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PERFORMANCE ANALYSES AND EVALUATION OF CO2 AND N2 AS COOLANTS IN A RECUPERATED BRAYTON GAS TURBINE CYCLE FOR A GENERATION IV NUCLEAR REACTOR POWER PLANT

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ABSTRACT

As demands for clean and sustainable energy renew interests in nuclear power to meet future energy demands, Generation IV nuclear reactors are seen as having the potential to provide the improvements required for nuclear power generation. However, for their benefits to be fully realised, it is important to explore the performance of the reactors when coupled to different configurations of closed-cycle gas turbine power conversion systems. The configurations provide variation in performance due to different working fluids over a range of operating pressures and temperatures. The objective of this paper is to undertake analyses at the design and off-design conditions in combination with a recuperated closed-cycle gas turbine and comparing the influence of carbon dioxide and nitrogen as the working fluid in the cycle. The analysis is demonstrated using an in-house tool, which was developed by the authors. The results show that the choice of working fluid controls the range of cycle operating pressures, temperatures and overall performance of the power plant due to the thermodynamic and heat properties of the fluids. The performance results favored the nitrogen working fluid over CO_2 due to the behavior CO_2 below its critical conditions. The analyses intend to aid *the development of cycles for Generation IV nuclear power plants (NPPs) specifically Gas-cooled Fast Reactors (GFRs) and Very High-Temperature Reactors (VHTRs).*

NOMENCLATURE

Notations

A	flow annulus area (m^2)
C	specific heat of gas at constant pressure (J/kg K)
M	Mach number
m	mass flow (kg/s)
N	rotational speed (rpm)
P	pressure (Pa)
PR	pressure ratio
PR_c	compressor pressure ratio
PR_t	turbine pressure ratio
R	specific gas constant (J/kg K)
T	temperature (°C and K)

Greek Symbols

η	efficiency
$\boldsymbol{\varepsilon}$	effectiveness
$\boldsymbol{\theta}$	referred temperature parameter
$\boldsymbol{\delta}$	referred pressure parameter
γ	ratio of specific heats
$\boldsymbol{\rho}$	density (kg/m^3)

Subscripts

C	compressor
C_{in}	compressor inlet
C_{out}	compressor outlet
DP	design point
HEX	heat exchanger
HPS	high-pressure side
LPS	*low*-pressure side
m	mechanical
Map	reference map
S	static
T	turbine
t_{in}	turbine inlet
t_{out}	turbine outlet
1-7	station number

Abbreviation

CIT	compressor inlet temperature
CMF	corrected mass flow
CMSF	corrected mass flow scaling factor
CSSF	corrected speed scaling factor
CS	corrected speed
CH	corrected enthalpy drop
CW	compressor work (W)
GH	gas heater
HPS	high pressure side

LPS	low pressure side
NTU	number of transfer units
PC	pre-cooler
PR	pressure ratio
PRSF	pressure ratio scaling factor
Q_{actual}	actual heat flux (W/m^2)
Q_{max}	maximum heat flux (W/m^2)
Q_g	heat gained from reactor input (W)
ReX	recuperator
SF	scaling factor
SOP	shaft output power (W)
SP	specific power (W/kg/s)
TET	turbine entry temperature
TW	turbine work (W)

INTRODUCTION

Nuclear energy is plays an important role in providing clean energy to mitigate the increasing world energy demand [1], with over 400 units of nuclear reactors in service around the world. In addition, various development projects are currently running [2,3] as part of efforts to improve on the limitations of currently deployed nuclear power plants. The on-going research into Generation IV (Gen IV) aims to improve the design and performance of the next generation nuclear reactor technologies [4]. However, for the benefits to be fully realised, the design and performance has to be explored. This is achieved using different closed gas turbine cycles, which utilise different working fluids over a range of operating pressures and temperatures. Hence, the foremost consideration as an initial step in to the successful development and deployment of this technology is performance simulations.

Performance simulation is a necessary step in the planning, execution, analyses and evaluation of operations specific to nuclear power plant designs. The purpose is to minimise risks and cost of development.

The Gen-IV systems applicable to this analysis are the Very High-Temperature Reactors (VHTR) and Gas-cooled Fast Reactors (GFR) concepts. Both reactors are high-temperature gas cooled, with core outlet temperatures between 750^{0}C and 950^{0}C. The GFRs uses a fast-spectrum core, while the VHTRs utilize graphite moderation in the solid state. With regards to a coolant such as helium, it brings several benefits to plant operations such as chemical inertness, single phase cooling and neutronic transparency [5–7]. However, adopting other working fluids and mixtures for reactor cooling such as carbon dioxide, nitrogen, argon have been proposed in different studies [8,9]. There are planned and on-going developmental projects for GFR and VHTR which focus on testing the basic concepts and performance phase validation [4,10,11].

The objective of this paper is to undertake performance analyses at the design and off-design conditions for a Generation IV nuclear-powered reactor with a recuperated closed-cycle gas turbine configuration. The effects of carbon dioxide and nitrogen as working fluid will also be analysed in the recuperated cycle loop. The analyses is carried out using an in-house modelling and simulation tool, which was developed by the authors for closed-cycle gas turbine simulations [12]. The results suggests that the choice of working fluid greatly influences the range of cycle operating pressures, temperatures and overall performance of the power plant due to the thermodynamic and heat properties of the fluids. However, the choice of working fluid for the proposed Gen IV system is dictated by availability, material compatibility, and thermal stability [13–15].

BRIEF DESCRIPTION OF THE POWER PLANT CASE STUDY

The Gen-IV system in this study utilises an indirect heat source configuration with the recuperated closed-cycle gas turbine as shown in Fig. 1. Using an indirect configuration provides flexibility that allows the same working fluid or a different fluid from that of the reactor to be used. . A growing research interest into the use carbon dioxide (CO_2) and nitrogen (N_2) [18–20] is prompted by the research whereby helium is utilised as working fluid for the closed-cycle gas turbine as noted in [10,11,16,17]. These studies show that the low molecular weight of helium affects the size and number of stages in the gas turbine turbomachinery set [6,15]. Furthermore, the aerodynamic and sealing design of its turbomachinery component presents challenges. Nonetheless, these have been mitigated as described in the referenced literature. The use of other working fluid alternatives provided additional mitigation and provided the justification to carrying out performance studies on both fluids selected. Concerns relating to the safety and operation of the plant using a working fluid that is different to the reactor coolant such as the chemistry and compatibility have been discussed in [21].

The recuperated closed-cycle that is shown in Fig. 1, utilises some of the heat from the turbine exhaust to preheat the working fluid prior to entering the gas heater. Thus, this allows more working fluid to pass through, thereby increasing the overall efficiency at every pressure ratio whereby recuperation is possible. The reference design point variables that were chosen for the plant system is listed in Table 1.

The studies assumed that the heat source transfers a fixed heat-rate to the working fluids at some specified temperature. Overall system pressure loss of 7% was assumed (recuperation (ReX) 3%, pre-cooler (PC) 2% and gas heater (GH) 2%. The mechanical efficiency was taken as 98%, and the heat sink temperature was assumed to be 21^{0}C. For consistency purposes, the same values for the turbomachinery and heat exchangers component efficiencies have been assumed for each working fluid.

Table 1 Reference Design Point Parameters

Parameters	Values
Compressor mass flow rate (kg/s)	441
Compressor inlet temperature (^{0}C)	28
Compressor inlet pressure (MPa)	2.5
Compressor isentropic efficiency (%)	85
Turbine inlet temperature (^{0}C)	750
Turbine exit pressure (MPa)	2.55
Turbine isentropic efficiency (%)	88
ReX, PC & GH effectiveness (%)	90

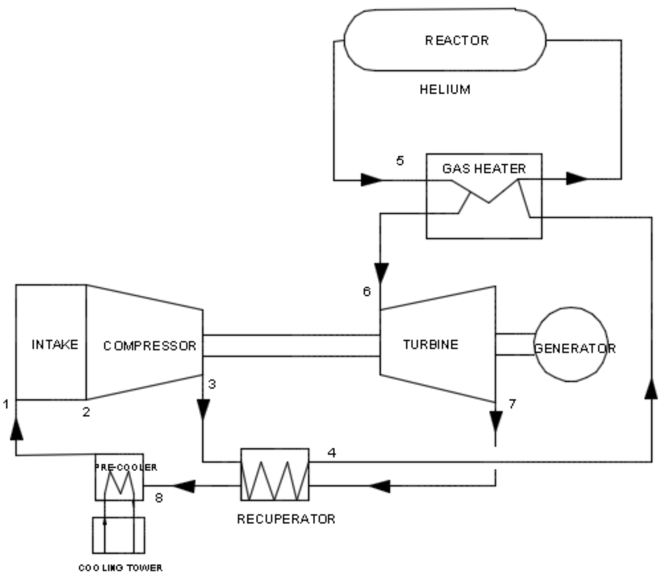

Fig. 1 Schematic Representation of the Gen-IV Reactor indirectly coupled with a recuperated closed-cycle gas turbine

THERMODYNAMIC ANALYSIS AND CYCLE MODELING

The overall performance is a function of the individual components of the Generation IV nuclear power plant [22]. The performance parameters were determined at design point. The off-design condition was simulated by changing operating design point variables such as compressor inlet temperature and pressures, turbine entry temperature or core outlet temperature, and turbine exit pressure. An estimation of the properties of the working was modeled using empirical correlations and coefficients which were compared with NASA SP-273 [23]. The thermodynamic equations implemented in within the tool for the assessment of the recuperated closed-cycle case study are given as follow:

Turbo-set: This includes the compressor and the turbine. The behavior of the turbo-set is described with dimensionless parameters such as corrected mass flow, corrected speed, pressure ratio, component efficiencies and work functions. These parameters are plotted on graphs with lines of pressure ratio against corrected mass flow for different corrected speed lines and contour lines of constant efficiency. It is essential when expressing these parameters that the properties of the working fluid are taken into consideration, which is expressed as:

$$CMF = \left(\frac{m\sqrt{\theta}}{\delta} \times \sqrt{\frac{R}{\gamma}}\right), \qquad CS = \left(\frac{N}{\sqrt{\theta R \gamma}}\right), \qquad CH = \left(\frac{\Delta H}{\sqrt{\theta R \gamma}}\right) \tag{1}$$

Where,

$$\theta = \frac{T}{T_{ref}}, and\ \delta = \frac{P}{P_{ref}}$$

The compressor exit temperature is given by the expression

$$Tc_{out} = Tc_{in} + \frac{Tc_{in}}{\eta_c}\left[\left(\frac{Pc_{out}}{Pc_{in}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right] \tag{2}$$

The compressor exit pressure is derived from the given pressure ratio as:

$$PR_c = \frac{Pc_{out}}{Pc_{in}} = f\ (CMF, CS) \tag{3}$$

The compressor work (CW), is a product of the mass flow, specific heat capacity at constant pressure and the overall temperature rise in the compressor. This is given as:

$$CW = mC(Tc_{out} - Tc_{in}) \tag{4}$$

Similarly, turbine exit temperature is given by:

$$Tt_{out} = Tt_{in} - Tt_{in}\eta_t\left[1 - \left(\frac{Pt_{out}}{Pt_{in}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right] \tag{5}$$

And turbine work (TW) is expressed as:

$$TW = mC(Tt_{out} - Tt_{in}) \tag{6}$$

The turbine discharge pressure ratio is calculated using Eq (7)

$$PR_t = \frac{Pt_{out}}{Pt_{in}} = PR_c\left[\frac{\sum(1 - \Delta P)_{HPS}}{\sum(1 + \Delta P)_{LPS}}\right] \tag{7}$$

Heat Exchangers: The heat exchangers which include the recuperator, gas heater and pre-cooler were modeled using the ε-NTU method and a counter-flow shell and tube configuration was assumed. The ε-NTU method was used since the inlet condition (temperature and pressure) of the fluid stream can be easily obtained and simplifies the iteration involved in predicting the performance of the flow arrangement. This method is fully described in references [24,25]. The approach also assumes that the heat exchanger effectiveness is known and the pressure losses are given.

Therefore, the effectiveness of the heat exchanger is the ratio of the actual heat transfer rate to the thermodynamically limited maximum heat transfer rate available in a counter flow arrangement.

$$\varepsilon = \frac{Q_{actual}}{Q_{max}} \quad \textbf{(8)}$$

$$= \frac{C_{hot}(T_{hotin} - T_{hotout})}{C_{min}\,(T_{hotin} - T_{coldin})}$$

$$= \frac{C_{cold}(T_{coldout} - T_{coldin})}{C_{min}\,(T_{hotin} - T_{coldin})}$$

Where,

$$C_{min} = \begin{cases} C_{hot}\, for\, C_{hot} < C_{cold} \\ C_{cold}\, for\, C_{cold} < C_{hot} \end{cases} \quad \textbf{(9)}$$

$$C_{hot} = (WC) hot\ fluid\ Stream \quad \textbf{(10)}$$

$$C_{cold} = (WC) cold\ fluid\ Stream$$

For counter flow shell and tube heat exchangers, number of transfer unit (NTU) is given by:

$$NTU = \frac{LOG_e \left[\frac{2 - \varepsilon(1 + C^* - \eta_{Hex}}{2 - \varepsilon(1 + C^* + \eta_{Hex}}\right]}{\eta_{Hex}} \quad \textbf{(11)}$$

Where,

$$C^* = Capacity\ rate\ ratio = \frac{C_{min}}{C_{max}} \quad \textbf{(12)}$$

$$\eta_{Hex} = (C^{*2} + 1)^{0.5} \quad \textbf{(13)}$$

The inlet and out pressures of the heat exchangers were calculated from the relative pressure losses given by:

$$P_{out} = P_{in}(1 - \Delta P) \quad \textbf{(14)}$$

Where ΔP is the percentage (%) pressure loss

Reactor Model: The reactor was modeled as a heat source supplying reactor thermal power at a specified temperature and efficiency. The heat gained is given by:

$$Q_g = mC_{(gas)}\Delta T \quad \textbf{(15)}$$

The heat source pressure loss is calculated in a similar way as shown in Eq. (14). The power plant thermodynamic states of temperature and pressure at all components were obtained by solving Eqs. (1) – (15)

Cycle Performance Calculation: The overall plant cycle assessment is represented as shaft output power (SOP), specific output power (SP), and cycle thermal efficiency. These are given by the following equations:

$$SOP = TW - CW/\eta_m \quad \textbf{(16)}$$

The capacity of the plant is represented as specific power (SP), given by:

$$SP = SOP/m \quad \textbf{(17)}$$

The cycle thermal efficiency (%) is given by:

$$\eta_{th} = SOP/Q_g \quad \textbf{(18)}$$

Component Matching: Component matching refers to the interactions between the gas turbine components which satisfies the engine matching conditions of mass and energy conservation to produce the system operating line. To be able to predict an accurate design and off-design point performance of the closed-cycle gas turbine would require matching of both the turbomachinery and heat exchangers. The following relationships in equations (19) – (23) are realized in order to obtain maximum matching in the recuperated closed-cycle system shown in fig.1.The matching process is comprehensively discussed in references [9,26,27]

$$\frac{m_T\sqrt{T_6}}{P_6} = \frac{m_c\sqrt{T_2}}{P_2} \times \frac{P_2}{P_3} \times \frac{P_3}{P_6} \times \sqrt{\frac{T_6}{T_2}} \times \frac{m_T}{m_c} \quad \textbf{(19)}$$

$$\frac{N_6}{\sqrt{T_6}} = \frac{N_2}{\sqrt{T_2}} \times \sqrt{\frac{T_2}{T_6}} \quad \textbf{(20)}$$

$$\frac{T_6 - T_7}{T_6} = \frac{T_3 - T_2}{T_2} \times \frac{T_6 - T_7}{T_3 - T_2} \times \frac{T_2}{T_6} \quad \textbf{(21)}$$

$$\frac{P_7}{P_6} \times \frac{P_2}{P_7} = \frac{P_2}{P_3} \times \frac{P_3}{P_6} \quad \textbf{(22)}$$

Map Scaling: The maps for different components were obtained using multi-fluid scaling methods which multiplies scaling factors derived at design point to the original component map at the off-design point. The following equations were used to obtain the scaling factor for off-design assessment.

$$CMSF = \frac{(CMF_{cs})_{DP}}{\left(CMF_{map}\right)_{DP}} \quad \textbf{(23)}$$

Where,

$$CMF = \left(\frac{m\sqrt{\theta}}{\delta} \times \sqrt{\frac{R}{\gamma}}\right)$$

Similarly, the pressure ratio scaling factor is obtained as:

$$PRSF = \frac{(PR_{DP} - 1)}{(PR_{DPmap} - 1)} \quad \textbf{(24)}$$

Component efficiency scaling factor is given by:

$$\eta_c SF = \frac{(\eta_c)_{DP}}{(\eta_c)_{DPmap}} \quad \textbf{(25)}$$

RESULTS AND DISCUSSION

Optimum Pressure ratio:

It can be observed from Fig. (2) that up to a certain point, there is a positive benefit in terms of cycle efficiency due to recuperation. After the limit is reached, a drop in cycle efficiency is observed regardless of increases in pressure ratio. The optimum pressure ratios for which the cycle efficiencies are maximum for both cycles are different for a given overall temperature ratio. The curves also show that the maximum feasible pressure ratio occurs when the compressor exit temperature equals the turbine inlet temperature.

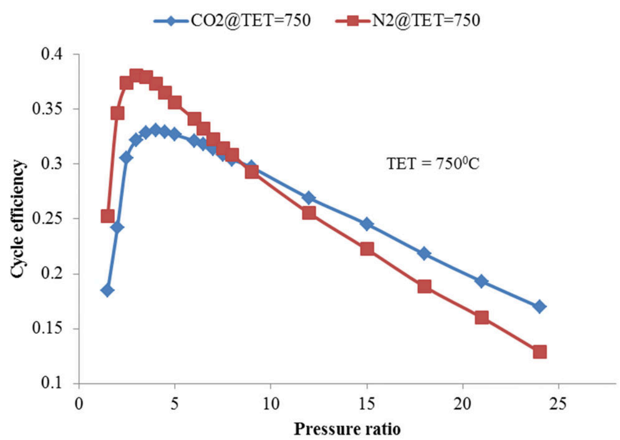

Fig. 2 Cycle efficiency against pressure ratio

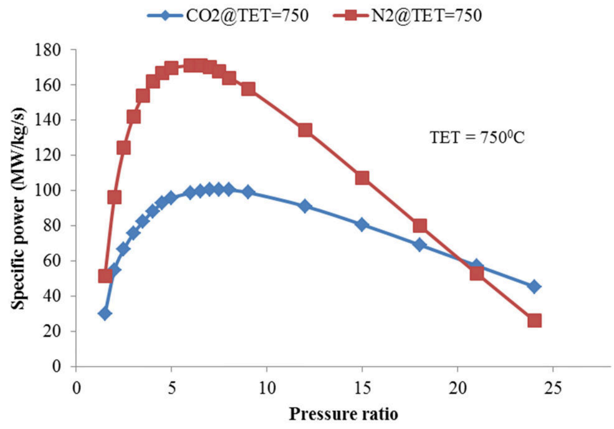

Fig. 3 Specific power against pressure ratio

The optimum pressure ratios for the maximum cycle efficiency occur at 3.0 for N_2 and 4 for CO_2. Similarly, the optimum pressure ratios for a maximum specific power shown in fig. 3 occur at 6.5 for N_2 and 7.5 for CO_2. The reason for this can be explained by considering the ratio of their heat capacities (gamma). N_2 with a higher ratio of heat capacity tend to have better performance at lower pressure ratio compared with CO_2. It can also be noted that the optimum pressure ratios when considering efficiency versus plant capacity (specific power) are different. This is because the recuperator improves the efficiency greatly meaning that less power is required to raise the temperature of the reactor coolant. Furthermore, the specific power or capacity of the plant is dependent on increasing the reactor thermal power. With regard to the pressure ratio, higher pressure ratio will pose higher design challenges. Thus, the pressure ratios obtained for each working fluid used in this study will require an advanced turbomachinery design to achieve optimal performance at design conditions. A compromise between the cycle efficiency, turbomachinery design challenges, size of plant and plant cost is required to meet the Gen-IV expectations. In reality, a slightly lower pressure ratio has been proven to be easier in terms of aerodynamic design, mechanical stresses and satisfactory level of efficiency for closed-cycle gas turbine [15,26,28–30].

Impact on efficiency and specific work

Fig. (2) and (3) show graphs of efficiency and specific work against pressure ratio respectively. From this analysis, the working fluid cycle efficiencies and specific power seem to peak at 38% and 33%, 171MWs/kg and 101MWs/kg for N_2 and CO_2 respectively at TET of 750^{0}C. The cycle efficiency of N_2 appears to be higher than CO_2 at lower pressure ratios due to its ratio of heat capacities. In addition, the system pressure and temperature at these points are higher than the critical temperature and pressure of N_2, hence, above this points its thermodynamic properties are usually stable at 3.35 MPa and 126.2K. CO_2 undergoes rapid changes in its thermodynamic properties due to variations of the system pressures and temperatures, which negatively influence the cycle efficiency and specific power, especially below its critical conditions. The design for an optimal CO_2 performance will mean that it operates above its critical points and the use of recompression within the cycle will be an added advantage for its selection [8,19].

Looking at the trend of existing nuclear power in operation and theoretical concepts, cycle efficiency above 40% will seem to be at a competitive advantage for future development and deployment. Hence, increasing the TET will be desirable to achieve a competitive efficiency and compact system.

Impact of Turbine entry temperature

The turbine entry temperature was increased to 850^{0}C, and 950^{0}C in repeated simulations performed. This was based on the limitation of material technology level and the nuclear reactor capability. The effects of the temperature increase on the turbine entry temperature and the impact on efficiency are illustrated in Fig. (4) and (5).

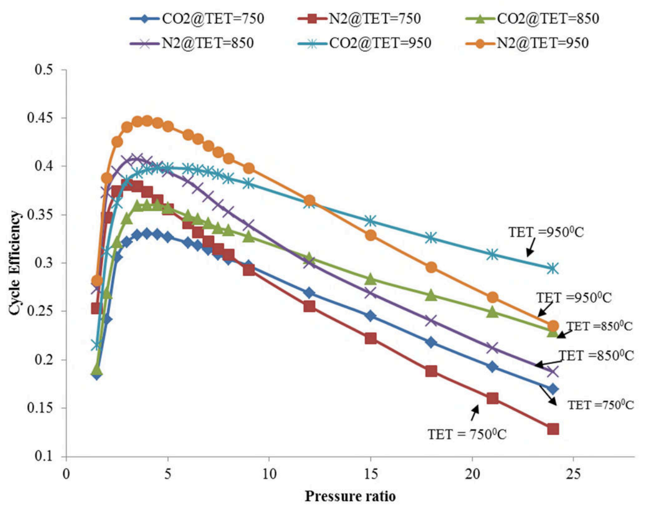

Fig. 4 Cycle efficiency against pressure ratio at different TETs

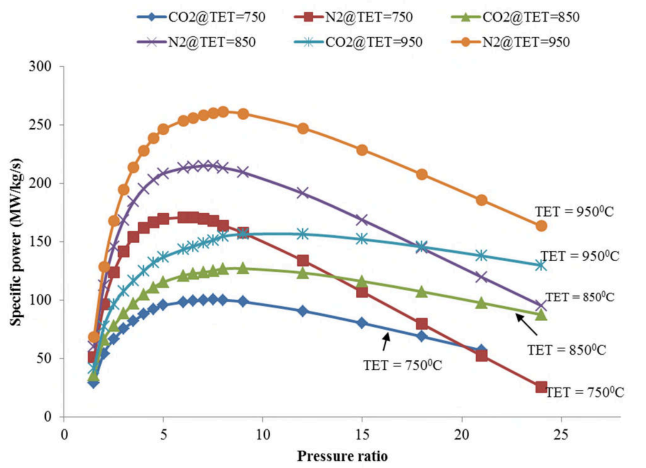

Fig. 5 Specific power against pressure ratio at different TETs

From an ideal thermodynamic stance, the overall cycle efficiency is independent of the turbine entry temperature; however, for this case, the gas properties were modeled as non-ideal, hence, changes in temperature and pressure have an effect on the working fluid properties and the cycle performance is impacted. Generally, a TET increase results in a corresponding increase in the cycle efficiency and specific power. Notwithstanding the benefits, operation at high TET always requires trade-offs, typically between capital cost and operational cost. This impact was observed to be more prevalent on CO_2 than on nitrogen. The specific power of CO_2 increased by 42% as TET moves from 750^{0}C to 950^{0}C, while N_2 increased by 36% at their respective optimum pressure ratios. Similarly, their cycle efficiencies increased by an average of 15% respectively.

Impact of compressor inlet temperature

The compressor inlet temperature (CIT) of the power plant is dictated by the environment in which the cycle waste heat is rejected. The effect of the CIT on the cycle efficiency and shaft power, in the temperature range of 27^{o}C to 67^{0}C is presented in Fig (6) and (7). The compressor pressure ratio was fixed at a design TET of 750^{0}C. The general trend from the results indicates that the cycle efficiency and power decreases as compressor entry temperature increases. This is due to increases in compressor work, meaning that the increase in temperature puts more demand on the turbine to able to drive the compressor. On average, a drop of 1% in efficiency was observed with corresponding increase in the entry temperature. These changes on the compressor inlet temperature can have a direct impact on the operational cost of the system.

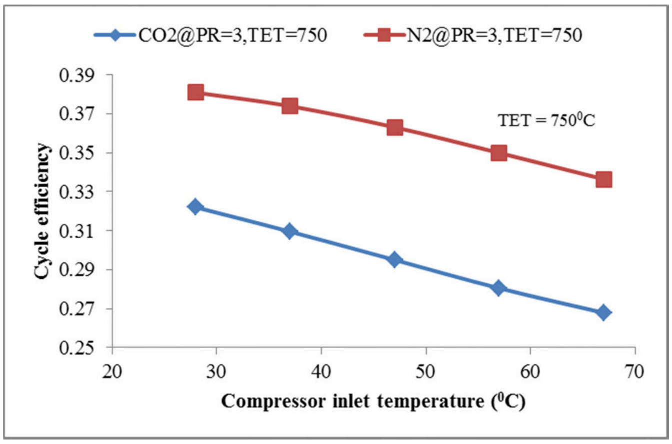

Fig. 6 Variation of cycle efficiency at different compressor inlet temperature

The preferred design criterion for closed-cycle gas turbine compressor inlet temperature is to ensure that the cycle is designed to peak at the fluid critical temperature in order to achieve the optimum performance from the system due to the stability of the thermodynamic properties above its critical temperature.

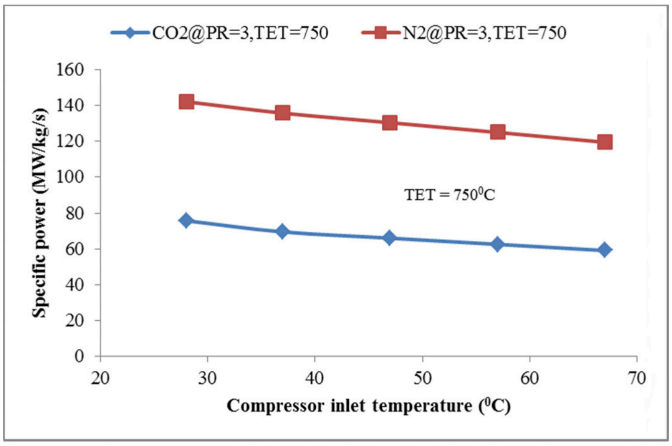

Fig. 7 Variation of specific power at different compressor inlet temperature

Impact of Compressor inlet pressure

Typically, the use of a high compressor pressure minimizes the system weight. Since the working fluids are non-ideal and its properties depend on the pressure and temperature prescribed for the system, changes in the compressor inlet pressure will have a slight impact on the cycle performance. In view of the results in Fig.(8), an increase in the compressor inlet pressure suggests a 0.1% increase in the overall cycle efficiency for nitrogen although it is expected to that this increase would have a significant impact on the structural integrity of its components. Similarly, the same trend is noted for CO_2; the efficiency gained due to increase in pressure is approximately averaged at 0.2%. This is because as the inlet pressure increases, it approaches the critical pressure of CO_2, and its thermodynamic properties are newar stability. For working fluids that behave like ideal fluid such as helium, changes in compressor inlet pressure do not have any significant impact on the cycle performance.

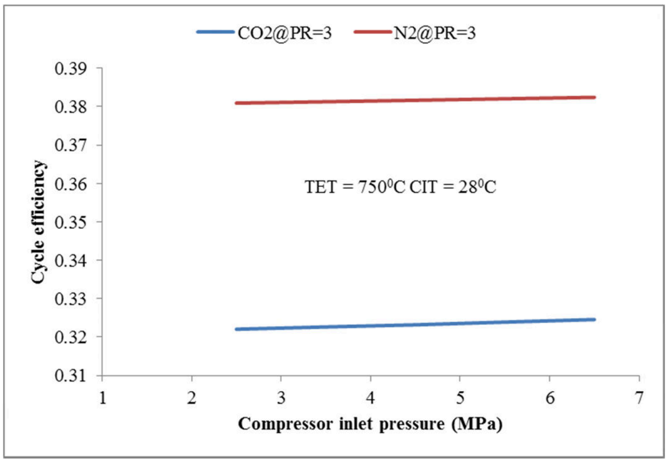

Fig. 8 Influence of compressor inlet pressure on the cycle efficiency

As the case is with the (CIT), designing for a closed-cycle gas turbine requires high compressor inlet pressure to achieve the working fluid critical properties. For CO_2, the objective is to achieve above its critical pressure of 7.38 MPa, while for N_2 the design should aim at achieving above 3.35MPa.

CONCLUSION

This paper presents a thermodynamic performance comparison between nitrogen and CO_2 as potential working fluids proposed for a Gen-IV nuclear reactor, which indirectly coupled to a recuperated closed-cycle gas turbine. The main findings are summarized below:

- Recuperation improves cycle efficiency at the optimised pressure ratio by utilizing exhaust gas at exit to the turbine and returning it back into the cycle.
- Selecting an optimum pressure ratio by design is based on reasonable compromise between cycle efficiency, component design constraints, and cost.
- In comparison, the results indicate that N_2 outperforms CO_2 at lower pressure ratio. This is due to the stable thermodynamic properties as it first approaches its critical point. However, the introduction of recompression for a CO_2 cycle could enable better performance.
- The cycle pressure ratios between 2 and 3 seem to be within the design constraints to achieve optimum performance for both fluids.
- The gas turbine Gen-IV cycle efficiency is greatest for working fluid with higher gamma (γ) at lower pressure ratio.
- The choice of working fluid for Gen-IV design considers the availability of the working fluid, safety measures and the impact its chemical properties on the system and environment.

- Increasing the TET has a significant influence on the cycle efficiency and specific power. However, as one of the major design constraints, the limit to which this is achieved is dependent on the material technology.
- Both compressor inlet temperature and pressure impact the performance of the working fluid since changes in these parameters have a slight impact on their thermodynamic properties. As a design constraint, the level of pressurisation within the cycle is dependent on the mechanical structural integrity of the system.
- Validation is recommended for tools such as the one developed for this study. This will enable optimisation to improve the applicability and accuracy, thereby encouraging it use and reducing the costs associated with extensive test activities.

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REFERENCES

[1] Abram, T., and Ion, S., 2008. Generation-IV Nuclear Power: A Review of the State of the Science, Elsevier Energy Policy, Vol. 36, No. 12, pp. 4323–4330. doi.org/10.1016/j.enpol.2008.09.059

[2] Kelly, J. E., 2014. Generation IV International Forum: A Decade of Progress through International Cooperation, Progress in Nuclear Energy, Vol. 77, pp. 240–246. doi.org/10.1016/j.pnucene.2014.02.010

[3] Locatelli, G., Mancini, M., and Todeschini, N., 2013. Generation IV Nuclear Reactors: Current Status and Future Prospects, Elsevier Energy Policy, Vol. 61, pp. 1503–1520. doi.org/10.1016/j.enpol.2013.06.101

[4] Forum Generation IV International, 2014. Technology Roadmap Update for Generation IV Nuclear Energy Systems, Washington D.C, USA, 66 pages.

[5] El-Genk, M. S., and Tournier, J., 2009. Performance Analyses of VHTR Plants with Direct and Indirect Closed Brayton Cycles and Different Working Fluids, Progress in Nuclear Energy, Vol. 51, No. 3, pp. 556–572. doi.org/10.1016/j.pnucene.2008.11.004

[6] Wang, J., and Gu, Y., 2005. Parametric Studies on Different Gas Turbine Cycles for a High-Temperature Gas-Cooled Reactor, Elsevier Journal of Nuclear Engineering and Design, Vol. 235, No. 16, pp. 1761–1772. doi.org/10.1016/j.nucengdes.2005.02.007

[7] Gad-Briggs, A., Pilidis, P., and Nikolaidis, T., 2017. Analyses of the Effect of Cycle Inlet Temperature on the Precooler and Plant Efficiency of the Simple and Intercooled Helium Gas Turbine Cycles for Generation IV Nuclear Power Plants, Applied Sciences, Vol. 7, pp. 319–338.

[8] Invernizzi, C. M., 2017. Prospects of Mixtures as Working Fluids in Real-Gas Brayton Cycles, Energies, Vol. 10, No. 10, 15 pages. DOI: 10.3390/en10101649

[9] Ulizar Alvarez., J. I., 1998. Simulation of Multi-Fluid Gas Turbines, PhD Thesis, Cranfield University.

[10] Gad-Briggs, A., and Pilidis, P., 2016, "Analyses of Simple and Intercooled Recuperated Direct Brayton Helium Gas Turbine Cycles for Generation IV Reactor Power Plants," ASME Journal of Nuclear Engineering and Radiation Science, Vol. 3, No. 1, 8 pages. doi: 10.1115/1.4033398

[11] Gad-Briggs, A., and Pilidis, P., 2017, "Analyses of Off-Design Point Performances Of Simple And Intercooled Brayton Helium Recuperated Gas Turbine Cycles," *Proceedings of the 2017 25th International Conference on Nuclear Engineering ICONE25*, ASME, Shanghai, China, July 2-6, Paper #ICONE25-67714, 12 pages. doi:10.1115/ICONE25-67714

[12] Osigwe, E. O., Pilidis, P., Nikolaidis, T., and Sampath, S., 2017. GT-ACYSS: Gas Turbine Arekret-Cycle Simulation Tool for Training and Educational Purposes, Working Paper, Cranfield University.

[13] Lee, J. C., J. Campbell, J., and Wright, D. E., 1981. Closed-Cycle Gas Turbine Working Fluids, ASME Journal of Gas Turbine Engineering and Power, Vol. 103, No. 1, pp. 220–228. doi:10.1115/1.3230701

[14] Robinson, S. T., 1957. Influence of Working-Fluid Characteristics on the Design of the Closed-Cycle Gas Turbine, Proceedingd of ASME Gas Turbine Power Conference, Detroit, Michigan, USA, March 18-21, 14 pages. doi:10.1115/57-GTP-13

[15] El-Genk, M. S., and Tournier, J., 2009. Effects of Working Fluid and Shaft Rotation Speed on the Performance of HTR Plants and the Size of CBC Turbo-Machine, Elsevier Journal of Nuclear Engineering and Design, Vol. 239, No. 10, pp. 1811–1827.

[16] Ainley, D. G., and Barnes, J. F., 1966. An Assesment of the Component Sizes of Nuclear Closed-Cycle Gas Turbines Using Argon and Helium, Proceedings of IMechE, IMechE, Pyestock, Farnborough, pp. 156–162.

[17] Xing, L. Yan, Lidsky, M. L., 1993. Design of Closed-Cycle Helium Turbine Nuclear Power Plants, ASME International Gas Turbine and Aeroengine Congress and Exposition, Ohio, USA, May 24-27, 8 pages. doi:10.1115/93-GT-196

[18] Jackson, A. J. B., and Audus, H., 2000. Gas Turbine Performance Using Carbon Dioxide as Working Fluid in Closed Cycle Operation, Proceedings of ASME Turbo Expo - Power for Land, Sea and Air, Munich, Germany, May 8-11, 8 pages. doi:10.1115/2000-GT-0153

[19] Dostal, V., 2004. A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors, Massachusetts Institute of Technology.

[20] Dyreby, J. J., 2014, "Modelling the Supercritical Carbon Dioxide Brayton Cycle with Recompression," University of Wisconsin-Madison.

[21] Alpy, N., Cachon, L., Haubensack, D., Floyd, J., Simon,

N., Gicquel, L., Rodriguez, G., Saez, M., and Laffont, G., 2011. Gas Cycle Testing Opportunity with ASTRID, the French SFR Prototype, Proceedings of Supercritical CO2 Power Cycle Symposium, Boulder, Colorado, USA, May 24-25, pp. 42–51.

[22] Shirakura, T., and Awano, S., 1977. Thermodynamic Performances of Closed-Cycle Gas-Turbine, Proceedings of the JSME/ASME Joint Gas Turbine Congress, Tokyo, Japan, May 22-27, pp. 260–270.

[23] Gordon, S., and McBride, B. J., 1996. Computer Program for Calculation of Complex Chemical Equilibrium and Applications: Part II, NASA, USA. https://ntrs.nasa.gov/archive/nasa/casi.ntrs.nasa.gov/19950013764.pdf.

[24] Shah, R. K., and Sekulic, D. P., 2003. Fundamentals of Heat Exchanger Design, John Wiley & Sons, Inc., New Jersey, USA, 976 pages.

[25] Kakac, S., and Liu, H., 2002. Heat Exchangers Selection, Rating and Thermal Design, CPC Press, New York, USA, 520 pages.

[26] W. M Crim, J., Hoffmann, J. R., and Manning, G. B., 1966. The Compact AK Process Nuclear System, ASME Journal of Engineering and Power, Vol.88, No. 2, pp. 127–138. doi:10.1115/1.3678495

[27] Walsh, P. P., and Fletcher, P., 1998. Gas Turbine Performance, Blackwell Science, Oxford, England, 646 pages.

[28] McDonald, C. F., 1978. The Closed-Cycle Turbine-Present and Future Prospectives for Fossil and Nuclear Heat Sources., Proceedings of ASME International Gas Turbine Conference and Product Show, Vol. 1B, London, England, April 9-13, 28 pages. doi:10.1115/78-GT-102

[29] Bammert, K., Burik, J., and Griepentrog, H., 1974. Highlights and Future Development of Closed-Cycle Gas Turbines, ASME Journal of Gas Turbine Engineering and Power, Vol 96, No. 10, pp. 342–348. doi:10.1115/1.3445856

[30] McDonald, C. F., 1985. Large Closed Cycle Gas Turbine Plants, Sawyer's Gas Turbine Engineering Handbook, Vol. 2, Sawyer, J.W Turbomachinery International Publications, Norwalk, Con., USA.

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