirements</td><td>Acceptable plant availability and replacement part costs require that hot end components achieve lifetimes in the 6-12 year range. Achieving this objective will require turbine disc cooling and, depending on turbine inlet temperature, cooling of the front-end blades, as well as the use of advanced materials.</td></tr>
<tr><td colspan="3">Rolls-Royce Comments:

We would agree with the risks identified in the report. Achieving both the long operational lives of six years, combined with the temperatures quoted in excess of 900ºC in the report provides a significant risk. Turbine disc cooling is necessary, and blade cooling has to be considered at >850ºC. Material choice is critical with only high temperature alloys being suitable for the blades. With incorporation of turbine blade cooling, Rolls-Royce has experience of long life applications at temperatures hotter than 900ºC, but the turbine aerofoil bladeshapes for the Gen IV application will be very different to existing gas turbine experience. Feasibility of incorporating blade cooling into these new aerofoil shapes needs to be assessed.</td></tr>
<tr><td>Turbine compressor blade path</td><td>Bypass leakage</td><td>Pressurized helium is a medium difficult to contain. The efficiency of the Joule-Brayton cycle is very sensitive to bypass leakage around compressor and turbine stages. The ability to achieve acceptable leakage can only be demonstrated by testing at or near full scale.</td></tr>
</table>

<table>
<tr><th>Component/ Subassembly</th><th>Risk Issue</th><th>Potential Consequence of Risk Issue</th></tr>
<tr><td colspan="3">Rolls-Royce Comments:

Maintaining adequate containment of helium is difficult within pressure vessels where leakages through seals are typically far higher than with other fluids such as air. However, within the blading of a gas turbine, the leakage over the tips or through the inter-stage seals is no more or less important than any other working fluid. This is because the critical parameter is not the absolute leakage, but rather the proportion of the working fluid that by-passes the blading rather passing through it. This is principally governed by the tip (or seal) clearance area compared to the open area in the blading.

However, the other properties of helium result in a design of turbine with a high blade speed, but low Mach number. In turn this tends to produce short blading with a high hub-to-tip ratio where the tip clearance is more important. This is mitigated by the vertical axis of the turbine and the use of electromagnetic bearings that allow the tip clearance to be better controlled.

In summary, the use of pressurised helium as a working fluid requires careful consideration of the leakage effects, but this is within usual gas turbine experience and can be mitigated.</td></tr>
<tr><td rowspan="2">Power conversion system vessel</td><td>Pressure boundary material and size</td><td>The vessel enclosing the power conversion system is a Class 1 boundary. Risks associated with its size and materials of construction are similar to those associated with the reactor vessel.</td></tr>
<tr><td>Bypass leakage</td><td>Because of differences in thermal expansion of various segments of the power conversion system, sliding (piston ring) or bellows seals between various subassemblies and the vessel wall will be necessary. Leakage past these seals degrades power conversion efficiency and may cause thermal striping of structural parts, leading to rapid fatigue.</td></tr>
</table>

Component/ Subassembly	Risk Issue	Potential Consequence of Risk Issue

Rolls-Royce Comments:

No specific comments on pressure boundary material and size - extent of risk should be less than reactor vessel, due to temperatures and radiation fluxes.

There are two classes of leaks: (A) within the structure inside the GT and the 'fixed' structures of the pressure vessel (B) the sliding seals between the two structures:

The fixed seals (A) are probably within normal engineering experience although the difficulty associated with high pressure piping to the recuperator must not be underestimated.

The sliding seals (B) seem to be a very novel feature. Not only because of their size, duty and effect on assembly risks of the plant but they also provide radial structural support especially for the inlet magnetic bearing.

For GT-MHR there are seven static seals each with a developed length of approximately 9.5 m.

GT efficiency is clearly sensitive to leaks across the seal between GT exhaust and reactor inlet and indeed this seal is subject to high temperatures and very high pressure difference.

The seals seem to be of a floating ring type with two separate ring assemblies.

The seal must be assembled 'blind' during the insertion of the GT module, but other seals appear to engage earlier thus probably reducing the risk damage during assembly.

Relative movements will occur in the axial and radial directions.

Axial movements will be small during normal operation. The distance between the seal and common 'grounding point' is about 9m but temperature profiles along the paths are probably similar

The outer seal surface will be subjected to transients during operation of the bypass valve. It seems likely that the outer will contract thus closing seal clearances.

The radial location of the GT assembly is transmitted through these seals to the case. So during start sequence fairly large vibration will need to be accommodated especially near GT inlet

All piston ring seals are subject to wear hence some development will be required. Cycling such a seal should be relatively easily in a test rig hence the design should be understood early in the development phase. Long term material surface properties in a helium environment may be the most significant change to current practice.

It is not obvious how the seal can be checked during or after assembly other than by measuring seal temperatures and cycle performance. A broken seal ring will be contained during operation and should be easily recovered during heavy maintenance. Although the depressurisation and disassembly of a PCU is expensive the failure of this seal should not compromise safe (and slow) shutdown of the plant.

Summary:

Risk Level : Medium

Consequence: Loss of performance extended maintenance

Mitigation: Extensive rig test and structural analysis. Develop maintenance procedure to allow quick(ish) seal replacement

<table>
<tr><th>Component/ Subassembly</th><th>Risk Issue</th><th>Potential Consequence of Risk Issue</th></tr>
<tr><td>Radial bearing systems</td><td>Reliability, consequences of loss of power to primary bearing system</td><td>To facilitate system tightness, proponents of direct cycle power conversion systems have proposed magnetic bearings. For system tightness, this choice is logical, but magnetic bearings for a 30-ton or more assembly rotating at 4400rpm (the single shaft design proposed by one proponent) are in an early stage of development. Magnetic bearings are .tuned to the dynamic characteristics of the rotating system that they support. A methodology that does not risk the hardware during initial startup is required for presetting them. The failure modes and effects of the bearing system, and the large clearance between the shaft and catcher bearings, must be analyzed and the design modified as necessary to reduce the risks to the rotating system to acceptable levels. Finally, an auxiliary (.catcher.) bearing system must be developed that is capable of allowing the rotating system to come to a stop without damage in the event of a loss of power to one or more of the magnetic bearings.</td></tr>
<tr><td colspan="3">Rolls-Royce Comments:

We agree with the ITRG view. The catcher bearings clearly represent a major risk especially the radial bearings of a vertically orientated shaft, which will require a concept to be developed.

Summary:

Risk Level : High

Consequence: High (bearing failure during operation at high power)

Mitigation: Extensive rig test and structural analysis</td></tr>
<tr><td>Axial bearing system</td><td>Reliability consequences of loss of power to primary bearing system</td><td>Some proponents of the direct cycle have proposed that the axial thrust of the turbine compressor system(s) be reacted by a magnetic bearing. This approach has similar but, for a vertical turbine compressor assembly, greater developmental challenges than the magnetic radial bearing system described above. In the vertical shaft configuration, the bearing must deal with the rotating system weight. For this reason also, the catcher thrust bearing is also a developmental challenge.</td></tr>
<tr><td colspan="3">Rolls-Royce Comments:

We agree with the ITRG view. The catcher bearings clearly represent a major risk, which will require a concept to be developed.

The loading on the thrust bearing is very high and thus extensive development of the catcher bearing will be required.

Summary:

Risk Level : High

Consequence: High (bearing failure during operation at high power)

Mitigation: Extensive rig test and structural analysis</td></tr>
</table>

<table>
<tr><th>Component/ Subassembly</th><th>Risk Issue</th><th>Potential Consequence of Risk Issue</th></tr>
<tr><td>Generator shaft seal</td><td>Leak tightness</td><td>At least one proponent of the direct cycle has proposed a high-speed dry helium seal between the turbine and the generator. This approach allows the use of oil bearings for the generator and an oil-lubricated thrust bearing for the turbine-compressor-generator assembly. Such a seal, while desirable, is still undeveloped and will require proof testing of a prototype.</td></tr>
<tr><td colspan="3">Rolls-Royce Comments:

Not applicable to the GT-MHR, however some general comments follow on the generator to turbomachinery seal, which is not designed to be leak-tight:

This seal only needs to prevent significant contamination of the generator. If it fails in service the helium purification flow will need to be increased to offset the risk of contamination

A large volume seems to have been allocated to these seals. Other mitigations may be possible by operating the magnetic bearings at low voltages and providing a second set of seals near to the generator. This may allow the rate of contamination of the intermediate chamber to be monitored. Dry seal design experience exists in the process industry (pipeline compressors etc. hydrogen seals within generators) hence the amount of technological development seems moderate

Summary:

Risk medium/low

Consequence: High (Generator failure due to silver 110 contamination)

Mitigation: Extensive review and rig test</td></tr>
<tr><td rowspan="2">Single shaft design</td><td>Vertical orientation, access for maintenance/ frequency of maintenance</td><td>One proponent of the closed cycle has proposed a design wherein the turbine, compressors, and the generator are located on a single vertical shaft. This assembly and a number of heat exchangers (recuperator, precooler, and intercooler) are all contained in a single power conversion vessel. Because of the relatively difficult, step-wise maintenance access problems created by this arrangement, the proponents have proposed a 6- to 7-year interval between maintenance outages for the power conversion system. This period appears impractically long, particularly for instrumentation and hot-end hardware.</td></tr>
<tr><td>Horizontal orientation, bearing system</td><td>The JAERI GTHTR design employs a horizontal power conversion module. This design overcomes many of the access-for-maintenance problems of the vertical design. It also reduces thrust loads, thereby making the design of the catcher thrust bearing less challenging. However, the radial bearing system, including the radial catcher bearings, must be designed to handle the gravitational load in this arrangement radial bearing system.</td></tr>
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Component/ Subassembly	Risk Issue	Potential Consequence of Risk Issue
Rolls-Royce Comments: *The life of the bellows seals on the hot inner reactor to GT return connection tube must be required to match the reactor life. Working so close to the reactor outlet connection must be a high exposure activity. The OKBM vertical insertion allows this connection to be made with minimal human exposure. I believe vertical orientation is driven by this assembly process. A mechanism to do this horizontally would be very costly and difficult (if indeed possible) to design.* *Clearly the issue of shaft bow could be dealt with by installing a baring system, but the vertical design probably minimises compressor blade tip clearance requirement* *It is supposed that the radial catcher bearing clearance issue can be addressed but this will require concept development.* *Crane height may be a significant issue but this orientation at least minimises combined maintenance and operating plant foot print* *Recuperator life is the key issue. It is a 'passive component' but subject to high pressure and thermal loading. The high pressure piping is complex and may require frequent access to plug leaking sub assemblies. It is not at all clear how this will be done but it is also not clear that a particular orientation of shaft would affect the process.* *Instrumentation and services needed by the GT will need to be dressed onto the GT. In particular the power supply to the bearing within the intake needs to run through the hot structure of the 'discharge nozzles and turbine* *Balance of risks: I guess I agree that the vertical catcher bearing is more feasible than radial bearings for a horizontal machine but both represent significant challenges.* *Summary:* *Cranage: Risk Low : Consequence: High Mitigation: Review and simulation* *Recuperator Maintenance Risk High: Consequence: High Mitigation: Review and simulation* *Shaft orientation - bearing design Risk High: Consequence: High Mitigation: Review, simulation rig test*		
Multi-shaft design	Turbine over-speed	One direct cycle proponent proposes to divide the rotating power conversion components into three subassemblies each on a separate shaft: two turbo-compressors and a turbine-generator. The advantage of this arrangement is that the rotational speed of the turbine-compressors can be chosen to optimize the size and thermodynamic performance of these machines. The speed of the generator turbine, on the other hand is selected to match the power system frequency. The power turbine assembly presents a developmental challenge with respect to the prevention of a destructive over-speed on loss of load. The proponent has proposed a large resistor bank that is connected by a fast-acting circuit breaker to the generator in the event of load loss. However, this measure does not deal with a loss-of-generator excitation, and its reliability and acceptability to power generating utilities remain to be demonstrated.

Component/ Subassembly	Risk Issue	Potential Consequence of Risk Issue
Rolls-Royce Comments: *Rolls-Royce do not it is possible to have two independent PCU's in parallel, because it will cause turbomachinery instability.*		
High-power bi-directional static frequency converters	Feasibility, reliability, induced torsional vibration of turbo machinery due to harmonic content	One of the direct cycle proponents has proposed a startup sequence that involves operating the generator in a synchronous-motor mode during startup, to provide starting power to the compressor and to facilitate synchronization (there is no control valve to admit hot gas to the turbine). If the generator design speed corresponds to the line frequency, this arrangement appears feasible. However, this direct cycle proponent has chosen a design speed above line frequency, to reduce the rotating mass. This arrangement requires a frequency converter on the generator output. A rotating frequency converter can be made bi-directional. However, the proposed concept uses a static converter. A bi-directional static frequency converter in the 300-MWe range requires development. Furthermore, static converters that use silicon-controlled rectifiers to generate sinusoids of the desired frequency generate harmonics in the power source. These harmonics could excite super-synchronous resonant torsional frequencies of the rotating assembly. [The multi-shaft direct cycle proponent also employs a static variable frequency converter to bring the power turbinegenerator (operating in a synchronous motor mode) up to line frequency. The power rating of this converter is low, and it is not adequately developed, although the arrangement is unusual.]
Rolls-Royce Comments: *We think that the power electronics to achieve the requirements is within current experience and would be reliable. The risks are therefore low. We do, however, think that the costs associated with these power electronics will be large – around $100M for a 300MW generator.*		
Control	Control of load during normal operation; control of overspeed on loss of load	The very long thermal time constant of the reactor and the impracticality of control valves in the reactor outlet (turbine inlet) duct lead to unusual control requirements. Rapid load control or frequency regulation does not appear feasible. One direct cycle proponent effects power changes by changing helium inventory, thereby changing its pressure and density. Reactor power is controlled in tandem with system pressure. Bypass valves are used to control over-speed on generator trip or load rejection transients. The control methodology is undeveloped and unfamiliar to utility operators.
Rolls-Royce Comments: *Outside of the scope of this Rolls-Royce study.*		

Component/ Subassembly	Risk Issue	Potential Consequence of Risk Issue
Contamination of turbine blade path and recuperator surfaces	Access for maintenance	Silver and other fission products will carry over and condense on the cooler parts of the blade path and the inlet end of the recuperator. Carryover of fission products has been observed in AVR and other gas cooled reactors. The amount of carryover depends on the integrity of the fuel kernels. The carryover results in radioactive contamination of the turbine assembly. Direct cycle proponents propose to have a spare rotating assembly that they will use to replace a rotating assembly pulled for maintenance. The radioactivity of the pulled assembly will require it to be decontaminated before inspection and refurbishment. If the radioactivity level is high, decontamination will require a large and costly on-site facility.
Rolls-Royce Comments: *The risk noted here is correct. The use of a spare rotating assembly will assist in allowing the PCU operation utilisation to be kept as high as possible, but risk mitigation activities are required through the turbine removal process to ensure contamination does not escape to atmosphere, and through the turbine contamination reduction, and repair and overhaul processes. The repair and overhaul process has to be considered against allowable dosage requirements for personnel using appropriate protective clothing, and any equipment that may be needed for remote operations. The turbine itself and its installation in to the GT may need specific design features to allow ease of removal, larger bolt and flange sizes etc. These have to be considered on a case by case basis against the contamination and the resulting decontamination and overhaul policies.*		