SECOND LAW ANALYSIS OF A NOVEL COMBINED COOLING AND POWER CYCLE WITH WATER HARVESTING

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ABSTRACT

The first and second laws of thermodynamics were used to analyze a novel cooling and power cycle that combines a semi-closed cycle called the High Pressure Regenerative Turbine Engine (HPRTE) with a vapor absorption refrigeration system (VARS). Waste heat from the recirculated combustion gas of the HPRTE is used to power the absorption refrigeration unit, which cools the high-pressure compressor inlet of the HPRTE to below ambient conditions and also produces excess refrigeration in an amount that depends on ambient conditions. The cycle is modeled using steady-state thermodynamics, with state-of-the-art polytropic efficiencies and pressure drops for the turbo-machinery and heat exchangers, and accurate correlations for the properties of the NH₃-water mixture and the combustion products. Exergy analyses were performed for all the components of the cycle to examine the losses and identify critical plant devices considering different operating conditions. Water produced as a product of combustion is intentionally condensed in the evaporator of the VARS, which is designed to provide sufficient cooling for: the inlet air to the high pressure compressor, water extraction and for an external load. The cycle is shown to operate with a thermal efficiency approaching 46 % for a turbine inlet temperature of 1400 °C while producing about 1.5 kg of water for each kg of fuel (propane) consumed. The thermal efficiency does not take into account the cooling effect produced in the evaporator of VARS. The combined cycle efficiency at the above operating condition was found to be 49%. Low emissions are also possible on liquid fuels and not just on natural gas. It should be noted that the values of efficiencies obtained are for a medium sized engine with conservative values of the design parameters. It is observed that the largest contribution to the total cycle irreversibility comes from the combustor and accounts for 85% of the total exergy loss. The generator of the vapor absorption refrigeration system is the next largest quantity, accounting for about 3.4% of the total exergy loss. The mass of water extracted from the system increases as the value of the low-pressure compressor ratio is increased. However, this rate of increase is more when the compressor ratio is increased from 1.0 to 2.0 and less as the compressor ratio is further increased. Based on these and prior results, which showed that the HPRTE is very compact and has inherently low emissions, it appears that this cycle would be ideally suited for distributed power and vehicle applications, especially ones with associated air conditioning loads.

(Key Words: High Pressure Regenerative Turbine Engine, Vapor Absorption Refrigeration System, Water Extraction)
INTRODUCTION

A schematic diagram of the HPRTE (High Pressure Regenerative Turbine Engine) model is shown in Fig. 1. Air enters the system at State 1 and is compressed by the low-pressure compressor (LPC, labeled C1). It is then adiabatically mixed with the recirculated combustion products from the recuperator at State point 2.9. The combined air and exhaust products then enter the high-pressure core where they are compressed in the high-pressure compressor (HPC, labeled C2), heated in the recuperator and combustor, and then expanded in the high-pressure turbine (HPT, labeled T1). After leaving the HPT, all of the combustion gases enter the recuperator, where they lose heat to the gases entering the combustor. At the exit of the recuperator, a portion of the gas recirculates to join the compressed air stream, and the remaining air is passed through a low-pressure turbine (LPT, labeled T2) before exiting to the atmosphere.

The Suzler brothers, Westinghouse and the US Navy first studied semi-closed cycles using gas turbine engines in depth in the 1940s and 50s [1]. However due to operational requirements and program cancellations, the above systems were not developed further. There is renewed interest in the above gas turbine cycles because of the improvements in technology and the availability of clean burning fuels. MacFarlane and Lear [2] reported that water could be extracted from the combustion products of the HPRTE system. This water can be used for several applications which include process cooling and potable drinking. They considered the effect of re-injection of water back into the system on the cycle’s performance. Nemec [3] showed that the HPRTE with a Rankine bottoming cycle could achieve over 60% gross thermal efficiency. Muley and Lear [4] showed that there was a reduction in thermal NOx of roughly five orders of magnitude in the HPRTE versus typical open-cycle engines due to the reduction in thermal NOx of roughly five orders of magnitude in the HPRTE versus typical open-cycle engines due to the dilution of the fresh mixture with the recirculated combustion products, providing lower flame temperatures. A study done at Universita degli Studi di Firenze by Facchini et al. [5] showed that a semi-closed cycle similar to the HPRTE possessed benefits in terms of exhaust gas treatment (CO2 sequestration) and peak-load shaving. In a joint study between the University of Florida and Uniistry Associates Inc., Lear and Crittenden [6] first showed the reductions in emissions that occur with the HPRTE. In another joint study between the University of Florida and NASA, Lear and Laganelli [7] showed that the HPRTE produced a constant-efficiency curve at part power and reduced emissions for an open-cycle engine.

Boza et al., [8] modeled the HPRTE with a vapor absorption refrigeration combined cycle using the zero-dimensional steady state thermodynamics. The cycle performance was then studied by producing parametric design curves that showed trends and gave an indication of potential optimized system performance. They considered two cases, a large engine with a nominal power output of 40 MW and a small engine with a nominal power output of 100 kW. The results were reported in terms of thermal efficiency, refrigeration ratio, and the combined cycle efficiency. Mostafavi et al. [9] showed that there was enough waste heat available in the exhaust of a twin–spool open-cycle system to pre-cool the inlet to 5 °C. There is however, a lack of literature that describes the performance of such a system. The combined HPRTE power cycle and the VARS refrigeration cycle has got many advantages over the other power and refrigeration cycles because of the following reasons:

1. The HPRTE power cycle can produce flat specific fuel consumption curves (SFC is fuel flow rate per unit power) over a wider power band (at least 70% of the power band) than even that of the IC/R (Inter cooled and recuperated open cycle engines) making part-load operation more economical. The specific power advantage comes from the high pressure section, where the mass flow is much higher than the intake air flow due to large internal recirculation.
2. The recuperator is compact (more than an order of magnitude) because both streams operate at high pressure.
3. Inherently low emissions of NOx, CO and soot.
4. For high recirculation and higher inlet compressor pressure ratios, large amount of heat can be provided to the generator of VARS resulting in large tonnage of refrigeration in the evaporator.
5. The presence of evaporator in the HPRTE power cycle will reduce the inlet temperature to the high pressure compressor of the power cycle, HPRTE. Thus improving its performance at different operating conditions.

The main application of the presented combined cycle includes military vehicles, isolated settlements in the desert, and transportation vehicles.

NOMENCLATURE

\begin{align*}
A_c & \quad \text{excess air (Eq. 2)} \\
COP & \quad \text{coefficient of performance} \\
C_p & \quad \text{specific heat of water} \\
E_{\text{refg}} & \quad \text{exergy associated with the refrigeration in the evaporator} \\
\text{HPRTE} & \quad \text{high pressure regenerative turbine engine} \\
h & \quad \text{enthalpy} \\
h_{3.2} & \quad \text{latent heat at temperature and pressure corresponding to State point 3.2} \\
h_{3.2,H2O(vapor)} & \quad \text{latent of water vapor at state point 3.2} \\
\dot{m} & \quad \text{mass flow rate} \\
MW & \quad \text{molecular weight} \\
M_R & \quad \text{ratio of number moles of water extracted to number of moles of fuel burned (Eq. 4)} \\
n & \quad \text{number of moles} \\
N & \quad \text{it represents the equation showing the number of moles of individual species (O}_2, N}_2, \text{CO}_2, \text{H}_2O) present at the particular state point of the cycle. Used in Eqs. 5-9.} \\
P & \quad \text{pressure} \\
PRC1 & \quad \text{low-pressure compressor pressure ratio} \\
\dot{Q} & \quad \text{Heat flow rate} \\
\dot{Q}_{\text{evap}} & \quad \text{heat load on the evaporator, kW} \\
\dot{Q}_{\text{ref}} & \quad \text{excess refrigeration, kW} \\
R_F & \quad \text{Ratio of mass flow rate of water extracted to mass flow rate of fuel (Eq. 12)} \\
T & \quad \text{temperature} \\
\text{VARS} & \quad \text{vapor absorption refrigeration system} \\
\beta & \quad \text{refrigeration ratio } \dot{Q}_{\text{ref}} / \dot{W}_{\text{net}} \\
\phi_r & \quad \text{relative humidity}
\end{align*}


\[ \Phi \]  combustor equivalence ratio  

\[ \eta_{th} \]  thermal efficiency \( \frac{W_{net}}{\dot{Q}_{LHV}} \)  

\[ \eta_{I} \]  combined cycle efficiency  

\[ \eta_{II} \]  second law efficiency  

\[ \omega \]  number of moles of water vapor present in the air entering the system at State point 1  

\[ \psi \]  availability  

Subscripts  

0  unrestricted dead state  
a  air  
amb  ambient  
comp  compressor  
cv  control volume  
dplg  dephlegmator  
evap  evaporator  
fuel  fuel (propane \( C_3H_8 \))  
gen  generation  
genr  generator  
in  inlet  
LHV  lower heating value of the fuel  
et  net work done  
out  outlet  
ref  refrigerant  
refg  refrigeration  
turb  turbine  
w  water  

**CYCLE CONFIGURATION**

The cycle configuration considered in this work is similar to that given by Boza et al., [8] except for the water removal step in the evaporator. The VARS cycle is shown in Fig. 2. The gas-turbine portion is the HPRTE, discussed above and shown in Fig. 1. The absorption refrigeration portion is a typical ammonia/water single-stage system. The generator was placed after the mixing junction of the recirculated exhaust and the incoming air in the gas turbine cycle. Another possible configuration considered was a two-stage generator that extracted heat first from the lower-quality incoming air, and then the higher-quality recirculated exhaust. This configuration would have had a thermodynamic benefit and could possibly have provided more refrigeration (the generator is the limit on how much refrigeration can be produced). However, it was decided that, for most applications, the added weight and complexity of such a configuration would outweigh its potential benefits in a real system. The evaporator of the system was placed after the heat exchanger MC and water is extracted in the evaporator. Also, evaporator temperatures were constrained to be above the freezing point of water due to the large amount of water vapor in the hot gas stream in contact with the evaporator. The turbine blade cooling for high turbine inlet temperatures is considered in the present study. The cooling flow required for turbine blade cooling is bled from the main airflow after the high pressure compressor (C2) at State point 4 in Fig. 1. The turbine blades are assumed to be cooled by latest turbine blade cooling technology. The percentage of the main air flow bled for turbine blade cooling depends upon the turbine inlet temperature, Massardo et al., [10] have given this percentage for different values of the turbine inlet temperature. Their data is used in the present study and it was found that the percentage varied from 10% to 14% as the turbine inlet temperature varied from 1000°C to 1400°C.

The cycle code was written in FORTRAN. Intercooling was, accomplished solely by heat exchanger MC. The following table shows base case values for input parameters for the system. These are the values held constant during the parametric studies, except where explicitly stated.

The turbine and recuperator inlet temperatures were determined based on materials limits for the turbine blades and the recuperator walls. These are conservative values; they are lower than true state-of-the-art limiting values so that they may be varied in the parametric study. The ambient temperature was determined based on a hot day in which air conditioning would typically be used. The evaporator temperature was constrained by the freezing temperature of water. Thermodynamically it would have been possible and favorable to set the evaporator at a significantly lower temperature. However, coil frosting would have then become an issue because of the large amount of water in the exhaust stream in contact with the evaporator. The cycle parameter values given in Table 1 are used for studying the combined system. It should be noted that some parameters and their values are underlined and they represent the input to the computer code of the combined cycle.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Inlet Temperature °C</td>
<td>1400</td>
</tr>
<tr>
<td>Recuperator Inlet Temperature °C</td>
<td>800</td>
</tr>
<tr>
<td>Ambient Temperature °C</td>
<td>30</td>
</tr>
<tr>
<td>Inlet air pressure MPa</td>
<td>0.101</td>
</tr>
<tr>
<td>Evaporator Temperature °C</td>
<td>5</td>
</tr>
<tr>
<td>Low-Pressure Compressor Ratio (PRC1)</td>
<td>2</td>
</tr>
<tr>
<td>Turbo-machinery polytropic efficiencies</td>
<td>90 %</td>
</tr>
<tr>
<td>Combustor Equivalence ratio, ( \Phi )</td>
<td>0.9</td>
</tr>
<tr>
<td>Effectiveness of the recuperator (( R )), ( \varepsilon_R )</td>
<td>0.85</td>
</tr>
<tr>
<td>Effectiveness of the heat exchanger, ( \varepsilon_{MC} )</td>
<td>0.9</td>
</tr>
<tr>
<td>Relative humidity of inlet air, ( \phi_r )</td>
<td>0.9</td>
</tr>
<tr>
<td>Pressure drop in the combustor, ( \Delta P / P )</td>
<td>0.05</td>
</tr>
<tr>
<td>Pressure drop in heat exchanger, ( \Delta P / P )</td>
<td>0.03</td>
</tr>
<tr>
<td>Pressure drop in the generator, ( \Delta P / P )</td>
<td>0.03</td>
</tr>
<tr>
<td>Pressure drop in the evaporator, ( \Delta P / P )</td>
<td>0.03</td>
</tr>
<tr>
<td>Pressure drop in the recuperator(Turbine (T1) exit side), ( \Delta P / P )</td>
<td>0.06</td>
</tr>
<tr>
<td>Pressure drop in the recuperator(Compressor C2) exit side, ( \Delta P / P )</td>
<td>0.02</td>
</tr>
<tr>
<td>Recirculation Ratio (R)</td>
<td>2.5</td>
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<tr>
<td>Temperature at State point 5 °C</td>
<td>820</td>
</tr>
<tr>
<td>High Pressure Compressor Ratio, PRC2</td>
<td>12.6</td>
</tr>
</tbody>
</table>

Table 1. Input and calculated parameters of the combined HPRTE/VARS cycle.
The low-pressure compressor (PRC1) pressure ratio was based on prior studies [2]. The turbo-machinery polytropic efficiencies were based on state-of-the-art values. The equivalence ratio was set to 0.9 based on previous studies with the HPRTSE. Those studies showed that performance improves as the equivalence ratio is raised. However, it has to be capped at 0.9 in a real system because it was hypothesized that at least 10% excess air was necessary to ensure complete combustion. Therefore, 10% excess air was used is as follows for typical fuels (see MacFarlane and Lear [2]).

$$
C_X H_Y + (1 + A_e) \left( \frac{Y}{4} + X \right) [O_2 + 3.76 N_2] \rightarrow X CO_2
$$

\[ + \left( \frac{Y}{2} \right) H_2O + A_e \left( \frac{Y}{4} + X \right) O_2 + 3.76(1 + A_e) \left( \frac{Y}{4} + X \right) N_2 \]


\[ (1)
\]

where \( C_X H_Y \) is the fuel and \( A_e \) is the excess air for combustion.

The equivalence ratio \( \Phi \) is commonly used to indicate quantitatively whether a fuel-oxidizer mixture is rich, lean, or stoichiometric. The equivalence ratio, \( \Phi \), is related to excess air, \( A_e \), by the following equation

$$
A_e = \left( \frac{1}{\Phi} - 1 \right)
$$

At steady state fresh air and fuel enter the cycle at a constant flow rate. Also, the exhaust products and extracted water exit the cycle at a constant flow rate. Continuity is used to determine the mass flow rate and species concentration at each point in the cycle. The recirculation ratio is defined as the mass flow rate of combustion products recirculated to the mass flow rate of air at the exit of the combustor.

$$
R = \frac{\dot{m}_{10}}{\dot{m}_1 + \dot{m}_{fuel} - \dot{m}_w}
$$

In the above equation \( \dot{m}_{fuel} \) is defined as the fuel flow rate and \( \dot{m}_w \) is the mass flow rate of water extracted. All state points referred to are clearly shown in Fig. 1.

Water is extracted in the evaporator after the mixing junction resulting in the decrease of the total mass flow rate of the system. The number of moles of water removed per mole of fuel burned is given by

$$
M_R = \frac{n_w}{\dot{m}_{fuel} / MW_{fuel}}
$$

The mass flow rate of water extracted (\( \dot{m}_w \)) can be calculated from the above expression using the values of \( M_R \) and \( \dot{m}_{fuel} \).

The mass flow rate of water removed is accounted for in the calculation and normalized with respect to the fresh air flow rate at the inlet (\( \dot{m}_1 \)). State 2.9 represents the recirculated portion of the combustion products. The known amounts in moles of fuel burned the number of moles of the fresh air at the inlet and \( M_R \) moles of water extracted allows continuity to determine the composition at States 2.9, 3, and 3.01. Thus the species present at these states can be written as,

$$
N_{2.9,3,3.01} = (X + \frac{Y}{4})(1 + A_e + R A_e) O_2 + (3.76)(1 + A_e) \]

\[ \ast (X + \frac{Y}{4})(1 + R) N_2 + R X CO_2 + (R \left( \frac{Y}{2} - M_R \right) + \omega) H_2 O \]

In the above equation \( \omega \) refers to the number of moles of water vapor present in the fresh air entering the system and its value is obtained from the humidity ratio of the fresh air entering the system. It should be noted that some water has already been removed from the control mass, which is reflected in the above equation. After the water extraction in the evaporator the species present at all State points from 3.2 to 5 can be written as

$$
N_{3.2,4.5} = (X + \frac{Y}{4})(1 + A_e + R A_e) O_2 + (3.76)(1 + A_e)(X + \frac{Y}{4})(1 + R) N_2 +
\]

\[ R X CO_2 + \left[ R \left( \frac{Y}{2} + \omega - M_R \right) - M_R \right] H_2 O \]
The addition of fuel and the combustion process occurs between States 5 and 6. The combustion equation mentioned previously in Eq. (1) can now be utilized to determine the composition at States 6, 7 and 9

\[ N_{6,7,9} = (1 + R)(X CO_2 + A_e(X + Y/4)O_2 + 3.76(1 + A_e)(X + Y/4)) \]

\[ N_2 = \left[ R(\omega + Y/2 - M_R) - M_R + Y/2 + \omega \right] H_2O \]

(7)

Using the definition of the recirculation ratio the composition at States 8 and 10 can be written as

\[ N_8 = X CO_2 + A_e(X + Y/4)O_2 + 3.76(1 + A_e)(X + Y/4)N_2 + \left[ \omega + Y/2 - M_R \right] H_2O \]

(8)

\[ N_{10} = RX CO_2 + RA_e(X + Y/4)O_2 + R(3.76)(1 + A_e) \]

\[ * (X + Y/4)N_2 + R \left[ \omega + Y/2 - M_R \right] H_2O \]

(9)

As mentioned earlier, water is extracted in the evaporator of the HPRTE system. It is assumed that the air coming out of the evaporator is saturated and the ratio of the saturated pressure of water vapor to the total pressure can be related to the mole ratio of the substances for the working fluid and can be written as (see MacFarlane and Lear [2])

\[ \Pi_{sat} = \frac{P_{3,2,sat}}{P_{3,2}} = \frac{n_w}{n_{total}} \]

(10)

Using the previous combustion and continuity analysis, the ratio of the number of moles of water to the number of moles of fuel \((M_{R})\) can easily be found, and is equal to

\[ R_{big} = R X + (1 + A_e + R A_e) \left( \frac{Y}{4} + X \right) + 3.76(1 + R)(1 + A_e) \left( \frac{Y}{4} + X \right) \]

\[ M_{R} = \frac{\Pi_{sat} \left( R_{big} - \{R \left( \frac{Y}{2} + \omega \right) + \omega \} \right)}{1 - \Pi_{sat}} \]

(11)

The number of moles of water removed from the working fluid depends upon the amount of cooling provided in the evaporator and the concentration of water at the evaporator inlet. The ratio of mass flow rate of water extracted to the mass flow rate of fuel is given by

\[ R_P = \frac{m_w}{m_{fuel}} = \frac{18.02}{12.011X + 1.0079Y} M_{R} \]

(12)

The combined cycle efficiency for the HPRTE and VARS system is defined as

\[ \eta_{t} = \frac{\dot{W}_{net} + E_{refg}}{\dot{Q}_{LHV}} \]

(13)

\[ E_{refg} = m_{ref} (\psi_{evap,in} - \psi_{evap,out}) \]

(14)

where \(\dot{W}_{net}\) is the net work output and \(E_{refg}\) is the exergy associated with the refrigeration output and \(\dot{Q}_{LHV}\) is the lower heating value (LHV) of the fuel. The above definition of the combined cycle efficiency takes into account both the mechanical work output of the HPRTE system as well as the refrigeration output of the VARS. The above combined cycle efficiency definition is consistent with the combined power and refrigeration cycle efficiency defined by Hasan et al. [12].

The thermal efficiency \(\eta_{th}\) will only be equal to ratio of \(\dot{W}_{net}\) and \(\dot{Q}_{LHV}\).

The exergy destruction for each component of the HPRTE/VARS cycle has been calculated using the following equation:

\[ \frac{d\psi}{dt} = \sum_{i=1}^{n} \left( 1 - \frac{T}{T_{i}} \right) \dot{Q}_{i} - \dot{W}_{cs} + \sum_{i=1}^{m} m_{i} \psi_{i} - \sum_{i=1}^{m} m_{i} \psi_{i} - T_{0} \dot{S}_{gen} \]

(15)

The term on the LHS of the above equation represents the accumulation of the nonflow exergy in the control volume. The first term on the RHS of the above equation represents the exergy transfer via the heat transfer into the control volume, the second term represents the exergy transfer by work done by the control volume, the third term represent the intake of flow exergy via mass flow into the control volume and the fourth term represent the release of flow exergy via mass flow out of the control volume. The last term represent the exergy destruction in the control volume. The flow availability \(\psi\) at each state point of the combined cycle is calculated by the equation

\[ \psi = (h - h_0) - T_0 (s - s_0) \]

(16)

The values of enthalpy \(h_0\) and entropy \(s_0\) are calculated at the unrestricted dead state where the mixture is in thermal, mechanical and chemical equilibrium with the environment as defined by Bejan [13].

Assuming steady state the exergy destruction in the various components of the combined cycle is found to be

\[ (T_{0,S}_{gen})_{comp} = \dot{m}_{in}\psi_{in} - \dot{m}_{out}\psi_{out} + \dot{W}_{comp} \]

(17)

\[ (T_{0,S}_{gen})_{turb} = \dot{m}_{in}\psi_{in} - \dot{m}_{out}\psi_{out} - \dot{W}_{turb} \]

(18)

\[ (T_{0,S}_{gen})_{recp} = \dot{m}_{4}\psi_{4} + \dot{m}_{3}\psi_{3} - \dot{m}_{5}\psi_{5} + \dot{m}_{0}\psi_{0} \]

(19)

\[ (T_{0,S}_{gen})_{2.9} = \dot{m}_{10}\psi_{10} + \dot{m}_{2}\psi_{2} - \dot{m}_{2.9}\psi_{2.9} \]

(20)

\[ (T_{0,S}_{gen})_{MC} = \dot{m}_{3}\psi_{3} - \dot{m}_{3.01}\psi_{3.01} \]

(21)

\[ \dot{m}_{w} C_{P}(T_{w,in} - T_{w,out}) - T_{0}(\dot{m}_{w} C_{P}) \ln \left[ \frac{T_{w,in}}{T_{w,out}} \right] \]

(22)

\[ (T_{0,S}_{gen})_{gen} = \dot{m}_{2.9}\psi_{2.9} - \dot{m}_{3}\psi_{3} + \dot{m}_{ref,gen,ou}\psi_{gen,ou,4V} \]

\[ (T_{0,S}_{gen})_{evap} = \dot{m}_{fuel}\psi_{fuel} + \dot{m}_{5}\psi_{5} - \dot{m}_{6}\psi_{6} \]

(23)

\[ (T_{0,S}_{gen})_{evap} = \dot{m}_{3.01}\psi_{3.01} - \dot{m}_{3.2}\psi_{3.2} \]

\[ - \dot{m}_{w} \psi_{w} + \dot{Q}_{ref} \left[ 1 - \frac{T_{0}}{T_{amb}} \right] + \dot{m}_{ref} (\psi_{evap,in} - \psi_{evap,out}) \]

(24)

In Eq. (24) \(\psi_{fuel}\) is the chemical availability of fuel (for the present study propane \(C_{3}H_{8}\)) as defined by Bejan [13] and \(\dot{Q}_{ref}\) represents the extra cooling load available in addition to the cooling load required for water extraction in the evaporator and is given by
\[ Q_{\text{ref}} = \dot{Q}_{\text{evap}} - (m_{3.01} h_{3.01} - m_{3.2} h_{3.2} + m_w h_{fg,3.2} - m_w h_{3.2,H,O(vapor)}) \]  

Moran [14] defines the second law efficiency as the ratio of net exergy rate of the product to the net exergy rate of the input. The net exergy rate of the product in the present study is the sum of the net power produced and the exergy associated with the refrigeration output \((E_{\text{refg}})\) and the net exergy rate of the input is the sum of exergy of the fuel and the exergy of the fresh air coming into the system.

\[ \eta_{II} = \frac{\dot{W}_{\text{net}} + E_{\text{refg}}}{m_{\text{fuel}} \psi_{\text{fuel}} + m_{\text{air}} \psi_1} \]  

SOLUTION METHOD

A computer code was written in FORTRAN to simulate the performance of the combined system. The combustion software HPFLAME provided by Turns [11] was used for calculating the adiabatic flame temperature at the exit of the combustor. Equations (5)-(9) provide the chemical composition of air at all of the state points of the system. The temperature of air at different state points is either given as input or obtained from the calculated enthalpy at that state point. Knowing the temperature and composition of gases at each state point, its enthalpy is calculated using the correlations given in the appendix of Turns [11]. The code is capable of calculating engine performance (net work output and efficiency) as a function of the input parameters (turbine inlet temperature, recuperator inlet temperature, generator temperature, evaporator exit temperature, low pressure compressor ratio, turbo-machinery efficiencies, heat exchanger effectiveness, equivalence ratio and pressure drops). The code iterates with respect to the recirculation ratio to get the required turbine inlet temperature. The program for simulating the working of NH3/Water VARS was written based on the system described in Kuehn et al. [15] and the properties of NH3/Water mixture were obtained using the equations reported by Tamm [16]. The condenser and the absorber of the VARS are assumed to be cooled by chilled water. A psychometric program is also written in FORTRAN and is added to the code which would calculate the values of relative humidity \((\phi_e)\) before the evaporator. The humidity ratio of air before the evaporator was calculated by using the values of the number of moles of water vapor and the number of moles of all other components. The pressure at each state point is calculated using the values of pressure drops (in case of turbines and heat exchangers) and pressure rise (in case of compressors) across the different components.

RESULTS

Effect of Turbine Inlet Temperature T6: Figure 3 and 4 shows the effect of, T6 on the combined cycle \((\eta_I)\) and the second law efficiency \((\eta_{II})\) of the system, respectively. As mentioned earlier for the values of T6 at 1000 °C and beyond a percentage of the total main air flow is bled for turbine blade cooling. The turbine blade cooling model, used in the present study is a conservative cooling model. It assumes a step jump in cooling flow rate from 0% at 999 °C to 10% at 1000 °C. This is equivalent to changing blade materials and maximum metal temperatures. Therefore there are discontinuities in Figs. 3, 4, and 5 at T6 equal to 1000 °C. The turbine exit temperature is fixed in drawing these plots and as T6 is increased the pressure ratios across the high pressure turbine T1 and the high pressure compressor C1 also increases. However, the net work output from the system increases, resulting in increase in the efficiencies for the values of T6 from 900 °C to 975 °C. The figure shows that after T6 equal to 1000 °C there is a little drop in the efficiencies for the following reasons: increasing the value of T6 beyond 1000 °C increases the percentage of mass flow rate bled from the main core flow for turbine blade cooling, reducing the net work output from the turbine T1 but also increasing the fuel consumption. The figure shows that at a particular turbine inlet temperature the efficiencies decrease with increasing values of low pressure compressor ratio PRC1. It is noted that as the value of PRC1 is increased from 2 to 10 the value of both, the work input into the compressor C1 and the work output from the low pressure turbine T2, increases. However, the increase in work input into C1 is more compared with the increase in the work output from T2. As a result the net work output from the combined cycle is reduced. Figure 5 shows the plot of the water extraction parameter \((R_F)\) defined in Eq. (12) versus T6. It should be noted that the water vapor which condenses is one of the products of combustion in the combustor. It is found that the recirculation ratio is decreased for higher values of turbine inlet temperature. If the amount of recirculated air decreases there will be less water vapor in the recirculated air entering the evaporator. Therefore decreases in the recirculation ratio due to higher turbine inlet temperature will result in lower values of \(R_F\). In addition the figure shows that for a given value of the turbine inlet temperature, the value of \(R_F\) increases as the values of PRC1 is increased. Figure 6 is a bar diagram showing the irreversibilities of different components of the combined cycle. The diagram shows that the irreversibility or the exergy destruction is the largest in the combustor and it is about 85% of the total exergy destroyed in the complete cycle. Among all other components which contribute to the overall exergy destruction, the most significant ones are the generator, the mixing junction (State 2.9) of the recirculated air and the fresh air (refer Fig. 1), the high pressure turbine (T1), the high pressure compressor (C2) and the water cooled heat exchanger (MC).

Effect of Low Pressure Compressor Ratios (PRC1): Figures 7 and 8 shows the effect of PRC1 on \(\eta_I\) and \(\eta_{II}\) of the system respectively. The plots are obtained for different values of T6. Figure 8 shows that, as the value of PRC1 increases, the values of both the efficiencies increase, reaches a maximum value, and then decrease. The figure shows that for all the three temperatures (T6) considered the thermal efficiency is found to be maximum at around PRC1 equal to 2. It is observed that \(Q_{\text{ref}}\), the work input to the compressor C2 and the work output from the turbine T1 are constant as the value of PRC1 is increased from 1.0 to 10.0. However, the work input to the compressor C1 and the work output from the turbine T2 increase as the value of PRC1 is increased from 1 to 10. The rate of increase of the above work of compression in C1 and the work of expansion in T2 are different, resulting in the optimum point at PRC1≈2 as shown in the figure. Figure 9 shows the variation of \(R_F\) with PRC1. The plots are drawn for different values of T6. The figure shows that at a particular turbine inlet temperature the value of \(R_F\) increases rapidly when the value of PRC1 is increased from 1 to 2 and the increase in its value is less significant when PRC1 is increased from 2 to 10.
CONCLUSIONS
A state-steady model was used for performing the first law and the second law analyses of a HPRTE combined with a NH3/water VARS. Turbine blade cooling was considered for turbine inlet temperature higher than 1000 °C. The following is the summary of the major findings:
1. The combined cycle is shown to operate with high thermal efficiency of 46% for a turbine inlet temperature (T6) of 1400 °C, in addition to providing refrigeration ratio (β) of 0.14. The combined cycle efficiency, which also takes into account the refrigeration effect produced in the evaporator, is found to be 49%. The cycle is also capable of producing about 1.5 to 1.8 kg of water for each kg of propane fuel consumed. The values of efficiencies obtained are for a medium sized engine with conservative values of design parameters.
2. Increasing the value of turbine inlet temperature for fixed values of all other design parameters increases the efficiency. Efficiency decreases by about 9 percentage points if turbine blade cooling is considered for T6 equal to 1400 °C, and this decrease is much higher at lower values of T6.
3. The values of combined cycle efficiency are less sensitive to low pressure compressor ratios (PRC1). They vary by about 4% as the values of PRC1 are increased from 2 to 10. The maximum efficiency occurs at about PRC1 equal to 2.
4. The water extraction from the system increases by about 5 percentage points when PRC1 is increased from 1 to 2. However this increase is only 3.7 percentage points when the value of PRC1 is increased from 2 to 10. Therefore, the desirable design regime, considering both efficiency and water extraction, is for PRC1 values greater than 2.

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REFERENCES
Figure 1: Sketch of the High Pressure Regenerative Turbine Engine (HRTPE)

Figure 2: Sketch of the Vapor Absorption Refrigeration System (VARS)

| C1        | Low-pressure compressor |
| C2        | High-pressure compressor |
| T1        | High-pressure turbine   |
| T2        | Low-pressure turbine    |
| MC        | Water cooled heat exchanger |
| R         | Recuperator             |
| CM        | Combustor               |
| cond      | Condenser               |
| abs       | Absorber                |
| evap      | Evaporator              |
| genr      | Generator               |
| dplg      | Dephlegmator            |
Figure 3: Combined cycle efficiency ($\eta_I$) versus the turbine inlet temperature ($T_6$) for different values of low pressure compressor ratio (PRC1).

Figure 4: Second law efficiency ($\eta_{II}$) versus the turbine inlet temperature ($T_6$) for different values of low pressure compressor ratio (PRC1).

Figure 5: Water extraction parameter ($RF$) versus the turbine inlet temperature ($T_6$) for different values of low pressure compressor ratio (PRC1).

Figure 6: Percentage of irreversibilities in the various components of the cycle.
Figure 7: Combined cycle efficiency ($\eta$) versus the low pressure compressor ratio (PRC1) for different values of the turbine inlet temperature (T6).

Figure 8: Second law efficiency ($\eta_{II}$) versus the low pressure compressor ratio (PRC1) for different values of the turbine inlet temperature (T6).

Figure 9: Water extraction parameter ($R_F$) versus the low pressure compressor ratio (PRC1) for different values of the turbine inlet temperature (T6).