Araştırma Makalesi
Bayburt Üniversitesi Fen Bilimleri Dergisi
Cilt 4, Sayı 2

Geliş Tarihi: 21.12.2021
Kabul Tarihi: 23.12.2021

dergipark.gov.tr/bufbd

Modelling and Performance Investigation of Semi-Closed Gas Turbine Cycle with Spray Cooling

Erman K. OZTEKIN*[1], William E. LEAR[2]

[1]Bayburt University, Mechanical Engineering Department, Bayburt, Turkey
[2]University of Florida, Mechanical & Aerospace Engineering Department, Gainesville, United States

Keywords:

Gas turbine
Ethylene glycol
Drying
Semi-closed
Thermal efficiency

Abstract

The performance of a novel system, Power, Water Extraction, and Refrigeration cycle (PoWER), has been under investigation recently which is a combination of high-pressure regenerative turbine engine cycle (HPRTE) and a vapor absorption refrigeration system cycle (VARS). After the burned gas expand in the turbine, some portion is mixed with the pressurized fresh air and taken back into the cycle. Some of the waste heat is used by VARS through several heat exchangers. By this way external cooling load can be efficiently obtained. The efficiency of the semi-closed gas cycle can be further improved by achieving below-freezing temperatures of pure water at the high-pressure compressor inlet. Solution of ethylene glycol and water can be directly sprayed into the gas stream to both dry and to lower the temperature of the gas flow simultaneously. The computer model of the PoWER cycle with simple VARS model by considering second-law thermal efficiency and glycol cycle are used to compare the thermal efficiencies of the system when cooling is only accomplished by VARS and cooling is accomplished with both VARS and spraying aqueous glycol solution into the gas stream. Ethylene glycol is used as desiccant since it has very low vapor pressure and has a relatively low freezing point. In this study, aqueous glycol solution is used to prevent possible crystallization in the gas stream which may seriously damage the compressor blades. Two systems with gas engines in different scales are investigated for this purpose. By enabling spray cooling the overall thermal efficiency of the system with medium size engine can be increased by 2.2%, whereas the efficiency of the system with the small size engine can be increased by 2.7% by using glycol-water solution with 60% ethylene glycol concentration.

1 INTRODUCTION

Gas turbines find a wide area of use in the military and industry. Generally ambient air is used as the working fluid and to react with fuel [1]. Gas turbines are a type of internal combustion engine and the operation of the gas turbine design fundamentally follows Brayton cycle principles [2]. The basic steps of the gas turbine cycle can be listed as: air-intake, compression via compressor, combustion in the burner, expansion via turbine and exhaust. Open gas cycles tend to release the working fluid after expansion in the turbine but the temperature of the released fluid may still be high [1]. This means that the gas is rejected without utilizing its remaining available thermal energy. In contrast to open cycle gas turbines, a basic semi-closed cycle gas turbine model was initially designed by Anxionnaz where the amount of fresh air-intake is significantly reduced [3, 4]. In the semi-closed gas cycle, instead of rejecting the whole gas stream directly to the ambient, a certain amount of burned gas is fed back to the exit of the compression stage and is mixed with a certain amount of compressed fresh air. In early development stages, semi-closed gas turbines were proposed to be used as power plants and propulsion units in submarines [5, 6]. Later, improvements in power and efficiency were shown in the High Pressure Regenerative Turbine Engine (HPRTE) which is a a new type of modified semi-closed gas engine [7]. In order to increase the efficiency of the gas turbine cycle and to use available waste heat, multiple cycles can be combined to produce power, as well as using waste heat in various ways and obtaining multiple energy products. For example, combining a vapor absorption refrigeration system (VARS) with the high-pressure regenerative turbine engine (HPRTE), a type of semi-closed gas turbine, results in higher thermal efficiencies and additionally provides external cooling capacity [8]. Moreover, it was initially suggested that fresh water can be obtained from this combined design [9]. The design is

*e-Posta: ermanoztekin@bayburt.edu.tr

named the Power, Water Extraction, and Refrigeration cycle (PoWER) and is aimed to produce power at high efficiencies as well as providing heat, cooling and fresh water output [10, 11].

Nemec et al. showed that using steam as a working at superheated state in the cycle can maximize the system efficiency of HPRTE cycle and Rankine cycle combination, though they did not analyze PoWER [12]. Boza calculated thermal efficiencies of HPRTE and VARS combination (ie PoWER) where lithium bromide - water was used as working fluid in the VARS system [13]. The HPRTE cycle was modeled with two different gas engines at different scales. Khan developed HPRTE model and presented the simulation results of the optimized model [11, 14]. Choon calculate the off-design air-conditoning and ice-producing capacity of PoWER under different loads by applying a constant second-law efficiency [15]. Over the years it was shown that system efficiency can be increased by increasing turbine inlet temperatures [16]. Material and cooling technologies play an important role to prevent material damage under high turbine inlet temperatures [17]. It is also known that the efficiency of the semi-closed gas turbine cycle can be improved if the compressor inlet temperature can be lowered [8, 14]. In the current study the performance of the PoWER system is simulated with compressor inlet temperatures below freezing. Thus, an extra binary solution of ethylene glycol and water is directly exposed to the gas mixture working in the power cycle as a direct contact heat exchanger. This aqueous desiccant sprayed directly into the gas flow path at the inlet of the intermediate compression stage prevents moisture from freezing. Aqueous solutions are commonly preferred in absorption dehumidification systems such as triethylene glycol (TEG), aqueous LiC and most glycol solutions [18, 19].

2 MATERIAL AND METHOD

2.1 Configuration of PoWER and Glycol Cycle

The flow path of the modified PoWER cycle and tags of the input and output states of the gas mixture can be seen from the Figure 1.

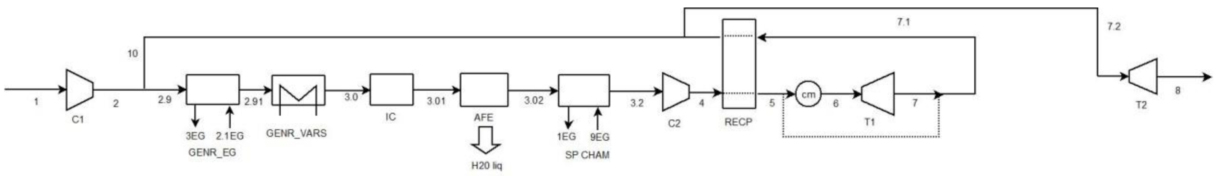

Figure 1. Flow path of the PoWER cycle.

Ambient air is entered at state 1 to the low-pressure compressor (C1). Then fresh air adiabatically mixes with recirculated burned gas coming from the recuperator output and continued to enter the several cooling steps. Initially, the mixed flow provides its high-grade thermal energy to the generator of the glycol cycle and heat exchanger element of Vapor Absorption Refrigeration System (VARS) cycle between stage 2.9 to stage 3.0. Then gas mixture rejects additional heat to the surroundings between stage 3.0 and stage 3.01 in an intercooler (IC). GENR_VARS and the above freezing evaporator (AFE) are components of the VARS cycle. Before entering the high-pressure compressor (C2), more thermal energy is extracted from the gas mixture at AFE and finally cold aqueous ethylene glycol solution is sprayed into the spray chamber and the temperature of the output mixture is dropped below freezing of water at stage 3.2. The binary solution composed of water and ethylene glycol prevents moisture in the gas to be crystallized before entering the high-pressure compressor. After compression the dried gas mixture is pre-heated in the recuperator (RECP) and burned with fuel in the combustor (cm). The combustion products with high-grade thermal energy are expanded in the first turbine (T1). The gas mixture leaving the turbine transfers some of its thermal energy in the recuperator (RECP). Some portion of the gas mixture follows the semi-closed path to mix with fresh air initially compressed in the low-pressure compressor. Leftover mixture expands in the second turbine (T2) and exits to the atmosphere. In this modified model, two heat exchangers are using high-grade thermal energy where initially heat is extracted for the ethylene glycol cycle (EG cycle) and then for the VARS cycle. Since the value of the bubble point temperature of the binary solution is relatively higher than the boiling point of water, high-temperature thermal energy is provided first to the EG cycle. The flow path of the glycol cycle can be seen from the Figure 2.

The EG cycle and PoWER cycle are connected at spray chamber where specified amounts of water and ethylene glycol solution are sprayed into the gas flow and collected back into the glycol flow path. Dried and colder gas leaves the spray chamber and enters the compressor. By doing so, possible crystallization in the gas flow flowing at subfreezing temperatures is prevented. The binary solution collected in the spray chamber is pre-heated in the heat exchanger (HEX) and expanded in the valve. The solution is heated above the bubble point at GENR_EG where heat is provided from mixed gas flow and water is extracted in the separator. The amount of water which has to rejected in the separator is directly related to mass flow of moisture in the gas mixture to keep the water balance. Then the concentrated glycol is cooled and pumped back into the spray chamber to close the cycle.

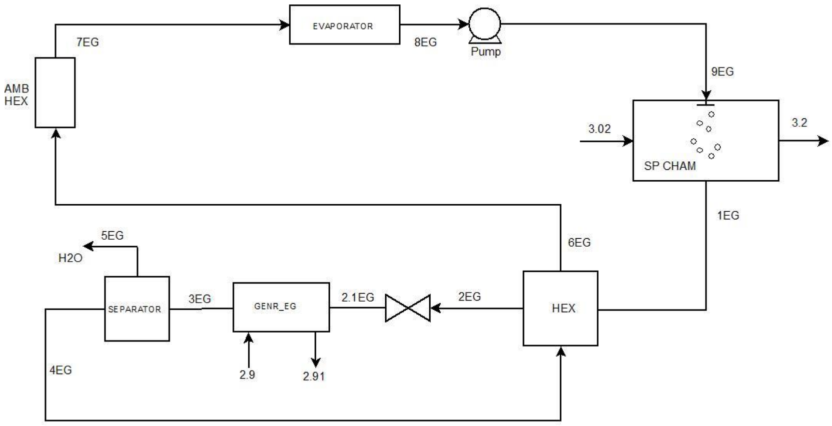

Figure 2. Flow path of glycol cycle.

2.2 Mathematical Model and Solution Method

Classical zero-dimensional steady state thermodynamics models are implemented for each components of the cycles. Polytropic efficiencies are used for turbines and compressors. Conversation laws, namely mass and energy are applied to heat exchangers. Curve fitting is used to calculate enthalpy changes by defining thermodynamic parameters at the input and the output of the control volumes. In addition to these some limitations are applied such as the exit temperature of the AFE is limited so that ice does not form based on various concerns for the VARS cycle.

Reference Fluid Thermodynamic and Transport Properties Database program of National Institute of Standards and Technology (NIST REFPROP) is used to obtain thermodynamic properties of the binary solution. Ethylene glycol is used to produce binary mixture since it has very low vapor pressure and leaks in the spray chamber and the separator are minimal. Effectiveness parameters are selected to characterize heat exchangers. The maximum temperature of the combustion is directly related with the system efficiency. So, 14% of the pressurized gas flow is bled to cool the first turbine where the value is consistent with the literature [8, 14, 20]. Adiabatic mixing is considered at the junction nodes. The performance of the VARS cycle is calculated by considering second-law thermal efficiency and Carnot engine working under similar thermal parameters. Simplified energy and mass balance equations is used for the spray chamber. Sprayed binary solution at state 9EG interacts with gaseous moisture in the gas flow and condensed water is added to the binary solution where all liquid is assumed to be collected inside the spray chamber at State 1EG. So, the dried gas mixture can move forward at State 3.2 with no water content. Total mass flow rate of the binary aqueous glycol solution can be obtained from Eq. 1

$$\dot{m}_{binarysol} = \dot{m}_{watermiosture} + \dot{m}_{sprayedgycol} \tag{1}$$

Mass flow rate of the water moisture carried by the gas flow can be estimated from humidity ratio since we know that gas mixture is saturated at given temperature.

$$\omega = 0.622\left(\frac{\phi P_{sat}}{P_{total} - \phi P_{sat}}\right) \tag{2}$$

$$P_{sat} = P_{sat}(T_{3.02}) \tag{3}$$

$$\dot{m}_{3.02W} = \omega \dot{m}_{3.02Gas} \tag{4}$$

Energy balance can be applied to the control volume taken around the spray chamber.

$$\dot{Q}_s - \dot{W}_s = \dot{m}_{1EG}\left(h_{1EG} + \frac{v_{1EG}^2}{2} + gz_{1EG}\right) + \dot{m}_{3.2}\left(h_{3.2} + \frac{v_{3.2}^2}{2} + gz_{3.2}\right) - \dot{m}_{3.02}\left(h_{3.02} + \frac{v_{3.02}^2}{2} + gz_{3.02}\right) - \dot{m}_{9EG}\left(h_{9EG} + \frac{v_{9EG}^2}{2} + gz_{9EG}\right) \tag{5}$$

Enthalpies of the of the gas mixture and binary solution can be calculated from energy curves. To calculate enthalpy of the gas two thermodynamic parameters should be entered on the other hand to calculate enthalpy of the gas three thermodynamic parameters should be entered. Mathematical expression can be written as follows:

$$h_{Gas} = h_{Gas}(P,T) \tag{6}$$

$$h_{solution} = h_{solution}(P,T,x) \tag{7}$$

The definition of effectiveness can be implemented to calculate the change in temperature in both streams of heat exchangers of glycol cycle by using Eq 8. The quality of the mixture has to be determined in the separator since three parameters are required to calculate temperature. Eq. 10 is used to determine temperature of the binary solution.

$$\varepsilon = \frac{(mC_p)_{hot}(T_{in} - T_{out})_{hot}}{(mC_p)_{\min}(T_{hot,in} - T_{cold,in})} = \frac{(mC_p)_{cold}(T_{out} - T_{in})_{cold}}{(mC_p)_{\min}(T_{hot,in} - T_{cold,in})} \tag{8}$$

$$q_{3EG} = \frac{\dot{m}_{3EG,vapor}}{\dot{m}_{3EG}} \tag{9}$$

$$T_{3EG} = T_{3EG}(P_{3EG}, q_{3EG}, x_{3EG}) \tag{10}$$

The work of the pump can be calculated by using isentropic pump and adiabatic efficiency parameter definition.

$$\dot{Q}_{pump} + \frac{\upsilon_{8EG}(P_{9EG} - P_{8EG})}{\eta} = \dot{m}_{9EG}h_{9EG} - \dot{m}_{8EG}h_{8EG} \tag{11}$$

$$\upsilon_{8EG} = \upsilon_{8EG}(P_{8EG}, T_{8EG}, x_{8EG}) \tag{12}$$

Excel and MATLAB are used to simulate PoWER cycle and EG cycles. The main code written in MATLAB is coordinating inputs and outputs from PoWER cycle written in Excel and property curves obtained from NIST REFPROP program. The code of PoWER cycle is capable of calculating water extraction rate and net power output by calculating all undefined thermodynamic states of PoWER cycle. Whereas code of the EG cycle is capable of calculating the amount of heat to separate sufficient water from the binary solution to keep the balance. AFE exit temperature is taken 3^0C to prevent water from undesired freezing. The calculation has converged when necessary heat is provided in the separator. Table 1 shows the design parameters for the engines characterized in different sizes. Several assumptions have been made for calculations. For instance, pressure drops are neglected in heat exchangers so the glycol cycle was assumed to work under two different pressures. Gas leaving the spray chamber is assumed to leave in a completely dry state.

Table 1. Base values for independent variables

	Medium Size Engine	Small Size Engine
Turbine Inlet Temperature [K]	1722	1500
Recuperator Inlet Temperature (Hot Side) [K]	1000	1000
Ambient Temperature [°C]	30	30
Generator Exit Temperature [°C]	100	100
Above Freezing Evaporator Temperature [°C]	3	3
LPC Pressure Ratio	3.5	3.5
Combustor Equivalence Ratio	0.9	0.9
Ratio of mass flow rate of water extracted to mass flow rate of fuel	1.51	1.51
Turbomachinery polytropic efficiencies	0.9	0.84
Effectiveness of recuperator	0.9	0.9
Effectiveness of intercooler	0.9	0.9
Pressure drop in recuperator hot side	3%	3.5%
Pressure drop in recuperator cold side	3%	3.5%
Pressure drop in combustor	5%	5%
Pressure drop in generator	3%	3.5%
Pressure drop in above freezing evaporator	3%	3.5%
Pressure drop in intercooler	3%	3.5%
Bleed fraction	0.14	0.14
Ratio of mass flow rate of solution to mass flow rate of air inside spray chamber	0.3	0.3
Concentration of ethylene glycol in aqueous solution	60%	60%
Low pressure value of glycol loop [kPa]	200	200
Effectiveness of heat exchanger in glycol loop	0.9	0.9
Isentropic efficiency of solution pump	0.8	0.8

3 RESULTS AND DISCUSSION

Two engines in different sizes are considered for simulations; the small one has a nominal power of output of 500 kW and the medium one has a nominal power of output of 10 MW. Thermal efficiency is the most important indicator of the advantage of spraying, since low temperatures at the inlet of the HPC has a direct effect on the overall thermal efficiency. This parameter is calculated by dividing net power output by lower heating value of the fuel. In this work VARS is not specifically designed but the ordinary coefficient of performance is taken from the literature where effective efficiency and ordinary efficiencies are not differ significantly. [Boza] Material durability and cooling technology are considered to limit burner exit temperature T6 and inlet temperature of recuperator T7.1. Here freezing point of glycol solution is very important since system operates at sub-freezing temperatures. The glycol-to-gas relative mass flow rate τ is defined in Eq. 13.

$$\tau = \frac{\dot{m}_{binary\ solution\ in\ the\ spray\ chamber}}{\dot{m}_{gas\ mixture\ in\ the\ spray\ chamber}} \tag{13}$$

Burner inlet temperature has a major effect on the entropy change which is a minimum temperature where heat is added to a cycle by an external heat source. The COP of the refrigeration cycle is affected by the inlet temperatures of the hot reservoirs and output temperature of the cold reservoirs as well as a temperature of the ambient. Parametric analyses are applied to small and medium sized engines. Recuperator inlet temperature (T7.1) is one of the most important parameters in the system since it can both affect the gas turbine cycle efficiency and available waste heat for cooling. The performance parameters are compared with the similar ones when the spray cooling is off with cooling only by the VARS.

3.1 Medium Scale Engine

Several parameters which are thought to have significant of efficiency of the system are selected and parametrically investigated. Figure 3 shows how efficiency of the system is changed with respect to recuperator inlet temperature.

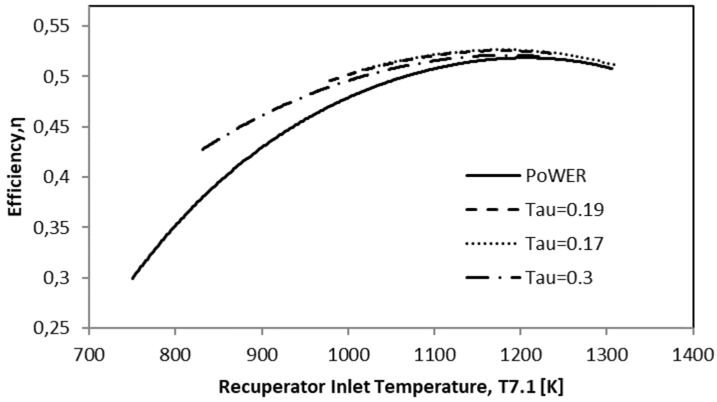

Figure 3. The variation in thermal efficiency of the PoWER system versus different recuperator inlet temperatures for different solution ratios for medium size engine.

Several plots are presented for different values of τ . The selection of τ depends on the limitations considered for recuperator. The blade cooling temperature and the recuperator inlet temperature are same so thermal properties of blade material are also taken into consideration. Although there is a relatively significant change in the system efficiency when the value of τ is selected as 0.3, the change is not at the same rate when this value is selected as 0.19. When the value of τ is 0.3, the system can operate in a wider recuperator inlet temperature range and the system efficiency increases by 1.6. Maximum system efficiency is observed when the recuperator inlet temperature is around 1600 K for medium scale engine. Figure 4 shows that the change in recuperator inlet temperature is directly proportional to high pressure compressor inlet temperature.

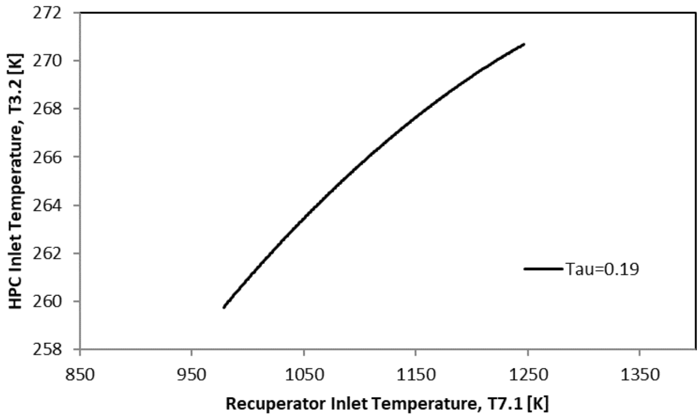

Figure 4. The variation in high pressure compressor inlet temperature with respect to different recuperator inlet temperatures for medium size engine

The observed behaviour is because the system has left with more useable waste heat to cool down the flow temperature when recuperator inlet temperature decreases. Figure 5 shows how the system efficiency changes with respect to different low-pressure compression ratios.

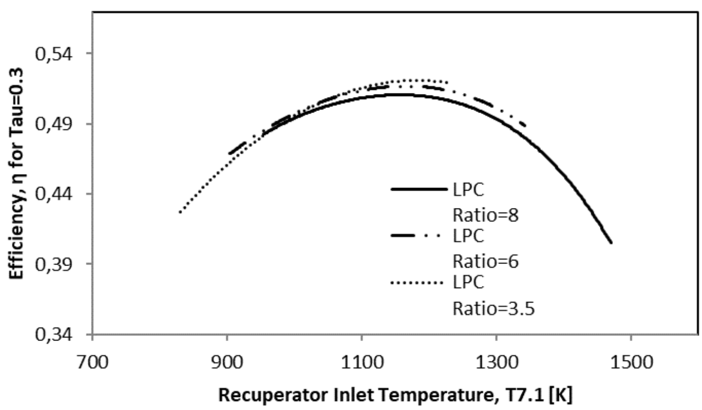

Figure 5. The variation in thermal efficiency of the system with respect to different recuperator inlet temperatures for different low-pressure compressor (LPC) ratios for medium size engine

Lower values of LPC ratios will yield higher thermal efficiencies since the work required to run compressor decreases. But when the output temperature of the compressor decreases at lower values of LPC ratio the available waste heat may not meet the energy required to bring the solution to above bubble point at given pressure and concentration for high values of τ. System efficiency decreases for low pressure compression ratios higher than 3.5 but the efficiency becomes more insensitive to pressure ratio changes around 1000K.

3.2 Small Scale Engine

The same investigation trend is followed for small scale engine. Figure 6 shows the variation of efficiency with respect to recuperator inlet temperature for small scale engine.

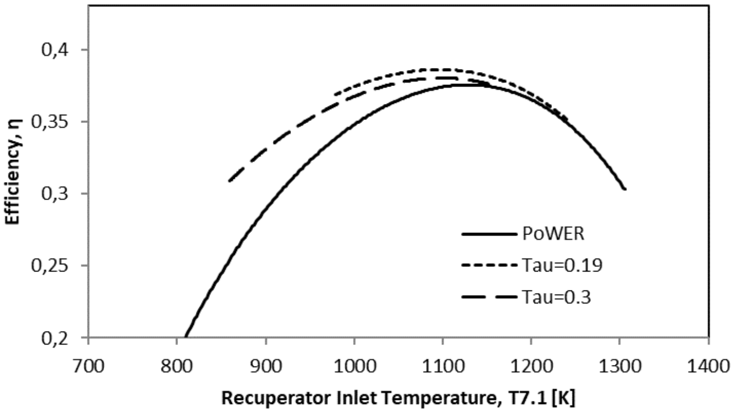

Figure 6. The variation in thermal efficiency of the PoWER system versus different recuperator inlet temperatures for different solution ratios for small size engine.

Again, the value of 0.19 yields a slightly higher efficiency profile than the value of 0.3 according to Figure 6. Efficiency tends to decrease under the the value of 1110 K of the recuperator inlet temperature. Cooling with glycol cycle increases the system efficiency around 2% at the value of recuperator inlet temperature is around 1000 K. Figure 7 shows direct relation between inlet temperature of high-pressure compressor and inlet temperature of recuperator for small scale engine.

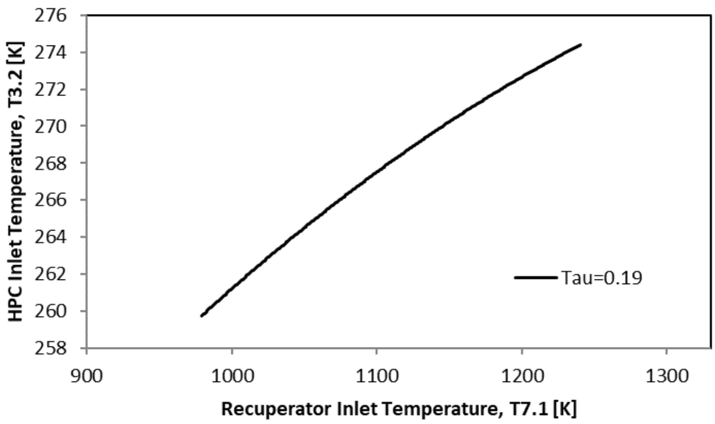

Figure 7. The variation in high pressure compressor inlet temperature with respect to different recuperator inlet temperatures for small size engine.

When pressure ratio of the low-pressure compressor increases, the system has more available waste heat but the thermal efficiency of the system decreases as expected which can be seen from Figure 8.

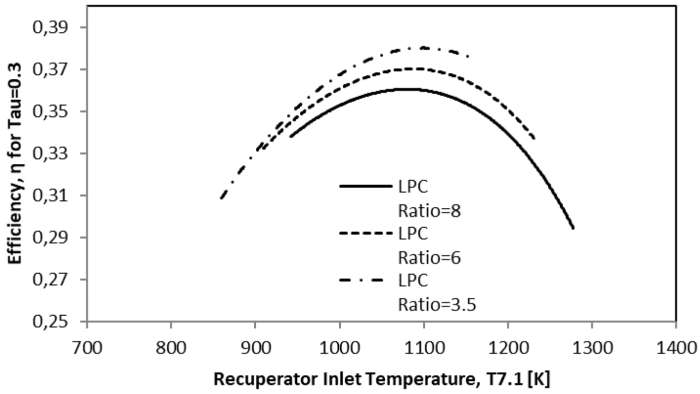

Figure 8. The variation in thermal efficiency of the system with respect to different recuperator inlet temperatures for different low-pressure compressor (LPC) ratios for small size engine

4 CONCLUSIONS

In this study, PoWER semi-closed cycle and glycol cycle are modelled by using zero-dimensional steady state thermodynamics. Moreover, VARS is modelled by using a second law efficiency approach. To check the benefit of the spraying procedure two different engines in different scales are modelled. Precooling of the gas is provided by the elements of the VARS system both directly and indirectly. The system temperature had been dropped to the ambient temperature at the high-pressure compressor inlet. Here, the improvement in efficiency of the system is investigated under sub-freezing temperature at the high-pressure compressor inlet. Spraying aqueous glycol solution into the gas mixture makes this investigation possible since the freezing point of the solution drops significantly with respect to freezing point of pure water. The following conclusions can be made according to the results:

- The overall thermal efficiency of the system with medium size engine can be increased by 2.2% up to efficiency value of 50% by spraying the binary glycol mixture, whereas with the small size engine it can be increased by 2.7% up to an efficiency value of 37.5%.
- System efficiencies are less affected with changes in recuperator inlet temperatures at certain working conditions when spraying is available. The increase in system efficiency with small scale engine is higher than the increase in system efficiency with medium scale engine when spraying is available.
- The mass flow rate of the binary solution has to be determined in advance, since the increase in mass flow rate of the solution results in decrease in system efficiency but higher mass flow rates prevent freezing.
- An increase in the system efficiency is demonstrated for the first time with the spraying method. Namely, aqueous glycol solution with relatively low freezing point is used to prevent possible crystallization in the gas stream.
- Since ethylene glycol poses a serious risk to human health and the environment, another liquid can be chosen instead. i.e. propylene glycol. Also, specific optimization method can be applied to choose optimal solution to gas flow mass ratio for the future studies.

Nomenclature

Variables		**Greek Letters**		**Subscripts**		**Abbreviations**	
COP	Coefficient of performance [-]	γ	Specific heat ratio [-]	amb	Ambient	AFE	Above Freezing Evaporator
f	Fuel air ratio [-]	ε	Heat exchanger effectiveness [-]	comp	Compressor	cm	Combustor
h	Specific enthalpy [kJ/kg]	η	Efficiency [-]	EG	Ethylene glycol	C1	Low pressure compressor
$\dot{m}$	Mass flow rate [kg/s]	ϕ	Relative humidity [-]	evap	Evaporator	C2	High pressure compressor
P	Pressure [kpa]	ω	Specific humidity [-]	gen	Generator	GENR	Generator
$\dot{Q}$	Heat transfer rate [kW]			in	Inlet	IC	Intercooler
R	Recirculation ratio [-]			max	Maximum	RECP	Recuperator
T	Temperature [K]			out	Outlet	T1	High pressure turbine
$\dot{W}$	Power [kW]			p	Polytropic	T2	Low pressure turbine
				sat	Saturation		
				turb	Turbine		
				W	water		

References

[1] Çengel, Y.A., M.A. Boles, and M. Kanoglu, *Thermodynamics: An Engineering Approach*. Boston: McGraw-Hill Higher Education, 2020.

[2] Smoot, L.D. and L.L. Baxter, 'Fossil Fuel Power Stations—Coal Utilization' in *Encyclopedia of Physical Science and Technology (Third Edition)*, Academic Press, 2003, pp. 121-144.

[3] Anxionnaz, R., 'Installation a turbines a gaz a circuit semi-ouvert.' French Patent 999–133, 1945.

[4] Anxionnaz, R., 'Improvements in or relating to gas turbine plant with semi-open circuit.' British Patent 651–166, 1948.

[5] Gasparovic, N., 'The Advantage of Semi-Closed Cycle Gas Turbines for Naval Ship Propulsion,' *Naval Engineers Journal*, 1968.

[6] DeWitt, S.H. and W.B. Boyum, 'Internally Fired Semi-Closed Cycle Gas Turbine Plant for Naval Propulsion,' *in ASME 1956 Gas Turbine Power Conference*, 1956.

[7] Lear, W., A. Laganelli, and P. Senick, 'High Pressure Regenerative Turbine Engine: 21st Century Propulsion,' NASAjCR-2001-210675, Hanover, MD(United States), 2001.

[8] Khan, J.R., 'Modeling and optimization of a novel pressurized CHP system with water extraction and refrigeration,' *International Journal of Energy Research*, vol. 32, pp. 735-751, 2008.

[9] MacFarlane, R., et al., 'System impact of H2O production and injection on a novel semi-closed cycle gas turbine,' in *35th Aerospace Sciences Meeting and Exhibit*, American Institute of Aeronautics and Astronautics, 1997.

[10] Lear, W.E. and S.A. Sherif, 'Combined cooling and power plant with water extraction,' US Patent, US7472550B2, 2009.

[11] Khan, J.R., et al., 'Testing and Modeling of a Semi-Closed Gas Turbine Cycle Integrated With a Vapor Absorption Refrigeration System,' in *ASME 2006 International Mechanical Engineering Congress and Exposition*, 2006, p. 137-148.

[12] Nemec, T.S. and W.E. Lear, 'Thermodynamic Design Point Study of a Semi-Closed Recuperated Intercooled Gas Turbine Combined With a Rankine Bottoming Cycle,' in *ASME 1996 International Gas Turbine and Aeroengine Congress and Exhibition*, 1996.

[13] Boza, J.J., W.E. Lear, and S.A. Sherif, 'Performance of a Novel Semiclosed Gas-Turbine Refrigeration Combined Cycle,' *Journal of Energy Resources Technology*, vol. 130, 2008.

[14] Khan, J.R., 'Modeling and experimentation of a novel pressurized CHP system with water extraction,' *International Journal of Energy Research*, vol. 32 pp. 1030-1046, 2008.

[15] Ryu, C.J., 'Modeling and control of a novel semi-closed gas turbine-absorption combined cycle,' University of Florida, Gainesville, Fla, 2011.

[16] Boyce, M.P., *Gas Turbine Engineering Handbook (Third Edition)*, Burlington: Gulf Professional Publishing, 2006.

[17] Soares, C., *Gas Turbines: A Handbook of Air, Land, and Sea Applications*, Boston : Butterworth-Heinemann, 2008.

[18] Chung, T.-W. and C.-M. Luo, 'Vapor Pressures of the Aqueous Desiccants,' *Journal of Chemical & Engineering Data*, vol. 44, pp. 1024-1027, 1999.

[19] Chen, S.-Y., A.N. Soriano, and M.-H. Li, 'Densities and vapor pressures of mixed-solvent desiccant systems containing {glycol (diethylene, or triethylene, or tetraethylene glycol)+salt (magnesium chloride)+water},' *The Journal of Chemical Thermodynamics*, vol. 42, pp. 1163-1167, 2010.

[20] Traverso, A., et al., 'A Novel Web-Based Approach for Thermoeconomic Analysis and Optimization of Conventional and Innovative Cycles,' in *ASME Turbo Expo 2004: Power for Land, Sea, and Air*, pp. 623-631, 2004.