Thermodynamics Cycle of Gas-Cooled Fast Reactor

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This paper deals with the thermo-hydraulic aspect of gas cooled fast 4 generation reactor. The paper is focused on the comparison of direct and indirect strategy of thermodynamics cycle of helium cooled reactor from the thermodynamics and turbomachinery point of view. The analyses respect pressure losses at all major part of the equipment - reactor, heat exchanger, pipe lines, etc. The compressor and gas turbines efficiency are included in calculation as well. The working fluid in primary circuit is helium. In the secondary circuit a mixture of helium and nitrogen is considered. The Cycle characteristic point and efficiency calculation reflects mixture properties of the real gas. Calculation points out the influence of mixture composition on the basic structural parameters of the turbines, compressor and heat exchangers. Thermodynamics cycle efficiency, specific heat input/output, heat flux and cycle work will be presented as characteristic parameters.

Keywords: Thermodynamic cycle, Generation IV reactor, Gas cooled reactor, Gas turbine.

1. Introduction

There are a lot of advantages on the side of gas-cooled fast reactors [11]. They do not suffer with weaknesses such as sodium cooled fast reactors. The gas is chemically inert and a single phase coolant eliminates boiling. Gas also allows high temperature operation without the corrosion and coolant radiotoxicity problems associated with heavy liquid metal reactors (Pb-Bi and pure Pb). On the other hand, gas coolants have little thermal inertia rapid heat-up of the core following loss from forced cooling. The motivation for this project was to determine the optimal effectiveness and specific work of direct (Figure 1) and indirect (Figure 2) cycle strategy of gas cooled reactors and propose an optimal solution for gas cooled fast reactors [1] [2].
Figure 1 Direct thermodynamics cycle

Figure 2 Indirect thermodynamics cycle
2. Model description

The computational models of the direct and indirect cycles are described by analytical and empirical equations of thermodynamics, fluid mechanics and turbomachinery [3] [6] [7]. Efficiency and specific work for arbitrary thermodynamics cycle can be determined as a follow function:

$$\eta = f(\lambda, \tau, \eta_t, \eta_c, \Sigma)$$ (1)

$$W_{\text{net}} = c_p \dot{m} T_{\text{min}} g(\lambda, \tau, \eta_t, \eta_c, \Sigma)$$ (2)

Our model respect thermophysical properties of ideal gas (used mainly for He), thermodynamics of isentropic, isobaric and polytropic transformations, thermophysical properties of real gas [5] (used mainly for $N_2$), pressure losses of reactor and heat exchangers, efficiency of turbine and compressor stages, and effectivity of heat exchangers and regenerator.

2.1. Non-dimensional parameters

Thermodynamics characteristics of different cycles can be describe by set of non-dimensional parameters [8] [9]:

- Isotropic Temperature ratio: $\lambda = \frac{T_2}{T_1}$
- Pressure ratio: $\epsilon = \frac{P_{\text{max}}}{P_{\text{min}}}$, respectively: $\lambda = \epsilon^{\frac{\kappa - 1}{\kappa}}$
- Temperature ratio: $\tau = \frac{T_{\text{max}}}{T_{\text{min}}}$, (in our case is $T_{\text{min}} = T_1$)
- Regenerator efficiency: $\Sigma_1 = \frac{T_{2}'' - T_2}{T_4 - T_2}$
- Heat exchanger efficiency: $\Sigma_2 = \frac{T_{6} - T_1'}{T_5 - T_1'}$
- Non dimensional specific cycle work: $w^*_\text{net} = \frac{W_{\text{net}}}{c_p \dot{m} T_{\text{min}}}$
- Non dimensional input specific heat: $q^*_\text{in} = \frac{Q_{\text{in}}}{c_p \dot{m} T_{\text{min}}}$
- Non dimensional output specific heat input/output: $q^*_\text{out} = \frac{Q_{\text{out}}}{c_p \dot{m} T_{\text{min}}}$

2.2. Cycle thermodynamics characteristics

Cycle efficiency as a function of different degree of regeneration, turbine and compressor efficiency and various thermodynamics transformation of Joule cycle (see Figure 3) is described by equations (3), (4), (5), (6) and (7) [8].

Cycle efficiency, ideal case, isentropic compression & expansion without regeneration:

$$\eta^\text{base} = \frac{\lambda - 1}{\lambda}$$ (3)
respectively with regeneration:

\[ \eta_{\text{reg}} = \frac{\tau - \lambda}{\tau} \] (4)

Cycle efficiency, ideal case, isometric compression & expansion without regeneration:

\[ \eta_{\text{base}} = \frac{(\tau - 1) \ln \lambda}{\tau \ln \lambda + \tau - 1} \] (5)

respectively with regeneration (equivalent to Carnot efficiency)

\[ \eta_{\text{reg}} = \frac{\tau - 1}{\tau} \] (6)

Cycle efficiency respecting devices efficiency (isentropic c&e)

\[ \eta = \frac{\lambda - 1}{\lambda} \frac{\tau \eta_t - \frac{\lambda - 1}{\eta_t}}{(1 - \Sigma)(\tau - 1 - \frac{\lambda - 1}{\eta_t}) + \Sigma \tau \eta_t (\lambda - 1)} \] (7)

Equation (7) can be modify by pipelines and reactor pressure losses and \( C_p \) correction, if \( C_p \neq \text{constant} \) etc. Regeneration is valid for:

\[ \lambda \in (1, \sqrt{\tau \eta_t \eta_c}) \] (8)

For higher \( \lambda \) is \( \Sigma = 0 \) (system without regeneration). Optimal temperature ration for max. cycle work is:

\[ \lambda_{\text{w,max}} = \sqrt{\tau \eta_t \eta_c} \] (9)

The heat fluxes are apparent on the Figure 3. The waste heat energy of the primary cycle can be used as an input energy for secondary thermodynamics cycle, or as technological heat for additional technology such as hydrogen production [10].

3. Results

3.1. Number of turbine stages

Number of turbine stages significantly depends on blade to gas speed ratio \( \sigma = u/c_{is} \) (schematically shown on Figure 4), where \( c_{is} = \sqrt{2h} \). The typical optimal value of impulse turbine stage is \( \sigma \approx 0.5 \).

Optimal number of turbine stages can be defined by equation:

\[ n_{\text{opt}} = f(RPM, C_p, \kappa, R_{\text{opt}}, \sigma_{\text{opt}}, \epsilon_{\text{turb}}) \] (10)

For total turbine efficiency we can use simplified function: \( \approx f(\eta_t^{n_{\text{stage}}}, ...) \) Typical number of stages and efficiency of conventional gas turbines diameter 2.5 m, RPM 3000 and different working gas are shown in Table 1.
Figure 3 Temperature-entropy diagram of Joule combined cycle and waste heat using

Figure 4 Blade to gas speed ratio $\sigma = u/c_{i,a}$

<table>
<thead>
<tr>
<th>Gas used</th>
<th>$He$</th>
<th>$N_2$</th>
<th>10%$He + 90%N_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Num. of stages</td>
<td>16</td>
<td>3</td>
<td>8</td>
</tr>
<tr>
<td>Total Turbine Efficiency</td>
<td>0.7</td>
<td>0.9</td>
<td>0.8</td>
</tr>
</tbody>
</table>
3.2. Effect of compressor and turbine efficiency

Figure 5 shows the effect of the compressor and the turbine efficiency on the specific cycle work $w_{net}$ and on the overall efficiency of the entire cycle. By decreasing the turbine and the compressor efficiency the $w_{net}$ and the cycle efficiency curves move to lower values of $\lambda$.

![Figure 5](image)

Figure 5 Specific work (left) and thermodynamic cycle efficiency (right) as a function of compressor and turbine efficiency, temperature ratio $\lambda = 4$

3.3. Combine cycle

One of the example is the indirect combined cycle with $He - N_2$ (20% - 80%) mixture for the intermediate gas cycle.

The efficiency of the Brighton cycle with regeneration and inter-cooling (secondary loop) is approximately 26% [9] and the efficiency of the Ranking cycle (tertiary loop) is 25% [9] (correspond with results [1]). Total efficiency of the combined power plant is based on equation (11). In our case the result is around ±45%. An additional water loop uses the waste heat from a secondary (Helium) loop. A part of this heat energy is moved to the conventional steam cycle, see Figure 6.

$$\eta_{c-e} = \eta_{HT} + \eta_{LT} - \eta_{HT}\eta_{LT}, \quad (11)$$

where $\eta_{HT}$ is efficiency of the high temperature primary cycle and $\eta_{HN}$ is efficiency of the low temperature secondary cycle. Example of thermodynamic parameters of the combine cycle is apparent on the Table 2.

4. Conclusion

The long-term development of gas turbine engines for aircraft and cogeneration has been in favour of the indirect cycle. The concept of a gas turbine for air (used for $N_2$ or $N_2$ and $He$ mixture) have higher efficiency numbers in comparison with the helium turbines today. Indirect cycle - decrease the energy flux in the
Table 2 Example of characteristic input/output parameters

<table>
<thead>
<tr>
<th>Characteristic parameters</th>
<th>values</th>
<th>units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Turbine input/output Temperatures</td>
<td>850 / 510</td>
<td>°C</td>
</tr>
<tr>
<td>Gas Turbine input/output Pressures</td>
<td>7.02 / 2.65</td>
<td>MPa</td>
</tr>
<tr>
<td>Compressor input/output Temperatures</td>
<td>178 / 362</td>
<td>°C</td>
</tr>
<tr>
<td>Compressor input/output Pressures</td>
<td>2.6 / 6.5</td>
<td>MPa</td>
</tr>
<tr>
<td>Compressor Pressure Ratio</td>
<td>2.5</td>
<td></td>
</tr>
<tr>
<td>Steam Turbine input/output Temperature</td>
<td>535 / 32</td>
<td>°C</td>
</tr>
<tr>
<td>Steam Turbine input/output Pressures</td>
<td>15 / 0.005</td>
<td>MPa</td>
</tr>
</tbody>
</table>

Figure 6 Scheme of indirect combine cycle

second working media approximately 2-3 times (depending on gas mixture). It cause robust construction of heat exchangers, dimensions of turbines, compressors pipelines and other cycle components, but required lower no. of compression and expansion stages. The thermodynamic cycle analysis presented here should help in the design process of the optimal variant according to the actual efficiencies of the components. Development of each component will push the limits forward.

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