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A Review of The Turbine Cooling Fraction for Very High Turbine Entry Temperature Helium Gas Turbine Cycles For Generation IV Reactor Power Plants

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Abstract

The potential for high Turbine Entry Temperatures (TET) turbines for Nuclear Power Plants (NPPs) require improved materials and sophisticated cooling. Cooling is critical to maintaining mechanical integrity of the turbine for temperatures >1000°C. Increasing TET is one of the solutions for improving efficiency after cycle optimum pressure ratios have been achieved but cooling as a percentage of mass flow will have to increase, resulting in cycle efficiency penalties. To limit this effect, it is necessary to know the maximum allowable blade metal temperature to ensure the minimum cooling fraction is used. The main objective of this study is to analyse the thermal efficiencies of four cycles in the 300 – 700 MW class for Generation IV NPPs, using two different turbines with optimum cooling for TETs between 950°C - 1200°C. The cycles analysed are Simple Cycle (SC), Simple Cycle Recuperated (SCR), Intercooled Cycle (IC) and Intercooled Cycle Recuperated (ICR). Although results showed that deterioration of cycle performance is lower when using improved turbine material, the justification to use optimum cooling improves the cycle significantly when a recuperator is used. Furthermore, optimised cooling flow and the introduction of an intercooler improves cycle efficiency by >3%, which is >1% more than previous studies. Finally, the study highlights the potential of cycle performance beyond 1200°C for IC. This is based on the IC showing the least performance deterioration. The analyses intend to aid development of cycles for deployment in Gas Cooled Fast Reactors (GFRs) and Very High Temperature Reactors (VHTRs).

Keywords: Gen IV, Efficiency, Specific Work, Cycle, Nuclear Power Plants, Performance, Simple, Intercooled.

Nomenclature

Notations

A	Area (m^2)
Cp	Spec. Heat of Gas at Constant Pressure (J/kg K)
CW	Compressor Work (W)
m	Mass Flow Rate (kg/s)
Q	Reactor Thermal Heat Input
q	Heat Flux (W/m^2)
P	Pressure (Pa)
PR	Pressure Ratio
SW	Specific Work/Power Output (W/Kg/s)
T	Temperature (K or °C)
TR	Temperature Ratio (T_4 / T_1; expressed in Kelvin)
TW	Turbine Work (W)
W	Work (W)
UW	Useful Work (W)

Greek Symbols

γ	Ratio of Specific Heats
Δ	Delta, Difference
ε	Effectiveness (Heat Exchanger; cooling)
η	Efficiency

Subscripts

$blade$	Turbine Temperature (also known as Blade Temp.)
c	Compressor
c_{in}	Compressor Inlet
c_{out}	Compressor Outlet
$cool$	Cooling
$coolant$	Compressor Exit Coolant
e	Power for Electrical Conversion
gas	Turbine Entry Temperature
he	Helium

he_{min}	Helium with minimum gas conditions
ic	Intercooled Cycle; intercooled coefficient
is_c	Isentropic (Compressor)
is_t	Isentropic (Turbine)
MHR	Reactor (Heat Source)
MHR_{in}	Reactor (Heat Source) Inlet
MHR_{loss}	Reactor (Heat Source) Pressure Losses
MHR_{out}	Reactor (Heat Source) Outlet
pc_{in}	Precooler Inlet (also applicable to intercooler)
pc_{loss}	Precooler Pressure Losses (same as above)
pc_{out}	Precooler Outlet (same as above)
re	Recuperator
re_{cold}	Recuperator cold side
re_{hot}	Recuperator hot side
re_{HPloss}	Recuperator High Pressure Losses
re_{LPloss}	Recuperator Low Pressure Losses
re_{real}	Recuperator Real (specific heat transfer)
re_{max}	Recuperator Max (specific heat transfer)
th	Thermal Power
t	Turbine
t_{out}	Turbine Outlet
t_{in}	Turbine Inlet

Superscripts

'	Recuperator inlet conditions

Abbreviations

C	Compressor
CH	Precooler
COT	Core Outlet Temperature
DP	Design Point
GEN IV	Generation Four
GFR	Gas-Cooled Fast Reactor
GIF	Generation IV International Forum
HP	High Pressure
HE	Recuperator
HPC	High Pressure Compressor
IC	Intercooled Cycle
ICR	Intercooled Cycle Recuperated
LP	Low Pressure
LPC	Low Pressure Compressor
M	Mixer (Figure 6)
NPP	Nuclear Power Plant
NTU	Number of Transfer Units
OPR	Overall Pressure Ratio
R	Reactor
RPV	Reactor Pressure Vessel
S	Splitter (Figure 6)
SC	Simple Cycle
SCR	Simple Cycle Recuperated
TET	Turbine Entry Temperature
VHTR	Very High Temperature Reactor

Introduction

Generation IV reactor performance is key to the design of Nuclear Power Plants (NPPs), with one of the key aspects being the improvement of cycle thermal efficiency in comparison to the incumbent designs [1]. Deriving better efficiencies is critical to the economics of the cycle. One key method of achieving improved efficiencies is to raise the TET in line with optimum pressure ratios. However, this requires knowledge of the maximum allowable turbine blade metal temperatures to deliver the minimum amount of cooling. The objective is to conduct a thermodynamic study using a performance simulation tool to analyse the cooling requirements of 4 different cycles at TETs in the range of 950°C - 1200°C, using 2 different turbines with different blade metal temperatures. The outcomes include demonstrating the effect of increased TET and optimised cooling flow on the cycle efficiencies for cycles in closed Brayton direct configuration, which use helium as the working fluid.

Generation IV (Gen IV) Systems

The Gen IV systems of interest are the Gas-Cooled Fast Reactor System (GFR) and Very-High-Temperature Reactor System (VHTR). The GFR is helium cooled with a high temperature reactor and a fast spectrum nuclear core. The Core Outlet Temperature (COT) of 850-950°C is made possible by utilising an efficient Brayton cycle. The benefits of using helium as a working fluid include single phase cooling in all circumstances, chemical inertness and neutronic transparency [2]. The VHTR is a high temperature thermal reactor, which is also helium cooled in gaseous phase and graphite moderated in the solid state. At a COT of 750-1000°C, a stable coolant such as helium is necessary to avoid induced chemical reaction with the moderator. Furthermore, graphite retains good mechanical properties at a high temperature. Several demonstrator projects planned for the GFR and VHTR are currently in the viability phase – this relates to testing of basic concepts or in the performance phase. Descriptions of demonstrator reactors are discussed in [1].

Applicable Cycles

The cycles of concern are the Simple Cycle (SC), Simple Cycle Recuperated (SCR), Intercooled Cycle (IC) and the Intercooled Cycle Recuperated (ICR). The importance of plant economics to cycle configuration means that the SC is not the cycle of choice for NPPs. Nonetheless, it has been included for completeness and comparison purposes. All of the aforementioned cycles require a compressor and a turbine as part of the turbomachinery. The compressor work is lower than the turbine work, thus the useful work can be used to drive the generator load but due to component inefficiencies, the compression and expansion phases are not isentropic. As a result, heating and cooling of the cycle is not achieved at constant pressure hence losses are observed in the cycle. The losses translate into more work input required for the compression process due to increase in temperature and results in a higher exit temperature. The heat addition into the cycle is not isobaric, which reduces total gas exit pressure, thus the total power extraction possible is reduced due to the reduced gas exit pressure and the reduced component efficiencies. The turbine exit heat is typically hotter than expected, which makes the compression inlet temperature hotter than ideal.

A precooler is a common component utilised within the applicable cycles in addition to the turbomachinery. The inclusion of a precooler ensures that the working fluid can be cooled by a cooling medium (usually seawater) at the compressor entry to achieve the necessary cycle inlet temperature. This reduces the compressor work but reduces the compressor exit temperature, which will increase the input thermal power. Due to the reactor thermal power being fixed for a given COT, the precooler alone will not yield the specific work required for the NPP, which devalues the economics of the plant. This is mitigated differently, depending on the cycle. For the SCR and ICR, the recuperator is introduced, whereby heat from the turbine outlet gas is used to preheat the working fluid downstream of the compressor, thus raising the temperature to reduce the amount of thermal heat input into the cycle, which positively impacts cycle efficiency.

The IC does not make use of a recuperator like the SCR and ICR, but the IC and ICR use an intercooler and a second compressor, which is downstream of the first compressor. With regard to the ICR, improving the specific and useful work in the ICR requires a reduction of the compressor work. The working fluid downstream of the first compressor is reduced to the same inlet temperature as the first compressor in the intercooler, prior to entry into the second compressor. The Pressure Ratio (PR) of each compressor in the ICR is determined by the square root of the Overall Pressure Ratio (OPR), and ensures an even compression split with negligible reduction in pressure at that stage. The IC also has the same requirement to improve the specific and useful work. The pressure ratio split between the compressors in the IC is not even but the split ensures that the downstream compressor (HPC) delivers the working fluid to the reactor at a much higher pressure and temperature than the upstream compressor (LPC), in the absence of a recuperator. The IC OPR is higher than the OPRs of the SCR and ICR and is required to be at its highest with an optimum split ratio between the LPC and HPC in order to achieve the maximum cycle efficiency, taking into account component losses.

Higher efficiency of the SC, which has neither an intercooler nor a recuperator, relies on obtaining high efficiencies of the compressor and turbine and obtaining minimum pressure losses in the precooler, reactor and flow ducts.

The benefits of changing from air to helium in a nuclear gas turbine, including the thermodynamic consequences, have been extensively covered in [3]. Although the study which is also documented in [4] and [5] focuses on off-design, control and transient operational modes of a helium nuclear gas turbine, it provides good bases for future off-design analyses, which will be applicable to the SCR, ICR and IC configurations.

Turbine Cooling

Gas turbine development, which allow operation in hot temperature conditions have been made possible due to advancements in cooling technology. Figure 1, which is based on data from [6], shows increases in TET as a function of progression in cooling technology. Sophisticated cooling technologies such as film impingement convection, are employed to bring about lower temperatures. Cooling requirements are dependent on blade life, material and cooling technology, radial temperature distribution, coolant temperature, blade reaction and centrifugal stresses.

Successive heat transfer lies in the use of multi passages with turbulators within the blade and film cooling to maintain the blade metal temperature for the turbine blade. Calculating the actual blade metal temperature requires knowledge of the cooling effectiveness (a non-dimensional parameter), the coolant temperature from the compressor exit and the TET. The cooling effectiveness (usually less than unity) defines the effect of the cooling technology (heat transfer capability) and determines the amount of cooling flow required to maintain the necessary blade metal temperature.

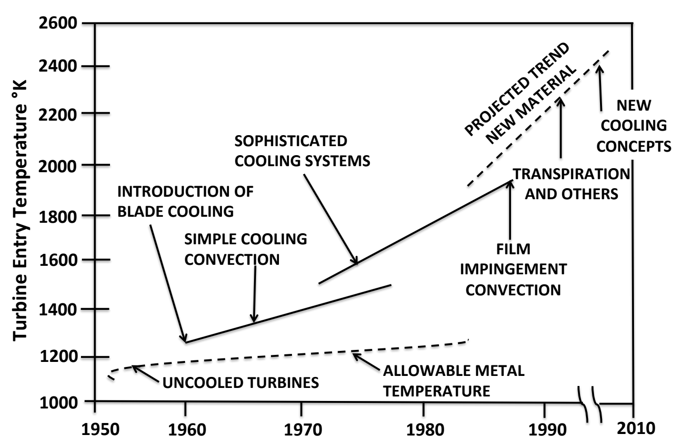


Figure 1 – Cooling Technologies [6]

Modelling of Nuclear Power Plants and Performance Simulation Tool

Figures 2 and 3 respectively illustrate typical schematics of the SCR and the ICR NPPs; figures 4 and 5 illustrate schematics of the SC and the IC respectively. Table 1 provides the key Design Point (DP) values for modelling, using the performance simulation tool. The TET was varied between

950-1200°C during the analyses. The optimum OPRs were unknown for the cycles because they were dependent on the cooling flows. The cooling flows were also unknown because they were dependent on the pressure at which the cooling air is bled out of the compressor, which is dependent on the OPR. Solving for 2 unknowns, whereby the OPR required solving (taking into account the significant range between 1 – 20) was a complex calculation task. The complexity was exacerbated when considering the IC has uneven split pressure ratios for the LPC and HPC, which were also unknown. The calculations, modelling and simulation were made possible through the use of a FORTRAN based tool developed as part of this study. The thermodynamic equations implemented within the code environment are described in the proceeding sections for steady state DP calculations against each component and for the applicable cycles. Tables 2 and 3 lists the OPRs and cycle power output for each cycle at the various temperatures.

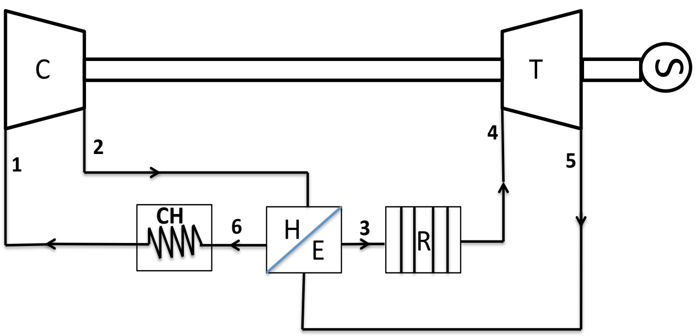


Figure 2 – Typical Simple Cycle with Recuperator (SCR) [7]

Compressor

Prerequisite parameters for performance design considerations of the compressor include the compressor pressure ratio, compressor inlet conditions (temperature, pressure and mass flow rate), component efficiency and the working fluid gas properties (Cp and γ).

The compressor outlet pressure (in Pa) is:

$$P_{c_{out}} = P_{c_{in}} \cdot PR_c \quad (1)$$

The isentropic efficiency of the compressor is $\frac{T_{rise_{ideal}}}{T_{rise_{actual}}}$ and is also indicative of the specific work input or total temperature increase.

Thus, the temperature (°C) at the exit can be derived from the inlet temperature, pressure ratio, isentropic efficiency and ratio of specific heats:

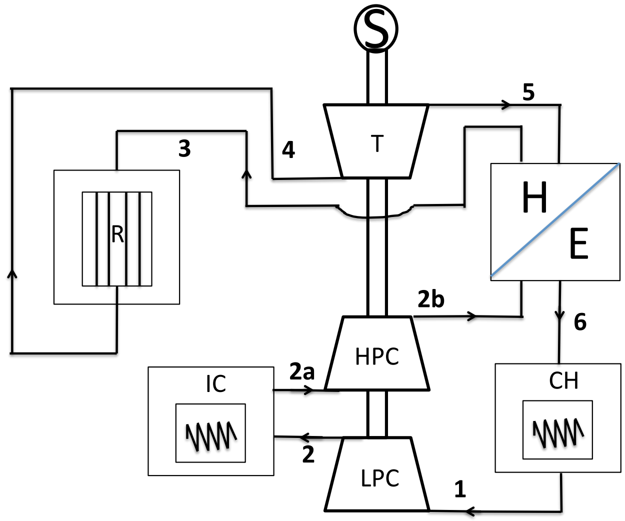


Figure 3 – Typical Intercooled Cycle with Recuperator (ICR) [8]

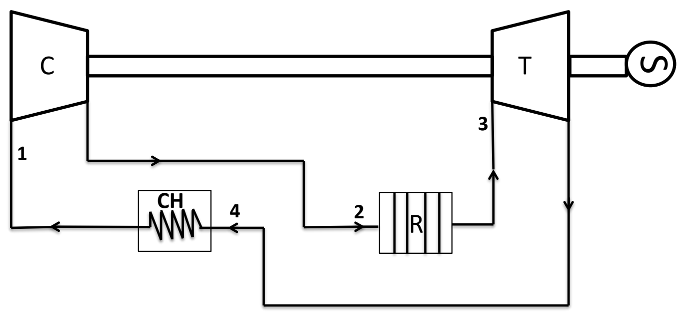


Figure 4 – Typical Simple Cycle without Recuperator (SC)

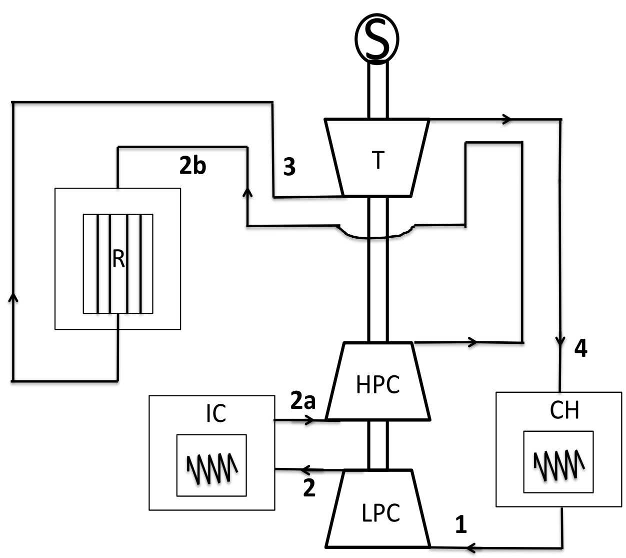


Figure 5 – Typical Intercooled Cycle without Recuperator (IC)

$$T_{c_{out}} = T_{c_{in}} \cdot \left[1 + \frac{\left(\frac{P_{c_{out}}}{P_{c_{in}}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{is_c}} \right] \quad (2)$$

Table 1 – Input Values for Modelling

Inputs	Values	Units
Inlet Temp. (T1)	**28**	**°C**
Inlet Pressure (P1)	**3.21**	**MPa**
Mass flow rate at inlet (m1)	**410.4**	**Kg/s**
Compressor Efficiency (Isentropic)*	**90**	**%**
Turbine Efficiency (Isentropic)*	**94.5**	**%**
Recuperator Effectiveness (SCR & ICR only)*	**96**	**%**
Pressure Loss (Precooler)	**2.5**	**%**
Pressure Loss (Intercooler ICR & IC only)	**2.5**	**%**
Pressure Loss (Reactor)	**2**	**%**
Pressure Loss (Recup. HP side, SCR & ICR only)	**6 combined**	**%**
Pressure Loss (Recup. LP side, SCR & ICR only)		
Reactor Cooling flow (% of Mass flow rate)	**0.25**	**%**

**Compressor and Turbine efficiencies and Recuperator effectiveness are based on technological improvements in* [9]

Table 2 – Optimum OPRs and Power at Various TETs (950 - 1050°C)

SIMPLE CYCLE						
TET (°C)	**950**		**1000**		**1050**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	9.6	9.8	10	10.6	10.4	11.4
Output Power (MW)	389	400	423	431	452	461

SIMPLE CYCLE RECUPERATED						
TET (°C)	**950**		**1000**		**1050**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	2.2	2.2	2.4	2.2	2.4	2.4
Output Power (MW)	322	325	375	348	399	405

INTERCOOLED CYCLE						
TET (°C)	**950**		**1000**		**1050**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	13.09	13.77	14.58	14.62	15.3	16.38
Output Power (MW)	475	475	534	517	550	559

INTERCOOLED CYCLE RECUPERATED						
TET (°C)	**950**		**1000**		**1050**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	2.6	2.6	2.8	2.6	2.8	2.8
Output Power (MW)	406	410	460	438	488	494

The mass flow rate (kg/s) at inlet is equal to the mass flow rate at outlet as there are no compositional changes:

$$m_{c_{out}} = m_{c_{in}} \quad (3)$$

The compressor work (W) is the product of the mass flow rate, specific heat at constant pressure and the temperature delta:

$$CW = m_c \cdot Cp_{he} \cdot (\Delta T_c) \quad (4)$$

Table 3 – Optimum OPRs and Power at Various TETs (1100 - 1200°C)

SIMPLE CYCLE						
TET (°C)	**1100**		**1150**		**1200**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	10.6	12	10.8	12.6	11	12.8
Output Power (MW)	484	496	511	526	535	567

SIMPLE CYCLE RECUPERATED						
TET (°C)	**1100**		**1150**		**1200**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	2.4	2.4	2.6	2.4	2.6	2.6
Output Power (MW)	423	430	479	455	503	514

INTERCOOLED CYCLE						
TET (°C)	**1100**		**1150**		**1200**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	15.66	17.1	16.72	18.18	16.91	19.08
Output Power (MW)	590	604	632	639	670	675

INTERCOOLED CYCLE RECUPERATED						
TET (°C)	**1100**		**1150**		**1200**	
Blade	Blade A	Blade B	Blade A	Blade B	Blade A	Blade B
Overall Pressure Ratio	3	2.8	3	2.8	3.2	3
Output Power (MW)	543	523	572	552	628	611

whereby $\Delta T_c = T_{c_{out}} - T_{c_{in}}$ (5)

Bypass splitters (S in figure 6) are incorporated within the performance simulation tool to allow for compressed coolant to be bled for reactor and turbine cooling.

Turbine

Prerequisite parameters of the turbine include the turbine inlet conditions (temperature, pressure and mass flow rate), the pressure at outlet, component efficiency and the working fluid gas properties (Cp and γ).

The temperature (°C) at the outlet is derived from the following expression:

$$T_{t_{out}} = T_{t_{in}} \cdot \left\{ 1 - \eta_{is_t} \left[1 - \left(\frac{P_{t_{out}}}{P_{t_{in}}}\right)^{\frac{\gamma-1}{\gamma}} \right] \right\} \quad (6)$$

As with the compressor, eqs (3) and (4) also apply to the turbine for mass flow rate (kg/s) conditions and turbine work (W) but:

$$\Delta T_t = T_{t_{in}} - T_{t_{out}} \quad (7)$$

A mixer (M in figure 6) is incorporated within the performance simulation tool to allow for the coolant to mix with the hot gas to simulate turbine cooling.

Recuperator (SCR and ICR only)

The calculation method for the rate of heat transfer is based on the Number of Transfer Units (NTU) method, which has been documented by [10] and applied for complex cross flow heat exchangers by [11]. The algorithm in the code ensures satisfactory results and numerical stability.

Prerequisite parameters include the recuperator effectiveness, hot and cold inlet conditions (pressure and temperature) and the delta pressures due to losses at the high and low pressure sides.

Effectiveness of the recuperator is given as:

$$\varepsilon_{re} = \frac{q_{re_{real}}}{q_{re_{max}}} \tag{8}$$

The maximum amount of heat flux (W/m^2) of the recuperator $q_{re_{max}}$, must consider the hot and the cold inlet conditions. It must also consider the minimum specific heat because it is the fluid with the lowest heat capacity to experience the maximum change in temperature. This is expressed as:

$$q_{re_{max}} = \frac{Cp_{he_{min}} \cdot \left(T'_{re_{hot}} - T'_{re_{cold}}\right)}{A} \tag{9}$$

and the real heat flux (W/m^2) is:

$$q_{re_{real}} = \frac{Cp_{he_{hot}} \cdot \left(T'_{re_{hot}} - T_{re_{hot}}\right)}{A} = \frac{Cp_{he_{cold}} \cdot \left(T_{re_{cold}} - T'_{re_{cold}}\right)}{A} \tag{10}$$

With helium as the working fluid, Cp is considered to be constant, thus $Cp_{he_{min}} = Cp_{he_{cold}} = Cp_{he_{hot}}$ in the energy balance equation. The temperatures at the hot and cold ends can be obtained when considering eq (10) (either hot or cold sides) and considering an arbitrary effectiveness.

The temperature for the cold end (°C) is then expressed as:

$$T_{re_{cold}} = T'_{re_{cold}} + \left[\varepsilon_{re} \cdot \left(T'_{re_{hot}} - T'_{re_{cold}}\right)\right] \tag{11}$$

With $Cp_{he_{min}} = Cp_{he_{cold}} = Cp_{he_{hot}}$, the energy balance is:

$$\left[m_{re_{cold}} \cdot \left(T_{re_{cold}} - T'_{re_{cold}}\right)\right] = \left[m_{re_{hot}} \cdot \left(T'_{re_{hot}} - T_{re_{hot}}\right)\right] \tag{12}$$

Thus, the hot outlet (°C) is:

$$T_{re_{hot}} = T'_{re_{hot}} - \left[\frac{m_{re_{cold}} \cdot \left(T_{re_{cold}} - T'_{re_{cold}}\right)}{m_{re_{hot}}}\right] \tag{13}$$

With regard to pressures, the exit conditions can be calculated if the pressure drops (%) across the hot and cold sides are known:

$$P_{re_{cold}} = P'_{re_{cold}} \cdot \left(1 - \Delta P_{re_{HPloss}}\right) \tag{14}$$

$$P_{re_{hot}} = P'_{re_{hot}} \cdot \left(1 - \Delta P_{re_{LPloss}}\right) \tag{15}$$

Due to no compositional changes, mass flow rate (kg/s) conditions are:

$$m_{re_{hot}} = m'_{re_{hot}} \tag{16}$$

$$m_{re_{cold}} = m'_{re_{cold}} \tag{17}$$

Precooler and Intercooler

Prerequisite parameters for the precooler and intercooler (ICR and IC only), take into account that the components are upstream of the first and second compressors respectively, thus compressor inlet temperature and pressure are of importance including the pressure losses. The conditions for the precooler are as follows:

$$T_{pc_{out}} = T_{c_{in}} \tag{18}$$

$$P_{pc_{in}} = P_{pc_{out}} \cdot \left(1 + \Delta P_{pc_{loss}}\right) \tag{19}$$

$$m_{pc_{out}} = m_{pc_{in}} \tag{20}$$

With regard to the intercooler, eqs (18), (19) and (20) also apply, but are differentiated for the intercooler. An addition of a second compressor for ICR only, means that the pressure ratio for both compressors is determined as:

$$PR_{ic} = \sqrt[ic]{PR} \tag{21}$$

whereby the ic coefficient denotes the number of intercoolers in the cycle +1, leading to a reduction in the pressure ratio per compressor (ICR only).

Modular Helium Reactor

The helium reactor is a heat source with pressure losses. The prerequisite are the thermal heat input from burning the fuel and the known reactor design pressure losses.

The heat source does not introduce any compositional changes, thus mass flow rate (kg/s) is:

$$m_{MHR_{out}} = m_{MHR_{in}} \tag{22}$$

Pressure taking into account losses (%):

$$P_{MHR_{out}} = P_{MHR_{in}} \cdot \left(1 - \Delta P_{MHR_{loss}}\right) \tag{23}$$

and the thermal heat input (Wt) is:

$$Q_{MHR} = m_{MHR_{in}} \cdot Cp_{he} \cdot \left(\Delta T_{MHR}\right) \tag{24}$$

whereby $\Delta T_{MHR} = T_{MHR_{out}} - T_{MHR_{in}}$ (25)

A mixer (see figure 6) is incorporated within the code to allow for coolant to be mixed with the heated fluid upstream of the reactor to simulate reactor vessel cooling.

Cooling Calculations

Prerequisites to calculate the cooling flow from the compressor exit, which is required for the cycle (cooling flow is taken as a percentage of mass flow rate) are the turbine metal temperature (simply known as blade metal temperature), compressor exit coolant temperature, TET (simply known as gas) and cooling effectiveness. The cooling effectiveness (<1) is expressed as:

$$\varepsilon_{cool} = \frac{(T_{gas} - T_{blade})}{(T_{gas} - T_{coolant})} \tag{26}$$

The cooling effectiveness as a function of the cooling flow (percentage of mass flow rate) has been empirically derived by NASA for various cooling technologies [12],[13],[14] . With regard to the choice of technology, film impingement forced convection is considered the technology for immediate and near term deployment based on current turbine cooling developments (see figure 1). With consideration of application, data from NASA studies were used to define the cooling effectiveness as a function of the cooling flow. The defined cooling conditions were verified against analysis, which featured empirical data for film impingement forced convection as published in [6]. The calculated results were comparable to the empirical results. The calculated results were judged to be satisfactory for this study based on good comparability.

Cycle Calculations

The useful work, specific work and thermal efficiency output values are of interests after executing each set of thermodynamic station parametric calculations. The useful work (We), that is the work available for driving the load is:

$$UW = TW - CW \tag{27}$$

whereby eq (27) is also applicable to the ICR and IC cycles but the CW is the summation of the LPC and HPC work requirements to be delivered by the turbine. The specific work or capacity of the plant (W/kg/s) is:

$$SW = UW/m \tag{28}$$

and the thermal efficiency (%) of the cycle is:

$$\eta_{th} = UW/Q_{MHR} \tag{29}$$

Figure 6 denotes the typical structure of the performance simulation code for the SCR. The structure is interchangeable for SC, ICR and IC but the calculation algorithms are tailored to the conditions driven by the requirements of each individual cycle. The tool was used to match DP conditions of known NPPs in open literature in order to verify its functionality. The matched results were considered satisfactory for the purpose of this study.

Effects of Turbine Cooling on Efficiency and Specific Work

Figure 7 shows the effect on cycle efficiency for SCR and ICR cycles, when 1% turbine cooling is introduced. The performance penalties for cooling are evident, when compared to an uncooled turbine.

Consequentially, the effects of turbine cooling flow are summarised below [15]:

- A parallel loss of turbine work is observed due to reduction in turbine mass flow rate. Mass flow rate effect (loss) is reflected as a reduction in turbine work of ~1%.
- The expansion process due to the cooling effect is no longer adiabatic; in addition negative reheat effect will be experienced, if the turbine design is multistage.
- There is a pressure loss and consequentially a reduction in enthalpy as a result of mixing bled coolant with hot gas in the gas stream.
- Heat exchange is less effective due to the reduced temperature of the exhaust gas.

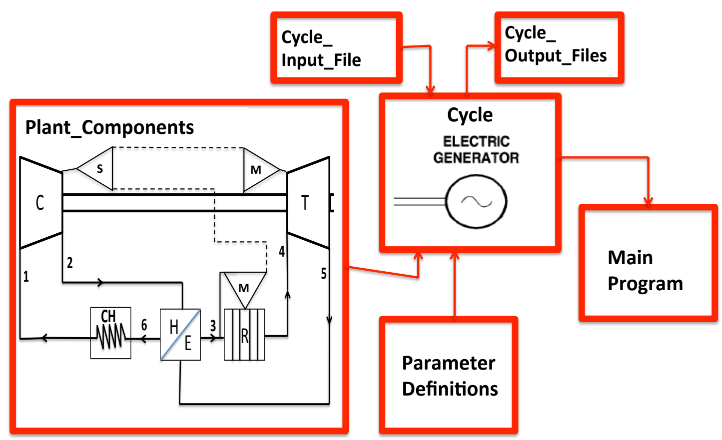


Figure 6 – Performance Simulation Tool Structure for SCR

The effect of varying turbine cooling flow and TET were analysed but focuses on a TET of 850°C. A 1% input in turbine cooling flow for a given TET requires an increase by ~30°C to maintain the same level of efficiency. This equates to a factor of 1.04, which can be considered as an increase in fuel costs. However, optimisation of the cooling flow is necessary to ensure the minimum cooling fraction is considered for a given TET.

Cycle Cooling Optimisation Results and Discussion

The cycle cooling optimisation analyses considered 2 types of turbine blades with different allowable blade metal temperatures. Blade A is derived from direct solidification casting with thermal barrier coating and film impingement, forced convection cooling technology. Blade A has an allowable blade metal temperature of 755°C because of the material, grain structure and casting process. Blade B is derived from a single crystal material with no grain boundaries and employs film impingement, forced convection cooling technology. Blade B has an allowable blade metal temperature of 870°C. The use of 2 blades with different blade metal temperatures demonstrates the effect of material on the minimum turbine cooling fraction and the overall effect on cycle efficiency for the various cycles.

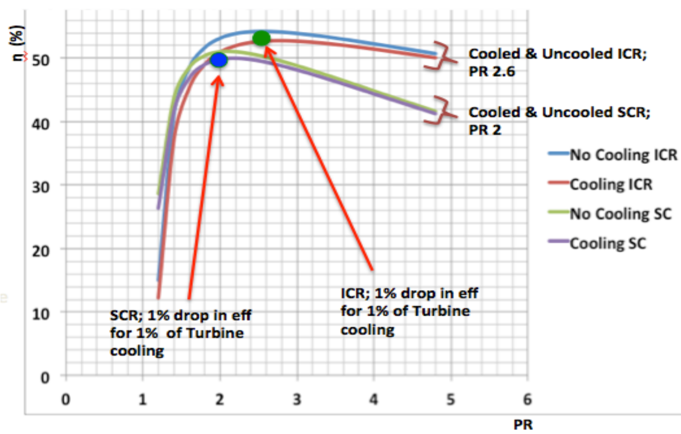


Figure 7 – Effect of Turbine Cooling on Efficiency (950°C) (Blade A)

Increasing TET and Optimising Turbine Cooling Fraction for SCR and ICR

With regard to SCR, blade B improves cycle efficiency by 1% at 950°C, with the improvement increasing to 1.7% at 1200°C. The same trend is noted for ICR, which is 0.82% at 950°C, increasing to 1.34% at 1200°C for blade B. No increases in OPR for the SCR and ICR due to the recuperator effect. There was also no 'near-stagnation' of the efficiency curves using blade A, as shown in figure 8. This is also the case for ICR and is due to the effect of the recuperator. The choice of turbine blade material on specific work is negligible for the SCR. In comparison, the ICR showed increases in specific work when using blade B (see figure 9). It is recommended to limit blade A to ≤950°C, while blade B can be specified for up to 1200°C for both cycles. This limit in TET is also based on cycle efficiency deteriorations, when the performance characteristics of both turbines with varying cooling fractions are compared. This limit recommendation is a 'soft' limit meaning that a turbine blade with a metal temperature and similar cooling technology comparable to blade A can still be used beyond 950°C because the difference between the two blades is not greatly significant. This will require a techno-economic analysis to understand the costs of both blades for each cycle.

With regard to cooling flows for SCR, blade A requires 4.3% cooling flow in comparison to 0.5% for blade B at 950°C. Blade A requires 15% cooling flow in comparison to 7% for Blade B at 1200°C. With regard to ICR, Blade A requires 3.75% cooling flow in comparison to 0.37% for blade B at 950°C. The effect increases at 1200°C due to blade A requiring 12.66% cooling flow in comparison to 6.48% for Blade B.

Increasing TET and Optimising Turbine Cooling Fraction for SC and IC

With regard to SC, blade B improves cycle efficiency by ~0.4% at 950°C, with the improvement increasing to 1.6% at 1200°C. The same trend is noted for IC which is ~0.36% at 950°C, with the improvement increasing to 1.24% at 1200°C. Other observations include both cycles showing increases in OPR (see table 2) when blade B is utilised in comparison to blade A, hence limiting the positive effect of blade B (in some cases) as the compressor work goes up with the OPR. The calculations for IC ensured the OPRs were based on optimum pressure ratio splits between the LPC and HPC. Both cycles showed pronounced cycle performance penalties for blade A due to the 'near-stagnation' trend in efficiency increase (see figure 10 for SC). SC showed increases in specific work when using blade B in comparison to IC, which showed negligible increases. The amount of cooling fraction required as the TET is increased means there is a need for limits to be imposed on blade A.

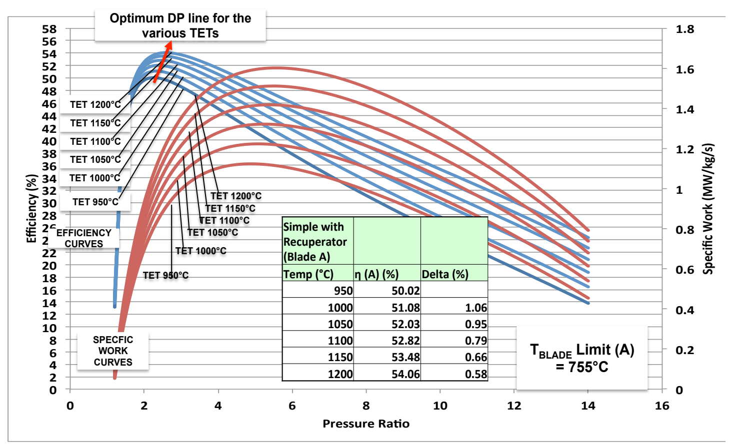
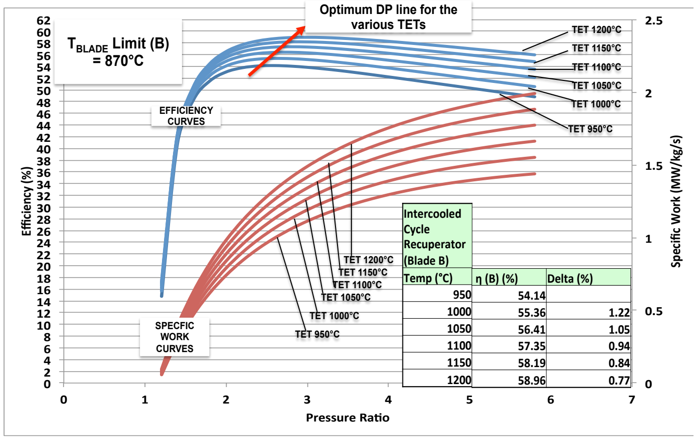


Figure 8 – SCR Optimum Efficiency and Specific Work Curves (Blade A)

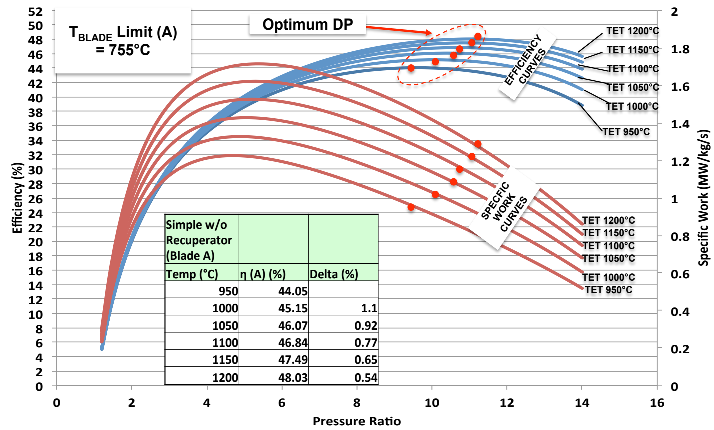


Temp (°C)	η (B) (%)	Delta (%)
950	54.14	
1000	55.36	1.22
1050	56.41	1.05
1100	57.35	0.94
1150	58.19	0.84
1200	58.96	0.77

Figure 9 – ICR Optimum Efficiency & Specific Work Curves (Blade B)

Due to cycle economics, SC is a rare proposal for Gen IV applications. However, its use will depend on demonstrating similar performance values for comparable turbine metal temperatures within the limits recommended. The recommendation for blade A is <u><</u>1000°C, with blade B specified for up to 1200°C. This limit is based on cycle efficiency deteriorations, when the performance characteristics of both turbines are compared, meaning limitations are imposed, if the benefit of using blade B with improved cooling, significantly increases the cycle efficiency. In the case of SC, this limit recommendation is a 'hard' limit. This means that a turbine blade with a metal temperature that is comparable to blade A, cannot be used beyond 1000°C if this cycle is considered for Gen IV. Furthermore, blade A requires 16% cooling flow in comparison to 2% for blade B at 950°C. This is further pronounced at 1200°C due to blade A requiring 73% cooling flow in comparison to 32% for Blade B.

With regard to IC, it is also rare proposal for Gen IV applications but studies in [16] show some viability in its use especially at high temperatures, due to an established pedigree of gas turbines operating at 1200°C. The recommendation for blade A is <u><</u>1050°C, with blade B specified for up to 1200°C, but due to the deterioration being negligible (above 1100°C) in terms of cooling demand (see figures 11 and 12), it is possible to investigate cycle performance beyond 1200°C for blade B. With reference to figure 11 (blade B), it is evident that the efficiency curves at optimum efficiencies show significant increases in OPRs between TETs in comparison to other cycles. The increases in OPR means more useful work (UW) for the plant but more importantly, the cooling fraction demanded 1100°C does not significantly penalise the cycle efficiency. As shown in figure 12, the cycle deterioration curve changes direction with the level of performance deterioration decreasing after 1100°C. The recommended limit on blade A for IC is a 'hard' limit (<u><1050</u>°C) because the efficiency benefits are significantly realised above 1050°C for blade B. Furthermore, blade A requires 11% cooling flow in comparison to 1.7% for blade B at 950°C. The difference at 1200°C is 48% cooling flow for blade A in comparison to 22.4% for Blade B.

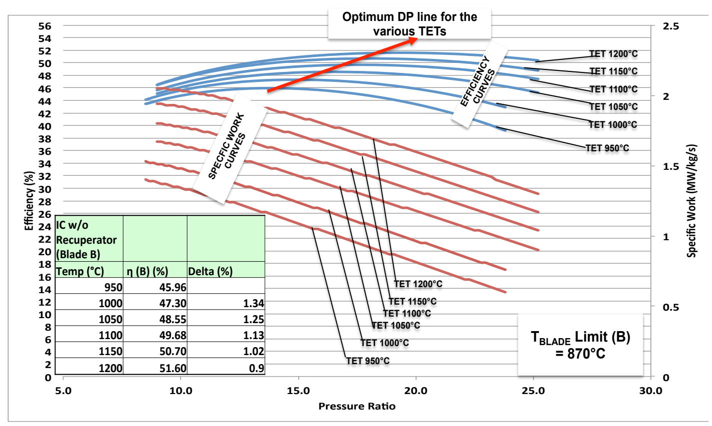


Temp (°C)	η (A) (%)	Delta (%)
950	44.05	
1000	45.15	1.1
1050	46.07	0.92
1100	46.84	0.77
1150	47.49	0.65
1200	48.03	0.54

Figure 10 – SC Optimum Efficiency and Specific Work Curves (Blade A)

<u>Effect of Recuperator and Cooling on Simple and Intercooled Cycles – Measurement of Efficiency Increase</u>

Figure 13 shows the effect of the recuperators on the simple and intercooled configurations for each turbine. The main observation is the recuperator has a significant effect on the cycles. The intercooled configuration has an average efficiency increase of 7.44% for blade A and 7.77% for blade B, whilst the simple configuration has an average increase of 5.98% for blade A and 6.33% for blade B. The justification to use optimum cooling flow based on maximum allowable turbine metal temperature, improves the cycle significantly when a recuperator is employed. The average cost of the recuperator for a typical NPP would need to be compared to the added value of an average efficiency increase for the life of the plant. In addition, the inclusion of a turbine with better metal temperature, increases the efficiency by 0.3%.



Temp (°C)	η (B) (%)	Delta (%)
950	45.96	
1000	47.30	1.34
1050	48.55	1.25
1100	49.68	1.13
1150	50.70	1.02
1200	51.60	0.9

Figure 11 – IC Optimum Efficiency and Specific Work Curves (Blade B)

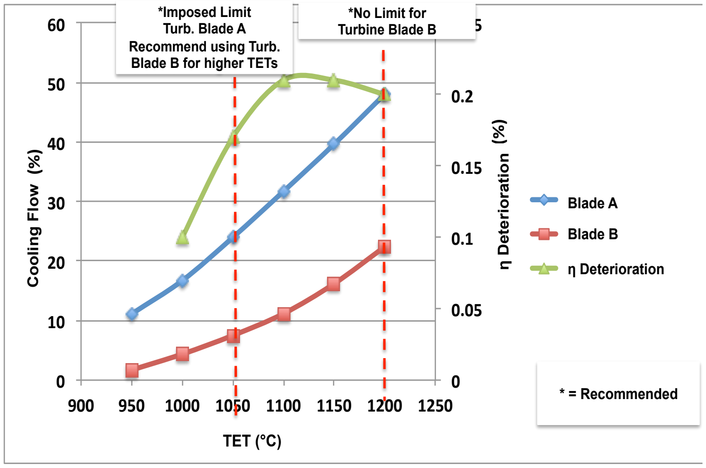


Figure 12 –IC Optimum Turbine Cooling Fraction and Effect on Efficiency

The cost of the improved turbine material over the life of the NPP would also need to be compared with the added value of the increase in efficiency.

Effect of Intercooler and Cooling on Cycles – Measurement of Efficiency Increase

The effect of intercooling on cycle efficiency was analysed for cycles with and without a recuperator. For cycles without a recuperator, there is an upward increase in efficiency as TET is increased, due to the reduced work effect when the compressor entry temperature is lowered. The average noted for blade A is an increase of 1.94% compared with an increase of 1.71% for blade B. The same upward trend is noted for the recuperated cycles but with a larger increase due to the heat exchange. For blade A, an average of 3.4% increase was noted compared to 3.15% for blade B. Previous studies such as those documented in [17] and [18] have stated that the average increase in cycle efficiency when an intercooler is incorporated is 2%. This holds true, if the same amount of cooling flow is used. Consequentially, maintaining the cooling flow means the life of the turbine increases, but maintaining the same blade metal temperature will optimise the amount of mass flow rate required for cooling and in most cases, improves the cycle efficiency, if the cooling mass flow rate (fraction) is reduced. The cost of introducing an intercooler and a second compressor for the life of the NPP needs to be assessed against the added value of efficiency increase for the life of the plant. In addition, the cost of using an improved blade material for the life of the turbine would appear to not have any benefit (in terms of increasing cycle efficiency) when focusing on the effect of the intercooler. The reason is less turbine cooling fraction is demanded by blade B, but results in higher mass flow rate at the intercooled temperature at the reactor inlet, which would normally have been bypassed to the turbine. The effect of higher mass flow rate in the reactor is higher reactor heat input Q_{MHR}, which reduces the cycle efficiency. Higher reactor heat input into the cycle has a greater effect in terms of reduced efficiency, when compared to pressure losses and reduced enthalpy as a result of mixing bled coolant with hot gas in the turbine gas stream. This observation with blade B becomes increasingly pronounced as TET is increased.

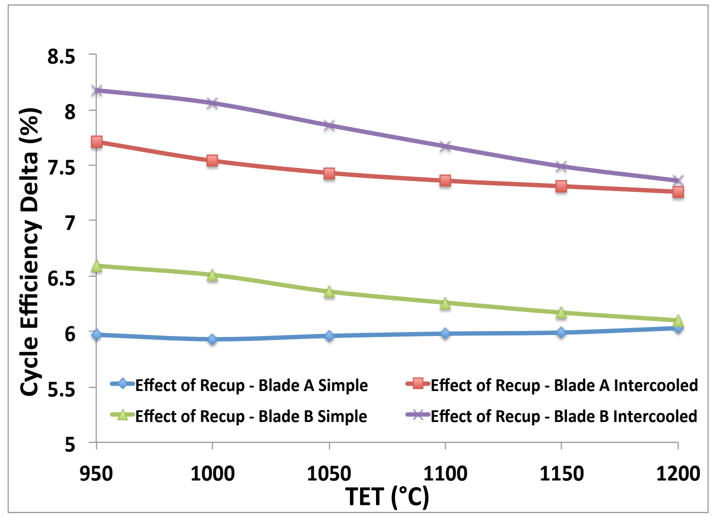


Figure 13 – Effect of Recuperator on Cycles

Technological Assessment

Figure 14 shows the cycle efficiencies for each cycle using blade A and blade B. TETs in excess of 1200°C are novel proposals that would require improvements in materials for recuperators. As such, immediate to near term goals stipulate limiting the SCR and the ICR to 950°C, whilst developing high temperature recuperators. The IC holds the answer for increases in TET beyond 950°C.

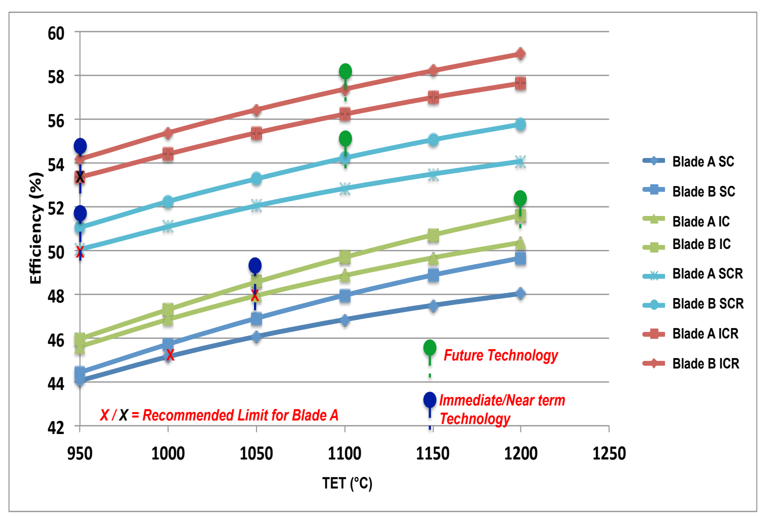


Figure 14 – Technological Assessment

An immediate to near term goal for the IC is to demonstrated capability at 1050°C through Gen IV development programmes, with the aim of increasing the TET to 1200°C and beyond. Current reactor development aim to deliver temperatures in excess of 1200°C for Gen IV applications [19], [20].The SC has not been recommended as a cycle for immediate to near term or future use because the efficiency figures, in comparison to other cycles do not support cycle economics but this will require a techno-economic assessment to justify this position.

Conclusion

In summary, the objective was to conduct a thermodynamic study using a performance simulation tool to analyse the cooling requirements for four different cycles in the 300 – 700 MW class at TETs in the range of 950°C - 1200°C using 2 different turbine blades with different blade metal temperatures. The cycles of interest were the Simple Cycle (SC), the Simple Recuperated Cycle (SCR), the Intercooled Cycle (IC) and the Intercooled Recuperated Cycle (ICR). The results provide a good basis to support preliminary design, testing, validation and verification activities of Gas Cooled Fast Reactors (GFR) and Very High Temperature Reactors (VHTR) for Generation IV NPPs. The main conclusions are:

- The performance penalties for cooling are evident when compared to an uncooled turbine. A parallel loss of turbine work was observed due to reduction in turbine mass flow rate. Mass flow rate effect (loss) was demonstrated as a reduction in turbine work of ~1% for a cooling flow of 1%.
- Limits have been proposed for blade A for SC and IC. These limits are based on cycle efficiency deteriorations when using both turbines. This means that limitations are imposed on blade A if the benefit of using blade B significantly increases cycle efficiencies.
- Simple Cycle has not been proposed for Gen IV applications because of cycle economics, but its use will depend a techno-economic assessment of the economics and demonstrating similar performance values as derived in this study.
- When a recuperator is employed, the intercooled configuration has an average efficiency increase of 7.44% for blade A and 7.77% for blade B, whilst the simple configuration has an average of 5.98% for blade A and 6.33% for blade B. The justification to use optimum cooling based on maximum allowable turbine metal temperature, improves the cycle significantly when a recuperator is used.
- The average efficiency increase due to use of an intercooler for blade A is a 1.94% increase in efficiency compared with a 1.71% increase for blade B. The same upward trend is noted for the recuperated cycles but with a larger increase due to the heat exchange. For blade A, an average of 3.4% increase was noted compared to 3.15% for blade B. Blade B has no benefit to cycle efficiency when considering the effect of the intercooler only. This is due to higher reactor thermal input in the cycle.
- IC is a rare proposal for Gen IV applications but studies have been undertaken in conjunction with this study, which concluded that there is some viability in its use especially at high temperatures. Due to the deterioration being negligible in terms of cooling demand for blade B, it is the only cycle with a possibility to investigate cycle performance beyond 1200°C. This is also based on established pedigree of turbines operating at such temperatures.
- Techno-economic assessments are required to understand the cost over the life of a plant of using blade B instead of Blade A. More importantly, techno-economic assessments are required to fully understand the cost over the life of a plant, if an intercooler configuration and/or a recuperator are employed. The costs will need to be compared against the added value of increase in efficiency.
- Immediate to near term goals stipulate limiting SCR and ICR to 950°C, whilst developing high temperature recuperators. IC holds the answer for increases in TET beyond 950°C. An immediate to near term goal for the IC is to demonstrate capability at 1050°C through Gen IV development programmes.

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