ASME Paper Title: Techno-economic study on implementation of inventory control requirements for a nuclear powered closed-cycle gas turbine power plant

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ASME Journal Title: Journal of Nuclear Engineering and Radiation Science

Volume/Issue: Volume 7, Issue 3

Date of Publication (VOR* Online): 23 October 2020

ASME Digital Collection URL: https://asmedigitalcollection.asme.org/nuclearengineering/article/7/3/031301/1089039/Techno-Economic-Study-on-Implementation-of

DOI: https://doi.org/10.1115/1.4048903

Journal of Nuclear Engineering and Radiation Science. Received September 15, 2019; Accepted manuscript posted October 23, 2020. doi:10.1115/1.4048903

Journal of Nuclear Engineering and Radiation Science,
Volume 7, Issue 3, July 2021,
Article number 031301

Techno-Economic Study on Implementation of Inventory Control Requirements for a Nuclear Powered Closed-Cycle Gas Turbine Power Plant

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Abstract

The use of an inventory control system offers a unique benefit of stable cycle thermal efficiency during part-load operation. This article focuses on the influence of initial inventory tank pressure on the control level using pressure differential as a driving force of the inventory control system. The study also considered the effects of using multiple tanks to increase the overall size of the inventory control tank and the use of insulation to reduce the impact of temperature variation between the compressor discharge temperature and the inventory tank temperature. The second part of this analysis is a cost comparison between the use of multiple tanks and the use of a transfer compressor to achieve high cycle efficiency at continuous part-load operation. The discussions in this paper accentuate the optimum benefit for utilizing an inventory control system for a single-shaft intercooled-recuperated closed-cycle gas turbine plant.

Keywords: Closed-cycle gas turbine, nuclear reactor, Inventory control, cost, techno-economic

1. Introduction

The nuclear energy system consists of the reactors, the fuel cell technology, the control systems, and the energy conversion systems (such as Rankine and Brayton Cycles). There has been lots of focus on nuclear reactors and fuel cell technology. However, this paper focuses on the energy conversion system using the closed cycle gas turbine and its control options.

In recent times, there has been a growing interest in the use of the closed-cycle gas turbine technology for nuclear power generation and utilities due to [1–4]: (a) its easy adaptability (b) flexibility in using many working fluids (c) high part-load performance (d) availability and low maintenance cost. Importantly, the closed-cycle gas turbine offers potential savings in operating costs due to its ability to relatively maintain better cycle efficiency compared with other advanced cycles under varying operating conditions [4]. This advantage can only be possible when the appropriate control strategies are in place during the operation of the power plant [5–11]. The goal of these control options when implemented would be (i) for the power plant to quickly adjust to a wide range of fluctuating load variation without significantly affecting the cycle thermal efficiency (ii) for prevention of thermal shocks on plant components during critical transients, and (iii) for providing automatic control manoeuvres during plant start-up and shut down and (iv) for the stability of the plant during operation. To this end, the objective of any operator would be to stably maximize the plant performance by synchronizing the different control options for high cycle thermal efficiency as the energy demand from the grid regulators fluctuates [12].

Among the various control options described in references [2,7,8,10], the inventory control system (ICS) provides distinct characteristics when modulating the gas turbine power output to match the required energy or load demand from the grid. It can maintain high cycle efficiency at different operating conditions compared with other control options when assessed singularly [3,4,8,10]. Consequently, a significant number of publications have focused on optimizing the capability of the inventory control systems for load-following [12–15]. For example, in reference [16], the authors described the use of multiple tank storage for better part-load capability. Their findings show that with

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Accepted manuscript posted October 23, 2020. doi:10.1115/1.4048903

additional inventory tanks, more working fluid could flow from the main gas turbine to enable an extended part-load range based on the pressure differential between the systems. The research also suggests an optimum number of tanks for which minimum part-load can be achieved.

Similarly, in the work of Bitsch et al., [17], the authors analyzed the relationship between inventory control range, total control volume, and the helium inventory in the helium coolant circuit. In their work, the minimum range of the energy demand limit was set at 50% of the gas turbine output power, and constant inventory tank temperature was assumed. The volume of the inventory tank was varied, which had an impact on the load control range. It is worth mentioning that the minimum range the part-load power could reach when the ICS is in use is determined by several factors [2,15]. Examples of such factors include the initial pressure of the inventory tank, the effect of centrifugal force on blade tip as a result of an increase in shaft rotational speed, as fluid inventory decrease, the location of inventory valves in the cycle loop, the availability of inventory transfer compressor, and the cost of implementing any of the listed options. Some of these factors have been discussed in references [18,19]. However, no research compares the use of natural pressure differential and with the use of transfer compressors as a driving mechanism of the ICS from a techno-economic point of view.

This paper presents a techno-economic study on the inventory control requirements at different load demand, with emphasis on the influence of initial tank pressure, the use of single, multiple inventory control tanks (ICT) driven by pressure differential compared with the use of transfer compressor. The principal goal for this study is to ascertain the technical and economic implications for choosing any of the three options under consideration: a single ICT, multi-ICTs, or inventory control with transfer compressor. In this study, the ICTs and pressure differential determined the minimum power range. However, for the transfer compressor, the system stability and integrity are constraints to the minimum power range. This paper also shows the effect of different control strategies on cycle efficiency to accentuate the advantage inventory control could offer compared with other options. This article discusses the influence of mass transfer on the inventory control tank (ICT) temperature and the cost of the system obtained from the design and weight analysis. An understanding of the techno-economic considerations presented in this paper will give an informed assessment of the options discussed.

2. Engine System Description

Figure 1 shows the main layout of the power plant cycle used in this study. The gas turbine (GT) power plant is a single-shaft inter-cooled recuperated cycle with air as a working fluid. The plant component consists of a low-pressure compressor (LPC), intercooler (IC), high-pressure compressor (HPC), a recuperator heat exchanger (RX), gas-heater (GH), the reactor coupled to the gas heater, turbine, pre-cooler (PC), inventory control valves (ICV) and a single ICT. A single shaft drives the LPC and HPC. Table 2 describes the conditions of the plant components. The power plant has all control options; however, the focus of this paper is on the ICT. The valve lift of the ICVs is actuated by an output controller, which has an input signal of the difference between the nominal and actual power output. This variation triggers the opening and closing of the ICVs. Section 3 of this paper provides a further explanation of the ICVs and ICS. In figure 1, the working fluid is discharged through the inventory control valve 1 (ICV1) and fed into the single inventory control tank. The extraction of the working fluid stops when the pressure at the ICT and the HPC is at equilibrium.

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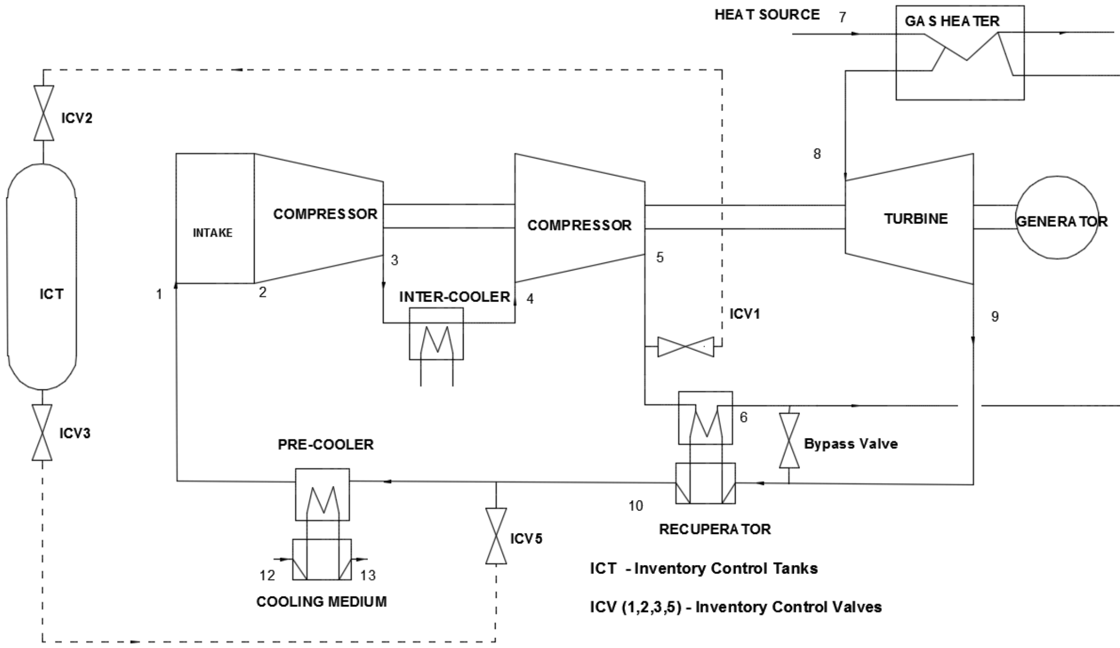

Figure 1: Scheme of Reference Plant – case 1: with a single storage tank

In figure 2, multiple ICTs are used to achieve further extraction of fluid from the gas turbine circuit via a natural pressure differential between the ICTs and the HPC. In figure 3, a transfer compressor (TC) extracts the working fluid from the gas turbine circuit to either a single or multiple tanks. The TC could be used alone or in combination with the natural pressure differential. In the later, the transfer compressor triggered when the pressure differential between the ICT and HPC is at equilibrium; hence, further extraction is achieved with the TC.

Table 1: ICS Design Assumptions

Description	Value
Tank Material	Carbon steel (SA-516-70)
Carbon steel density	7850 kg/m^3
Material cost factor	1.37
Volume of Tank	2100 m^3
Volume of GT-Cycle Loop	1500 m^3
L/D of Valve	2

Table 2: Summary of Power Plant Design Point Description

Description	Unit
Heat Source Temp.	827 (℃)
LPC Pressure ratio	1.65
LPC Inlet Pressure	830.8 (kPa)
LPC Inlet Temperature	17 (℃)
HPC Pressure ratio	2.40
LPC& HPC efficiency	86 (%)
Turbine efficiency	90 (%)
Flow rate at LPC	230 (kg/s)
Plant Thermal Efficiency	41.2 (%)
IC effectiveness	90 (%)
RX & GH effectiveness	90 (%)
Rated power	40.8 (MW)
Working Fluid	Air

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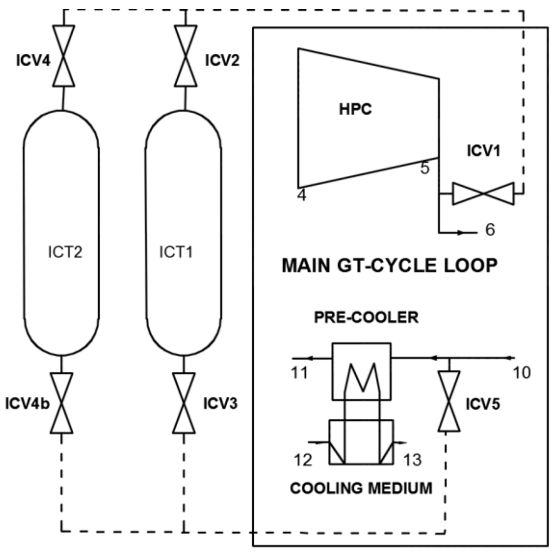

Figure 2: Case 2 - Inventory Control with Multi – Storage Tank using Natural pressure differential

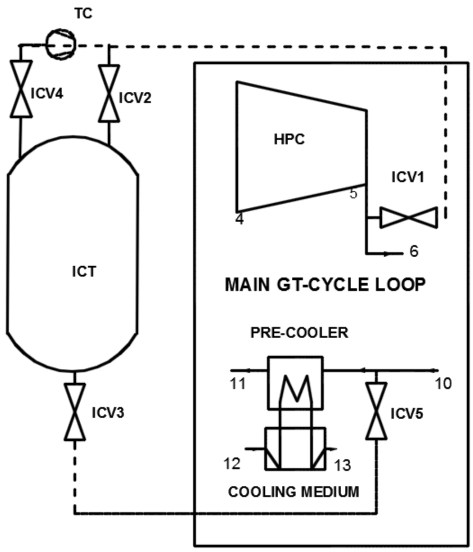

Figure 3: Case 3 - Inventory Control with Transfer Compressor (TC)

3. Philosophy of Inventory Control System

The inventory control system has been extensively used in most nuclear powered closed-cycle gas turbine plants [4,17] because it enables the plant to maintain high cycle performance at varying conditions. The operational logic of the ICS is that the power plant can store or save energy during off-peak periods and replenish this energy during peak load demand using the inventory control tanks (ICT). With the ICS in place, the working fluid can be extracted or injected into the gas turbine circuit, resulting in a related change in system pressure, change in density, change in mass flow rate and, change in the gas turbine output power [20]. Throughout the process, the plant cycle efficiency remains almost the same as its design point efficiency because the gas path temperatures ratios, and compressor pressure ratio stays the same during the part-load operation.

As shown in Figures (1) – (3), during the plant operations, the daily changes in power demand because of varying operating conditions are adjusted using the inventory control system (ICS). The ICS comprises of the inventory control tank (ICT), and the inventory control valves (ICV). A decrease in load demand from the grid will trigger the controllers and lift-flap of the ICV1 to open, and working fluid is discharged from the HPC exit into the ICT. Once the ICV opens, the pressure differential

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between the HPC and ICT or transfer compressor facilitates the flow of the working fluid into the ICT(s). The use of pressure differential (PD) or transfer compressor (TC) has its merits and demerits, which are discussed in this article. The pressure differential is regulated based on the pressure variation between the plant cycle and the inventory tank. When the HPC and ICT pressure is at equilibrium, the minimum power range of part-load using pressure differential is achieved. The TC can be used to obtain further extraction bearing in mind the system stability and integrity. The working fluid stored in the ICT(s) can be injected back into the gas turbine circuit by the opening of ICV3 if the grid demand increases again. Figure 4 shows a flow of command in the ICS system. When the ICS is initiated, the temperature input from the reactor and the gas turbine shaft speed is controlled to remain constant.

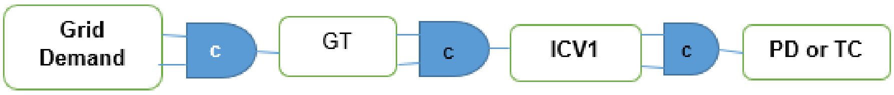

Figure 4: Flow command for the ICS System

4. Method of Analysis

A modified Bitsch et al., [17] model was implemented in the development of a physical procedure to represent the inventory load control system of the reference plant. The storage tank utilized for this study followed the American Society of Mechanical Engineering (ASME) standard pressure vessel design. The design and technical considerations for developing the ICTs are documented in references [21–24]. Figure 5 is an overview of the steps used in this analysis.

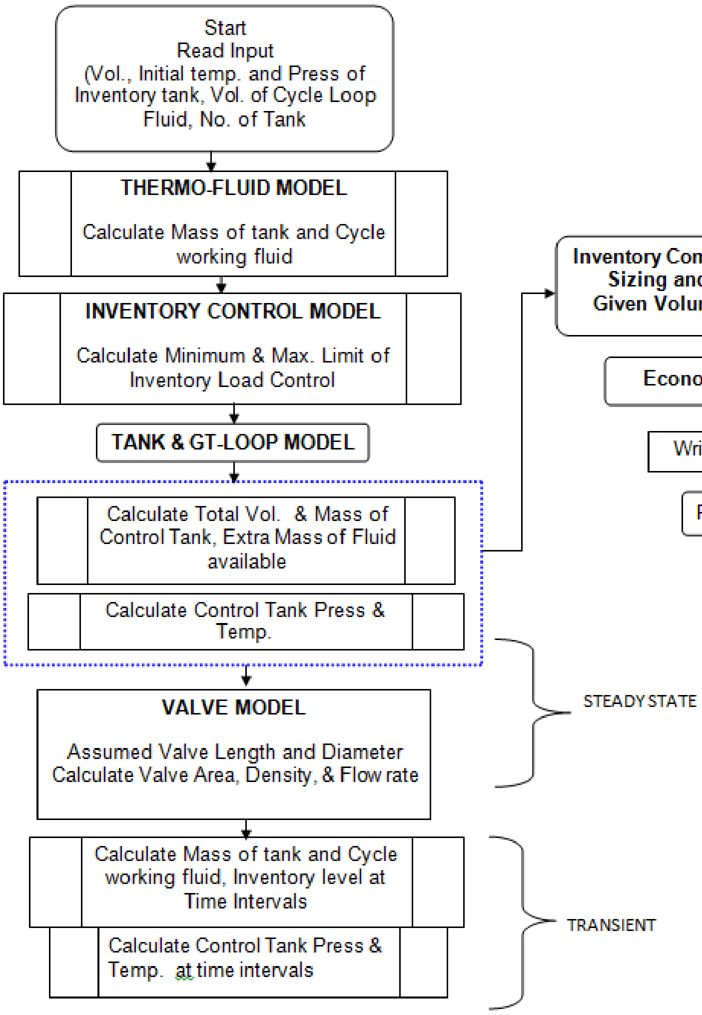

Figure 5: Procedure for Analysis

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The procedure for this analysis was modeled in a closed-cycle gas turbine performance simulation and design assessment tool developed by the authors at Cranfield University [20]. The input starts with an assumption on the initial volume, pressure, temperature, and number of the inventory control tank. These inputs are then passed on to the various models described in section 4.1 to 5.

4.1 Inventory Load Control Model

The minimum equilibrium inventory – control power range is given as

$$\Omega_{min} = \frac{1 + \frac{M_{T1}}{M_{GT5}}}{1 + \psi\left(\frac{M_{T1}}{M_{GT5}}\right)} \tag{1}$$

As the number of ICTs increases, the minimum equilibrium becomes

$$\Omega_{min} = \Omega_{min}{}^{\frac{1}{n}} \tag{2}$$

where n > 1

The maximum equilibrium inventory – control power range is given by

$$\Omega_{max} = \frac{1 + \frac{M_{T1}}{M_{GT5}}}{1 + \left(\frac{\psi}{OPR}\right)\left(\frac{M_{T1}}{M_{GT5}}\right)} \tag{3}$$

Where

$$\psi = \frac{P_5}{P_{T1}} \tag{4}$$

$$n = number\ of\ storage\ tank$$

$$OPR = overall\ cycle\ pressure\ ratio$$

4.2 Thermo-fluid model

The thermo-fluid model describing the behavior of the ICS follows the law of conservation of mass, conservation of momentum and conservation of energy to balance the fluid extraction and injection in the GT-cycle and storage tank relationship [16].

$$\Delta M_T = \Delta M_{GT} \tag{5}$$

Where

$$\Delta M_T = \frac{\Delta P_T \times V_T}{RT_t} = \frac{\Delta P_{GT} \times V_{GT}}{RT_{GT}} = \Delta M_{GT} \tag{6}$$

Since the storage tank is a stationary closed system and no boundary work is done, it means that $\delta KE = \delta PE = 0$ where KE is the kinetic energy and PE is the potential energy**.**

4.3 Storage Tank and GT-cycle loop

During low power demand (part-load operation), the ICV1 is opened for the extraction of working fluid into the inventory tanks. The HPC discharge temperature remains constant during this process because the reactor is controlled to maintain a steady supply of heat to the GT. Also, the temperature across the gas turbine gas path remains constant. There is a significant variation of the HPC exit temperature to the storage tank's initial temperature, which could have some heat transfer effect on

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the system. The impact of the rapid heat and mass transfer could be minimized by making the process to intermittently transfer the working fluid into the ICT, such that the ICT can cool by some degree before further extraction. The total mass and volume of the ICT is calculated using equation (8) and (9)

$$\Delta M_T = (1 - \Omega_{min}) \times M_{GT5} \tag{7}$$

$$V_T = \left[\frac{(1 - \Omega_{min}) \times M_{GT5} \times (RT_{t2} - RT_{t1})}{(P_{t2} - P_{t1})}\right] \tag{8}$$

The final tank temperature is calculated as

$$T_{t2} = \left[\frac{T_{t1} C_v M_{T1} + C_{pGT}(\Delta M_{GT}) T_{GT}}{C_v M_{T2}}\right] \tag{9}$$

Perforated insulation is implemented on the internal of the storage tank to absorb the heat from the gas turbine circuit fluid, thus lowering the tank temperature. Hence, the ICT final temperature becomes.

$$T_{t2} = T_{cap} + \Delta T_{lag} \tag{10}$$

Where T_{cap} is the temperature of insulation and ΔT_{lag} is the temperature difference between the cycle working fluid flowing to a storage tank and T_{cap}

4.4 Valve Model

The mass flow through the ducting valve during inventory change was determined using equation (11)

$$W = \sqrt{\frac{\Delta P \times \rho}{2f\left(\frac{L}{D}\right)}} \tag{11}$$

4.5 Sizing and weight model

In this study, a cylindrical pressure vessel was selected. This type of pressure vessel is generally preferred because it presents a simple manufacturing problem and better use of available space. The ASME VIII code was used as a reference guide [25] for the pressure vessel design. The ASME code design criteria consist of basic rules specifying the design method, Load, allowable stress, acceptable materials, and fabrication.

The pressure vessel consists of cylindrical shells, two ellipsoidal head, saddle supports, valves and reinforcement, and an insulator perforated on the walls of the cylindrical shell [24,26]. The procedure for determining the optimum size and weight of the storage pressure vessel based on references [27,28] was as follows

First, the tank capacity factor F is determined using equation (12)

$$F = \frac{P_{tDesign}}{C \times S_a \times E}, \qquad 0 \le C \le 0.125, \qquad 0.85 \le E \le 1 \tag{12}$$

Where

$$P_{tDesign} = (P_5 + P_{t1}) \times J \tag{13}$$

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Using the Value of F and Total Volume of Fluid in the storage tank, the diameter D is obtained using the chart in reference [27,28], and the total length of pressure vessel storage tank is given by

$$L = \frac{4V_T}{\pi D^2} \tag{14}$$

The thickness of the cylindrical shell (CST_T) is obtained as follows

$$CST_T = \frac{P_{tDesign} \times r_i}{\left[S_a E - 0.6P_{tDesign}\right]} + C \tag{15}$$

The internal volume of the cylindrical shell (V_{cs}) is given as

$$V_{cs} = \pi {r_i}^2 h \tag{16}$$

Where r_i is the storage tank internal radius

The minimum wall thickness of the ICT is determined from the maximum allowable stress and the material properties given in references [25,27]

Similarly, the required thickness of the dished head ellipsoidal (EH_T) was obtained by

$$EH_T = \frac{P_{tDesign} \times D \times K}{\left[2S_a E - 0.2P_{tDesign}\right]} + C\ , \qquad K = \frac{[2 + (D/2h)^2]}{6} \tag{17}$$

Where K is the stress intensification factor

The volume of the ellipsoidal (V_e) is given by

$$V_e = \frac{2\pi {r_i}^2}{3} \tag{18}$$

The total volume in terms of ellipsoidal and cylindrical shell volume gives

$$V_T = V_{cs} + V_e \tag{19}$$

Considering the internal volume the perforated insulator occupies, the insulator volume (V_c) is given by

$$V_c = V_T - \frac{w_c}{\rho_c} \tag{20}$$

Thus, the weight of the cylindrical shell is obtained from [27]

$$W_{CS} = \pi \times D \times CST_T \times L \times \rho \tag{21}$$

The weight of the ellipsoidal head (W_{EH}) is given as [27]

$$W_{EH} = 1.084D^2 \times EH_T \times \rho \tag{22}$$

And the weight of the insulator (W_c) is given as

$$W_C = V_T \times HC_{ratio} \times \rho \tag{23}$$

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Therefore, the total weight of the ICT is given as

$$Total\ Weight_{ICS} = (W_{CS} + W_{EH} + W_C) \tag{24}$$

5. Economic Model

Developing the inventory systems component size and weight was necessary to estimate the capital cost model. This article assumes the use of carbon steel (SA-516-70) as material for the ICTs. The cost model utilizes the weight estimates, length estimate, and the ICT material to forecast the cost of the ICS, according to reference [29].

Therefore, the total cost of the inventory control system was estimated as follow

$$Total\ Cost_{ICS} = F_M C_b + C_a + \text{Cost of Valves} \tag{25}$$

Where $F_M = Material\ cost\ factor$

$$C_b = exp[9.100 - 0.2889(In\ Total\ Weight_{ICS}) + 0.04333(In\ Total\ Weight_{ICS})^2] \tag{26}$$

C_b is an empirical value based on the weight of the ICS

$$C_a = 246D^{0.7396}L^{0.7068} \tag{27}$$

The cost of the valve was estimated based on the diameter of the valve using reference [30]

The cost of the transfer compressor was sized based on the minimum part-load power required and the pressure differential.

$$TC\ Power\ Capacity = (P - P_E) - P_{lmin} \tag{28}$$

Where $TC = Transfer\ compressor,\ P = Design\ Point\ Power,$

$P_E = Power\ at\ equilibruim\ pressure\ differential$

$P_{lmin} = Minmium\ required\ partload\ Power$

Therefore, the cost of transfer compressor was obtained using a power-law scaling factor

$$Cost = KA_a^n \tag{29}$$

Where $K = \frac{C_r}{A_r^n},\ n = cost\ exponent,\ A_a = TC\ Power\ Capacity$

$A_r = reference\ compressor\ power\ capacity, \qquad C_r = Reference\ compressor\ cost$

6. Results and Discussion

Figure 6 provides the cycle efficiency of the reference plant accomplished with the different control strategies applicable to the closed-cycle gas turbine. In this analysis, each control option was assumed to operate independently at the same power level. This article will only focus on the ICS in figure 6. During the operation mode of the ICS, the mass of air is withdrawn from the gas turbine cycle almost proportional to the power output. This withdrawal of mass flow reduces the density and pressures across the cycle. However, the pressure ratios of LPC and HPC remains constant because the operating point N/√T and the non-dimensional mass flow, does not change significantly; hence, the cycle efficiency remains relatively constant. At 50% output power, the cycle thermal efficiency is 40.1%. The slight drop in the cycle efficiency is a result of a small change in the working fluid properties. Nonetheless, this control option brings about a rapid acceleration of the shaft as the Load is reduced and is usually balanced with other control options.

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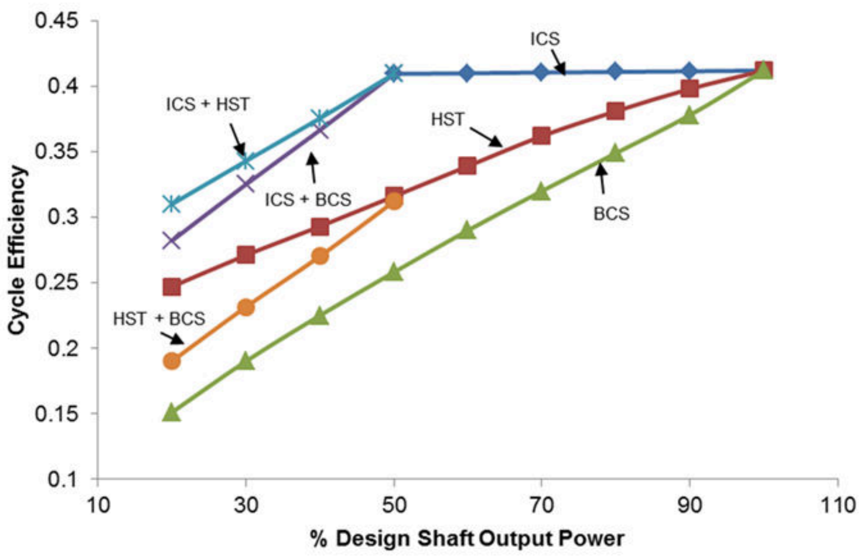

Figure 6 Comparison of efficiency for different control strategy at part-load

Figure 7 describes the impact of ICT initial pressure on the ICS power range using a single ICT. The blue line represents the minimum part-load range achievable during a reduction in shaft power to meet the grid requirement as a function of the ICT initial pressure. The red line represents the equivalent maximum part-load range possible when returning the working fluid from the ICT into the cycle loop through the pre-cooler inlet when the grid requires an increase in shaft output power. In both cases, the driving force of the flow is the pressure differential between the HPC exit and the reservoir tank. At an initial ICT pressure of 1 atmosphere (101.325 kPa), the shaft power is reduced by natural pressure differential (between the HPC discharge pressure and ICT pressure) to an optimal minimum of 32% obtained using equation (1). The minimum occurs when the HPC exit pressure and the ICT pressure reaches equilibrium. On the other hand, during the return of the shaft power (injection of fluid back to the GT-cycle loop), to full capacity using pressure differential, the maximum power recovered is 53%. This is because, at this point, the ICT pressure and the cycle intake pressure reaches equilibrium, which implies that a transfer compressor will be required for more fluid to be returned into the GT-cycle loop to achieve full power capacity (100% load).

As the initial tank pressure increases, the minimum part-load control decreases, which means that the level at which part-load operation can be attained at 10 atmospheres (1013.25 kPa) without incorporating an external compressor will be 52% shaft power. Similarly, an increase in the initial tank pressure will increase the maximum return to the full load range. As the initial tank pressure becomes equal or higher than the LPC inlet pressure, there are greater chances that the nominal value of 100% load can be reached. From this study, the minimum load-range at which full power capacity can be returned using pressure differential on a single ICT occurs at 50%. It is possible to achieve lower power output the single ICT using pressure differential; however, returning the fluid to full capacity will depend on the ICT initial pressure. The other alternatives are to use multiple ICT or to support the system with a transfer compressor. Increasing the number of ICT increases the total volume of the ICT and allows for more pressure differential downstream. The auxiliary transfer compressor will support further extraction of fluid from the GT circuit or injection into the GT circuit. Figure 6 also shows a decrease in the ratio of the HCP discharge pressure to the ICT initial pressure, and this could be used to determine the location of the auxiliary transfer compressor.

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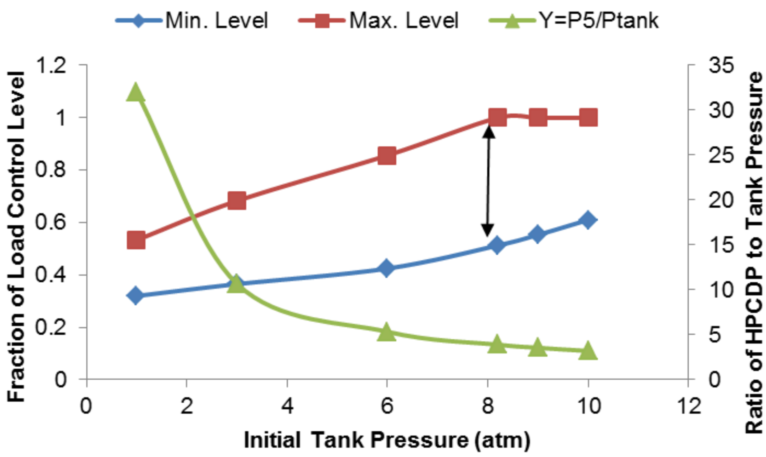

Figure 7: Effect of Tank Pressure on Inventory Control Level

Figure 8 represents the influence of the multiple ICTs on the load control level. During this analysis, the initial pressure of the ICT was assumed to be equal to the LPC inlet pressure, so that at any point in part-load operation, 100% output power return can be achieved (during injection of working fluid back into the gas turbine cycle). As the number of tanks increases, it becomes possible for the ICS to substantially increase the minimum part-load level and the ratio of extra mass to the cycle loop mass increases. Comparing with the results of the single tank at similar initial tank pressure, the minimum inventory load level for single ICT is 50%. Increasing to two (2) ICTs, we can obtain a 26% minimum load level. As the first ICT pressure reaches equilibrium with the HPC discharge pressure, the inventory control valve 4 (ICV4) in figure 3 opens, allowing more working fluid to flow out of the GT circuit. Increasing to three (3) ICTs shifts the minimum load level to approximately ~14%.

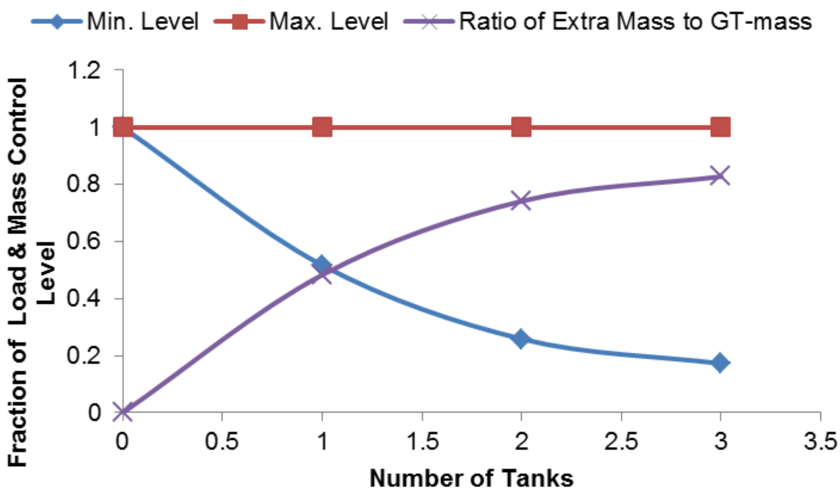

Figure 8: Effect of Number of Tanks

Inadequate cooling of the extracted fluid before flowing into the ICT will impact on the volume of fluid flowing into the tank due to an increase in the ICT pressure and temperature. Pre-cooler could be installed to keep the extracted fluid temperature close to the ICT temperature. However, introducing a pre-cooler will be an additional cost; hence, using a perforated insulator on the ICT internal walls could reduce the thermal effect and allow the ICT(s) to accommodate more volume of fluid. The valve size affects the rate of response of the ICS and cost. As valve diameter increases, the rate of ICV response increases.

Figures 9 and 10 represents a breakdown of the weight of ICT components and their respective cost. From the results, the cylindrical shell contributed over 70% of the total weight of the ICT. As the

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number of tanks increases, the weight per tank decreases, because of the decrease in volume per tank. The cost of the inventory control system was estimated based on the weight of the individual components. The total cost of ICT also increases with the number of ICTs; however, the proportion of increase reduces when the third ICT was installed. An economic trade-off on which approach to consider in terms of the capital cost is described in Figure 11. The decision to take any approach may also require maintenance, operations, and contingency cost considerations. However, the results represented in figure 11 favours the use of multiple tanks to achieved optimal minimum load range as compared with the cost of installing auxiliary transfer compressor by a factor of 0.08 at 17% load control level. This is so because as the number of tank increases, the material cost compared with a single ICT is reduced per kg.

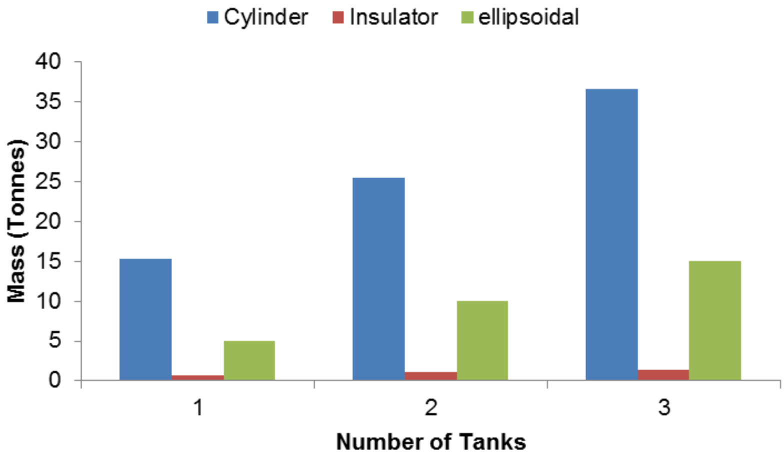

Figure 9: Mass Break down of Inventory Control Tank

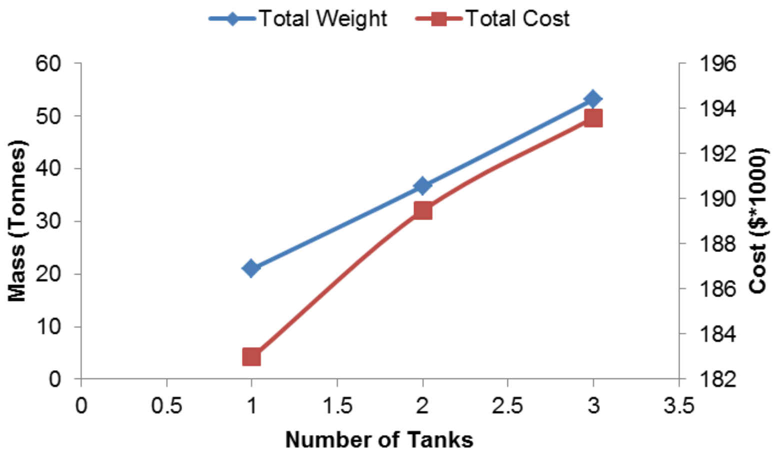

Figure 10: Weight and Cost Analysis of ICS

Journal of Nuclear Engineering and Radiation Science. Received September 15, 2019;
Accepted manuscript posted October 23, 2020. doi:10.1115/1.4048903

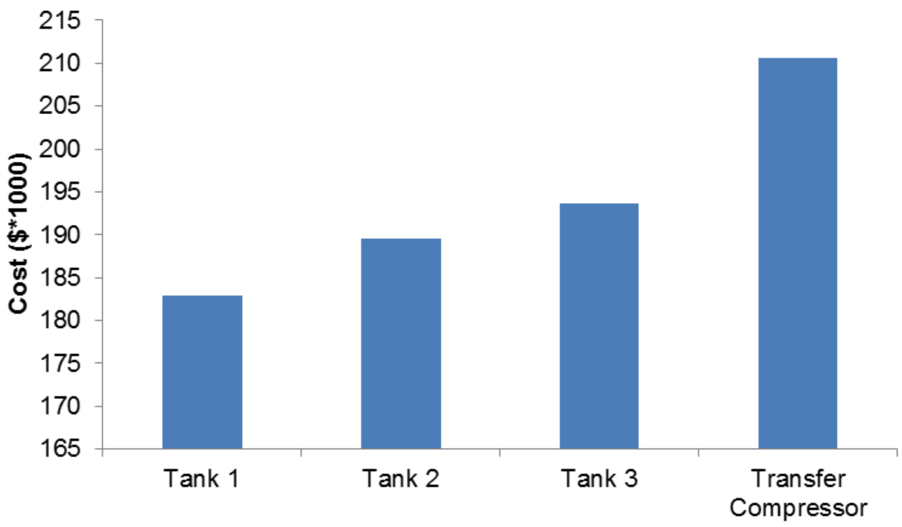

Figure 11: Trade-off Cost Economics of Inventory Control Options

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7. Conclusions

The objective of this article was to ascertain the techno-economic benefit of utilizing pressure differential on a single ICT, multi-ICTs, or introducing a transfer compressor to support the ICS. The merit of using a single ICT is for simplicity and cost purpose; however, achieving the minimum part-load level without a TC will be its downside. The use of multiple inventory control tanks gives better part-load handling using pressure differential. Still, one must factor in the rate of the system response as the number of ICT increases, since grid fluctuation requires rapid system response. The merit of using the transfer compressor is the response time and better performance of the ICS.

As the tank, initial pressure becomes higher than the LPC inlet pressure, the minimum level of part-load operation that is achievable using pressure differential reduces. If the ICS is to be designed such that 100% inventory can be restored without the use of an auxiliary transfer compressor, then the ratio of the cycle peak pressure to the ICT initial pressure should be relatively equal to the cycle overall pressure ratio. The capital cost of implementing an increased ICT volume provides a better option compared with the auxiliary transfer compressor cost on a long-term assessment. The pressure differential provides a preferred mechanism as it takes advantage of the existing pressure variation between the GT-cycle loop and ICT without reliance on additional turbomachinery, which is potentially expensive.

Nomenclature

A_r	reference compressor power capacity
BCS	bypass control system
C	corrosion allowance
C	controller
C_b	empirical cost of inventory control tank
C_r	reference compressor cost
CST_T	thickness of the cylindrical shell, m
C_p	specific heat of the gas at constant pressure, J/kg K
C_v	specific heat of the gas at constant volume, J/kg K
D	diameter, m
E	joint efficiency
EH_T	thickness of dished head ellipsoidal, m
F	tank capacity factor
F_M	material cost factor
GT	gas turbine
GH	gas heater

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H	enthalpy J/kg K
HST	heat source temperature
h	height, m
HPC	high-pressure compressor
IC	inter-cooler
ICS	inventory control system
ICT	inventory control tank
ICV	inventory control valve
K	stress intensification factor
L	length, m
LPC	low-pressure compressor
M_{GT}	mass of GT-cycle loop, kg
M_T	mass of tank, kg
OPR	overall pressure ratio
P	pressure of the tank, Pa
P_o	design point power, W
P_E	power at an equilibrium pressure
PD	pressure differential
PR	pressure ratio
P_T	pressure of the tank, Pa
R	gas constant J/kg K
RX	recuperator
S_a	stress value of the material, Pa
T	temperature, K
TC	transfer compressor
V_{CS}	volume of the cylindrical shell, m^3
V_e	volume of ellipsoidal, m^3
V_T	volume of the tank, m^3
V_{GT}	volume of fluid in gas turbine cycle loop, m^3
W	mass flow rate, kg/s
W_C	weight of insulator, kg
W_{CS}	weight of the cylindrical shell, kg
W_{EH}	weight of the ellipsoidal head, kg

Greek Symbols

ε	effectiveness
η	efficiency
γ	ratio of specific heats
ρ	density, kg/m^3
f	friction factor
j	factor of safety
Δ	difference
Ω	power control range
Ψ	ratio of HPC exit pressure and tank pressure

Subscripts

1	initial
2	final
5	HPC exit
1-7	*station number*

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2020-10-23

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Osigwe EO, Gad-Briggs A, Obhuo M, Pilidis P. (2020) Techno-economic study on implementation of inventory control requirements for a nuclear powered closed-cycle gas turbine power plant. Journal of Nuclear Engineering and Radiation Science, Volume 7, Issue 3, July 2021, Article number 031301

https://doi.org/10.1115/1.4048903