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PCS options for NGNP Generation IV nuclear reactor

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Summary#60

Following on from Next Generation Nuclear Plant phase 1 pre-concept work carried out in 2007 looking at the Power Conversion System (PCS), Rolls-Royce was tasked with investigating combined cycles further, to identify the best PCS cycle option. In particular Rolls-Royce's was asked to evaluate an indirect combined cycle PCS for a commercial plant, to compare this with a direct cycle PCS, and to make a Rolls-Royce recommendation on the best option for a commercial plant. Furthermore Rolls-Royce was also tasked to advise, based on the findings and recommendations, how technology required for this option can be demonstrated in the NGNP programme.

To perform this work, an integrated team was pulled together from all five parts of Rolls-Royce (Civil Aerospace, Defence Aerospace, Marine systems, Energy and the new Civil Nuclear group). The integrated team is shown in Figure 1.

Rolls-Royce studies of an indirect combined cycle have concluded that there are very significant reductions in risk/cost for this arrangement compared to a direct combined cycle:-

The Rolls-Royce studies compared the costs and risks between direct and indirect and concluded that, although the direct cycle was predicted to be more efficient (50.2% versus 48.6% for the indirect at 850°C reactor outlet temperature), an indirect combined cycle would be a cheaper and lower risk option for a commercial electricity plant.

Furthermore, in comparing the indirect combined cycle plant with a pure steam cycle, the addition of an intermediate heat exchanger (IHX) and a gas turbine provides excellent value for money increasing the cycle efficiency from 42.6%. to 48.6% which is a massive benefit in terms of both plant capacity and fuel/waste processing costs.

The Rolls-Royce study therefore recommends an indirect combined cycle as a better choice than a direct combined cycle for a medium term commercial electricity plant. In the longer term, evolution of the technology base may bring other cycles into consideration.

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1 Executive Summary

Following on from NGNP phase 1 pre-concept work carried out in 2007, two constraints on the NGNP power conversion system (PCS) configuration were made. These were firstly, that the NGNP pilot plant would use an indirect cycle to reduce risk, and secondly that the NGNP pilot plant PCS must produce steam. This rules out a pure gas turbine cycle, but a combined cycle (which Rolls-Royce favoured in studies carried out during phase 1) is still an option. In phase 3, Rolls-Royce was tasked with investigating combined cycles further, to identify the best PCS cycle option. Rolls-Royce's work in phase 3 has been broken down into three main sections. These are:

(1) Evaluate an indirect combined cycle PCS for a commercial plant.

(2) Compare this with a direct cycle PCS, and make a Rolls-Royce recommendation on the best option for a commercial plant.

(3) Given the Rolls-Royce cycle recommendation above, how should technology required for this option be demonstrated in the NGNP programme?

To perform this work, an integrated team was pulled together from all five parts of Rolls-Royce (Civil Aerospace, Defence Aerospace, Marine systems, Energy and the new Civil Nuclear group). The integrated team is shown in Figure 1.

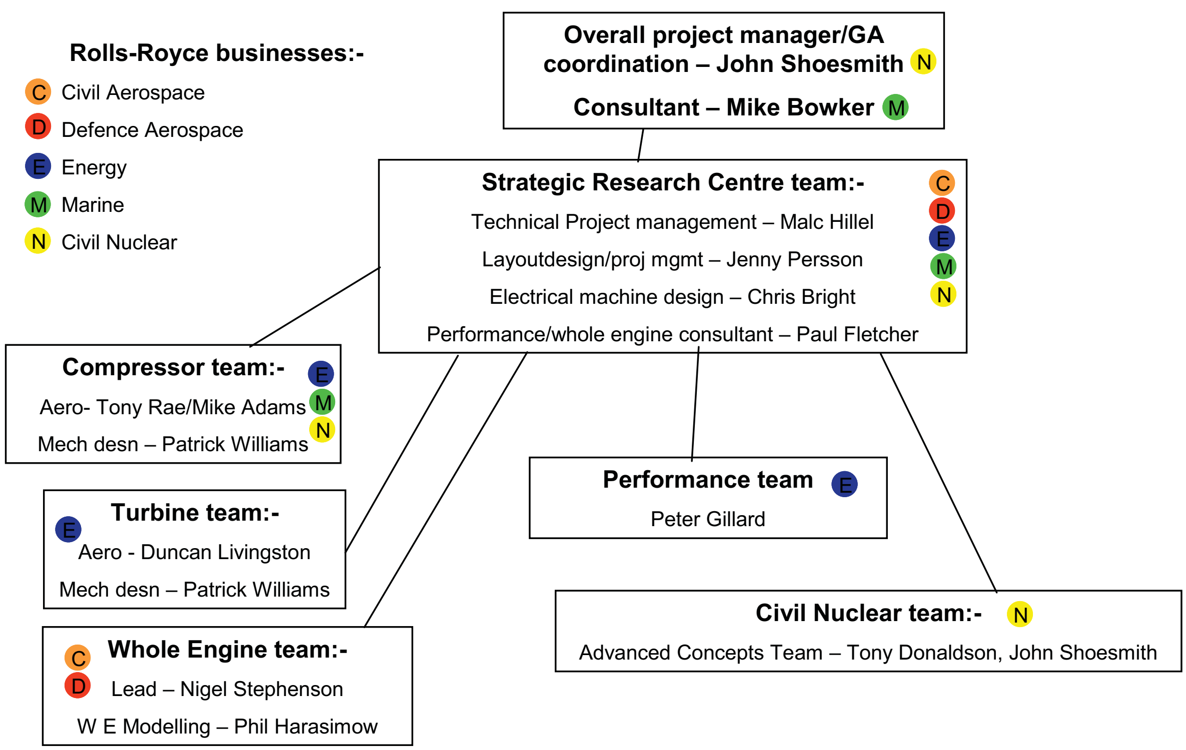


Figure 1 Rolls-Royce Integrated Team to Work on Power Conversion System

Rolls-Royce studies of an indirect combined cycle have concluded that there are very significant reductions in risk/cost for this arrangement compared to a direct combined cycle:-

(1) Conventional bearings can be used – no Active Electromagnetic Bearing risk.

(2) Working fluid in gas turbine can be more 'air-like' – reduced risks of the unknowns associated with Helium aerodynamics and leaks.

(3) Compressor looks like a conventional aeroengine compressor – only 6 stages compared with 18 for Helium direct combined cycle. Turbine is only 2 stage compared with 5 for Helium direct combined cycle. This would allow large cost savings in the turbomachinery compared with a direct combined cycle.

(4) Gearbox can be used instead of power electronics for gas turbine generator providing a significant cost saving.

(5) Turbine blade cooling is not likely to be required, even at the elevated 950°C reactor outlet temperature.

(6) Turbomachinery maintenance work much easier and cheaper with indirect cycle. Turbomachinery does not get contaminated with radioactivity. There is an opportunity to replace turbine blades that are creep life – expired when the reactor is refuelled every 18 months.

Balanced against these significant benefits in cost and risk for an indirect combined cycle, a large intermediate heat exchanger would need to be introduced which brings large associated risks and costs. The Rolls-Royce studies compared the costs and risks between direct and indirect and concluded that, although the direct cycle was predicted to be more efficient (50.2% versus 48.6% for the indirect at 850°C reactor outlet temperature), an indirect combined cycle would be a cheaper and lower risk option for a commercial electricity plant.

Furthermore, in comparing the indirect combined cycle plant with a pure steam cycle, the addition of an intermediate heat exchanger (IHX) and a gas turbine provides excellent value for money increasing the cycle efficiency from 42.6%. to 48.6% which is a massive benefit in terms of both plant capacity and fuel/waste processing costs.

The Rolls-Royce study therefore recommends an indirect combined cycle as the best choice for a medium term commercial electricity plant.

In the longer term, a direct combined cycle may merit reconsideration because it is slightly more efficient and this effectively translates as more electrical capacity for a given reactor. For the benefits of this higher efficiency to be realised, the turbomachinery (running on pure Helium as a working fluid and utilising advanced active magnetic bearing technology) would have to be absolutely, dependably reliable. In a direct cycle, the pure gas turbine option whilst not quite as efficient as a combined cycle has the lowest footprint and is the most elegant. This option has an especially challenging active magnetic bearing and also requires a large robust recuperator with a delicate internal structure.

2 Performance (choice of cycle and off design)

The proposed indirect combined cycle and its operation at off-design conditions are discussed in the following sub-sections. Section 2.1 describes the development of the indirect cycle and Section 2.2 suggests how the chosen cycle will perform during operation at off-design conditions.

2.1 Choice of Performance Cycle

2.1.1 *Previous Work*

Two cycles have previously been developed as potential solutions for the power conversion unit of the current reactor design. Both were direct cycles, hence included a gas turbine in the primary circuit. The earlier of the two designs, the GT-MHR cycle, was an intercooled and recuperated closed Brayton cycle with helium as the working fluid. The minimum helium pressure in the cycle was 25 times atmospheric with an overall pressure ratio of 2.8. Power was generated from a gas turbine (~300MW) in the primary circuit. The cycle had a high net electrical efficiency of **48%** at a reactor outlet temperature of 850°C.

In 2007, a second cycle was developed by Rolls-Royce. This was done in an attempt to further improve the efficiency and to mitigate some of the risks perceived in the GT-MHR design. The cycle proposed was a direct combined cycle. The reactor outlet temperature was 850°C and to reduce the temperature to a level acceptable to the steam equipment, a gas turbine was installed in the primary circuit. This gas turbine had an overall pressure ratio of 1.87 giving a steam turbine inlet temperature of 580°C. The gas turbine power was ~50MW and the steam turbine power ~250MW. The predicted net electrical efficiency was **50.2%**.

2.1.2 *Development of the Indirect Cycle*

Following from the work described above, the current project is to develop an indirect cycle of optimum efficiency. An indirect cycle is achieved by introducing an intermediate heat exchanger (IHX) and moving the turbo-machinery into a secondary circuit. The coolant in the primary circuit is now driven by a motor driven circulator.

The move to an indirect cycle provides the flexibility to select a more appropriate working fluid for the gas turbine circuit. To the first order the choice of gas in the gas turbine circuit does not affect efficiency from a cycle viewpoint. There are second order effects, such as the pressure drop in the heat exchangers, which for a heavier gas, with a lower specific heat capacity will be higher. However, the choice of the working fluid is dominated by turbo-machinery considerations and as described in Section 3, the most appropriate working fluid was decided to be a mix of nitrogen and helium in an 80/20 proportion by mass. The primary coolant remains helium.

In the development of an optimum cycle, several different power conversion unit designs were considered. These included an indirect Brayton cycle, an indirect pure steam cycle and several indirect combined cycles; one of which was an indirect version of the 2007 direct combined cycle. The potential of each cycle was evaluated and the more promising designs modelled using the performance package IPSEpro. From this down-selection process, it was decided that efforts should be concentrated on the development of an indirect version of the 2007 direct combined cycle.

During the development process, the efficiency was seen to be highly sensitive to the temperature difference at the ‘pinch point’ in the steam generator economiser. From an assessment of the steam generator in 2007, 22°C was determined to be an achievable temperature difference with a reasonably compact steam generator. It will therefore be important for the 22°C economiser temperature difference to be achieved in order for the target efficiencies to be reached in the operational plant.

The introduction of the IHX also has an effect on the cycle efficiency. This must be considered in the IHX design since for the same type of heat exchanger, the size is directly related to the temperature and pressure drops across it. A larger unit should provide smaller temperature and pressure drops and a more efficient cycle for a fixed reactor outlet temperature. Of course, a larger unit will be more expensive, especially considering that the IHX must be located within the containment building. A 50°C temperature drop was considered a reasonable cycle assumption offering a compromise between efficiency and IHX size and cost.

The indirect combined cycle enables the primary and secondary circuit mass flow rates to be controlled independently. It was therefore possible to optimise the design points for reactor outlet temperatures of 850°C, 900°C and 950°C by varying the primary circuit mass flow rate. For each case, the reactor temperature difference was restrained to remain in the range 360°C - 460°C. It was found that the optimum efficiency is achieved where the reactor temperature difference is close to the minimum allowable.

2.1.3 *Description of the Optimised Indirect Combined Cycle*

The description of the proposed indirect combined cycle presented here is based on a reactor outlet temperature of 850°C, which provides a net electrical efficiency of **48.6%**. 900°C and 950°C reactor outlet temperatures were also modelled; the associated efficiencies for the direct and indirect cycles are shown in Table 1.

The cycle is similar to the direct combined cycle described in the phase 1 report[1], but includes an IHX as described in section 6. The cycle operates with a minimum pressure of 36.6 bar and an overall pressure ratio of 1.85. For a fixed reactor outlet temperature, the indirect cycle has a 50°C lower gas turbine inlet temperature compared to the direct cycle. Therefore, in order to maintain the steam turbine inlet temperature at 580°C, the work done and power generated in the gas turbine is less than for the direct cycle. However, the steam circuit mass flow rate is increased slightly compared to the direct equivalent because of the heat introduced to the primary circuit by the circulator. This serves to slightly increase the proportion of work done in the steam circuit, which takes more advantage of the phase change in water.

Controlling the primary and secondary circuit mass flow rates independently enables a higher reactor inlet temperature to be achieved compared to the direct cycle. For the indirect cycle an inlet temperature of 490°C can be achieved (compared to 437°C for the direct cycle). The mass flow rate through the gas turbine is larger than for the direct cycle due to the lower specific heat capacity of the nitrogen/helium mix compared to helium. This increases the pressure drop in the heat exchangers compared to the direct cycle to 1.8 bar; up from 0.6 bar.

Reactor Outlet (°C)	Direct (%)	GT Inlet (°C)	Indirect (%)	Indirect Cycle Penalty (%pts)
850	**50.2**	800	**48.6**	1.6
900	**51.4**	850	**49.3**	2.1
950	**52.4**	900	**49.8**	2.6

Table 1 Direct combined and indirect combined cycle efficiencies for a range of reactor outlet temperatures. Direct combined cycles values quoted from DNS133534 v1.0.

[1] Preliminary Assessment of the GT-MHR Power Conversion System, Rolls Royce, DNS133534 v1.0, 10 July 2007

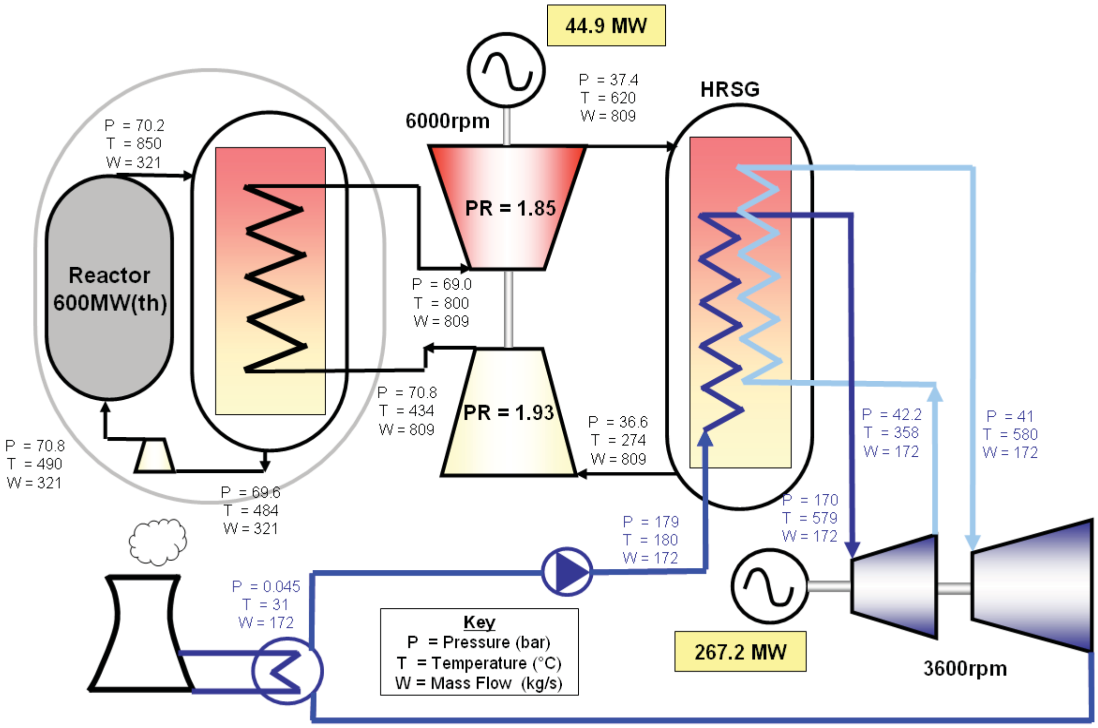


Figure 2 Diagram of the proposed indirect combined cycle with optimised operating conditions shown for 850°C reactor outlet temperature.

2.1.4 *Evaluation of the Indirect Combined Cycle compared to Direct Combined Cycle*

For the 850°C cycle, the 1.6%pts reduction in the efficiency of the indirect cycle compared to the direct cycle can be attributed to the following factors:

- Disadvantage of a reduced gas turbine inlet temperature: -0.7%pts
- Disadvantage of Increased pressure drop in heat exchangers: -0.6%pts
- Disadvantage of the power consumption of the primary circuit circulator: -0.83%pts
- Advantage of the optimisation of the primary circuit mass flow rate: +0.53 %pts

Overall, the total reduction in efficiency is smaller than may have been expected. The expected losses due to the above items are offset in part by the advantage gained in being able to control the primary and secondary circuit mass flow rates independently.

2.2 Off-Design Performance

2.2.1 *Introduction*

The IPSEpro tool used for modelling the design point of the cycle is incapable of modelling the cycle at off design conditions. Nevertheless, it is possible to predict the likely performance of the indirect combined cycle at off-design conditions by drawing on information from two sources. The first of these is the documentation on off-design performance of the GT-MHR cycle that was supplied during the 2007 Rolls-Royce study. The second source is the Rolls-Royce Aero-engine Performance synthesis program (RRAP) modelling work that was done in 2007 on both the GT-MHR and direct combined cycles. The indirect combined cycle has enough in common with these other cycles that useful conclusions can be drawn and issues that require further attention can be highlighted.

2.2.2 *Review of 2007 work on GT-MHR and Direct Combined Cycles*

The 2007 modelling of the GT-MHR and direct combined cycles drew the following key conclusions:

- Performance of the GT-MHR cycle on a hot day was compromised as it appeared that, in addition to the normal 'carnot' effects, the reactor power would have to be reduced. Similar effects were also observed if it were necessary to derate the reactor to run at reduced outlet temperature.

- In comparison, the direct combined cycle appeared to offer better hot day performance. This is because the steam mass flow can be varied to control the helium temperature at compressor entry and hence the reactor is 'unaware' of the ambient temperature change. It was speculated that this extra control variable would also provide benefits at other off-design conditions.

- The GT-MHR cycle bypass valve would need to pass around 45% of the main flow in order to reduce the net power on the gas turbine shaft (i.e. the power developed in the turbine less the power absorbed in the compressor) to zero. This action would be required during the start sequence and also in the event of the power station being dropped by the grid.

- The starting sequence proposed in the supplied GT-MHR documentation appeared to be feasible and there were no threats to compressor stability at part load.

2.2.3 *Indirect Combined Cycle at Part Load*

It is suggested that inventory control should be used as the main strategy for running at part load. This has the advantage of maintaining turbo-machinery non-dimensional operating points and cycle temperatures and thus keeps cycle efficiency high. To both minimise the pressure differential across the IHX and maintain the required relationship between reactor power and reactor temperature rise, the inventory would also need to be reduced in parallel in the reactor circuit.

It is believed that it will be essential to have control over the gas turbine pressure ratio. This is so both the net power on the gas turbine shaft and the gas temperature into the top of the boiler can be controlled. Since the gas turbine is to be connected to the main steam turbine/generator shaft (through a clutch) the gas turbine shaft will, when the machine is synchronised, run at constant speed. Controlling pressure ratio across a compressor running at constant speed requires the use of either variable geometry in the compressor or a bypass valve (similar to that proposed in the GT-MHR cycle). It is considered very unlikely that variable compressor geometry alone would offer the required control. It is therefore concluded that a bypass valve, between compressor outlet and turbine outlet, will be required.

By extrapolation from the 2007 RRAP work on the GT-MHR cycle it is clear that the bypass valve would need to pass a substantial flow. It would need to be of sufficient size that the net power on the gas turbine shaft could be reduced to zero during the startup sequence and in the event of the power station being dropped by the grid. The pressure ratio of the indirect combined cycle is lower than that of GT-MHR and this means the bypass flow would probably need to be in excess of 45%

of the 809kg/s secondary circuit flow. This is because, at lower pressure ratios, the ratio of compressor power to turbine power is lower. Therefore the net power is a large fraction of the turbine power and the turbine power thus needs to be sharply reduced to reduce the net power to zero. The practicalities of the bypass valve passing such a large flow, whilst also being able to exert fine control over the speed of the shaft when it is not synchronised, will need careful consideration.

2.2.4 *Indirect Combined Cycle on Hot Day/Cold Day*

On a hot day the steam flow can be varied and this means that the compressor inlet temperature can be controlled. This in turn means that the primary and secondary circuits would be unaffected by day temperature and there would be no need to reduce reactor power on a hot day. Although the net output and efficiency of the cycle would be reduced on a hot day, due to the increased condenser back-pressure, it is expected that reductions would be in line with those of conventional fossil fired combined cycle plant. Similarly, on a cold day, improvements in efficiency and net electrical output would be expected to be broadly in line with conventional combined cycle plant.

2.2.5 *Indirect Combined Cycle - Transients*

The most onerous transient is loss of grid. In order to prevent gas turbine/steam turbine shaft over-speed the net torque on the shaft will need to be dropped to zero very rapidly. For the steam turbine this can be done in the conventional way by slamming a stop valve closed and dumping steam. For the gas turbine the bypass valve will need to be fully opened rapidly. This places a further design requirement on the bypass valve in addition to those noted above.

For other transients a combination of inventory control and bypass flow can be used to vary power output whilst maintaining compressor stability. An advantage of the indirect combined cycle over the direct combined cycle is that the IHX acts as a ‘damper’ between the reactor and gas turbine circuits. This means that the reactor is somewhat isolated from gas turbine transients and also, the gas turbine is isolated from reactor transients. This should make control of transients, and especially fault transients, easier to manage.

2.2.6 *Indirect Combined Cycle - Starting*

A starting sequence has been proposed based on RRAP modelling of the GT-MHR starting sequence. In general, the indirect combined cycle has more variables which can be controlled than was the case for the GT-MHR cycle. For example the reactor, gas turbine and steam circuit flows can all be independently varied. It is concluded that the availability of extra control variables must make starting easier than for GT-MHR. Since no issues were identified which would make starting the GT-MHR difficult, it is further concluded that starting the indirect combined cycle is unlikely to present insurmountable problems.

2.3 Technology Demonstration in NGNP Programme

The exact configuration of the NGNP demonstrator plant is yet to be fully defined but it is believed that the plant is likely to be configured to produce process heat and steam. It is clear, however, that a commercial plant for generating electricity would almost certainly have a gas turbine included to maximise plant efficiency. It would therefore be desirable if the gas turbine technology could be demonstrated in some way in the demonstrator plant.

This section seeks to answer the following questions:

1. What would be the efficiency of a pure steam turbine plant with no gas turbine?

2. What is the relationship between reactor outlet temperature and efficiency for an indirect combined cycle plant. Down to what reactor outlet temperature is it worthwhile or feasible to include a gas turbine in the plant?

3. How could a gas turbine be included into the NGNP demonstrator plant

2.3.1 *Efficiency of a Pure Steam Turbine Plant*

The current state of the art for steam turbines is for steam inlet temperatures of 600°C, the limiting factor being the properties of the materials used in the thick-walled headers, pipes, valves and rotors. The indirect combined cycle presented above assumes a slightly conservative steam inlet temperature of 580°C, with a maximum steam pressure of 180bar. The cycle is reheated, at 40bar, also to 580°C.

When a pure steam cycle is modelled in the IPSEpro tool at the same steam conditions a net electrical efficiency of 42.6% is predicted. The efficiency of a steam cycle is very dependent on the condenser pressure. This in turn depends on how the condenser is cooled and the temperature of the air or water used to cool it. The steam cycle model was developed during the 2007 study and the aim was to compare against the Brayton cycle concept on a basis that would not unfairly favour either. The Brayton cycle assumed that cooling water was available which could cool the helium in the precooler and intercooler to 26°C. Achieving this would probably require a coastal location in a fairly cool climate. To give a fair comparison a condenser pressure of 45mbar, achievable in a similarly cool coastal location, was specified in the steam cycle.

It is of interest to compare this efficiency prediction with modern fossil-fired steam plants. A literature search has shown that state of the art steam plants can have net electrical efficiencies well in excess of 40%. Denmark, in particular, has a number of very high performing plants which benefit from coastal locations and low sea temperatures which allow very low condenser pressures (down to 23mbar). A net electrical efficiency of 47% is claimed for the Nordjylland 3 plant with a double reheat steam cycle operating at 'ultra-supercritical' steam conditions of 580°C and 290bar. There are a number of other examples around the world where net electrical efficiencies well in excess of 40% are claimed. It is therefore concluded that the steam cycle efficiency of 42.6% modelled here is reasonable.

When comparing the efficiencies of the various cycles the relative values are more important than the absolutes. It is explained above why the steam cycle is considered to be modelled on a fair basis against the original intercooled and recuperated closed Brayton cycle. When modelling steam only and combined cycles care has been taken to ensure that the steam parts of the cycle are consistent, i.e they produce the same power output per unit heat input. Therefore, although the efficiency of the steam plant would vary depending on climate and how the condenser is cooled, it is expected that the relative efficiencies of steam only and combined cycle plants would stay constant.

In the IHX options study report[2] a steam inlet temperature of 540°C was assumed. IPSEpro modelling suggests that, at a fixed steam pressure, this would penalise the net electrical efficiency by around 0.3%pts compared to a 580°C inlet temperature.

[2] Engineering Services for the Next Generation Nuclear Plant (NGNP) with Hydrogen Production, NGNP IHX and Secondary Heat Transport Loop Alternatives Study, General Atomics, 911119/0, 2008/04/23

2.3.2 *Reactor Outlet Temperature and Value of Gas Turbine*

The reactor is capable of high outlet temperatures but these cannot translate into high electrical efficiency if only a steam turbine is used for power conversion. It is of interest to understand the relationship between reactor outlet temperature and cycle efficiency so that the point at which it becomes economically worthwhile to add a gas turbine can be judged.

The relationship between net electrical efficiency and reactor outlet temperature is given in the figure below, In all cases the steam inlet temperature is 580°C.

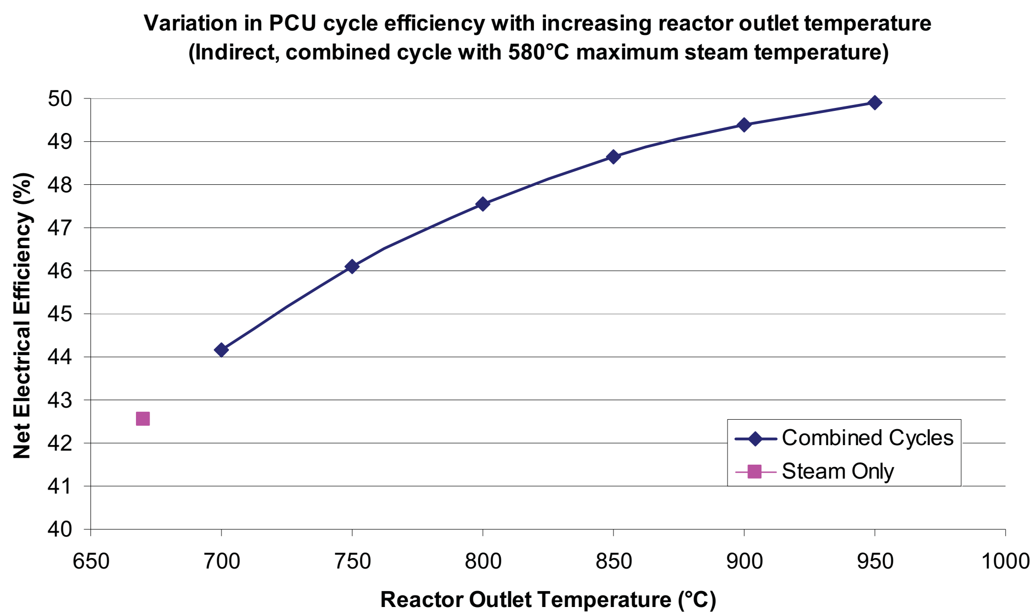


Figure 3 Cycle Efficiency vs Reactor Outlet Temperature for Both Steam Only and Combined Cycles

It is clear that the combined cycle curve and the steam only cycle point form a continuous family. Including a gas turbine in the cycle can be seen to be worth around 3.5%pts of efficiency at a reactor outlet temperature of 750°C, 5%pts at 800°C and 6%pts at 850°C.

2.3.3 *Feasibility of Including a Gas Turbine in the NGNP Demonstrator Plant*

The IHX options study report by GA presents two possible layouts for the NGNP demonstrator plant, the serial HTS (Heat Transport System) configuration and the parallel primary loop configuration. Both of these assume the inclusion of hydrogen production plant and therefore have high reactor outlet temperature (900°C). Both cycles assume the inlet temperature to the steam plant is 540°C.

A plant with a gas turbine inserted would ideally have the following characteristics:

- The gas turbine would operate at conditions (i.e temperature, pressure, pressure ratio) consistent with those foreseen for the commercial plant.

- The gas turbine would operate with the helium/nitrogen mix proposed for commercial plant

- If the gas turbine were not available, this should not prevent the running of the demonstrator plant in other modes.

- The plant should be designed so that a gas turbine is not required at the outset but could be fitted part way through the demonstrator programme.

- The plant should be designed so the gas turbine could be inserted at minimum additional cost.

These requirements have been reviewed against a number of possible layouts for the demonstrator plant which include a gas turbine. The most promising of these is based on the parallel primary loop configuration and is described in the sub-section below.

2.3.4 *Proposal for Including a Gas Turbine in the NGNP Demonstrator Plant*

The parallel primary loop configuration is shown in Figure 4. There are two primary loops, one which provides 65MW of heat to a hydrogen plant and the second which provides steam. The second of the primary loops exchanges heat through an intermediate heat exchanger to a helium secondary circuit. The intermediate heat exchanger drops 200°C, and a further 160°C is dropped across the steam generator in order to give 540°C steam at steam turbine inlet. Helium is circulated at 70bar in the secondary circuit by an 11MW circulator.

It is proposed to make the following changes to allow a gas turbine to be included:

1. To replace the helium in the secondary circuit with a helium/nitrogen mix, so that when the gas turbine is included, it can run on the correct working fluid.

2. To insert a larger/more efficient intermediate heat exchanger between the primary and secondary circuits to reduce the temperature drop across it. This opens up a temperature difference between which the gas turbine can operate.

3. To insert a larger/more efficient steam generator. This again opens up the temperature difference between which the gas turbine can operate.

4. To insert a cooler in the secondary circuit between the intermediate heat exchanger and the steam generator. The purpose of this is to reduce the temperature at steam generator inlet to a satisfactory level in the event that the gas turbine is not operational.

The gas turbine can then by installed by bypassing the cooler on the hot side with the turbine and bypassing the circulator on the cold side with the compressor. The layout is shown in Figure 5. The layout and conditions when the gas turbine is operational are shown as solid lines. The dashed lines and boxes show the layout and conditions when the gas turbine is not operational.

This layout would cost more than the existing parallel primary loop configuration even without the cost of the gas turbine and its ducting. This is because the heat exchanger and steam generator would need to be larger (to give smaller temperature differences and because the working fluid is no longer pure helium), and the cooler is an additional component. The heat exchangers would also need to be carefully designed so that they could operate satisfactorily in both operating regimes.

The pressure on the cold side of the intermediate heat exchanger would be 70bar when the gas turbine was in operation. This means that the pressure differential introduced by the gas turbine around the secondary circuit does not affect the intermediate heat exchanger. When the gas turbine is not operational, the whole secondary circuit can be pressurised to 70bar. When the gas turbine is operational the pressure at the steam generator will of course be lower (around 37bar) but it is not unusual for steam generators to operate at pressure differentials such as this.

The proposed layout is very flexible because it allows the plant to operate either with or without the gas turbine in place. The addition of the cooler allows the temperature at steam generator inlet to be controlled, which means that the steam inlet temperature can be controlled independently of the reactor outlet temperature. This gives extra flexibility when commissioning the plant and would allow it to be tested over a wider range of conditions.

The layout also allows the gas turbine plant to be tested at exactly the conditions envisaged in a commercial plant, with temperatures and pressures being reproduced correctly. The inclusion of the bypass valve allows that component to be properly tested and fault conditions, such as dropped load, could be simulated.

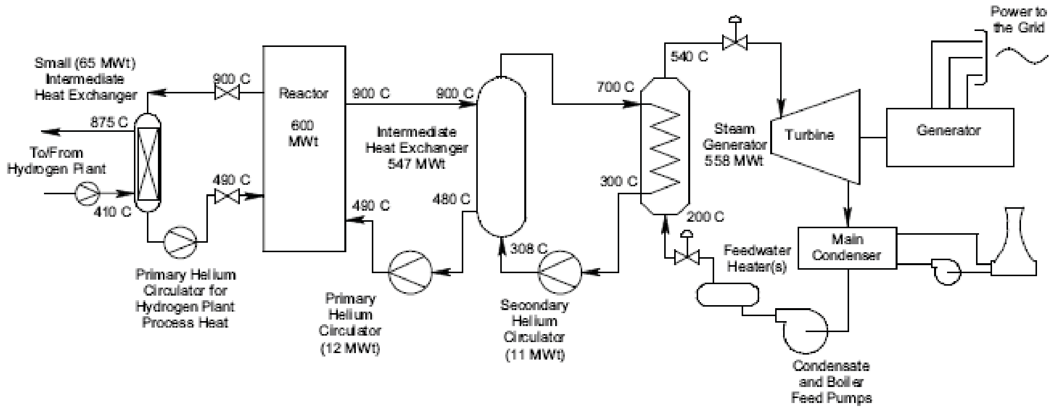


Figure 4 Parallel Primary Loop Configuration from HXoptionsStudy report 911119/0

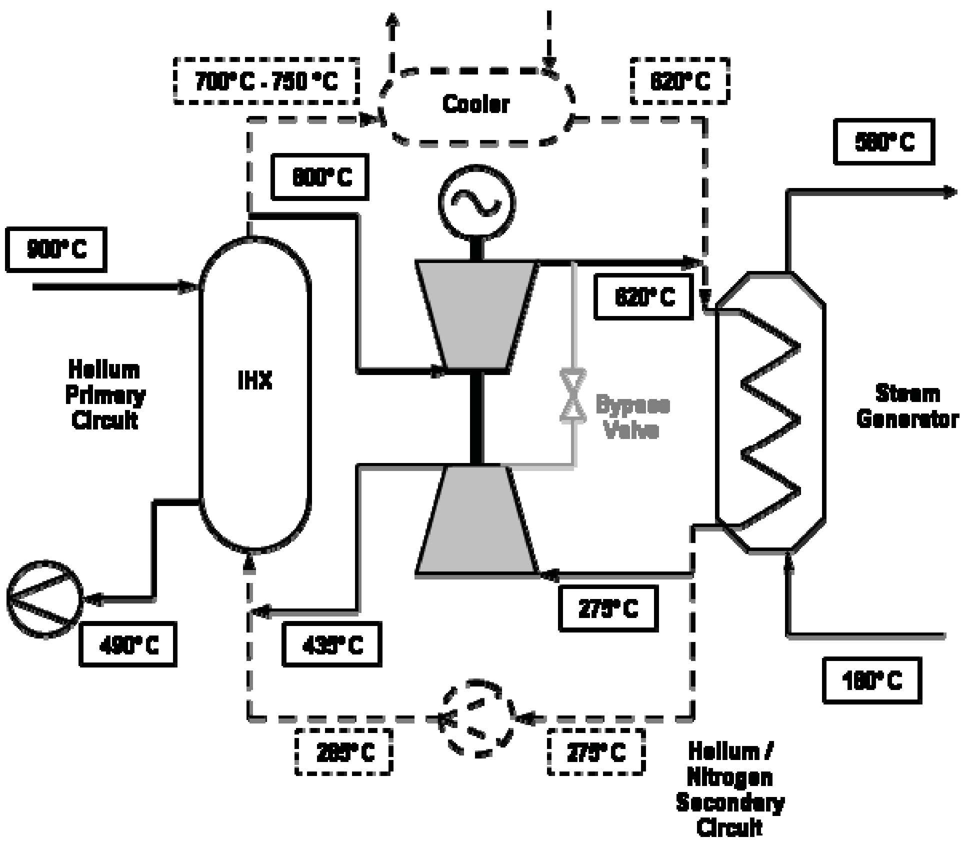


Figure 5 Proposal for Inclusion of Gas Turbine in NGNP Demonstrator Plant

3 Secondary working fluid

A number of pure gases and gas mixtures have been assessed as options for the secondary working fluid. It can be relatively simply shown that the “ideal” gas requirements for the turbo-machinery and intermediate heat exchanger are contradictory; compact and efficient turbo-machinery requires a higher density gas for lower volumetric flow rate while a heat exchanger requires a gas with good thermal conductivity. Issues such as corrosion, oxidation and nitriding can also have a significant impact on gas selection. Preliminary investigations into a number of pure gases suggested that most would result in complex turbomachinery or an excessively large IHX.

3.1 Choice of Working Fluid

A study was conducted into the choice of secondary working Fluid. This study involved looking at the size and nature of the turbine for each of the following fluids. From a turbomachinery point of view the relative merits of each are briefly summarised.

- Helium
  A helium turbine is large, has many stages and a low expansion through the turbine. The development of a helium turbine has additional costs due to the lack of experience of designing turbines for noble gases.

- Argon
  Argon is similar thermodynamically to helium, but more dense. It would result in a much smaller turbine with one third the number of stages. It is widely available as an industrial gas. The development of an argon turbine would be similarly expensive.

- Xenon
  Xenon is also similar to helium and would result in an even smaller turbine. The development of a xenon turbine would be similarly expensive. Furthermore, xenon may have disadvantages due to the possibility of poisoning the reactor in the event of a leak.

- Carbon Dioxide
  This is widely used in current reactor designsas a primary coolant, however there is little experience in gas turbine turbomachinery. It would result in a large turbine with a high expansion ratio and would be difficult to design.

- Air / Nitrogen / Nitrogen-Helium Mixtures
  An air turbine is roughly half the size and half the number of stages of a helium turbine. There is a wealth of experience in designing air turbines and therefore it would be a much lower risk option. Adding helium to the system has not got any advantages to the turbo-machinery but has significant advantages elsewhere in the secondary system.

As a result of this study a mixture of Helium and Nitrogen by mass was selected as the secondary working fluid. From a turbomachinery perspective a mixture of nitrogen and helium behaves in exactly the same way thermodynamically as pure nitrogen, however an overall system benefit was found to be achievable by introducing some helium. This benefit arises because of the contradictory nature of the dependence of the intermediate heat exchanger and turbo-machinery components on gas properties.

Selection of an approximate 80/20 by mass nitrogen/helium gas mixture is recommended on the basis of balancing risk in turbo-machinery and the intermediate heat exchanger and reducing cost. Introduction of a small mass inventory of helium results in turbo-machinery key parameters similar to existing Rolls-Royce aero and energy products, but significantly reduces the intermediate heat exchanger size. Further discussion is detailed in sections 4 and 6. It should be recognised that the exact specification of the working fluid will vary depending on the final detailed cycle and component design, however the completed analysis indicates that a gas composed of 20-30% helium by mass offers significant benefits.

3.2 Gas mixture properties

Detailed consideration has been given to gas mixtures of helium and nitrogen. Some academic literature identifies an inconsistently large improvement (relative to the helium inventory) from the addition of helium into other gases. Data from selected academic papers has been analysed to determine underlying non-dimensional heat transfer functions, and it has been demonstrated that the perceived performance improvements result from systematic error in the experimental method. Independent models of helium/nitrogen gases based on approximation methods of Wilke and Wassiljewa at various helium inventories were produced to estimate transport properties, and the effect on both turbo-machinery and intermediate heat exchanger assessed.

An overall system benefit was found to be achievable by introducing approximately 20% helium by mass. This benefit arises because of the contradictory nature of the dependence of the intermediate heat exchanger and turbo-machinery components. Mass and density dependant properties of the gas rise linearly with the mass inventory of helium in the gas mixture. Conversely, expansion ratio (γ) and thermal conductivity are dominated by gas kinetics and hence rise linearly with the helium molar fraction.

Mass and density effects primarily affect the turbo-machinery stage count and achievable velocity ratios, hence a linear increase in turbo-machinery stages is observed relative to the helium mass inventory. Compressor annulus area is governed by expansion ratio hence rises linearly with helium molar fraction. This increase in area can be effectively managed to maintain blade aspect ratios by increasing compressor mean line. Figure 6 demonstrates the dependence of compressor stage count on gravimetric helium inventory.

Heat exchanger sizing is affected by both volume flow rate and thermal conductivity, however it is apparent that thermal conductivity dominates within the examined working range. Intermediate working fluids are exchanged on the basis of constant fluid heat capacity (mc_p). Figure 7 illustrates the normalised reduction in heat exchanger size from the introduction of helium to the working fluid.

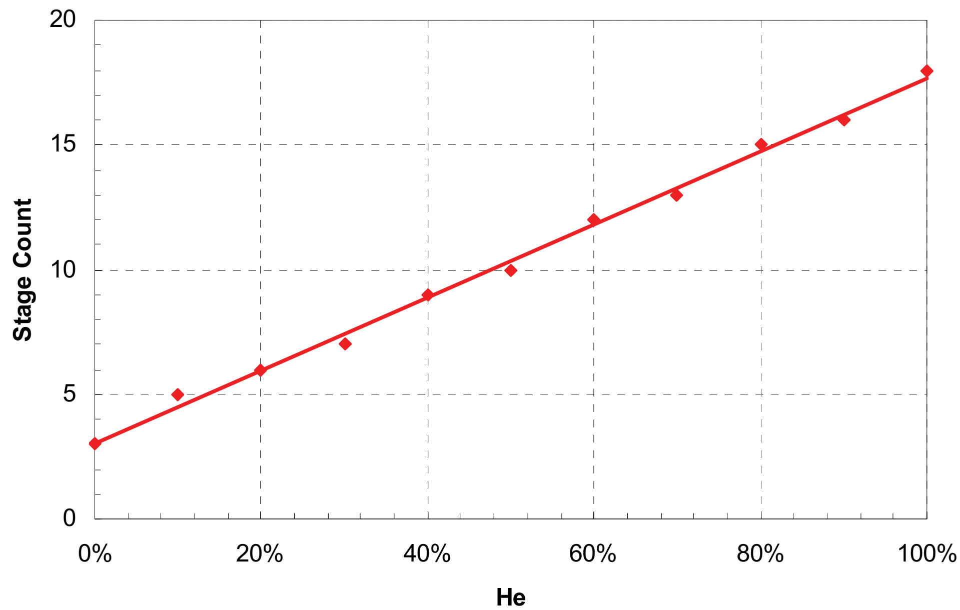


Figure 6 **Compressor stage-count relative to gravimetric helium inventory.**

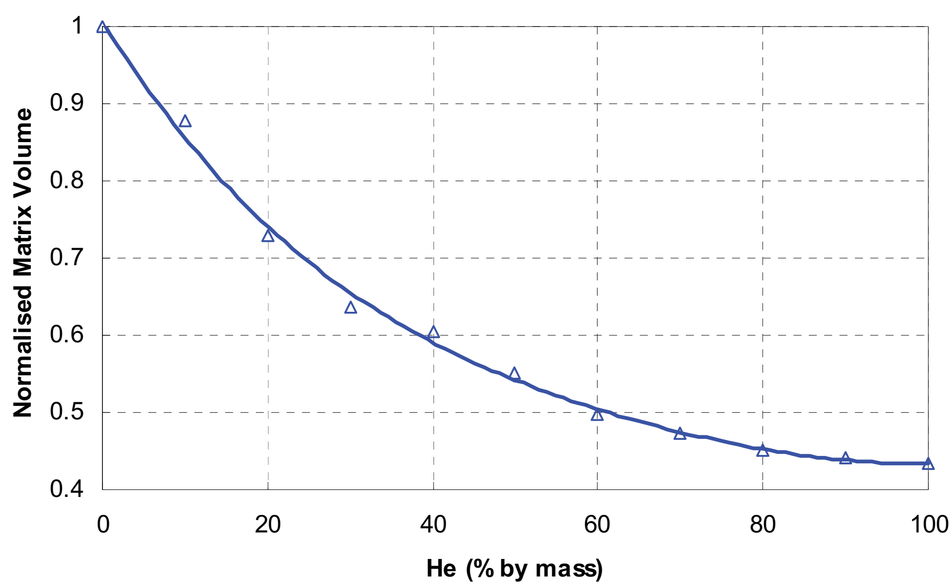


Figure 7 **Normalised matrix volume relative to gravimetric helium inventory.**

4 Compressor and Turbine Aerodynamic design

The compressor and turbine aerodynamic design are discussed in this chapter. Section 4.1 describes the compressor aero design and Section 4.2 the turbine aero design. Following chapter will then discuss the mechanical design aspects. Choosing a turbomachinery working fluid that is much more similar to air than the Helium required for a direct cycle gives turbo machinery designs that are very similar to Rolls-Royce gas turbine experience. The design task is also much less of a stretch for the well established aerodynamic design tools that Rolls-Royce has. The turbomachinery risks are therefore significantly reduced with the indirect combined cycle.

4.1 Compressor Aerodynamic design

The philosophy of the aerodynamic design of the compressor is to base the NGNP Phase 3 design on an existing design that has been developed for an aero engine application; for this study we have chosen the 6 stage Trent aeroengine family style High Pressure Compressor. Using an existing and well proven design will significantly reduce risks.

A conventional Trent compressor uses air as the operating fluid. The change from air to Helium/Nitrogen mixture changes the gas properties of operating fluid which will result in the compressor having less overall compressor pressure ratio (CPR) capability, this though is not a problem as the cycle requires a lower CPR so fits well with this design concept. To have a compressor that is strongly based on an existing design you have to maintain the basic aerodynamic properties. These are individual stage enthalpy rise divided by the square of stage mean blade speed ($\Delta H/U^2$) and the stage inlet axial velocity divided by the stage mean blade speed (Va/U). By maintaining these parameters the blade mean radius geometries will be very similar to the original compressor but will be operating at lower Mach numbers as a result of the speed of sound of the mixed gas being higher than in air.

Once the above design is completed the compressor is scaled such that it will operate at the required inlet non-dimensional flow parameter, massflow times the square of the temperature divided by the total pressure ($M\sqrt{T}/P$) and the revolutions per minute (RPM) is changed to maintain the original hub line blade speed (Uhub). A further correction is made to achieve the required design speed of 6000RPM. This is achieved by moving the hub line of the compressor to a new position such that Uhub is maintained. With this new hub line the casing line is moved to maintain the flow areas through the compressor.

	Trent Compressor	Mixed Gas Compressor
Mean $\Delta H/U^2$	0.443	0.443
Mean Va/U	0.6	0.596
Mean Rotor deflection	19.4	19.4
Mean Stator deflection	32.1	32.1
Mean axial Mach number	0.42	0.275
Inlet hub/tip radius	0.813	0.806
Exit hub/tip radius	0.921	0.851
Mean Aspect Ratio	0.6	0.888

Table 2 Comparison of compressor parameters between a Trent compressor and suggested He/N mixture compressor

The table above illustrates that the above process does give a compressor that is very similar to a Trent compressor. This can be seen by the mean aerodynamic parameters and blade deflections being the same. The mean axial Mach numbers are lower for the mixed gas design, because of the gas properties being different; this will give a slight efficiency advantage. The different gas properties and operating point between the two compressors does result in a lower exit hub/tip radius ratio and high mean blade aspect ratios[3]; these again will give an efficiency advantage.

In summary, because this compressor is so closely related to a well developed gas turbine compressor, there would be very high confidence of a 'right first time' design. Indeed rig validation testing may well not be required to be certain of achieving design point efficiency and surge margin. This would save significant technology development costs compared with a helium designed direct cycle gas turbine.

4.2 Turbine Aerodynamic design

4.2.1 *Previous Helium Turbine Design for the direct cycle*

Rolls-Royce have previously studied turbomachinery options for a helium cycle gas turbine. This included an evaluation of the OKBM GT-MHR direct cycle concept, a simpler more efficient Rolls-Royce design for the GT-MHR and a Rolls-Royce combined cycle concept. To understand the benefits of the current design for a Nitrogen-Helium turbine it is important to understand the design drivers behind the previous work.

[3] Ratio of the height of the blade to the blade chord where the chord is the length of the blade between the leading edge and trailing edge

Helium has a very high specific heat capacity and a high ratio of specific heats (gamma). Together, these properties mean that the speed of sound in helium is very high. However, it has a very low molecular weight and hence is not very dense. This choice of working fluid therefore drives the design in the following ways.

- As the helium is not dense, a large flow area is required and the axial velocity of the helium is very high.
- As the velocity of the helium is high, the blade speed needs to be high. This pushes the design towards a fast shaft speed and a high diameter turbine.
- The high gamma means that a large amount of work can be extracted from the fluid with like density change. This means there is a small change in flow area through the turbine.
- The low density gas means many blades are required for each stage and the shape of each blade is unconventional.
- The large, high speed turbine means the mechanical design of the blades and disks is challenging. This forces design choices such as blade cooling, thermal barrier coating and exotic materials.

Despite these challenges, a suitable design for the direct combined cycle was found to be a five stage turbine rotating at 5000rpm with a diameter of around 1.5m. Although the cycle design has changed, this turbine performs roughly the same power output as the current design. It is therefore useful for comparison with the Nitrogen-Helium design described in section 4.2.2.

4.2.2 *Turbine Design for indirect cycle*

A realistic, achievable concept design for the turbine has been completed. The design has the following features.

- To provide the smallest turbomachinery and to match the requirements of the compressor design, a high shaft speed of 6000rpm was selected.
- Two stages
- Diameter of 1.2m
- Shroudless blading is used. This keeps costs down and simplifies the mechanical design for a very small performance penalty.
- The outer hade line is parallel, giving good sealing and good tolerance to large axial movement.
- The vortex is optimised to keep the first stage cool and to help the exit diffuser.
- The aerodynamics are chosen to keep blade stress to a reasonable level. Hence no thermal barrier coating or blade cooling is required.
- The turbine is lightly loaded and an efficiency of 90% should be possible.
- The blade design is conventional and would look very similar to an aero engine.

Hence, in comparison to the helium turbine described in 4.2.1 above, the nitrogen-helium turbine is one eighth of the weight, with only 2 stages (compared to five), considerably simpler with no blade cooling and existing design tools are available. It is therefore a very favourable design.

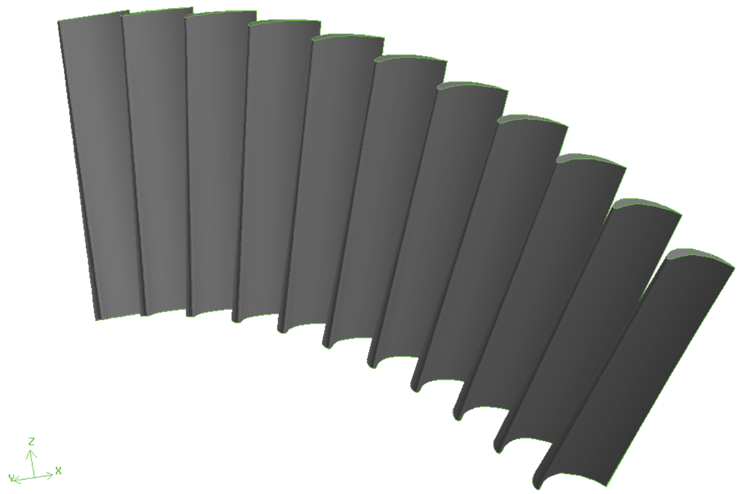


Figure 8 Suggested turbine blade design

In summary, similar to the compressor design the turbine design is closely related to a conventional well developed gas turbine, there would be very high confidence of a 'right first time' design. Indeed rig validation testing may well not be required to be certain of achieving design point efficiency and surge margin and again would save significant technology development costs compared with a helium designed direct cycle gas turbine.

5 Compressor and Turbine mechanical Design

The compressor and turbine mechanical design are discussed in this chapter. Section 5.1 describes the compressor mechanical design, section 5.2 the turbine mechanical design and section 5.3 discusses the selected compressor and turbine materials.

5.1 Compressor Mechanical Design

For the compressor mechanical design Rolls-Royce has completed an assessment of a six stage compressor operating at 6000RPM, see Figure 9. In both cases the design and stress analyses used gas paths from Rolls-Royce compressors which apply Trent style 'architecture'. The latter is the preferred compressor solution aerodynamically operating within a N_2/He gas closed circuit.

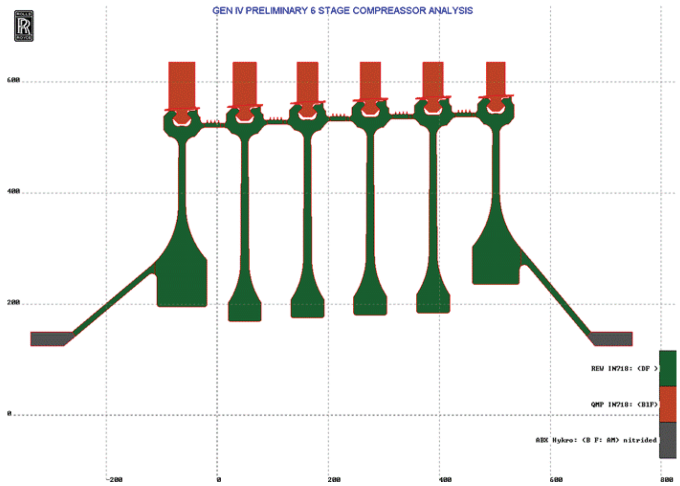


Figure 9 Six stage compressor operating at 6000RPM

The design of both the turbine and compressor is driven by the creep life requirements to achieve 18 months operation which allows repair and overhaul activity during the period when the Nuclear plant is being refuelled. This requires a life of 13,000 hours and an assumed minimal number of cycles. Therefore the objective with the compressor has been a life of 13,000 hours or a multiple (e.g 26,000 hours) to allow a modular maintenance regime.

For the compressor temperatures and Creep Factor (CF) loads a range of materials have been considered, also noting the cost implications. For the blades INCO 718 has been considered as the most suitable material, with discs either in INCO 718 or possibly Titanium IMI834 at the forward end. Titanium is a more expensive solution and would lead to a more complex bolted rotor with the added risk of difficulties in designing out the consequential thermal fight at the joint. Hence an all INCO solution would be preferable.

For the six stage compressor with INCO blades and discs, an initial Finite Element (FE) analysis has been completed. This has shown that suitable RF values for burst, rim peel and creep over the defined temperature gradients are achievable within the design concept. The preliminary rotor-dynamic analysis results indicate consideration is needed in the next design phase regarding the bounce and bending mode frequencies (from Campbell diagram analysis around 100Hz) created with the 6000RPM operation. This would have to be re-assessed during a refinement of the rotor support bearing stiffness in the design.

From the assessment of the I_P/I_D (Polar Inertia/Diametral Inertia) relationship the compressor ideally should be lengthened slightly. The addition of a balance piston for rotor thrust compensation would also help to improve the result.

Hence in summary the six stage compressor concept is mechanically viable to suit this application.

5.2 Turbine Mechanical design

The aerodynamic design for the Turbine favours a two stage solution at 6000RPM, see Figure 10. A mechanical study has been completed with Turbine inlet temperatures from 750°C to 850°C, with an expected load of 226MN and a life of 13000 hours. The Turbine blade material is expected to be 50°C lower than the Turbine Entry Temperature.

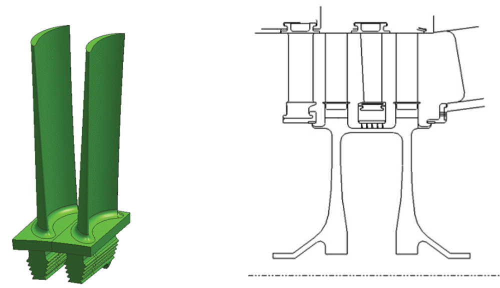


Figure 10 Two stage turbine solution at 6000RPM

In order to maximise the chances of having acceptable creep life without blade cooling, an unshrouded turbine blade has been selected. With this configuration, with the slightly cooler turbine operating temperatures than with a direct cycle, acceptable creep life looks possible without the need for blade cooling, even at 950°C reactor outlet temperatures. The proposed design has 79 blades for each stage. The blade has been supported by a Trent style root and a layout familiar to this type of design.

The blade material selection process has been completed in relation to the range of temperatures and the creep requirements. In completing this assessment creep of 0.1% and 0.2% have been considered. For energy applications 0.2% is considered to be appropriate. In all cases to achieve these requirements aerospace type Nickel alloy material solutions have been identified as the most appropriate using either Direction Solidification blade (DS) or single crystal solutions. The objective has been to use un-cooled blades to allow a less complex design solution. At 750°C DS (M002) materials may be used however beyond 800°C Single Crystal solutions (CMSX4) are more suitable and at 850°C CMSX4 or higher temperature RR3000 (a Rolls-Royce developed high temperature material) may be considered.

Udimet 720 Li has been applied as the most suitable material for the Turbine disc in the expected temperatures. The Turbine blades and disc have been assessed using FE modelling against the initial layouts and material choices shown. The assumed temperature distribution (Figure 11) and worst X Y principal stresses at 6000RPM (Figure 12) are shown below.

This shows that the basic turbine layout is acceptable at this stage of the concept analysis. The Reserve Factors (RF) associated with; Burst, Rim peel are acceptable with RF's of 2.47 and 1.13, and creep RF's of 1.14, 1.18 and 0.68 from the disc bore, diaphragm and rim. The Turbine disc will need cooling which would require some investigation when generating a more detailed design of the disc.

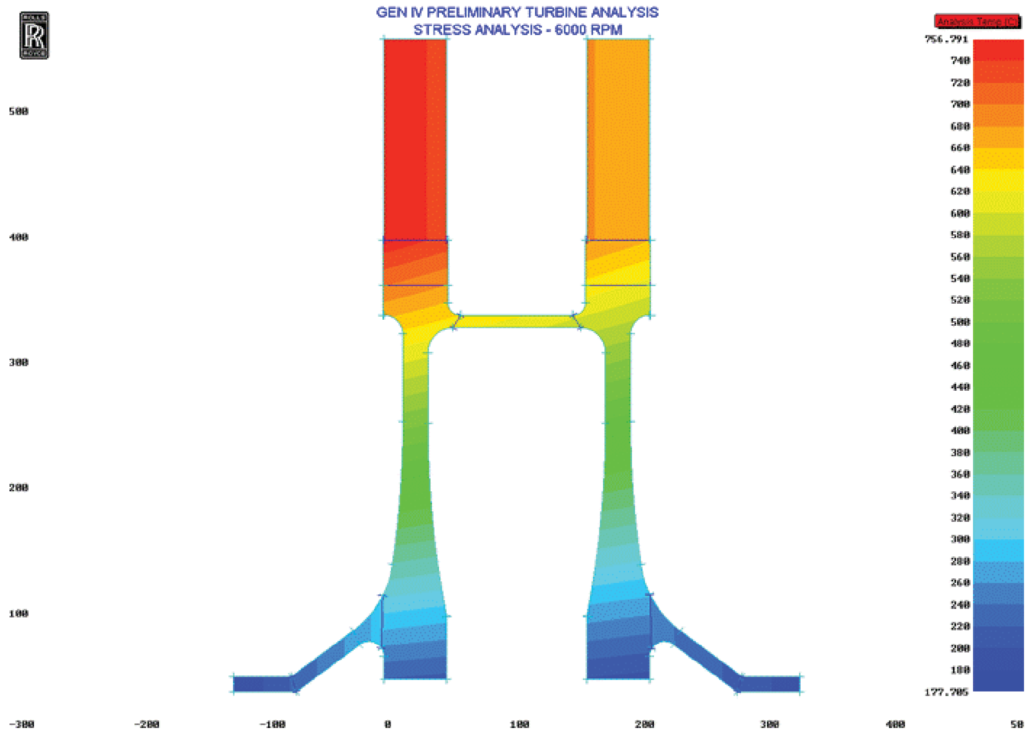


Figure 11 Temperature distribution

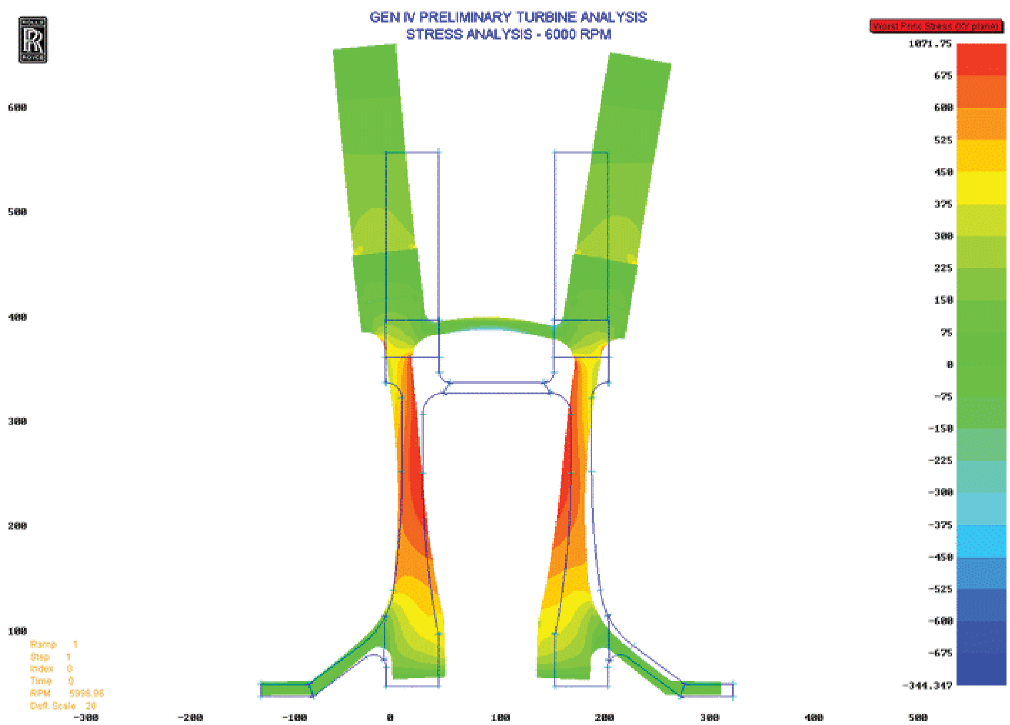


Figure 12 Worst X Y principal stresses

The rotor dynamics of the Turbine with the associated design layout and Campbell analysis have shown an acceptable solution with stiff bearings at this stage of the concept. The first engine mode at 6000RPM from the analysis is below the pitch, bending and bounce modes.

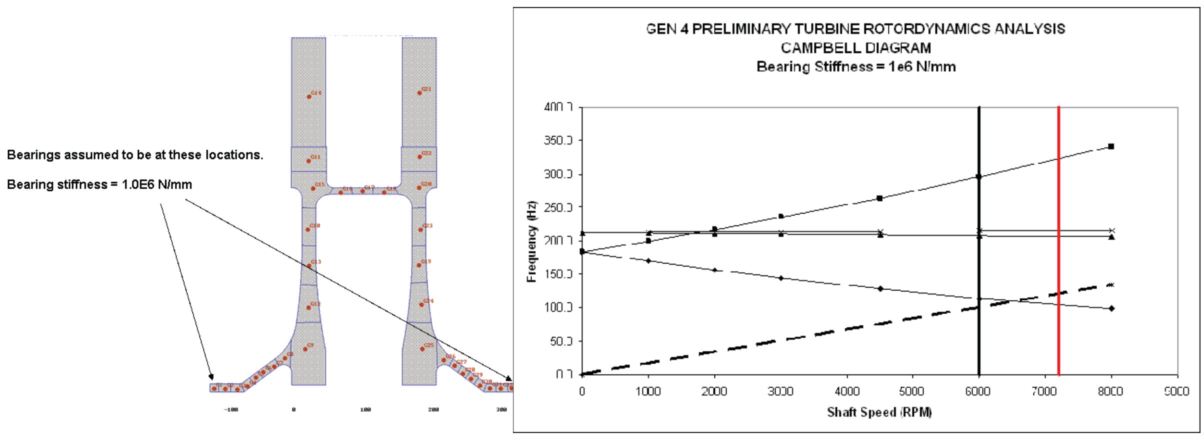


Figure 13 The first engine order at 6000RPM, pitch, bending and bounce modes

5.3 Compressor and Turbine Materials

Consideration has been given to the materials being applied to both the Compressor and Turbine, the implications of use within a N_2/He closed cycle environment and when mixed with oil within the closed cycle. With the solutions being proposed Nickel is a significant constituent but also with major traces of Chromium, Iron and Cobalt. Other minor traces include Tantalum, Tungsten, Niobium, Titanium, Molybdenum, Aluminium, Carbon, Boron and Zirconium. The assessment completed by a literature search of academic material considered the implications with oxidation and corrosion.

It was found that the oil, helium, and nitrogen would not contribute to the corrosion of the compressor and turbine materials. Greater corrosion risks are presented by contaminants or impurities such as steam or air, with the main corrosion mechanism being carburization and decarburization. Because the nitrogen/Helium mix will have less impurities than air corrosion and nitriding is considered likely to have less impact in this environment than in a normal land based or aero gas turbine.

Further consideration would be needed if a direct cycle were to be used because of the implications of turbomachinery contamination by fission products.

6 IHX - choice and sizing

Although the IHX was not part of the Rolls-Royce workscope, some preliminary investigations proved necessary to reach conclusions on the choice of best cycle and best working fluid for the secondary cycle.

The IHX design options are discussed in this chapter. Section 6.1 discusses the heat exchanger style, section 6.2 material selections and section 6.3 the sizing of the IHX for various Nitrogen/Helium gas mixtures.

6.1 Heat Exchanger Style

The design operating conditions derived from the cycle performance model infer that several heat exchanger styles may be appropriate for the IHX. During design-point operation the heat exchanger is subjected to a relatively small pressure differential at elevated temperatures thus plate-fin and tube-shell designs were considered.

Cursory analysis of transient conditions and failure cases, however, suggests that more rigorous design criteria exist to handle development of a severe pressure differential across the heat exchanger boundary arising from a loss of secondary coolant. The severity of this pressure differential is likely to exceed the design pressure capability of a plate-fin heat exchanger and result in significant damage to the matrix and loss of primary coolant boundary integrity.

Nuclear Regulatory Commission (NRC) requirements for inspection during manufacture of heat exchangers providing the primary coolant boundary within high-temperature gas reactors are presently unknown. Plate-fin exchangers require significant welds and lack capability for full inspection of jointed sections. Conversely, modern welding techniques for tube-shell exchangers produce a welded joint that is fully inspectable.

It is therefore suggested that a tube-shell style heat exchanger is the most robust solution capable of meeting both operational, regulatory and safety design requirements. In the absence of detailed information on primary / intermediate circuit protection systems, it must be assumed that the heat exchanger should withstand the full differential pressure of either a primary or intermediate circuit failure, thus compact heat exchangers are excluded from selection. A tube-shell heat exchanger designed to withstand the full pressure differential is therefore required.

6.2 Materials

At the elevated temperatures and long service duration required, progressive inclusion of nitrogen into heat exchanger materials may result in embrittlement and a reduction in structural integrity of duplex steels. The significant material volumes in both the pressure vessel and header assembly may be sufficient such that nitriding results in only a marginal and manageable reduction in overall design strength, hence the use of duplex steels may be permitted to reduce cost. The relatively thin walls of the tube assembly may suffer a significant reduction in strength, and thus it may be necessary to utilise nickel alloy materials (such as Alloy 800) in their construction.

6.3 Sizing

Preliminary estimates of heat exchanger size have been completed on the basis of the indirect 850˚C cycle. Estimations assume a heat exchanger composed of numerous 1” outer diameter tubes with a 5% wall thickness. The heat exchanger is based upon a counter flow arrangement using the ε-NTU method and no specific modelling of entrance or exit losses has been conducted. The specified design fluid containing 20% helium by mass reduces the heat exchanger volume by 27% from pure nitrogen relative to a 56% reduction for a pure helium working fluid. It has also been demonstrated that heat exchanger size is strongly dependant upon allowable pressure drop, selection of which has implications for the performance of the plant cycle. The figure below illustrates variation of normalised matrix volume with pressure loss. Data from the Areva PCS study is marked on for reference, as well as the Rolls-Royce assumption. It should be noted that a some differentiation occurs from the use of helical tubes in the Areva study against straight tubes in the simplistic Rolls-Royce model. Further assessment is required to strengthen the confidence in these numbers.

It would also be possible to adopt a modular heat exchanger arrangement consisting of a number of smaller heat exchangers. Due to the timescale and that the IHX design is not included in the Rolls-Royce workscope for this project this has not been investigated further.

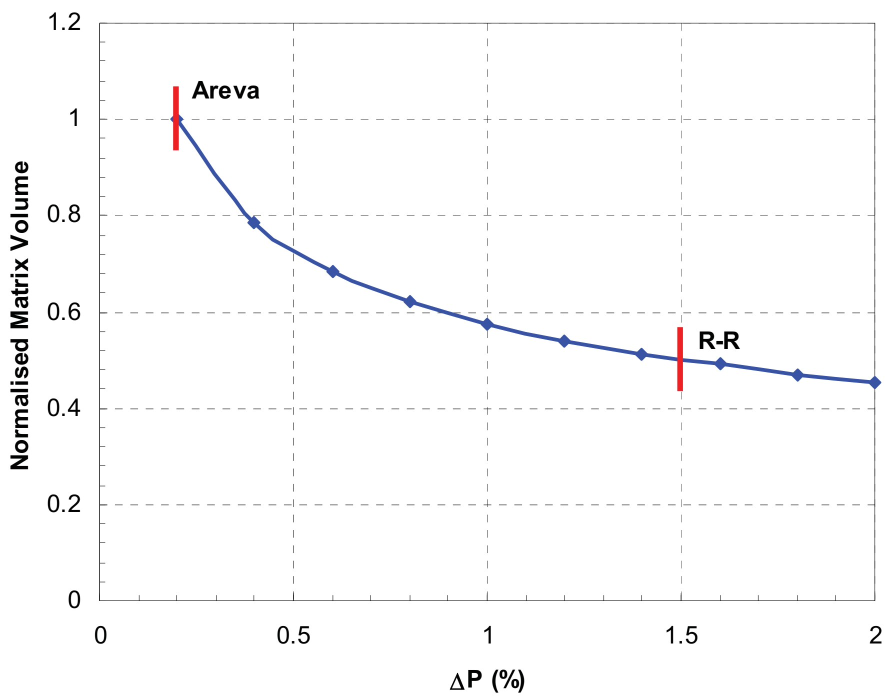


Figure 14 Normalised matrix volume scaling.

7 Layout design

The suggested plant layout and arrangement is discussed in the below sub-sections. The overall site layout is shown in section 7.1. Other sections discuss layout of the ducting, Reactor, IHX, Containment Building, power generation equipment and water system.

7.1 Site Layout

An illustration of a potential plant layout is shown in Figure 15. Although full optimisation of the layout was considered beyond the scope of this study, it is considered that the overall layout proposed is sensible. Where sufficient information was available to approximate equipment dimensions quickly and easily, this has been done.

A workshop, stores, control room, office building is included for completeness but no electrical substation has been included. Also, no refuelling management and spent fuel handling buildings have been included.

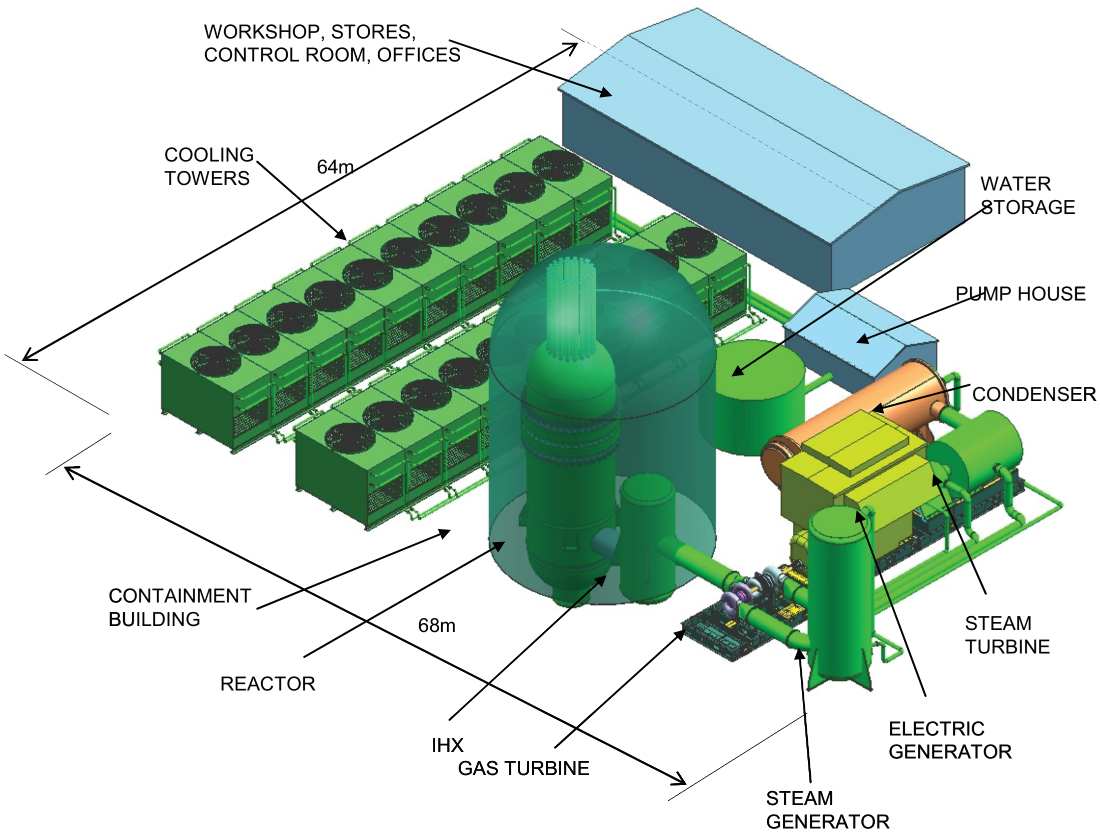


Figure 15 Site Layout

7.2 Ducting

Concentric ducts have been assumed to carry the helium to and from the reactor although the benefits of this have not been assessed.

Pressure losses in the system are considered to be critical in achieving the required performance so the ducting has been sized to give velocities below Mach 0.1 where possible.

The ducts to and from the IHX are concentric with the lower temperature compressor exit flow on the outside. Further study is required to assess if concentric ducts is the optimum solution with stresses in the ducts being weighed against the complexity and stresses in the transition casing.

The pipes from the low pressure (LP) steam turbine have been limited to 1.5m diameter for practicality which results in a velocity of Mach 0.4. This is considered acceptable as the pipes are straight and quite short resulting in a small pressure loss.

7.3 Reactor, IHX, Containment Building

The reactor dimensions have been taken from the direct cycle design study and measures approximately 24m tall (not including the control rods) and 7m diameter

The IHX dimensions have been approximated to suit the matrix dimensions defined in section 6 and is approximately 11m tall and 4m diameter to accommodate the pressure vessel and headers.

A booster compressor/circulator, not shown, is required between the IHX and reactor to circulate the helium and recover the pressure from 69.6 bar at IHX outlet to 70.8 bar at reactor inlet. This may necessitate separation of the ducts. Both the reactor and IHX will be installed within a containment building. The containment building has been sized to fit over the reactor and IHX leaving some space for additional equipment if required including multiple IHX's.

7.4 Power Generation Equipment

The compressor and turbine dimensions approximate to initial gas path estimates but no detailed design has been undertaken.

The compressor outlet scroll will need to incorporate an effective diffuser at the front end and some further diffusion in the scroll is still probable. Careful design will be required to minimise losses. Similarly the turbine exhaust scroll and diffuser will need careful design to ensure pressure recovery is maximised and will need to be structural to withstand the 37bar pressure. The compressor and turbine inlet scrolls will have some contraction so design should be more straight forward.

A double ended generator is assumed where the gas turbine drive is at one end and the steam turbine drives from the opposite end via a synchronous self shifting clutch (SSS clutch) to prevent the gas turbine from trying to drive the steam turbine. The gas turbine is connected to one side of the electrical generator via a gearbox and the steam turbine to the other. The gearbox is shown diagrammatically mounted onto front of generator to give the 6000 to 3600 rpm speed reduction between the gas turbine and generator. Vertically offset parallel shaft gears are used to allow for the differing centreline heights of the gas and steam turbines.

The generator is scaled to 300MW size from a Brush 100MW machine

The steam generator (HRSG) shown is based on the direct cycle design and measures approximately 5m diameter and 14m tall

The steam turbine comprises a high pressure (HP) turbine and twin LP turbines. Steam, having passed through the HP turbine is re-circulated through the HRSG before passing to the LP turbine. The length of the steam turbine has been estimated at approximately 10 metres.

7.5 Water Systems

Steam from the LP turbine outlet passes through a condenser with the outlet water being re-pressurised before returning to the HRSG.

The condenser shown is a 'Basco' type shell and tube cross flow design but no analysis has been done to determine the size.

The cooling water system includes 20off Baltimore AirCoil (BAC) closed circuit coolers, water tank and treatment/pump house.

8 Technology readiness and development requirements for key technologies – direct versus indirect combined cycles

For each key technology, a summary table is given that shows current estimated Technology Readiness Levels (TRL) using the definitions given in INL/EXT-08-14251 (from which the figure below was taken). The reasoning behind the estimate is given, together with development work required to advance the TRL further.

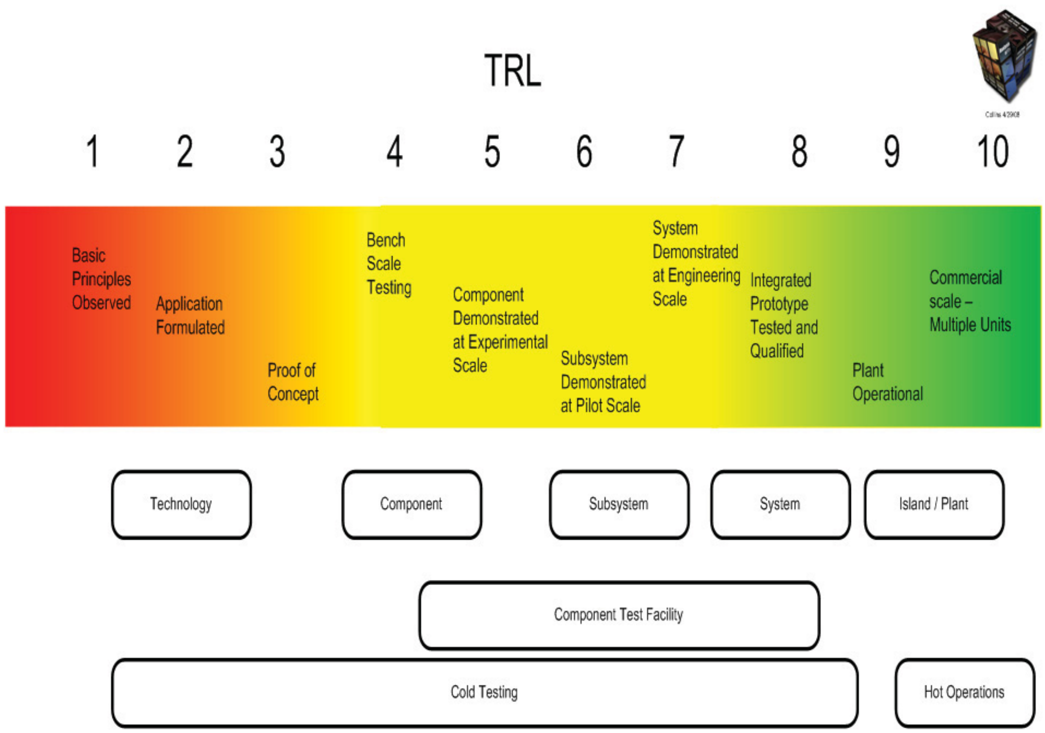


Figure 16 Summary of TRLs from INL/EXT-08-14251

8.1 Turbomachinery aerodynamics

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**3 to 3.5**	**5**
Reasoning	Helium working fluid a long way from current experience	Nitrogen/Helium mix is much closer to air in properties
Requirements to develop technology to higher TRL levels	Closed loop single row cascade testing with Helium would enable TRL4. To get higher is very difficult – best strategy would be to test whole PCS at reduced pressure with large non-nuclear heat source.	Because turbomachinery is not in primary cycle, some development would be possible in the pilot plant. Compressor and turbine could be stripped down and rebuilt with modified blading.

Table 3 Turbomachinery aerodynamics TRL

8.2 Turbomachinery mechanicals

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**3.5**	**4**
Reasoning	Turbomachinery will be exposed to radioactive contaminents (especially Silver). Also, the high pressure Helium environment may cause embrittlement.	Amount of Helium is less. No radioactive contamination. The effects of oil build up in the circuit need to be understood.
Requirements to develop technology to higher TRL levels	Exposing material samples to high pressure Helium environment and performing tensile tests would verify effects. Nuclear contamination is harder to assess – will not know concentrations until reactor is operating for some time.	Exposing material samples to high pressure Helium/Nitrogen mixture environment and performing tensile tests would verify effects.

Table 4 Turbomachinery mechanicals TRL

8.3 Electrical Generator

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**3**	**10**
Reasoning	High pressure Helium cooled. Exposed to high temperatures and harsh environment – novel insulation may be required	Commercial off the shelf.
Requirements to develop technology to higher TRL levels	Insulation tests at high temperatures. Scale rotor tests in high pressure helium to verify windage losses.	

Table 5 Electrical Generator TRL

8.4 Power Electronics

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**8**	**Not required**
Reasoning	Commercially available, but all 'one-offs' rather than off the shelf.	
Requirements to develop technology to higher TRL levels		

Table 6 Power Electronics TRL

8.5 Duct work

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**6**	**6**
Reasoning	Losses in Helium pipework – theoretically calculated, but not experimentally verified	Losses in Nitrogen/Helium pipework – theoretically calculated, but not experimentally verified
Requirements to develop technology to higher TRL levels	Simple flow tests at correct Reynolds Number.	Simple flow tests at correct Reynolds Number.

Table 7 Duct work TRL

8.6 Steam Generator

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**8**	**8**
Reasoning	AGR experience shows manageable, but not Commercial off the shelf.	AGR experience shows manageable, but not Commercial off the shelf.
Requirements to develop technology to higher TRL levels		

Table 8 Steam Generator TRL

8.7 Steam plant

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**10**	**10**
Reasoning	Commercial off the shelf	Commercial off the shelf
Requirements to develop technology to higher TRL levels		

Table 9 Steam plant TRL

8.8 Active Magnetic Bearings and catcher bearings

	Direct Combined cycle	Indirect Combined cycle
Current TRL level	**4.5**	**Not required**
Reasoning	Challenging combination of environment, loads and rpms. Not commercially available for this type of application	Not required.
Requirements to develop technology to higher TRL levels	Small scale rig tests followed by full scale tests.	

Table 10 Active Magnetic Bearings and catcher bearings TRL

9 Direct versus Indirect Cycle - Risk Assessment

The proposed indirect combined cycle and the direct combined cycle are compared and discussed in this section and the next. The comparison has been done from a risk point of view in Section 9, and from a cost point of view in section 10.

A detailed comparison between the indirect combined cycle and the direct combined cycle to evaluate best commercial option has shown that a significant risk reduction will be achieved for the indirect cycle compared to a direct cycle. A direct cycle is still seen to be an achievable option for long term future applications when development of high risk and high cost components such as magnetic bearings are more mature and the risks associated reduced.

9.1 Advantages and risks reduced for the indirect cycle

For the indirect cycle the secondary loop is not in direct contact with the fuel and there is no or only very small risk of contamination. It is therefore assumed that the turbomachinery can be placed outside the containment building and a more suitable working fluid for the turbomachinery can be used. This allows the turbomachinery to adopt a much smaller more conventional arrangement using conventional oil lubricated bearings instead of electromagnetic bearings, and conventional gearbox instead of power electronics. Following advantages with the indirect cycle compared to the direct cycle were identified;

Turbomachinery close to Trent style

One advantage of the indirect cycle is the possibility of using Nitrogen (with 20% Helium by mass) as the working fluid in the secondary loop. Because of its gas properties (close to properties for air) Nitrogen will give a much smaller device with relatively low development risk due to well proven design tools. Furthermore Nitrogen is a cheap gas that could potentially very easily be produced on-site.

Conventional bearings instead of electromagnetic bearings

The development of a viable electromagnetic bearing including catcher bearings and adequate stiffness control required for a direct cycle is an area of great concern. The cost of such a system is also believed to be significant. For an indirect cycle the turbomachinery can be placed outside the containment area allowing conventional oiled bearings to be used. Rolling element a bearings are common in both the aerospace and energy sector and existing off the shelf solution significantly reduces the risks. It should be pointed out that using oil lubricated bearings is likely to cause small oil contamination in the secondary cycle. Some form of filtering would probably be needed in the secondary cycle however this is at this stage not seen to cause any major problems.

More frequent maintenance intervals

The extent of radioactive contamination of serviceable turbomachinery components for the direct cycle is unclear however it is in no doubt that the indirect cycle with a turbomachinery outside the containment area can be accessed for maintenance more easily and more frequently. This has two main benefits; more frequent planned maintenance intervals can be allowed and quicker less costly unplanned maintenance will be achievable.

Reduced size containment building

For the indirect cycle the turbomachinery and the HRSG can be placed outside the containment building. However, instead the Intermediate Heat exchanger (IHX) will be placed inside the containment area. It is likely that a decrease of the containment building size, leading to a cost reduction can be made for the indirect cycle.

Conventional gearbox instead of power electronics

One of the key commercial risks for the direct cycle includes the anticipated very high cost of the power electronics. For the indirect cycle conventional gearbox can be used reducing costs significantly.

Double ended generator

The two separate generators (gas turbine and steam generator) used for the direct cycle can be replaced in an indirect cycle by a single ‘double ended generator’ where the gas turbine drives one end and the steam turbine drives the opposite end via synchronous self shifting (SSS) clutch. This will reduce the cost of electrical transformers, switch gear etc. and also increases inertia connected to the Gas Turbine, which would help in preventing overspeed for electrical dropped load events.

More design and off-design flexibility

The indirect cycle has the flexibility to be designed for a more optimum reactor inlet temperature. Optimum reactor inlet temperatures are close to the upper limit for each reactor outlet temperature (i.e. close to reactor ΔT of 360°C). Furthermore the indirect cycle has more off-design flexibility to vary steam flow to maintain reactor inlet temperature and maintain a better off-design efficiency.

9.2 Disadvantages and additional risks for the indirect cycle

Although the risks are significantly reduced for an indirect cycle for the reasons given above it should be acknowledged that there are additional disadvantages and risks for the indirect cycle associated with the Intermediate Heat Exchanger. The following disadvantages with the indirect cycle compared to the direct cycle were identified;

Pressure drop in IHX and decreased TET reduces performance

Introduction of an IHX have two disadvantages on cycle performance. The first one is the penalty associated with the decreased turbine inlet temperature and the second one is the pressure drop across the IHX. The efficiency drop for the indirect cycle compared to a direct cycle with same reactor outlet temperature is approximately 1.6%pts.

Cost and risks associated with the introduction of an Intermediate Heat Exchanger

The introduction of the IHX for the indirect cycle does have a cost and risk associated. Although the turbomachinery and the HRSG can be placed outside the containment building the IHX will be placed within that area. More detailed design and sizing of the IHX is required to determine the full cost implication. The main risks identified for the IHX are; leakage of primary or secondary fluid, material damage due to nitriding, failure due to a pressure differential across the IHX. The implication of this has not been fully assessed.

9.3 Risks that apply to both the indirect and the direct cycle

There are still risks associated with transient that apply to both the direct and indirect cycle. The bypass valve, which needs to pass 45% of the main flow in order to reduce the net power on the gas turbine shaft, will need additional investigation. The practicalities of the bypass valve passing such a large flow, whilst also being able to exert fine control over the speed of the shaft when it is not synchronised, will need careful consideration. The design of such a valve is at this stage believed to be possible however the difficulties with the design should not be ignored.

Furthermore the probability of component failure (shaft failure, blade-off, disk-off failures etc) is still the same for both the indirect and the direct cycle. For the indirect cycle a component failure could damage the IHX. For the direct cycle same failure would cause damage to reactor. The impact, cost and plant down time, of the latter one is likely to be higher.

It has been assumed that the PCS can be placed outside the containment area for the indirect cycle, this is believed to be a reasonable and likely assumption. If this for any reason would not be possible and the PCS would have to be placed within the containment building then the risks discussed in 9.1 would not be valid and these risks would remain similar as for the direct cycle.

10 Direct versus Indirect Cycle - Cost Comparison

A preliminary cost comparison of different cycle configurations for the PCS has been done. This comparison has been done using information available at the time and refinement once more detailed designs have been established is recommended. The cost comparison shows cost benefit to the indirect combined cycle compared to the direct cycle and also compared to a pure steam cycle. It should be kept in mind when examining the results below that is likely that in the long term the cost of electromagnetic bearings and power electronics can be reduced significantly which will bring the cost of electricity (COE) for the direct cycle down. However, the significant risk reduction of an indirect cycle compared to a direct cycle weigh higher than the cost benefit and should be first priority.

NOTE: To enable to do a cost comparison a number of assumptions had to be taken. It should be pointed out that there is still high uncertainty in these numbers and should only be used to identify the relative cost trends between the different plant configurations and not as a firm absolute number.

10.1 Cost comparison for different working fluid mixtures

By introducing approximately 20% helium by mass to the nitrogen in the secondary circuit an overall cost reduction of the PCS can be seen. This benefit arises because of the contradictory nature of the dependence of the IHX and turbo-machinery components. A cost comparison of the turbomachinery and the IHX for pure nitrogen, a Nitrogen/Helium mix and pure Helium was done. The comparison is based on the size and cost of the turbo-machinery, the IHX and the steam plant. The results show that there is a cost benefit to the Nitrogen/Helium mixture.

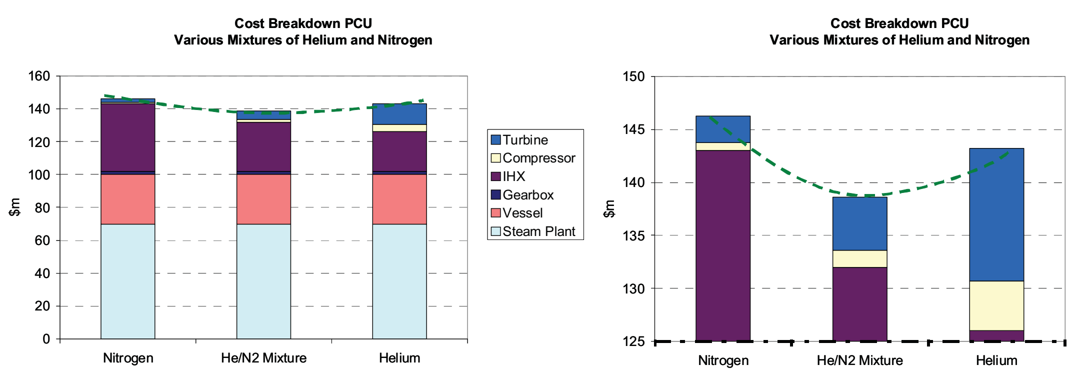


Figure 17 Capital cost of PCS for different Helium/Nitrogen mixtures. Picture to right is a zoom in of left picture.

10.2 Cost comparison for different PCS configurations

A cost comparison was then done between the indirect combined cycle gas turbine (CCGT), a direct CCGT, an indirect gas turbine cycle with pre-cooler and recuperator and a direct gas turbine cycle (OKBM). Both the indirect cycles have a Helium/Nitrogen mixture as working fluid while the direct cycles are constrained to pure helium as the working fluid. The results show that both indirect cycles have lower capital cost than the direct cycles. This is mainly down the high cost of the electromagnetic bearings, the power electronics and the more costly turbomachinery required for using pure helium as the working fluid.

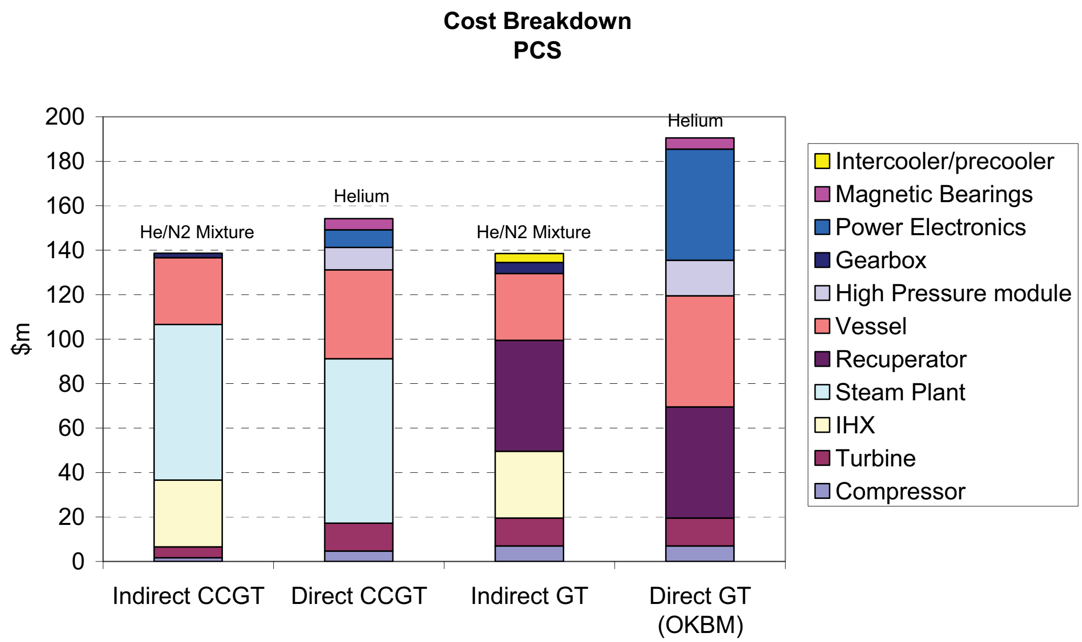


Figure 18 Capital Cost comparison of PCS for various cycle configurations

The above configurations all have different plant efficiencies and to make a cost comparison this needs to be taken into account when comparing the costs.

The cost shown in Table 11 is the COE associated with the capital cost of the PCS (IHX included for indirect cycles) and does not include the costs associated with the reactor and containment building. Furthermore all cases above assume plant availability of 90% but do not include any cost for maintenance. Payback period set to 30 years with a discount rate of 10% (Bank practice is to use a rate of 10 or 12 percent to calculate the net present value of a project).

Capital cost for pure steam plant has been estimated to \$120m which derives from removing the gas turbine from the indirect cycle and scale up the steam plant. A pure steam plant was modelled using IPSEpro (section 2.3.1) in the same way as the indirect CCGT cycle was model but with removed gas turbine. For a steam temperature of 580°C at 170 bar pressure the maximum plant net efficiency is 42.6%. This was considered as the fairest comparison with the other cycles.

	Direct CCGT 850°C	Indirect CCGT 850°C	Indirect CCGT 950°C	Direct GT (OKBM)	Indirect GT	Pure Steam
Plant Efficiency	50.2%	48.6%	49.3%	48.0%	46.0%	42.6%
COE	Reference	-3.0 %	-4.4%	+16%	+1.7%	+2.5%

Table 11 Cost comparison of PCS for various plant configurations

The results show that the indirect CCGT has the best COE when comparing the electricity output with the cost of the PCS. Although the direct CCGT has better efficiency the higher capital cost results in a COE higher than the indirect CCGT, it does not compensate for the higher capital cost compared to the indirect CCGT.

10.3 Whole Plant Cost Comparison and Parametric Study

When the total plant COE is calculated, including cost of reactor, containment building, operation and maintenance etc (these costs have not been available in this study) the plant efficiency will have a far greater impact on the COE. This chapter will discus the main factors that will impact the total plants COE.

The cost model used for comparison of various PCS options is an excel based tool developed within Rolls-Royce. The tool compares the cost of various electricity generation sources such as coal fired, nuclear and CCGT plants and also renewable energy sources such as on-shore and off-shore wind. Part of the tool could be used to compare the various NGNP options in this study. The inputs required can be divided into three different components; capital, finance and operating costs. Capital and financing costs make up the project cost. Capital cost here includes the reactor installed cost (including land, infrastructure, buildings, site works, licences etc) and the installed cost for the pure Power Conversion System, PCS design installed cost. Financing costs will depend on the set assumed discount rate. Operating costs are O&M costs plus fuel cost and any connection charge to grid.

Four different case studies where done for different financial scenarios. The costs are proposed for n end-of-a-kind plant. Following assumptions were taken;

- Cost of PCS for the various configurations are same as used for section 10.2.
- Capital cost of the entire plant (excluding the PCS) is assumed to be fixed for a 600MW plant and therefore the cost per produced kilowatt of electricity will vary depending on plant efficiency. Various sources suggest that a plant cost between \$500m and \$1000m is a reasonable assumption. Two cases have been run; one in the lower end of this range and one for the high end of the range.
- The fuel cost is assumed to be between \$1700 and \$3400 per kg of UO2. The cost of the fuel includes processing, enrichment and fabrication of the Uranium into fuel elements. It also includes management of radioactive spent fuel and the ultimate disposal of the spent fuel or the wastes separated form it.
- Plant efficiency for each configuration is based on results from modelling the specific thermodynamic cycle using IPSEpro. The plant efficiency for each cycle are shown in Table 11.
- Plant availability assumed to be 90% for all plant configurations and is based on 20 days outage every 18 months due to refuelling.
- O/M costs include predicted operation, planned and unplanned maintenance. The O&M cost per kWh produced electricity for the indirect cycle has been assumed to 2 c/kWh. O&M costs are assumed to be 20% higher for a direct cycle due to the higher component costs and risks and 20% lower for a pure steam cycle for the opposite reason. This has been kept constant for all four cases.

- The discount rate is the percentage by which the value of a cash flow in a discounted cash flow valuation is reduced for each time period by which it is removed from its present. In this study it has been set to 10% for 2 of the cases and 5% for the other 2 cases. A payback period of 30 years is assumed. It should be noted that this is not the same as the operational plant life but the financial project life.

- Lead time (time interval between the start of an activity or process and its completion, here the time between starting manufacturing and building the plant and its completion) has been estimated to 48 Weeks. The lead time has been kept constant throughout this study.

The results of the COE for each of the four cases are presented in Figure 19, Figure 20, Figure 21 and Figure 22. It splits the cost between equipment/installation costs, fuel cost and O&M costs. Two different discount rates have been used. The final result is sensitive to the choice of discount rate and is often the most difficult and uncertain parameter to set. A discount rate of 10% is the more common to use however for comparison two cases with a lower discount rate of 5% was also run. For each of the two discount rates one scenario for the upper range on capital and fuel cost and one scenario for the lower range of the capital and fuel cost. The capital cost will have a much greater impact on the COE than the Fuel cost and therefore the combination high capital cost/low fuel cost and vice versa has not been presented.

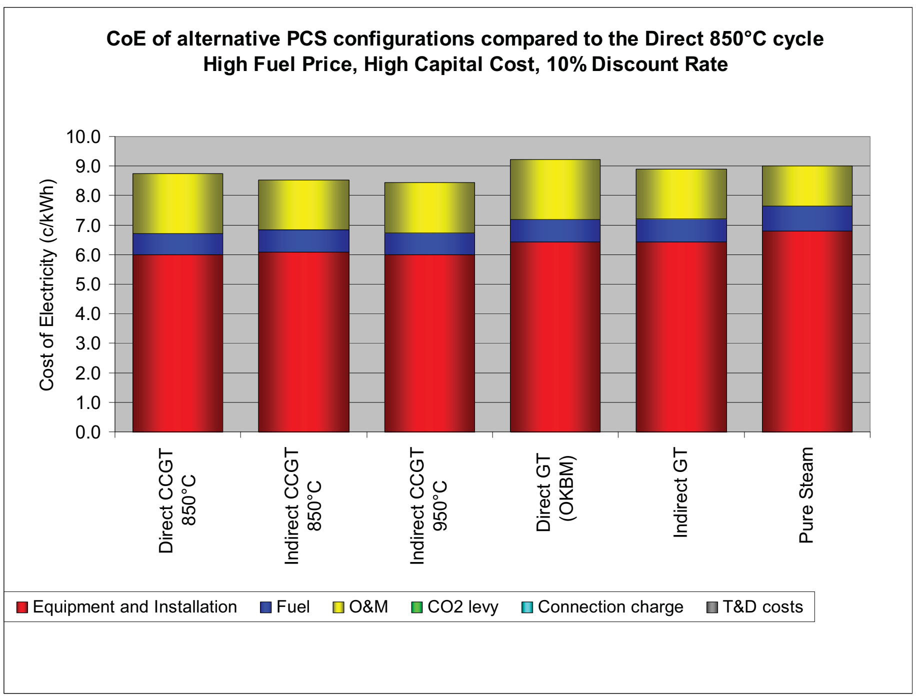


Figure 19 Estimated COE of alternative configurations for high fuel price, high capital cost and a 10% discount rate.

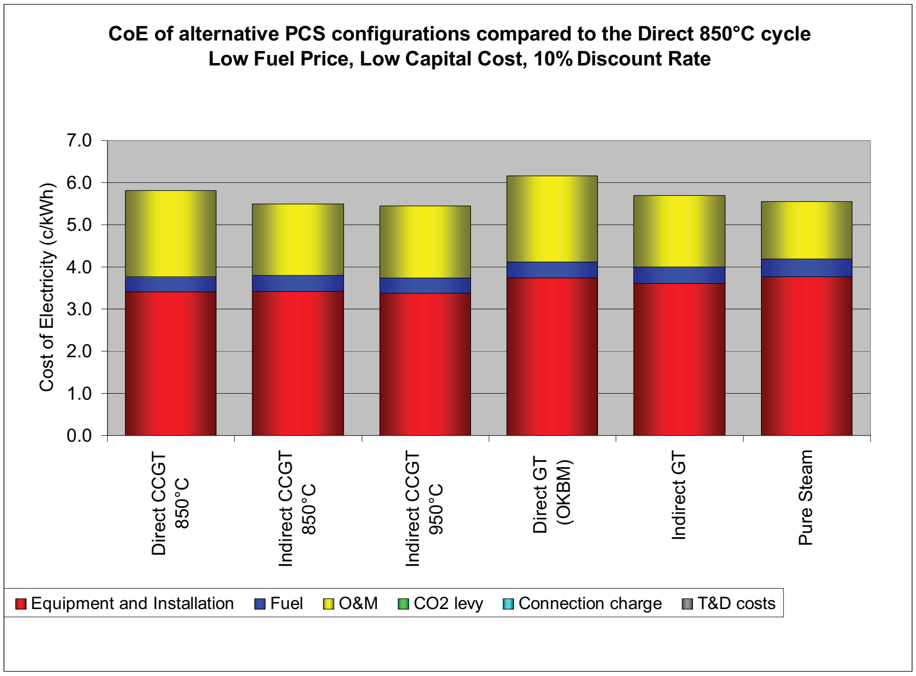


Figure 20 Estimated COE of alternative configurations for low fuel price, low capital cost and a 10% discount rate.

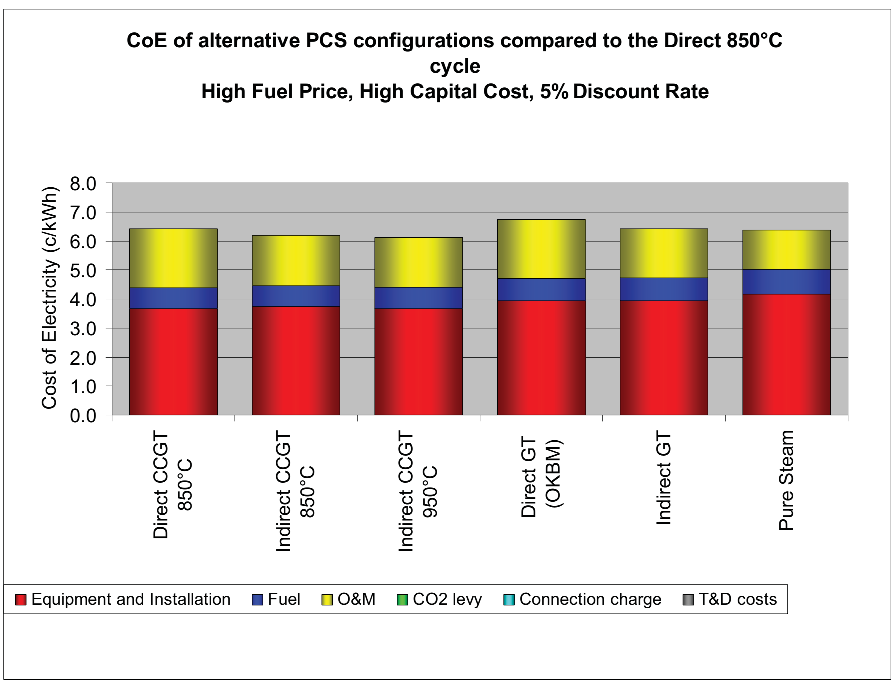


Figure 21 Estimated COE of alternative configurations for high fuel price, high capital cost and a 5% discount rate.

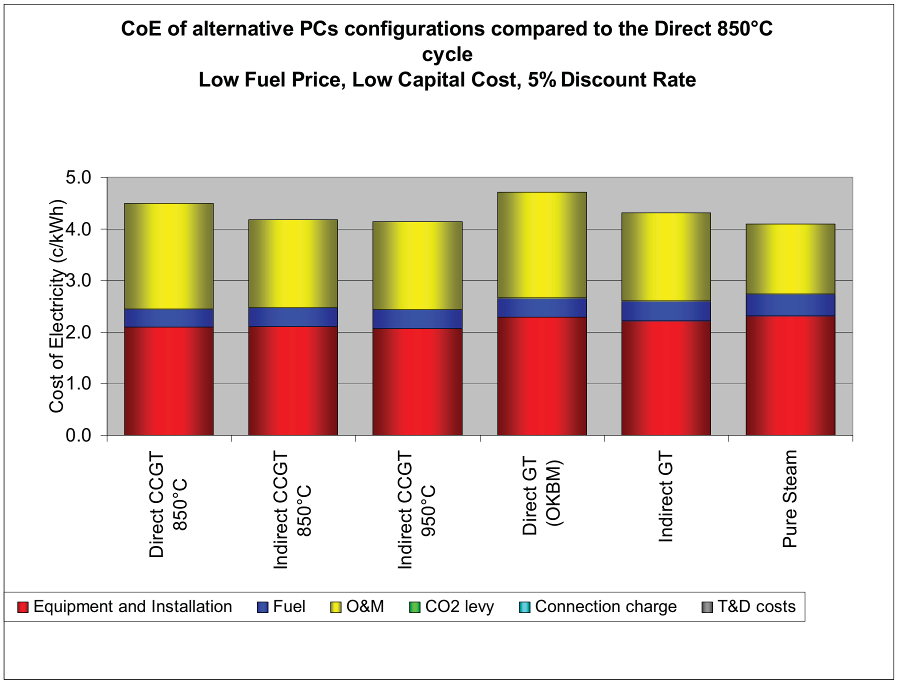


Figure 22 Estimated COE of alternative configurations for low fuel price, low capital cost and a 5% discount rate.

Equipment and installations cost for a nuclear plant is know to be the main cost driver, this is also the case for all of the configurations compared here. Second highest contribution to the COE is the O&M cost while the Fuel price only represents a smaller fraction of the cost. The lower capital cost and discount rate the higher share the O&M costs represents. The COE for the different cases varies significantly. As the project develops the certainties in the cost figures should increase but it should also be acknowledge that both the costs can vary significantly depending on the country and location of the plant. A summary of the four cases with the relative change to the direct CCGT cycle is shown in Figure 23.

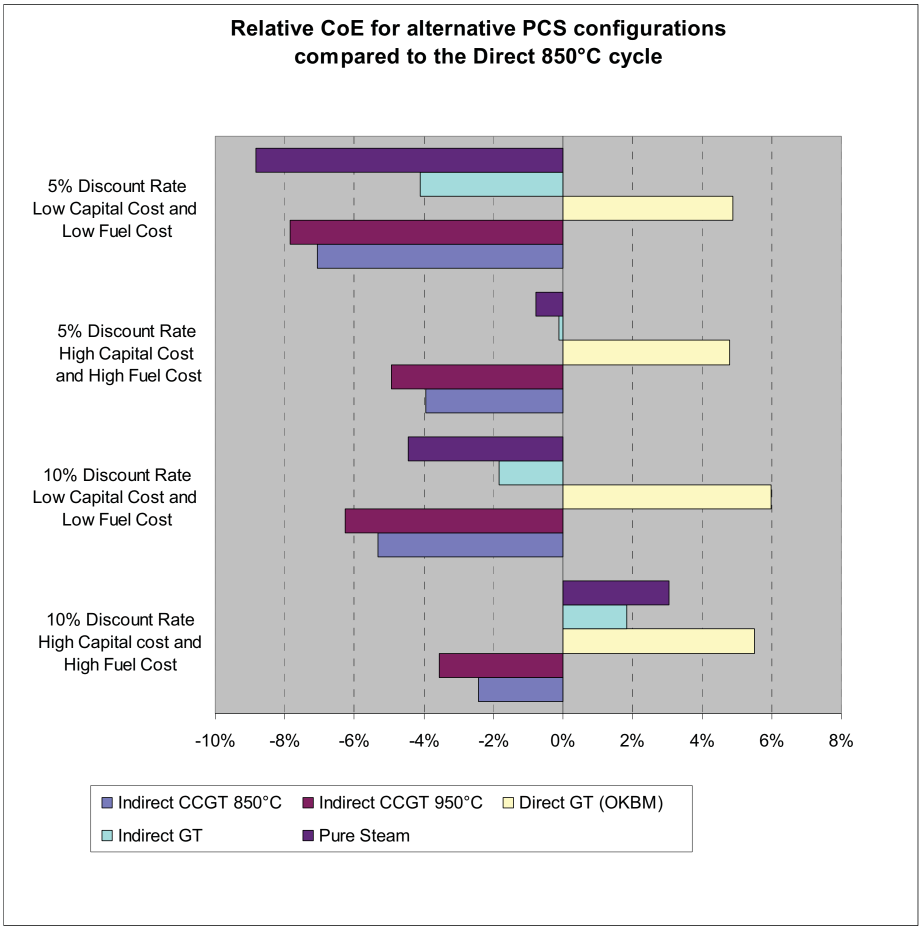


Figure 23 Whole plant cost comparison for various plant configurations

With the assumptions made it shows that the indirect cycle still gives the lowest COE for 3 out of the 4 cases. If the capital cost is high then a high plant efficiency becomes more important. This is why the indirect and also direct CCGT will give better COE than a pure GT or Steam cycle for a high capital cost. For a low capital cost and low discount rate the pure steam cycle gives the best COE. The indirect CCGT still shows a very good COE in comparison the direct CCGT and the GT cycles. When comparing the 4 cases it can be observed that the indirect CCGT is less sensitive to change in capital cost and discount rate compared to the pure steam cycle. This because the indirect cycle has a much higher efficiency making it less sensitive to capital cost increase.

An uncertainty that has not been investigated is the plant availability for the above compared cycles. Plant availability has been assumed the same for all configurations. However the risks associated with the direct cycle is likely to reduce the plant availability, particular in the early development stage of these technologies.

11 Conclusions

An indirect combined cycle has been found to be the best choice for a medium term commercial electricity plant. This cycle is shown to have the best efficiency of any indirect cycle and to achieve the lowest cost of electricity generation among the systems compared in this study.

The indirect combined cycle conforms to the requirements specified for NGNP, in that it is compatible with the provision of high temperature process heat and with provision of process steam, to support demonstration of hydrogen production technologies.

The direct Brayton cycle was the subject of a previous study. Its efficiency has been shown to be lower than that of the direct and indirect combined cycles, although only slightly lower than that of the indirect combined cycle. This is an elegant concept, for which the principal drawbacks were found to be development and maintenance risks (particularily the very challenging active magnetic bearing and large recuperator), together with some potentially high costs. In the longer term, if these major challenges can be overcome, the direct Brayton cycle may merit reconsideration because its lack of a steam cycle makes it simpler so there is less to fail.

The direct combined cycle was also a subject of the previous study, achieving the highest efficiency of the systems considered here. Its greatest advantage over the direct Brayton cycle was considered to be its lower development risk (reduced risk active magnetic bearing and no recuperator requirement). However, the risk remains high in comparison with the indirect combined cycle. If future evolution of the technology base resolves the risk issues, the direct combined cycle may become a preferred option for longer-term commercial electricity generation plants.

The indirect Brayton cycle is outside the terms of reference of the present study and has not been given detailed attention. It has been shown to be a lower efficiency cycle than the direct cycles and the indirect combined cycle. Turbine risks are higher than for the indirect combined cycle, although much lower than for the direct Brayton cycle and might be resolved by future evolution of the technology base. Plant simplicity, there being no steam cycle is a benefit this cycle shares with the direct Brayton cycle.

All the cycles that incorporate a gas turbine have the potential for gaining improved performance from raised reactor operating temperatures (950°C and beyond), although some materials development and/or blade-cooling system would be required for the direct cycles.

The combined cycle systems are less susceptible than pure Brayton cycles to loss of output in off-design conditions such as reduced heat sink effectiveness.

The pure steam system is the lowest efficiency option of those considered for comparison, although there is limited scope for improvement by adoption of supercritical technology. A pure steam cycle is a disappointing match with the characteristics of a high temperature reactor, because the temperature limitations of the steam system prevent effective utilisation of primary temperatures above about 650ºC. Part-load conditions that produce steam at above design point temperatures may incur some materials development issues.

12 Appendix A – Cycle Modelling Assumptions and Net Electrical Efficiency Calculations

Two tables are presented in this appendix. The first of these (Table 12) details the assumptions used in modelling the intercooled and recuperated Brayton cycle, the direct combined cycle and the indirect combined cycle. The Brayton and direct combined cycles were both modelled using the commercial code Thermoflex. This was not available during the current study and hence the modelling was done using a different commercial code, IPSEPro. Because of differences in the way that components are represented in the two codes, it has not been possible to model them using identical parameter assumptions. Nevertheless, great care has been taken that the assumptions between the cycles and codes are as consistent as possible in order to avoid unfairly favouring any of them.

The second table (Table 13) shows how the net electrical efficiency numbers are calculated from the shaft powers, the generator efficiencies and the auxiliary losses.

Parameter	Intercooled and Recuperated Brayton Cycle	Direct Combined Cycle	Indirect Combined Cycle
Reactor Power	600MW	600MW	600MW
Reactor Outlet Temperature	850°C	850°C	850°C
LPC efficiency	89% poly (87.8% isen)	89% poly	89% poly
HPC efficiency	89% poly (87.8% isen)		
Turbine efficiency	91.8% poly (93.2% isen)	91.8% poly	90.9% poly [1]
Recuperator ΔP/P	2% on both hot and cold sides	N/A	N/A
Reactor ΔP/P	0.85%	0.85%	0.85%
Pre-cooler ΔP/P	0.13%	N/A	N/A
Pre-cooler outlet temperature	26°C	N/A	N/A
Intercooler ΔP/P	1.6%	N/A	N/A

Parameter	Intercooled and Recuperated Brayton Cycle	Direct Combined Cycle	Indirect Combined Cycle
Intercooler outlet temperature	26°C	N/A	N/A
GT Generator efficiency	95%	96%	0.96
GT mechanical efficiency	100%	99% (based on turbine power)	99% (based on turbine power)
Steam Turbine efficiency	N/A	91.8% (poly)	91.8% (poly)
Steam Turbine Wetness Loss	N/A	Automatically calculated	Efficiency of wet part of expansion set to match that calculated in direct combined cycle.
Steam Turbine Mech efficiency	N/A	99%	99%
Steam Turbine Gen efficiency (assumed synchronous)	N/A	98.5%	98.5%
Condensate Pump efficiency	N/A	85% (isen)	80% (isen)
Reheater steam side $\Delta P/P$	N/A	3%	3%
OTSG Water Side $\Delta P/P$	N/A	5%	5%
OTSG + Reheat Gas side $\Delta P/P$	N/A	2.2%	2.1%
OTSG Superheater & Reheater ΔT	N/A	40°C	40°C
OTSG Economiser exit 'pinch'	N/A	22°C	22°C
Condenser Pressure	N/A	45mbar	45mbar
IHX Hot Side $\Delta P/P$	N/A	N/A	0.85%
IHX Cold Side $\Delta P/P$	N/A	N/A	2.5%

Parameter	Intercooled and Recuperated Brayton Cycle	Direct Combined Cycle	Indirect Combined Cycle
Helium Circulator efficiency	N/A	N/A	89% (poly)

Table 12 Cycle Modelling Assumptions

Notes on Table 12

[1] Turbine efficiency was reduced by 1% to account for the use of unshrouded blades.

[2] The steam turbine was modelled using a number of steam turbine modules in series. The isentropic efficiency of each of these was set to match the expansion which resulted from the 91.8% polytropic efficiency turbines specified in the Thermoflex model of the direct combined cycle in the 2007 study.

Parameter	Intercooled and Recuperated Brayton Cycle	Direct Combined Cycle	Indirect Combined Cycle
Reactor Power	**600MW**	**600MW**	**600MW**
GT Shaft Power	303.2MW	68.3MW	46.8MW
GT Electrical Power	288.0MW	65.6MW	44.9MW
ST Shaft Power	N/A	247.2MW	271.3MW
ST Electrical Power	N/A	243.5MW	267.2MW
Gross Electrical Power	**288.0MW**	**309.1MW**	**312.1MW**
Cooling Tower Fan	N/A	1.8MW	N/A[Ψ]
Water Cooled Condenser Pump	N/A	1.8MW	2.1MW
Condensate Pump Power Consumption	N/A	4.6MW	5.7MW

Primary Circuit Circulator Power	N/A	N/A	9.9MW
Auxiliary Power Correction			2.3MW$^{\Psi}$
Auxiliaries Total	**0MW***	**8.2MW**	**20.0MW**
Net Electrical Power	**288.0MW**	**300.9MW**	**292.1MW**
Net Electrical Efficiency	**48.0%**	**50.2%**	**48.6%**

Table 13 Calculation of Net Electrical Efficiency

Notes on Table 13

* Auxiliaries were not modelled for the intercooled and recuperated Brayton Cycle. A generator efficiency of 95% was modelled when the available references (p75 of 08.03-006.01A1-e) suggested a value of 97.7%. It was considered that the difference between these values was sufficient to account for auxiliary losses and therefore that the claimed 48% efficiency was valid. No attempt was made to model the auxiliary losses in detail. It is noted, however, that 2.7% of the 303.2MW gross shaft power equals 8.2MW, which is almost identical to the value used in the direct combined cycle. This is reasonable since the cooling duty is very similar.

Ψ Cooling Tower Fan was not modelled directly in the IPSEPro model. Its effect was accounted in an 'auxiliary power correction' term. This was calculated using the following equation: Additional power consumption (kW) = 2000x(steam mass flow rate(kg/s))/152.

13 Appendix B - More details from compressor and turbine mechanical work.

For completeness, here are more details from the turbomachinery mechanical design activity. Included here is also analysis information for the 10 stage compressor variant that we considered earlier in the study. This variant was dropped during the course of this phase of work in favour of the 6 stage 'Rolls-Royce Trent like' compressor which was discussed earlier in the report.

13.1 Introduction.

An assessment of mechanical solutions for the Gen IV Gas Turbine Compressor and Turbine has been carried out.

The Gas Turbine features as part of the combined energy generation capability of the Gen IV combined cycle plant defined in section - 2.1

The Gas Turbine functions at 6000RPM in normal operation, with the inlet and outlet conditions as specified.

The objective therefore has been for a Gas Turbine mechanical concept that satisfies these conditions with an operational period that aligns with the refuelling cycle of 18 months with the proposed Nitrogen/Helium closed cycle solution and a technology readiness that would allow the system to be available at acceptable levels of risk.

The Nitrogen/Helium mixture has been selected as an enhanced combination of heat transfer thermo dynamic characteristics whilst having some similarities to those of air. This provides a combination of beneficial performance characteristics with features that satisfy this type of mechanical solution.

13.2 Gas mixture implications on the mechanical system.

A study has been completed using available literature on the potential implications of these gases in a closed cycle on the types of material that would be used for the Compressor and Turbine. The study included Noble gases Xenon and Argon as well as Helium and Nitrogen.

The list of research papers used in this assessment are included at the end of this Appendix.

The main findings are as follows.

For Helium it was found that the actual elements involved in the corrosion process were the impurities in the helium environment, not the Helium itself. Two important impurities were H_2O and CO, in which their concentration largely determined the partial pressure of oxygen and the activity of carbon. It was shown by Quadakkers and Schuster that the corrosion behaviour in a flowing gas could be described by defining a steady state carbon activity and oxygen partial pressure, which were determined by the kinetics of the different elementary gas metal reactions.

The main mechanisms of corrosion in this situation were corrosive oxidation, carburization and decarburization. The alloy surface, temperature and gas composition determined which mechanism would dominate. For mechanical stability, carburization and decarburization were of particular significant. Carburization is associated with low temperature embrittlement due to the transfer of carbon from the atmosphere to the metal, while decarburization is linked to reduced creep rupture strength.

There were two types of oxidations, one beneficial and the other detrimental. The beneficial oxidation was the formation of a layer of oxide on the surface of the material (sometimes known as oxide scale formation). This scale formation was the main factor behind the corrosion resistance of the material. The scale would normally be chromium oxide, which was formed from the chromium traces in the material. More information of the kinetic of the scale formation can be found in the paper by Wright, Nagarajan and Stringer The detrimental oxidation was that of internal oxidation from two types of oxidations noted in the paper from Wright.

The newer alloy Haynes 230 was considered as an alternative to Inconel 617 due to its equivalent creep properties and less susceptibility to internal oxidation.

In addition, there was also the possibility of the loss of part of the oxide scale due to spallation, thermal cycling or mechanical damage, exposing the alloy surface already depleted in chromium due to the formation of chromium oxide earlier to the environment. Detailed study of this can be found in papers by Wright, Nagarajan and Stringer.

Little information appears to exist on the effects of Argon and Xenon at high temperature. However, from the understanding of the corrosion mechanisms with helium as the working fluid where the corrosions were due to the impurities in the working fluid, it is believed that similar explanations applied to the case of Argon and Xenon.

13.3 Design criteria.

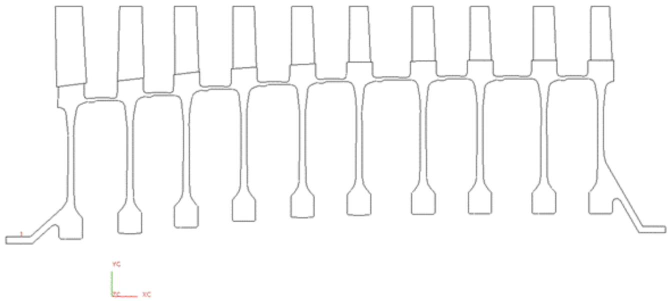
For the Gas Turbine 18 months operation equates to 13000 hours. It has been assumed within this period that the number of start and stops will be minimal, hence potentially a low cyclic use with a long duration of continual operation was the most likely operating regime. In these conditions the most likely failure mode is creep.

Ideally no significant maintenance or repair action should be required within the 13000 hours and hence a modular approach to any replacement of the compressor or turbine should be required at less than 13000 hours, and if possible these actions should be at multiples of this i.e 13000 or 26000 hours.

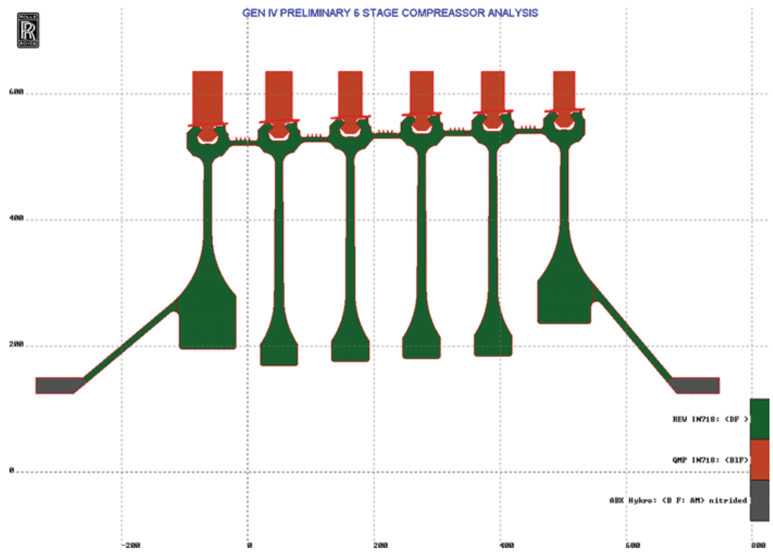
13.4 Compressor.

13.4.1 *Compressor concept and materials*

To achieve the thermo and aerodynamic properties required from the Gas Turbine two compressor concepts were identified. These are a ten stage and a six stage solution. Both were assessed to confirm whether a mechanical solution was viable. In both cases the solution has been based on aerodynamic solutions based on Rolls-Royce Trent engine technology.



Ten Stage Compressor concept



Six stage compressor concept with materials choice.

To assess the concepts design models were created along with Stress analysis models. Material selection was defined based on the environment and the creep life.

The material selected is INCO 718 for both the Compressor blades and discs. For the discs Titanium IMI834 has also been considered for the front of the compressor. Both are used extensively for Gas Turbine designs, but for cost reasons as well as mechanical properties in this environment INCO 718 is considered more appropriate for the discs. It is also possible that a combined INCO and Titanium arrangement may cause thermally induced stresses which would make the design more complex across any bolted joints.

13.4.2 *Stress and rotor dynamic analysis*

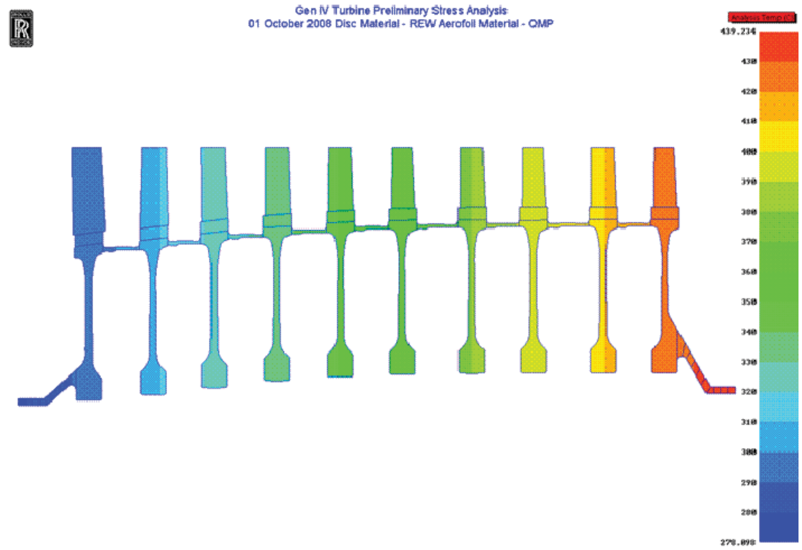
An assumed thermal gradient across the compressor has been made and FE stress and vibration analysis completed for both solutions. Analysis of burst, rim peel and the relationship to the creep conditions have been completed. The analysis output below shows acceptable values for the concepts proposed.

The outputs shown for both solutions are as follows.

Ten Stage Compressor analysis.

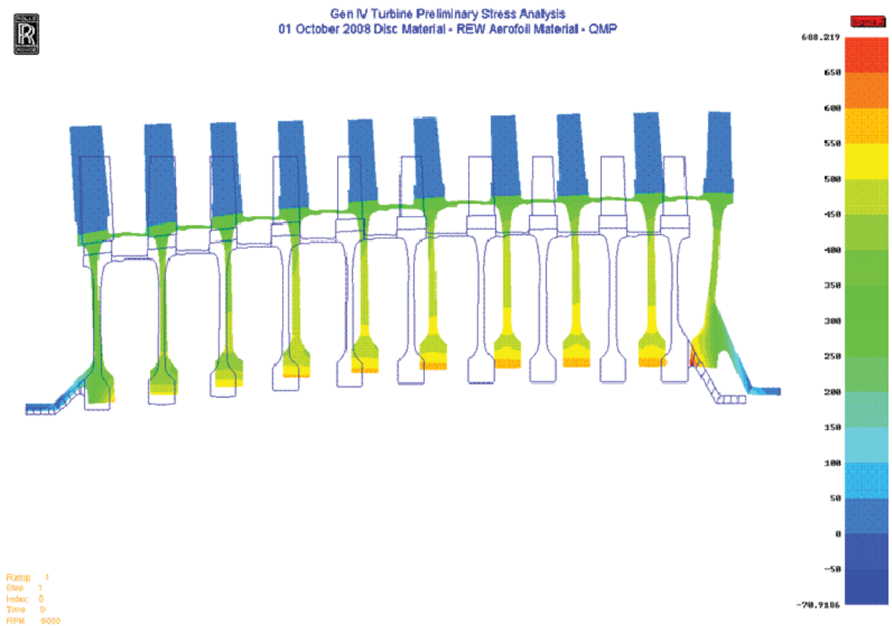
Compressor, materials, and mechanics

Assumed Temperature Distribution – deg C

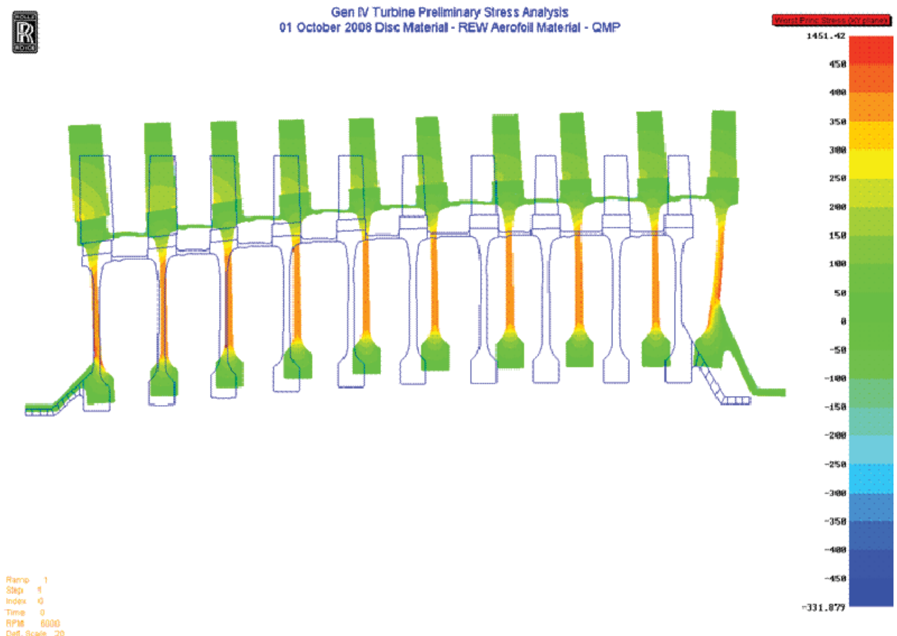


Compressor, materials, and mechanics

Hoop Stress – 6000 RPM



Worst XY Principal Stress – 6000 RPM



Compressor, materials, and mechanics

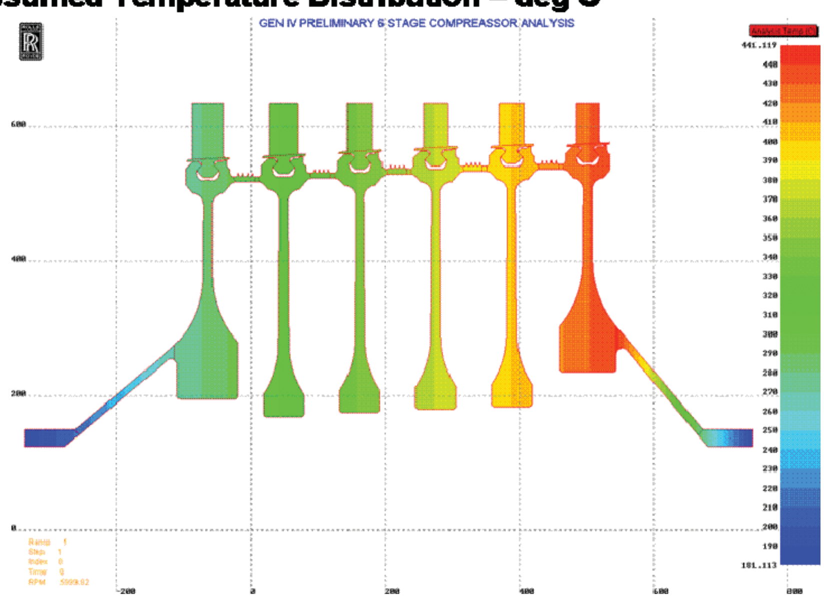
Assessment Summary

REW Material Data at 434 deg C (Mpa)		
UTS*	1180	Mpa
UTS	1268	Mpa
0.2*	965	Mpa
0.2	1076	Mpa
Allowable disc burst stress	1106	Mpa
Allowable rim peel stress	1106	MPa
Allowable 0.1% creep stress	900	MPa
Peak bore hoop stress (Max Speed - 6000 RPM)	688	MPa
AW-HMS (Max Speed - 6000 RPM)	469	MPa
Worst Mean Radial Stress (Max Speed - 6850 RPM)	374	MPa
AW-HMS (1.20 * Max Speed)	675	MPa
Worst Mean Radial Stress (1.20 * Max Speed)	539	MPa
Burst margin RF	1.64	
Rim peel margin RF	2.05	
Creep RF	1.31	

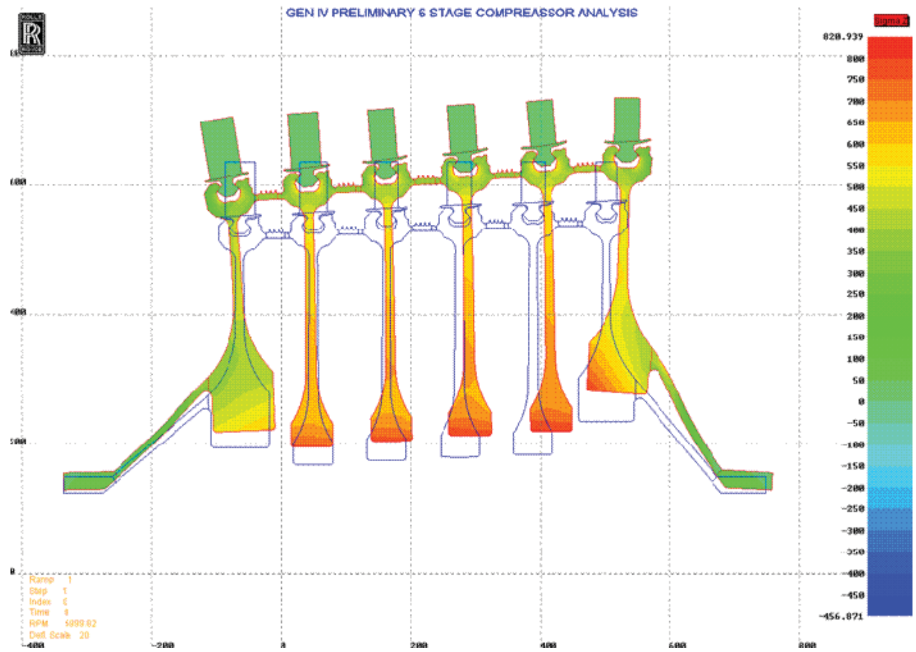
<u>Six Stage Compressor analysis</u>

Compressor, materials, and mechanics

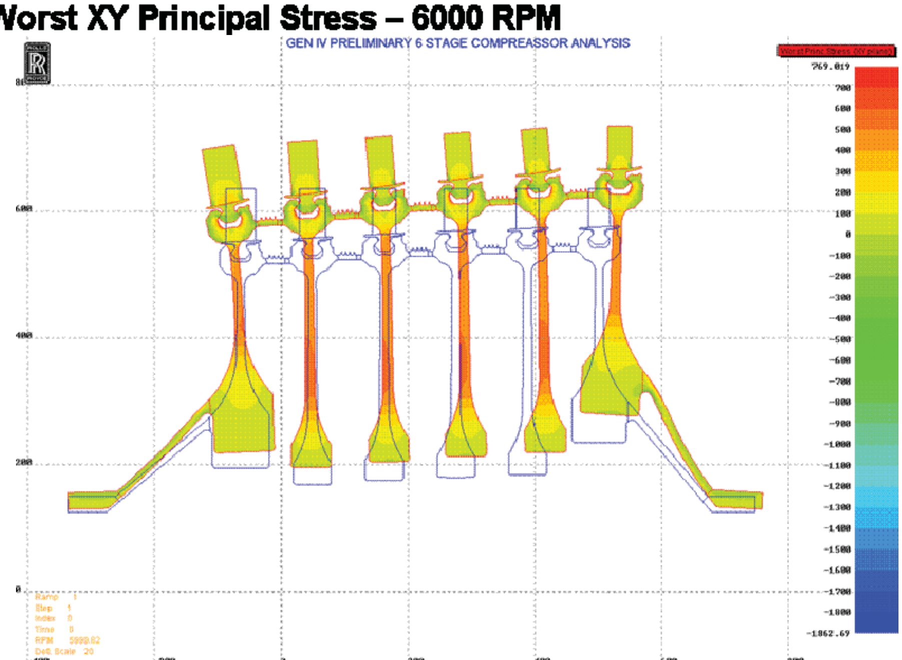
Assumed Temperature Distribution – deg C



Compressor, materials, and mechanics
Hoop Stress – 6000 RPM



Compressor, materials, and mechanics
Worst XY Principal Stress – 6000 RPM



Compressor, materials, and mechanics

Assessment Summary – Modified Disc Profiles

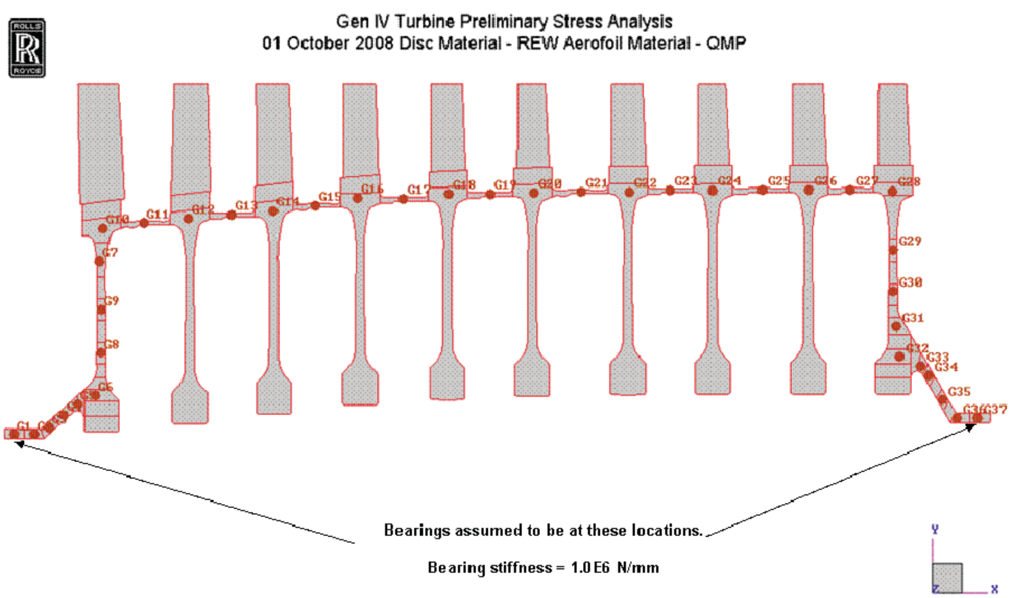
STAGE		1	2	3	4	5	6
REW Material Data	Assumed Temp (°C)	280	310	340	370	400	430
UTS*	MPa	1206	1198	1193	1189	1185	1181
UTS	MPa	1299	1286	1283	1278	1273	1269
0.2*	MPa	981	977	974	971	968	964
0.2	MPa	1096	1092	1088	1084	1080	1076
Allowable disc burst stress	MPa	1127	1122	1118	1115	1111	1107
Allowable rim peel stress	MPa	1127	1122	1118	1115	1111	1107
Allowable 0.1% creep stress (1300 HOURS - 550 DEG C)	MPa	730	730	730	730	730	730
Peak bore hoop stress (Max Speed - 6000 RPM)	MPa	619	661	662	677	695	606
Peak worst XY principal stress (Max Speed - 6000 RPM)	MPa	621	576	570	572	577	613
AVHMS (Max Speed - 6000 RPM)	MPa	431	516	517	526	542	516
Worst Mean Radial Stress (Max Speed - 6660 RPM)	MPa	501	562	543	541	552	465
Mean Von Mises stress at bore	MPa	571	666	661	767	693	685
Mean Von Mises stress at diaphragm	MPa	465	569	566	561	571	517
AVHMS (1.20 * Max Speed)	MPa	621	734	744	760	780	743
Worst Mean Radial Stress (1.20 * Max Speed)	MPa	721	765	762	778	795	696
Burst margin RF		1.82	1.53	1.50	1.47	1.42	1.49
Rim peel margin RF		1.56	1.41	1.43	1.43	1.40	1.58
Creep RF - Disc bore		1.28	1.12	1.10	0.95	1.05	1.10
Creep RF - Diaphragm		1.51	1.31	1.31	1.30	1.28	1.41

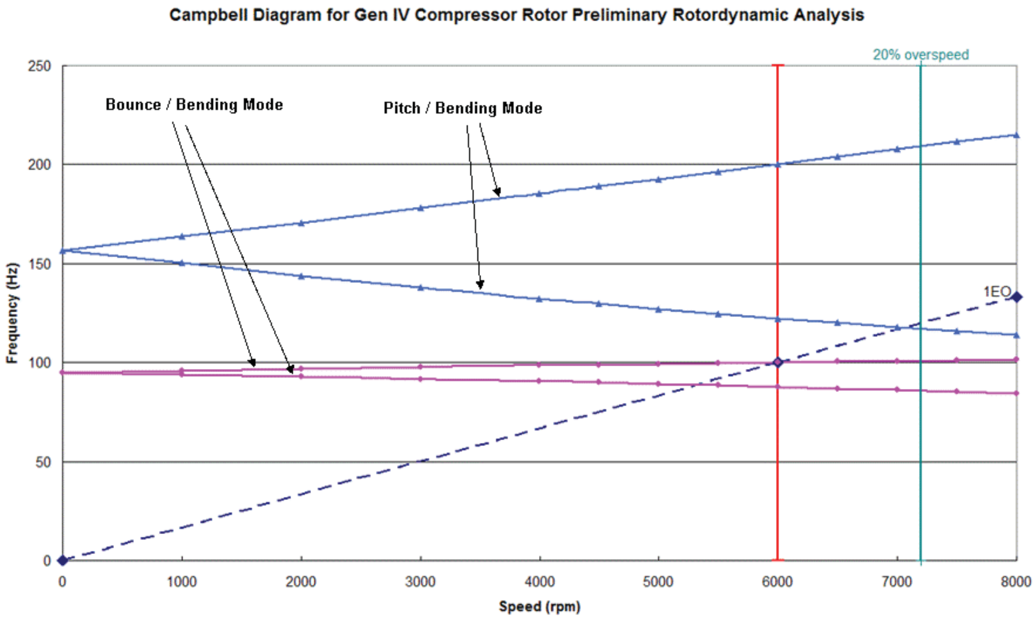
Note that creep data is available for temperatures above 550 deg C only.
Hence all creep RF's are conservative.

Rotor dynamic assessment for Ten Stage Compressor.

Compressor, materials, and mechanics

Rotordynamics



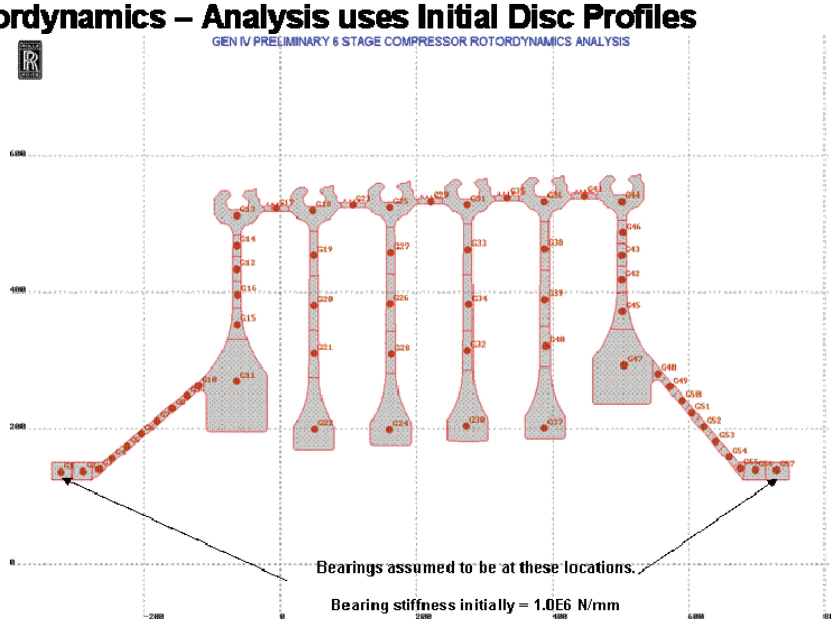


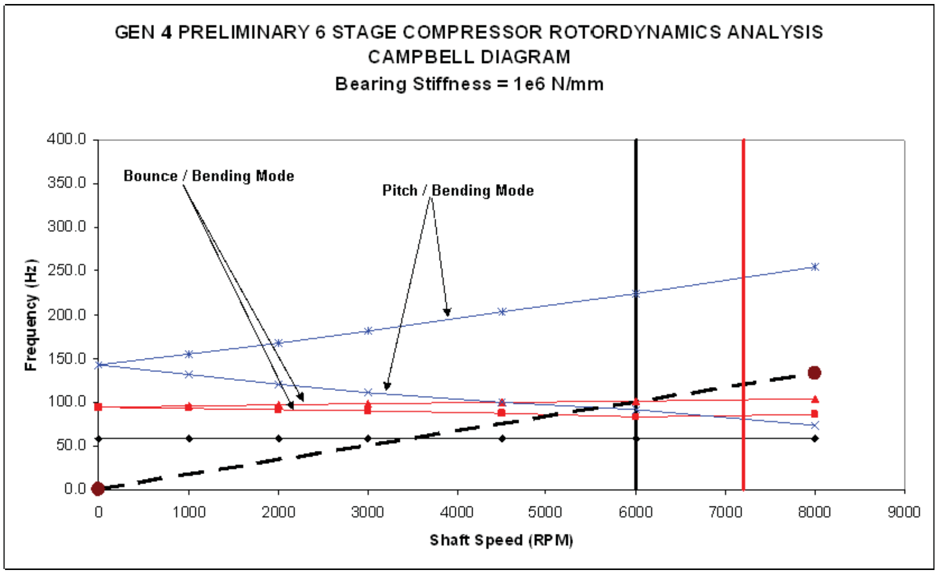
The initial Campbell diagram indicates some potential issues with the bounce/bending mode at 6000 RPM at 100 Hz. This is considered to be as a result of the initial bearing support in the concept which can be refined to improve.

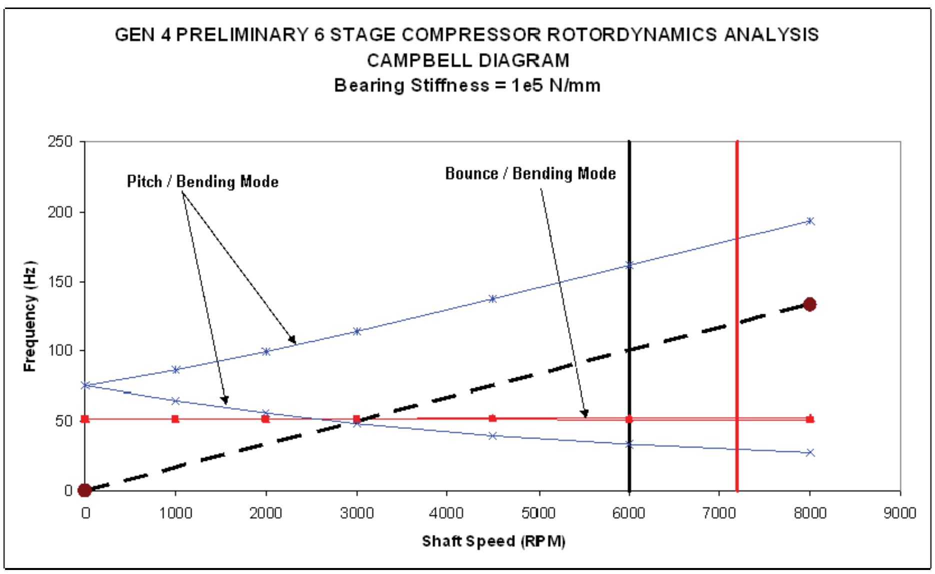
<u>Rotor dynamic assessment for Six Stage Compressor</u>

Compressor, materials, and mechanics

Rotordynamics – Analysis uses Initial Disc Profiles







Refinement of the bearing stiffness gives a more acceptable bounce/bending mode performance which should be added in to the design refinements in future development of the design in any follow on work.

13.4.3 *Conclusion from this stage of activity for Compressor design.*

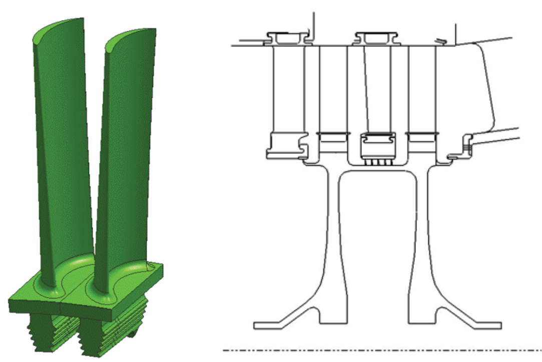
- The design conclusion is that the Six Stage solution is preferable for performance reasons.
- A mechanical concept is viable based on the Rolls-Royce Trent technology and design 'architecture'.
- INCO 718 material is preferable for both blades and disc.
- Acceptable steady state stresses are achievable with the 13000 hour life requirement at 6000PRM and temperatures and pressures identified in the cycle.
- Certain elements of the design such as the Compressor drive arms and the bearing stiffness can be refined, but the concept for this stage of the design programme is acceptable.
- From the assessment of the Polar Inertia to Diametral Inertia relationship ($I_P/I_{D)}$ relationship the compressor ideally should be lengthened slightly. The addition of a balance piston for rotor thrust compensation would also help to improve the result. These features should be included as part of the next stage of the programme to refine the design concept.

13.5 Turbine.

13.5.1 *Turbine concept design layout.*

The aerodynamic design for the Turbine favours a two stage solution operating at 6000RPM. A mechanical study has been completed with Turbine inlet temperatures from 750°C to 850°C, with an expected centrifugal load of 226MN based on the calculated weights and speed, combined with a life of 13000 hours. The Turbine blade material is expected to be 50°C lower than the Turbine Entry Temperature. Considerations have been given to the materials and stresses generated associated with 0.1% and 0.2% creep. In this operating environment 0.2% creep is considered as acceptable.

The Turbine blade design takes the aerodynamic and thermodynamic requirements defined from the cycle studies to give a 0.65m outer annulus diameter and a 0.4m inner annulus diameter. A Rolls-Royce Trent technology approach has been taken fixed by a typical large fan Gas Turbine root. A total of 79 blades have been applied to both stages of the turbine to achieve the performance required resulting in the design concept as shown for the blades and the two stage turbine.



To determine the most suitable materials consideration has been given to materials from the aerospace and non aerospace gas Turbines sector. A review of material available however has indicated that the alloy properties for Turbines operating in the temperature range required are primarily from Aerospace origins.

Ideally for lower cost solutions, un-cooled blades solutions or blades without complex thermal barrier coatings should be identified.

13.5.2 *Materials.*

A range of material solutions based on either directionally solidified (DS) cast blades or single crystal cast blades have been assessed in relation to the temperature operating range for Stress against creep performance.

Materials	Max. Operating Temperature (°C)
IN713LC MSRR7048	950

M002CC MSRR7080	1050
IN738 (Non HIP) MSRR7125	950
IN738 (HIP) MSRR7125	950
IN738LC (Non HIP) MSRR7168	950
IN738LC (HIP) MSRR7168	950
M002DS MSRR7150	1050
SRR99 MSRR7210	1050
RR2000 MSRR7225	1050
CMSX4 MSRR7248	1150
CMSX4 (HIP) MSRR7257	1150
RR3000 MSRR7260	1175

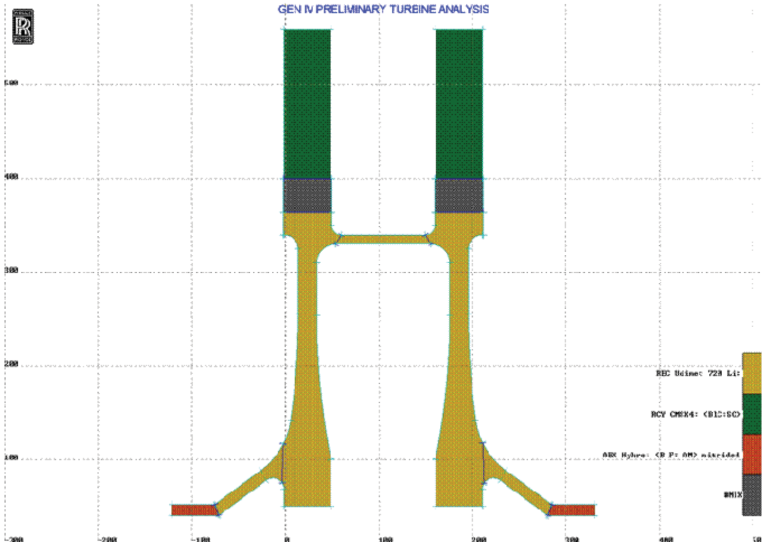
Blade temp v materials	**750°C**	**800°C**	**850°C**
Directional Solidification blades	M002CC M002DS	M002DS M002CC Marginal solution.	
Single Crystal blades	SRR99 CMSX4 (HIP) CMSX4 RR3000	SRR99 CMSX4 (HIP) CMSX4 RR3000	CMSX4 CMSX4 (HIP) RR3000

If 0.1% creep is considered essential only RR3000 can be applied if un-cooled blades are to be used.

Above 800°C CMSX4 or RR3000 blades are suitable if un-cooled, but CMSX 4 would be less costly. The use of DS blades is marginal at this temperature range.

The disc material selected for these temperatures and loads is Udimet 720 Li which would be a cooled design, CMSX4 blades and ABX Hykro Nitriding on the end of the discs.

Assumed Materials

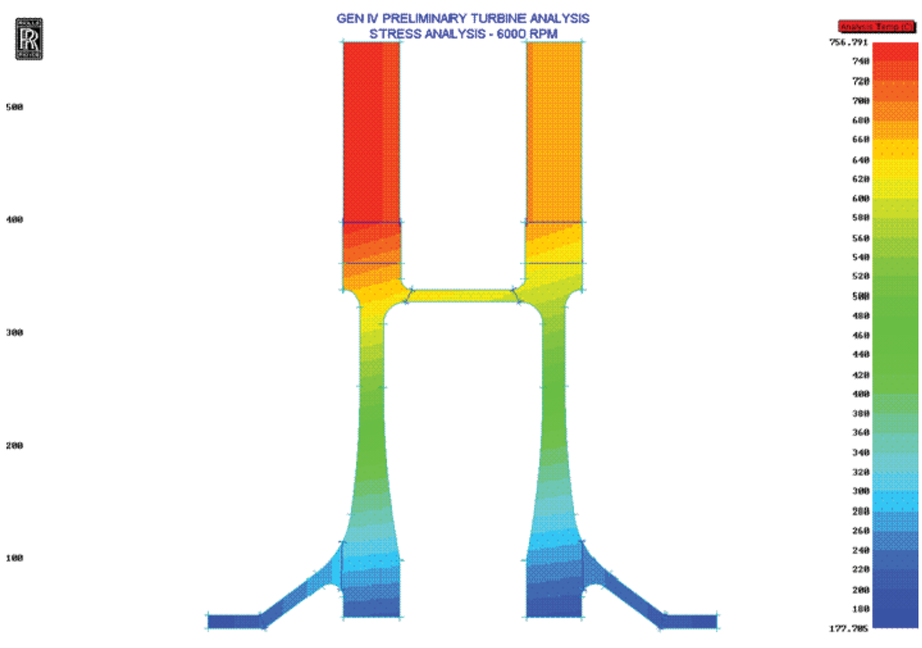


13.5.3 *Turbine thermal distribution applied, and stress analysis*

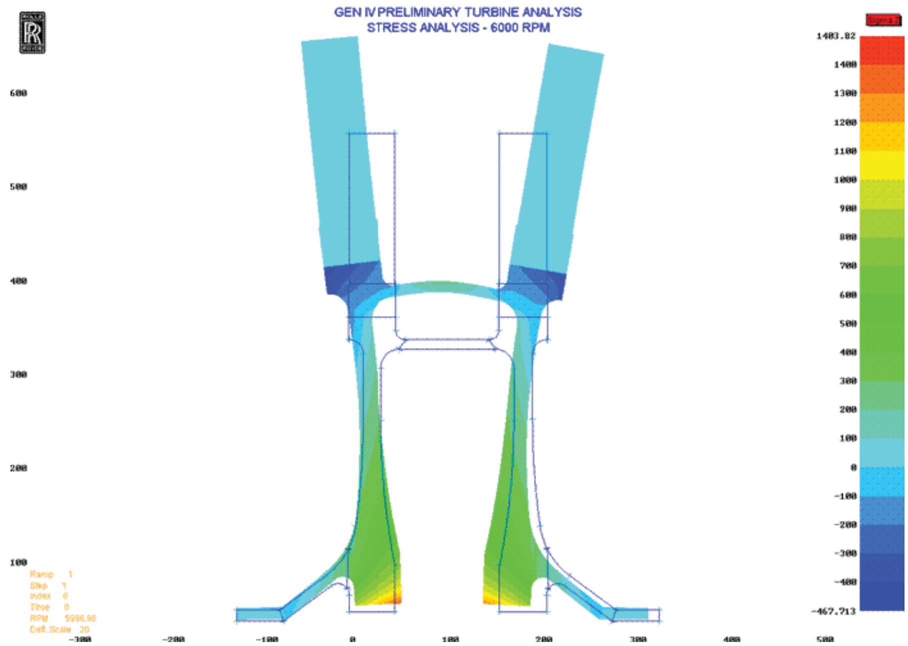
The thermal distribution and stress analysis has produced the following results operating at 6000RPM with the anticipated loads.

Turbine, materials, and mechanics

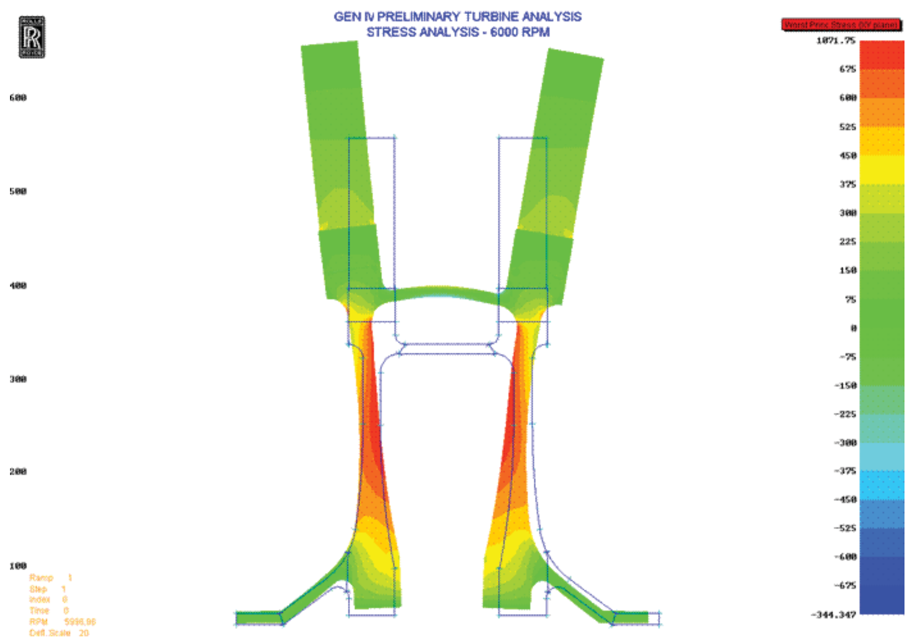
Assumed Temperature Distribution – deg C



Turbine, materials, and mechanics
Hoop Stress – 6000 RPM



Turbine, materials, and mechanics
Worst XY Principal Stress – 6000 RPM



Turbine, materials, and mechanics

Assessment Summary

REC Material Data	Assumed Temp (°C)	400	500	600	700	800	
UTS*		1396	1389	1343	1147	836	MPa
UTS		1526	1518	1468	1254	914	Mpa
0.2*		992	969	976	948	705	Mpa
0.2		1084	1081	1067	1037	772	Mpa
Allowable disc burst stress		1113	1111	1097	1054	781	Mpa
Allowable rim peel stress		1113	1111	1097	1054	781	Mpa
Allowable 0.1% creep stress		1303	1303	703	242	0	MPa
Peak bore hoop stress (Max Speed - 6000 RPM)			1403				MPa
Peak worst XY principal stress (Max Speed - 6000 RPM)				606			MPa
AWHMS (Max Speed - 6000 RPM)				308			MPa
Worst Mean Radial Stress (Max Speed - 6850 RPM)				674			MPa
Mean Von Mises stress at bore			1147				MPa
Mean Von Mises stress at diaphragm				596			MPa
Mean Von Mises stress at rim					355		MPa
AWHMS (1.20 * Max Speed)				444			MPa
Worst Mean Radial Stress (1.20 * Max Speed)				971			MPa
Burst margin RF				2.47			
Rim peel margin RF				1.13			
Creep RF - Disc bore	500		1.14				
Creep RF - Diaphragm	600			1.18			
Creep RF - Rim	700				0.68		

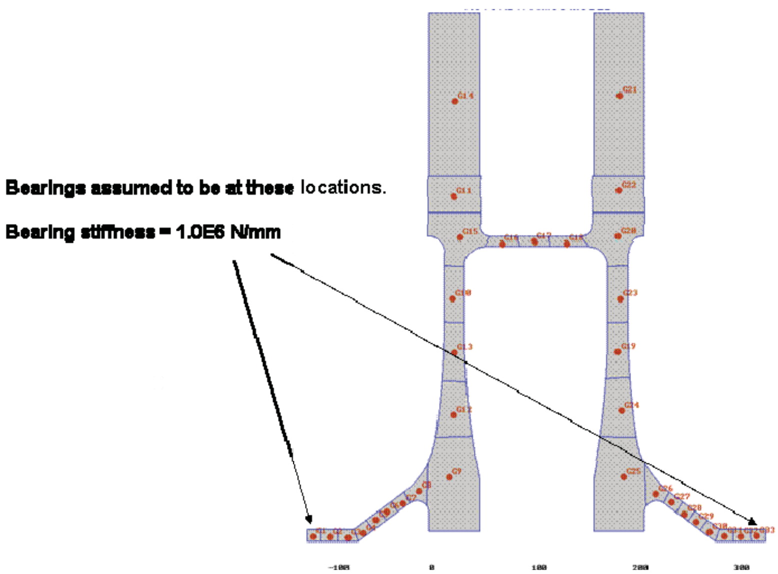
The initial stress results with the loads and temperatures identified indicate that the Turbine concept is viable. Some consideration may be needed for disc rim cooling in further design development and the drive arms can be further optimised.

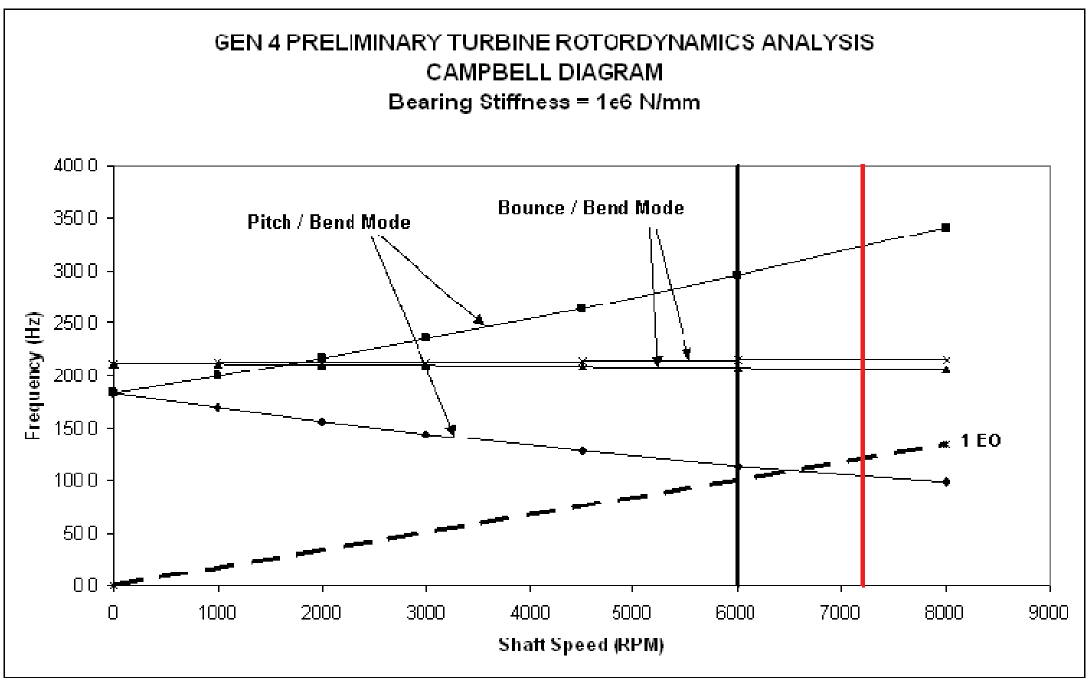
Further refinement of the two stage Turbine would be required, as the design progresses in to more detailed assessment of the solution in the next stage of this programme.

13.5.4 *Turbine rotor dynamic assessment.*

The rotor dynamic assessment shown below has provided a set of results to confirm the concept is viable with no significant vibration or harmonics issues at this concept stage.

Rotordynamics





13.5.5 *Conclusion from this stage of activity for Turbine design.*

- A two un-shrouded stage design is mechanically viable using blade design features comparable to current generation aerospace and energy sector Gas Turbines.

- An un-cooled single crystal alloy is suitable for the expected design conditions to achieve the required design life based on a creep requirement of 13000 hours..
- A cooled Udimet 720 disc is a suitable concept to support the blades in these conditions.

13.6 Secondary versus Primary circuit considerations.

This concept design and analysis has focussed on the Gas Turbine being part of the secondary cycle. The safety implications of this allow significant benefits to the design, operation and maintenance of the Gas Turbine which are comparable to a normal modular Gas Turbine using technology comparable to those used currently in either energy or aero sectors. It is only in the event of a significant failure of the heat exchanger that any potential radiation may enter the gas Turbine which would require different recovery techniques, and require any safety considerations.

If the Gas Turbine were part of a primary circuit than it is assumed the temperatures would need consideration as part of a wider overall thermodynamic cycle study, but the implications of radiation and the operation and handling of the Gas Turbine would be significant. Further consideration would also be needed if a direct cycle were to be used because the implications with radioactive He (3 or 4 He with extra electrons on the p or d orbital) from nuclear fission.

Consideration would be needed for Gas Turbine handling with radiation, the implications of storage and half life in relation to potential activity to either safely dispose, clean and refurbish components and replacement of others where the life has expired.

The costs associated with such an operation would need assessment with potentially more complex design solutions creating additional development and unit production costs plus more complex maintenance repair and overhaul processes and storage facilities. Hence it is likely the overall life cycle costs would be higher than the secondary circuit system proposed which has considerably more cost effective energy generation features for significantly lower cost.

13.7 Overall conclusions.

There are considerable advantages with this Gas Turbine solution based on a secondary circuit.

The requirements specified by the aero and thermo dynamic cycles can be achieved by the mechanical design concepts defined in this study. The technology being applied is generally comparable to those technologies currently applied in Gas Turbine design, operation and support. Hence the design work would be refinements of the concepts presented as the process from concept to production design variant progresses.

Some further assessments of the implications with the working gases specified may be required. Some further consideration would also be recommended of the mechanical design associated with the control and operation of the Gas Turbine as part of a normal technology and design development process with these materials and gases operating within this combined cycle system and the reactor.

13.8 List of papers/references used for the gas and materials research with Gas Turbines.

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[1] Christ, H-J, Schwanke, D, Uihlein, Th, Sockel, HG, *Mechanisms of High-Temperature Corrosion in Helium Containing Small Amounts of Impurities. I. Theoretical and Experimental Characterization of the Gas Phase*, Oxidation of Metals, **Vol. 30**, **1-2**, 1998, pp. 1-26

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[1] Cohen, H, Rogers, GFC, Saravanamuttoo, HIH, Gas Turbine Theory, 4th Ed., Addison Wesley Longman Limited, 1996, pp. 160, 180.