ABSTRACT

The Supercritical CO2 (S-CO2) Brayton power cycle has the advantages of relatively high efficiency at moderate heat source temperature compared to a steam Rankine cycle or a helium Brayton cycle and compact components size. In nuclear energy industry, Small Modular Reactors have potentials in many aspects, flexibility of construction, and applicability for the distributed power source and reduced construction cost. Therefore, KAIST research team developed an S-CO2 cooled Small Modular Reactor namely KAIST Micro Modular Reactor (MMR), using both advantages of S-CO2 cycle and SMR. Currently, radial turbine is being used for the output of 12MWe in MMR. The original KAIST-MMR was designed to utilize radial turbine but the 12MWe power output of MMR can adopt axial turbine as well. Therefore, this paper explores the potential of improving both steady state performance and transient response by switching from radial type to axial type turbine. The implications of the transient analysis results show the advantages of using the axial turbine over the radial turbine because the axial turbine operates at relatively higher efficiencies for wider off-design ranges.

INTRODUCTION

The supercritical CO2 (S-CO2) cycle is attracting attention as it has the potential to replace the existing steam cycle in the process of developing the next generation nuclear power plants. In addition, the supercritical CO2 cycle has a higher thermal efficiency than the steam Rankine cycle and the helium Brayton cycle when the turbine inlet temperature is above 500 degree Celsius [1, 2]. It also has the advantage that the components of the cycle, such as turbines and compressors, are more compact than other cycles, making them well suited for small modular nuclear reactors. Fig. 1 compares the efficiency of a supercritical CO2 Brayton cycle, a steam Rankine cycle, and a helium Brayton cycle [1].

Fig. 1. Comparison of efficiency of S-CO2 power cycle [1]
SMR and gas turbine technologies [3]. MMR is sized such that it can be transported via truck and the layout of MMR is shown in Fig. 2. The design values of MMR are summarized in Table 1. These values were obtained from the design process of a nuclear system by using validated in-house codes. The design was conducted by using the following well-validated programs: KAIST-TMD [5] and KAIST-HXD [6].

The KAIST TMD code for the compressor was validated in the previous study [5]. In addition, for the same turbomachinery, equivalent conditions can provide a basis for comparing different working fluids [7]. Furthermore, sCO$_2$ turbine operates where the properties are behaving similar to an ideal gas. Therefore, the KAIST TMD code was validated using NASA's air radial turbine data, which is equivalent to sCO$_2$ conditions for a radial turbine case [8]. For the axial turbine, the loss set of the following reference was used [9]. In ref. [9], the author selected the GTHTR 300 design of JAERL [10], a direct cycle using helium gas, as a reference model to validate the axial turbine code. Thus, KAIST-TMD is validated in radial compressor, radial turbine, and axial turbine with the available data.

The KAIST-HXD code was developed by the KAIST research team using a new thermal hydraulic correlation for sCO$_2$ heat exchanger and the code is validated by a laboratory-scale PCHE test at the KAIST supercritical CO$_2$ pressurizing experiment (SCO2PE) facility [6].

Table 1. Summary of design results of MMR

<table>
<thead>
<tr>
<th></th>
<th>Thermal power</th>
<th>Net electric power</th>
<th>12MWe</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal efficiency</td>
<td>34.09%</td>
<td>Mechanical efficiency</td>
<td>98%</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>180.0 kg/s</td>
<td>Total-to-total Pressure ratio</td>
<td>2.49</td>
</tr>
<tr>
<td>Turbine total-to-total efficiency</td>
<td>92%</td>
<td>Compressor total to total efficiency</td>
<td>85%</td>
</tr>
<tr>
<td>Generator efficiency</td>
<td>98%</td>
<td>Rotating speed</td>
<td>19,300 rpm</td>
</tr>
<tr>
<td>Recuperator efficiency</td>
<td>95%</td>
<td>Compressor inlet pressure</td>
<td>8.0 MPa</td>
</tr>
<tr>
<td>Design point of recuperator</td>
<td>440.7°C, 8.2 MPa</td>
<td>Hot side inlet</td>
<td>Cold side inlet</td>
</tr>
</tbody>
</table>

**DESCRIPTION OF KAIST-CCD, TMD, HXD CODE**

**KAIST-CCD**

The KAIST-Closed Cycle Design (KAIST-CCD) code is based on MATLAB, and it is an in-house code developed by KAIST research team. Fig. 3 shows the main algorithm of the KAIST-CCD code. The monitored error value for the iteration is defined below.

\[
    Error = \left[ \frac{heat\ input(n) - heat\ input(n-1)}{heat\ input(n)} \right]
\]

**Fig. 3. Main algorithm of KAIST-CCD code for simple recuperated Brayton cycle** [4].

Values of material properties such as enthalpy and entropy used in the code are provided by NIST-REFPROP (transport properties database) [11]. To illustrate how KAIST-CCD code works, the algorithm for the simple recuperated Brayton cycle analysis is shown in Fig. 3. The numbers in algorithm in Fig. 3 refer to the node numbers in Fig 4. First, read input data: fluid type, cycle layout, Total heat received by the cycle, max, min temperature of the cycle, max pressure of the cycle, efficiency, flow split and pressure ratio of the components. Second, the
recuperator cold side inlet flow condition and the heater hot side inlet flow conditions are assumed. Next, the inlet and outlet conditions of each component are obtained using the models of the components and the assumed values. Finally, the heat input is obtained from the calculated condition values.

Since the heat source is prescribed, the error is estimated by comparing the prescribed heat source value to the calculated value. If the error is greater than or equal to 1E-5, the assumed values are updated to the calculated values.

Fig. 4 shows the nodes of the KAIST MMR for the KAIST CCD code. KAIST MMR is a system with a simple recuperated cycle as a power cycle and the shaft connects compressor and turbine to rotate at the same RPM. The CCD code is used to calculate the temperature, pressure, enthalpy, and entropy of each node. Therefore, the original MMR was optimized using KAIST-CCD when the turbine efficiency was changed.

Table 2 Summary of the loss model of each turbomachineries

<table>
<thead>
<tr>
<th>Turbomachinery</th>
<th>Loss Model</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial turbine</td>
<td>Profile loss</td>
<td>Balje-Binsley [14]</td>
</tr>
<tr>
<td></td>
<td>Secondary loss</td>
<td>Kacker-Okaapu [15]</td>
</tr>
<tr>
<td></td>
<td>Tip clearance loss</td>
<td>Dunham-Came [16]</td>
</tr>
<tr>
<td>Radial compressor</td>
<td>Incidence loss</td>
<td>Boyce [17]</td>
</tr>
<tr>
<td></td>
<td>Blade loading loss</td>
<td>Coppage et al. [18]</td>
</tr>
<tr>
<td></td>
<td>Skin friction loss</td>
<td>Jansen [19]</td>
</tr>
<tr>
<td></td>
<td>Clearance loss</td>
<td>Jansen [19]</td>
</tr>
<tr>
<td></td>
<td>Disk friction loss</td>
<td>Daily and Nece [20]</td>
</tr>
<tr>
<td></td>
<td>Mixing loss</td>
<td>Johnston and Dean [21]</td>
</tr>
<tr>
<td></td>
<td>Recirculation loss</td>
<td>Oh et al. [13]</td>
</tr>
<tr>
<td></td>
<td>Leakage loss</td>
<td>Aungier [22]</td>
</tr>
<tr>
<td>Radial turbine</td>
<td>Incidence loss</td>
<td>Balje [23]</td>
</tr>
<tr>
<td></td>
<td>Rotor passage loss</td>
<td>Balje [23]</td>
</tr>
<tr>
<td></td>
<td>Clearance loss</td>
<td>Jansen [19]</td>
</tr>
<tr>
<td></td>
<td>Disk friction loss</td>
<td>Daily and Nece [20]</td>
</tr>
</tbody>
</table>

Using the models summarized above, a turbomachinery design in-house code, KAIST-TMD, was developed which is again written in MATLAB. The main code structure is shown in Fig. 6.

The main design variables are defined by the designer when designing the turbine with KAIST-TMD. Then the off-design values and the geometry values are calculated through the iteration process with the selected loss models suitable for the
selected turbomachinery. The property values are referred from REFPROP developed by National Institute of Standards and Technology (NIST) [11].

**KAIST-HXD**

The KAIST Heat eXchanger Design (HXD) code is based on MATLAB environment and was developed to design a printed circuit heat exchanger (PCHE) for the S-CO$_2$ power system applications. To overcome dramatic changes of properties of CO$_2$ near the critical point, the discretized design and analysis method is adopted in place of widely used conventional heat exchanger design methods such as LMTD method. In KAIST-HXD, the energy and momentum governing equations are solved numerically to calculate properties of CO$_2$ in finite number of volumes [6]. It performs analysis on the unit channel representing the entire heat exchanger. The total heat transfer for one control volume is calculated as follows.

$$ Q = UA\Delta T = \frac{1}{R_{hot}^3 R_{cond}^3 R_{cold}^3} \frac{1}{i} \frac{1}{R_{hot}^3 R_{cond}^3 R_{cold}^3} \Delta T \quad (4) $$

To obtain pressure drop for one control volume, the friction factor must be determined. Friction pressure drop equation is as follows:

$$ \Delta P = f \frac{\rho V^2}{D_e} \quad (5) $$

**MOTIVATION OF AXIAL TURBINE DESIGNED UNDER THE CONDITION OF KAIST-MMR**

The original MMR is designed to have an output of 12MWe, and the type of turbine satisfying these conditions is shown in Fig. 8 [24], which shows that both radial type turbine and axial turbine can be appropriate. In the previous MMR design, the design choice was using a single stage radial turbine.

Using the above equations, the inlet and outlet temperatures and pressures for one channel can be determined by the enthalpy change due to heat transfer and the pressure drop due to friction. The cold side can be calculated as above, but since the flow direction of the hot side is opposite to that of the cold flow (i.e. counter current), the calculation starts from the outlet of the cold side to the inlet. Therefore, the property of the cold side outlet should be assumed first. If the cold side inlet result obtained from the assumed cold outlet does not satisfy the cold inlet temperature calculated in the heat transfer equation and pressure boundary conditions, the new cold outlet temperature and pressure are assumed and the calculation is conducted again until the convergence conditions are met.

**Fig. 8. Component and technology options for S-CO$_2$ cycles [24]**

The existing MMR radial turbine was designed at the boundary between the radial turbine and the axial turbine as shown in Fig. 8. In other words, it is designed to have the maximum capacity that can be covered by the single stage of radial turbine. If the capacity of the existing MMR becomes larger, the number of stages for the radial turbine has to increase or using axial turbine can be another choice. However, in the case of radial turbine, it is not recommended to increase the number of stages because the inter-stage flow path can induce large pressure drop, so an axial turbine can be more appropriate [25]. Before designing an axial turbine for the larger MMR, it is possible to evaluate the performance of an axial turbine compared to a radial turbine for existing MMR.
Therefore, the purpose of this paper is summarized as the following: 1. Design an axial turbine suitable for MMR. 2. Evaluate the potential for using an axial turbine for MMR by comparing off design performance with originally designed single stage radial turbine. The design is performed by using KAIST CCD and TMD codes. For the isolated grid application, timely transient response is imperative, therefore the newly designed MMR was evaluated with GAMMA + code and compared to the radial turbine based on MMR.

**COMPARISON OF RADIAL AND AXIAL TURBINES**

An axial turbine was designed to replace a radial turbine under the conditions of original MMR using KAIST-TMD code described above. Using the KAIST-TMD code, the following results were obtained.

A turbomachinery map was generated for the range of possible mass flow rates by raising the rpm from 60% to 120% of the design point for performance evaluation. The range of possible mass flow rates depends on the rpm. The minimum possible mass flow rate is 24% of the design point at rpm = 11,580 and the maximum possible mass flow rate is 126% of the design point at rpm = 17,370.

Table 3. The range of performance evaluation and design point of turbine

<table>
<thead>
<tr>
<th>The range of the mass flow rate (min to max)</th>
<th>24-126%</th>
</tr>
</thead>
<tbody>
<tr>
<td>The range of the rpm</td>
<td>60-120%</td>
</tr>
<tr>
<td>The rpm of design point</td>
<td>19300</td>
</tr>
<tr>
<td>The mass flow rate of design point</td>
<td>180kg/s</td>
</tr>
</tbody>
</table>

Figs. 9 and 10 show the pressure ratio-mass flow rate map and the efficiency-mass flow rate map of the axial turbine and the radial turbine. Results are generated from KAIST-TMD code. The performance map shown in Figs. 9 and 10 show that the slope is smooth compared to the radial turbine in both pressure ratio and efficiency for the axial turbine when the mass flow rate deviates from the design point. On the other hand, under the same conditions, the radial turbine shows a rapid change in efficiency and pressure ratio, which means that the axial turbine can operate in more stable performance than the radial under the off design operating conditions. In order to analyze these results, loss of radial turbine and axial turbine of KAIST-TMD code were obtained respectively.

Fig. 9. Comparison of Pressure ratio map for MMR radial and axial turbines

Fig. 10. Comparison of efficiency map for MMR radial and axial turbines

Fig. 11. The cumulative loss of the axial turbine – mass flow rate graph
The losses of radial turbine and axial turbine of KAIST TMD code were obtained as follows. Comparing Figs. 11 and 12, the overall cumulative loss for axial turbine is smaller than that of the radial turbines. Because the incidence loss for radial turbine which is generated during the off-design condition due to the mismatch of the direction of relative velocity of fluid at inlet and inlet blade angle is larger than that of the axial turbine. In addition, the incidence loss decreases with the approach to the design point, but since the vaneless space loss which occurs in a vaneless space between the impeller vane and the diffuser vane becomes larger as the mass flow rate increases, the efficiency of the radial turbine is higher than that of the radial turbine in the wide-range of off-design condition.

Therefore, these results show that the axial turbine has potentially better performance than the radial turbine under the off-design operation, and it is a motive to evaluate the potential of MMR using an axial turbine. As a result, the following design values were obtained and shown in Table 4.

Table 4. TMD code result of axial turbine MMR cycle

<table>
<thead>
<tr>
<th>Number of stages</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbomachinery work</td>
<td>21.53MW</td>
</tr>
<tr>
<td>Total-to-total-Pressure Ratio</td>
<td>2.44</td>
</tr>
<tr>
<td>Total-to-total efficiency</td>
<td>91.57%</td>
</tr>
</tbody>
</table>

Using the above results, the cycle was optimized by using KAIST-CCD code. It should be noted that the axial turbine for MMR developed conceptually in this paper is designed to be similar to the CSP turbine which is four stages axial turbine used in the Sunshot project. The developed turbine is for 10 MW power output CSP solution [26].

The CSP turbine was developed by aero design tools and full scale CFD tool was also used. A 4-stage design with 27000 RPM is selected due to the reduction of mechanical stress and the improvement of efficiency as shown in Fig. 13.

![Fig. 12. The cumulative loss of the radial turbine – mass flow rate graph](image)

![Fig. 13. Preliminary flow path layout of the 4-stage axial turbine](image)

Although it is designed with similar concepts, the CSP turbine produces 10 MW of power output with 27,000 rpm, four stages, while the axial turbine designed in this paper produces 12MW of power output with 19,300 rpm, 8 stages. From the viewpoint of the number of stages, it seems reasonable that the CSP turbine has better performance than the axial turbine designed in this paper. However, the CSP turbine has an rpm of 27,000, while the MMR axial turbine developed in this paper has an rpm of 19,300 due to the synchronized compressor. Therefore, if the rpm is increased to about 27,000, a turbine with four stage having similar performance can be produced by using KAIST_TMD code. This is left as future works for the further design optimization of the MMR.

**OPTIMIZATION OF CYCLE FOR MAXIMIZING EFFICIENCY**

In this paper, MMR using axial turbine was optimized under the original MMR condition by KAIST-CCD for selecting the highest efficiency. In other words, in case of MMR using axial turbine, the efficiency and pressure ratio of the turbine are changed from the input values of the original MMR using radial turbine and used as input values in the KAIST CCD code. The new input values of MMR using axial turbine are as follows.

Table 5. The input values of MMR using axial turbine for KAIST CCD

<table>
<thead>
<tr>
<th>Thermal power (MW)</th>
<th>36.2</th>
<th>Cycle layout</th>
<th>Simple brayton recuperation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mechanical efficiency</td>
<td>98%</td>
<td>Total-to-total Pressure ratio</td>
<td>2.44</td>
</tr>
<tr>
<td>Turbine total-to- total efficiency</td>
<td>91.57%</td>
<td>Compressor total to total efficiency</td>
<td>85%</td>
</tr>
<tr>
<td>Generator efficiency</td>
<td>98%</td>
<td>Rotating speed</td>
<td>19,300rpm</td>
</tr>
<tr>
<td>Max Pressure</td>
<td>20Mpa</td>
<td>Recuperator effectiveness</td>
<td>95%</td>
</tr>
<tr>
<td>Max Temperature (Turbine inlet)</td>
<td>550 °C</td>
<td>Min Temperature (Compressor inlet)</td>
<td>60 °C</td>
</tr>
</tbody>
</table>

*Jeong Ik Lee*
The T-S diagram obtained from the KAIST-CCD code and the property values such as temperature and pressure are shown in Table 6. T-S diagrams for new MMR with an axial turbine are shown in Fig. 14 to compare with the thermodynamic property values of original MMR with the radial turbine. The numbers in Fig. 14 correspond to the node numbers in MMR shown in Fig. 4, respectively.

Table 6. The optimization design value of new MMR using axial turbine

<table>
<thead>
<tr>
<th>Point</th>
<th>Mass Flow rate</th>
<th>Temperature(°C)</th>
<th>Pressure(Mpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>182.15 kg/s</td>
<td>550</td>
<td>19.93</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>440.72</td>
<td>8.17</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>156.47</td>
<td>8.10</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td>60</td>
<td>8.01</td>
</tr>
<tr>
<td>5</td>
<td></td>
<td>141.96</td>
<td>20</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>388.48</td>
<td>19.98</td>
</tr>
<tr>
<td>7</td>
<td></td>
<td>550</td>
<td>19.93</td>
</tr>
</tbody>
</table>

As mentioned above, since the results obtained by using KAIST CCD code and the original value of MMR show no marked changes, GAMMA+ code of MMR can be adopted identically except for the turbine section, which are replaced by newly formed axial turbines. The design points under steady state of the main part of the cycle obtained by GAMMA+ code is shown in Fig. 15.

![Fig 14. Comparison of T-s diagram of MMR and result of CCD code](image)

As a result, it was confirmed that there is almost no difference in properties between two MMRs. This means that there is almost no difference in the T-s diagram between the cycle using the axial turbine with the best efficiency and that of the radial turbine. Therefore, since the GAMMA+ code of original MMR using radial turbine is already constructed, only the turbine will be modified in the GAMMA+ transient simulation.

The KAIST research team modified GAMMA+ code to become applicable to the original MMR. GAMMA+ code is originally developed for a gas cooled reactor transient analysis by KAERI. However the original GAMMA+ code is designed to calculate the CO₂ properties with simple correlation, but it was necessary to calculate the CO₂ property values near the critical point more accurately. The modified GAMMA+ code used the REFPROP program developed by NIST to solve the above problems [11]. The REFPROP program accurately calculates the thermal and transport properties of various fluids, including CO₂. The modified GAMMA+ code with NIST database for CO₂ properties near the critical point was validated using experimental data from SCO2PE [27].

![Fig 15. Comparison of design parameters and code result in the main part of the cycle](image)
The results of the transient analysis show that thermal power of the MMR with an axial turbine are similar to those of the MMR with radial turbine as shown in Fig. 16, and the cycle efficiency of MMR with axial turbine comparable to the MMR with radial turbine in the off-design point in Fig. 18.

Displayed from the efficiency maps of radial turbine and axial turbine in Fig. 10, the efficiency graphs of the axial turbine remain unchanged from the design efficiencies, whereas the graphs of the radial turbine drop abruptly as the mass flow rate decreases as shown in Fig. 19.

In Fig. 20, it can be seen that there is almost no difference between the radial and axial turbine works. This is because the same radial compressor was used and the thermal power and efficiency in axial and radial cases are similar, as shown in Figs. 16 and 17.

\[ W_{t,turbine} = Q_{core,thermal} + W_{t,compressor} \]  

Where \( W_{t,turbine} \) = Turbine work, \( W_{t,compressor} \) = Compressor work, \( Q_{core,thermal} \) = Thermal power of MMR, \( \eta_{thermal} \) = thermal efficiency of cycle.

The radial and axial turbine works are similar, but the mass flow
rate is relatively different. Therefore, the relationship between turbine work and mass flow rate has to be considered.

The formula for turbine work is:

\[
W_{turbine} = m \Delta h_{turbine}
\]  

Where \( \Delta h_{turbine} = h_{o2} - h_{o1} = U_2V_{\theta 2} - U_1V_{\theta 1} \), Enthalpy difference between the turbine inlet and outlet

![Fig. 21. Enthalpy change between the turbine inlet and outlet of MMR of Radial and Axial turbines](image)

At the off-design point, the enthalpy difference (\( \Delta h_{turbine} \)) of the MMR with axial turbine is larger than that of the MMR with radial turbine as shown in Fig. 21 because the mass flow rate (\( m \)) of the axial turbine is relatively reduced and turbine work of the axial and radial is no difference.

The implications of the results show the advantages of using the axial turbine over the radial turbine. Although the axial turbine operates at comparable efficiencies to radial turbine for off-design ranges, it retains its high \( \Delta h_{turbine} \) compared to the radial turbine which consequently leads to having lower mass flow rate under a given turbine work. As a result, the specific work of the turbine is increased at lower load conditions when the axial turbine is adopted as shown in Fig. 21. The advantage will be amplified as the system capacity increases, and thus, the axial turbine will prove beneficial for larger size systems.

**SUMMARY AND FUTURE WORKS**

In this paper, an axial turbine was designed to replace the radial turbine of the KAIST MMR and the transition analysis was performed.

First, axial turbine was designed with KAIST_TMD code by using MMR design conditions for radial turbine system (turbine inlet temperature and pressure, turbine outlet pressure, rpm, mass flowrate) developed by KAIST research team. As a result, an axial turbine with eight stages was designed which has a turbine work of 21.53 MW, a pressure ratio of 2.44, and a turbine efficiency of 91.57%.

Second, an optimization was performed with KAIST_CCD code using the newly designed system with axial turbine. As a result, the T-s diagram shows that there is almost no difference between the properties of the original MMR and the new MMR. Therefore, the code for performing the transition analysis with GAMMA + code was created by using the operating conditions of the existing MMR, except for the newly developed axial turbine.

Third, in order to perform the transient analysis of MMR with axial turbine, GAMMA + code is implemented at the scenario where load is changed from 100%-70%-100%. As a result, it was found that the efficiency of the MMR with axial turbine is comparable to the near design point and wider operating range of MMR is possible with axial turbine. Furthermore, axial turbine shows high specific work compared to the radial turbine which leads to having lower mass flow rate for the given turbine work. As a result, the specific work by the turbine can be increased at lowered load conditions when the axial turbine is adopted in the system.

In this study, the potential of the axial turbine was confirmed in terms of specific work under the off-design conditions when axial turbine or radial turbine is used. The axial turbine is more advantageous with respect to the specific work compared to the radial turbine under off-design conditions and these advantages will be amplified as the system becomes large. In addition, when the new system is bigger than the system of MMR, the advantages of axial turbine is maximized because axial turbine has better efficiency than the radial turbine for larger system.

Finally, the planned future works are to design the MMR with 4-stage axial turbine to operate at higher efficiency.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Surface area of the heat transfer [m²]</td>
</tr>
<tr>
<td>A'</td>
<td>Flow area [m²]</td>
</tr>
<tr>
<td>D</td>
<td>Diameter [m]</td>
</tr>
<tr>
<td>D_e</td>
<td>Equivalent diameter [m]</td>
</tr>
<tr>
<td>K</td>
<td>Form loss</td>
</tr>
<tr>
<td>P</td>
<td>Pressure [Pa]</td>
</tr>
<tr>
<td>Q</td>
<td>Thermal power [W]</td>
</tr>
<tr>
<td>R</td>
<td>Chemical reaction</td>
</tr>
<tr>
<td>T</td>
<td>Temperature [K]</td>
</tr>
<tr>
<td>U</td>
<td>Impeller tip speed [m/s]</td>
</tr>
<tr>
<td>V</td>
<td>Flow velocity [m/s]</td>
</tr>
<tr>
<td>Y</td>
<td>Mass fraction</td>
</tr>
<tr>
<td>W</td>
<td>Relative velocity [m/s]</td>
</tr>
<tr>
<td>W_t</td>
<td>Specific work [W/kg]</td>
</tr>
<tr>
<td>f</td>
<td>Friction factor</td>
</tr>
</tbody>
</table>

*Jeong Ik Lee*
<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>g</td>
<td>Gravitational acceleration [m/s²]</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy [J/kg]</td>
</tr>
<tr>
<td>h'</td>
<td>Heat transfer coefficient [W/m²K]</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate [kg/s]</td>
</tr>
<tr>
<td>q</td>
<td>Specific heat [J/kg]</td>
</tr>
<tr>
<td>q''</td>
<td>Heat flux [W/m²]</td>
</tr>
<tr>
<td>j</td>
<td>Diffusion flux [mol/m²s]</td>
</tr>
<tr>
<td>t</td>
<td>Time [sec]</td>
</tr>
<tr>
<td>z</td>
<td>Height [m]</td>
</tr>
<tr>
<td>ρ</td>
<td>Density [kg/m³]</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>[26] Chiranjeev Kalra, et al., Development of high Efficiency hot gas turbo-expander for optimized CSP supercritical CO₂ power block operation, The 4th International Symposium - Supercritical CO₂ Power Cycles</td>
</tr>
</tbody>
</table>
Published in: 3rd European sCO2 Conference 2019

This text is made available via DuEPublico, the institutional repository of the University of Duisburg-Essen. This version may eventually differ from another version distributed by a commercial publisher.

DOI: 10.17185/duepublico/48896

This work may be used under a Creative Commons Attribution 4.0 License (CC BY 4.0).