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Design Parameters Optimization of an Axial-type Turbine for N_2 Brayton Cycle coupled with KALIMER-600 using a Mean-line Prediction Method

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1. Introduction

A sodium-cooled fast reactor (SFR) has been acknowledged as one of the most matured reactor types among various concepts of Generation IV (Gen-IV) nuclear systems [1]. As a part of Gen-IV international collaborations, the Korea Atomic Energy Research Institute (KAERI) completed the conceptual design of KALIMER-600 reactor in 2000s [2].

However, due to an essential risk of a sodium-water reaction, a steam Rankine cycle has possibility of severe accidents if there is any crack in steam generator tubes. For this reason, N_2 Brayton cycle has been considered to be a power conversion system of KALIMER-600 [3]. This is mainly because nitrogen is an inert gas that enables to remove sodium reaction accidents essentially. A purpose of this work is to find optimum design parameters of an unshrouded axial-type turbine in terms of isentropic efficiency obtained from parametric studies using one-dimensional analysis code and suggest the reasonable turbine design in terms of the selected optimal design point.

2. Design Methods

2.1 Turbine Cascade and Stage Configurations

An axial-type turbine is considered in this paper. One repeating stage consists of one stator cascade and one rotor cascade (Fig. 1). An axial-type multi-stage turbine consists of more than two repeating stages in the axial direction (Fig. 2).

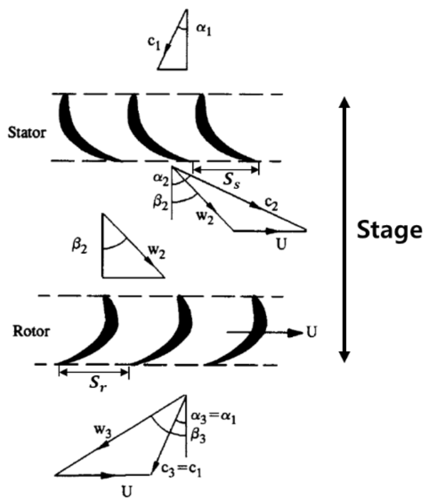

Fig. 1. Repeating stage configuration and velocity diagrams of an axial-type turbine [4]

Stator cascade is a group of stationary vanes. The purpose of stator vanes is to control the flow direction and accelerate the working fluid to enhance the kinetic energy that is an important power source for work output in rotor cascade. Rotor cascade is a group of moving blades. By using the accelerated fluid energy, rotor blades rotate and generate the power. IGVs (Inlet Guide Vanes) and EGVs (Exit Guide Vanes) are special stator vanes that control the flow directions at the turbine inlet and outlet, respectively.

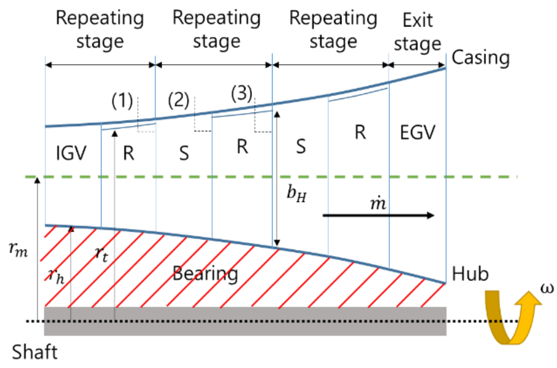

Fig. 2. Schematic of an axial-type multi-stage turbine

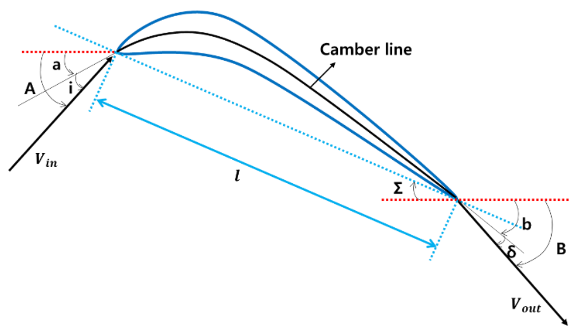

Fig. 3. Vane/blade with a parabolic-camber line

Some geometric dimensionless parameters can be defined from Fig. 1 to 3:

$$\text{Solidity} \quad \sigma = l/S \tag{1}$$

$$\text{Aspect ratio} \quad \mathrm{AR} = b_H/l \tag{2}$$

$$\text{Hub-tip ratio} \quad \zeta = r_h/r_t \tag{3}$$

Large solidity means that the vanes or blades are located densely. This parameter is inversely proportional to lift coefficient and also can be used to calculate the number of vanes or blades in a cascade. Aspect ratio and hub-tip ratio determine the geometric features of vanes or blades.

The mean radius, blade tangential velocity, mass flow rate and average density are given as follows:

$$r_m = \frac{r_t + r_h}{2} \tag{4}$$

$$U = r_m \omega \tag{5}$$

$$\dot{m} = \overline{\rho} C_x A = \overline{\rho} C_x (2\pi r_m b_H) \tag{6}$$

$$\overline{\rho} = (\rho_1 + \rho_3)/2 \tag{7}$$

2.2 Turbine Design Parameters

There are three dimensionless variables which are usually used for designing turbomachinery: flow coefficient, stage loading coefficient, degree of reaction [4]. Flow coefficient (ϕ) is defined as the ratio of axial flow velocity to blade tangential velocity:

$$\phi = C_x/U \tag{8}$$

This parameter is related to flow rate and turbine blade span. High flow coefficient means that a working fluid flows fast and accordingly blade span becomes short for the same mass flow rate condition.

Stage loading coefficient (ψ) is defined as the ratio of flow velocity tangential variation to blade tangential velocity. From the Euler turbine equation, specific work output can be obtained by the following equation:

$$w = U(C_{\theta 2} - C_{\theta 1}) \tag{9}$$

So the stage loading coefficient can be written in terms of turbine work:

$$\psi = \frac{C_{\theta 2} - C_{\theta 1}}{U} = \frac{w}{U^2} = \frac{h_{o1} - h_{o3}}{U^2} \tag{10}$$

This parameter explains how much the enthalpy drop occurs in a stage. If the stage loading coefficient is high enough, the number of turbine repeating stage decreases for the same overall pressure ratio. Therefore, it is possible to design a smaller multi-stage turbine.

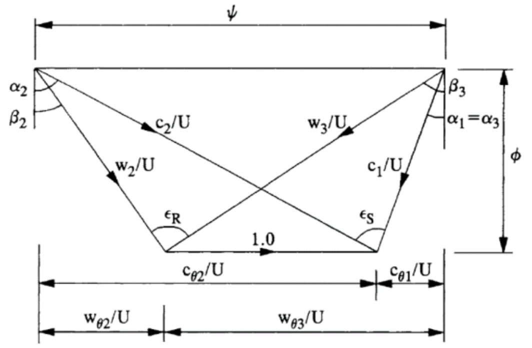

Fig. 4. Turbine design parameters from an assembled and normalized velocity diagram [4]

Degree of reaction (R) is defined as the ratio of static enthalpy drop in a rotor cascade to static enthalpy drop in a stage:

$$R = \frac{\Delta h_{rotor}}{\Delta h_{stage}} = \frac{h_2 - h_3}{h_1 - h_3} \tag{11}$$

This parameter accounts for the fractions of enthalpy drop which are in charge of rotor cascade within overall enthalpy drop in one stage. Turbine design parameters are given in Fig. 4 with the assembled and normalized velocity diagram.

2.3 Turbine Efficiency

Turbine efficiency is defined as the ratio of actual process work to isentropic process work. In adiabatic and steady state processes, applying the 1st law of thermodynamics, the work can be calculated from stagnation enthalpy change if there is no elevation. Therefore, the total-to-total turbine efficiency can be expressed as follows:

$$\eta_{TT} = \frac{w_a}{w_s} = \frac{h_{o1} - h_{o3}}{h_{o1} - h_{o3,s}} \tag{12}$$

The turbine efficiency depends on the thermodynamic variables (e.g. turbine specific work), shaft rotational speed, turbine size, velocity diagram, properties of working fluid and turbine losses [4]. The following equation is obtained from the dimensional analysis to reduce these variables [4]:

$$\eta_{TT} = [1 + \frac{1}{2\psi}\left\{\left(\phi^2 + \left(\frac{\psi}{2} + 1 - R\right)^2\right) Y_{stator} + \left(\phi^2 + \left(\frac{\psi}{2} + R\right)^2\right) Y_{rotor}\right\}]^{-1} \tag{13}$$

Stagnation pressure loss coefficient for stator and rotor is defined as follows, respectively [4]:

$$Y_{stator} = \frac{P_{o1} - P_{o2}}{P_{o2} - P_2} \tag{14}$$

$$Y_{rotor} = \frac{P_{o2} - P_{o3}}{P_{o3} - P_3} \tag{15}$$

2.4 Mean-line Prediction Method

Tournier et al. [5] summarized stagnation pressure loss coefficient prediction models. A mean-line prediction method is to predict turbine losses at the mean radius of vanes or blades. This is simple but powerful for predicting turbine performances at the first step of turbine design processes. However, this gives less accurate results when the blade span is too long. On the other hand, too short blade causes severe tip clearance losses. Thus, it is recommended to maintain the hub-tip ratio between 0.75 and 0.9 [1].

2.5 Simple Calculation Code for Turbomachinery-Turbine(SCCOT-T)

One-dimensional analysis code (SCCOT-T) for determination of the optimum turbine design parameters has been developed by FORTRAN. NIST REFPROP version 7.1 program was used for the nitrogen properties. This code follows the procedures in Fig. 5. All calculations are conducted by assuming shock-free condition $i = 0°$, rotor deviation angle $\delta_r = 2°$, zero stator deviation angle and constant mean radius.

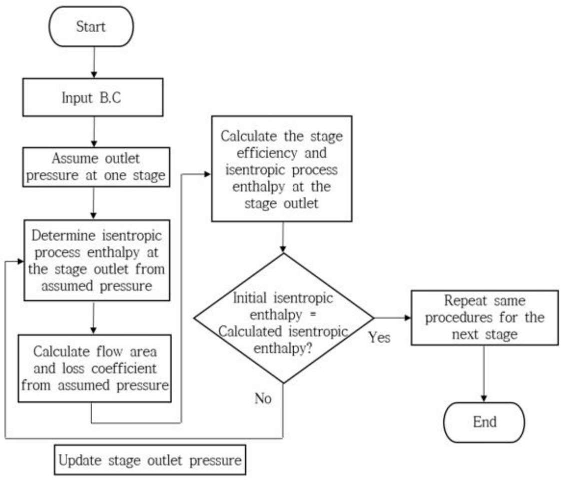

Fig. 5. Flow chart of SCCOT-T

The developed code initiates calculation from the first repeating stage to the last exit stage in a regular sequence. Input conditions for running the SCCOT-T are summarized in Table 1. The purpose of the code is to predict turbine isentropic efficiency from the given chord lengths and turbine inlet conditions. Then, after parametric studies by changing turbine design parameters, the optimum point can be selected from the obtained performance distributions.

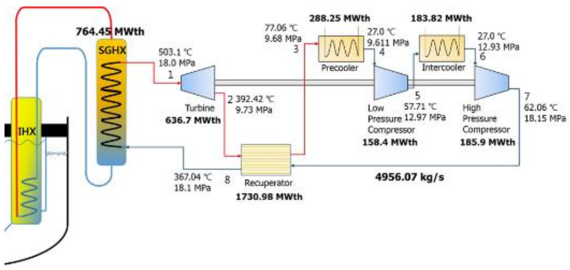

Fig. 6. Nitrogen Brayton cycle coupled with KALIMER-600 [3]

Parametric studies of turbine design parameters have been conducted for N_2 Brayton cycle coupled with KALIMER-600 given in Fig. 6 [3]. The stator chord and rotor chord length are extracted from the single shaft N_2 turbine design results by Olumayegun et al. [1]. The mean radii of all stages were constrained constant as 0.6m.

Table 1. Inputs for SCCOT-T

Parameters	Input Conditions
Number of repeating stage	4
Shaft rotational speed (rpm)	3600
Mean radius (m)	0.6
Mass flow rate (kg/s)	4956.07
Turbine pressure (inlet/outlet, MPa)	18.0/9.73
Turbine inlet temp. (℃)	503.1
Chord length (stator/rotor, cm)	26.4/28.4
Phi	0.3-0.75 (interval 0.05)
Psi (50% reaction)	1.05-1.85 (interval 0.05)
Psi (45%, 55% reaction)	1.15-1.9 (interval 0.05)

3. Results

Turbine performance curves are shown in Fig. 7 to 9. Comparing these three figures, there are no distinctive performance differences according to the degree of reaction. The case of R=0.45 has slightly better efficiency distributions than other two cases.

Considering the turbine efficiency only, it is profitable to select the flow coefficient as low as possible. However, too low flow coefficient causes too long blade span, which is undesirable. To maintain the hub-tip ratio larger than 0.75, the flow coefficient should be of at least 0.65.

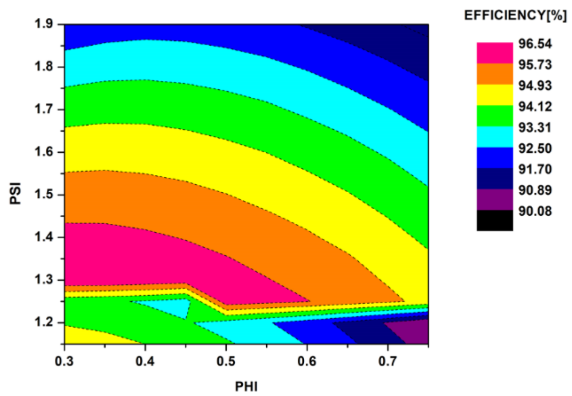

Fig. 7. Turbine efficiency distributions for R=0.45

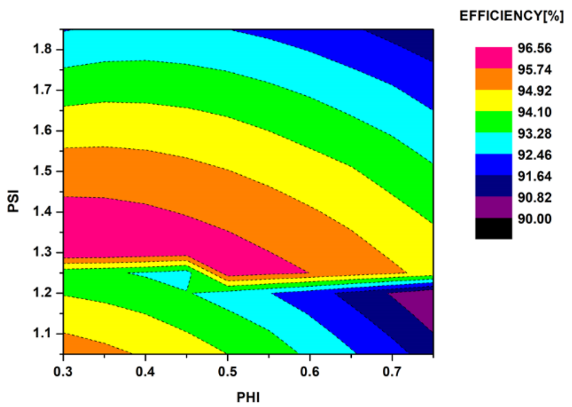

Fig. 8. Turbine efficiency distributions for R=0.5

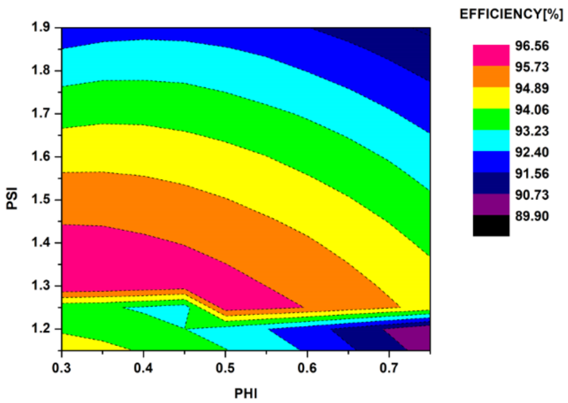

Fig. 9. Turbine efficiency distributions for R=0.55

There is a critical efficiency change region between $1.2 < \psi < 1.3$. This change is due to the decrease of the number of repeating stage. In the region of $\psi \leq 1.2$, the number of repeating stage is two and decreases to one in the region of $\psi \geq 1.3$. This results from the turbine overall pressure ratio constraint. The stage loading coefficient also shows similar tendency with the flow coefficient that is inversely proportional to turbine isentropic efficiency. Therefore, the optimum flow coefficient and stage loading coefficient are chosen as $(\phi, \psi) = (0.65, 1.3)$.

Then, the optimal degree of reaction should be found. The results of efficiency sensitivity test for the degree of reaction are given in Fig. 10. Finally, the optimum turbine design parameters are chosen as $(\phi, \psi, R) = (0.65, 1.3, 0.38)$. The design results of turbine at this point are given in Table 2.

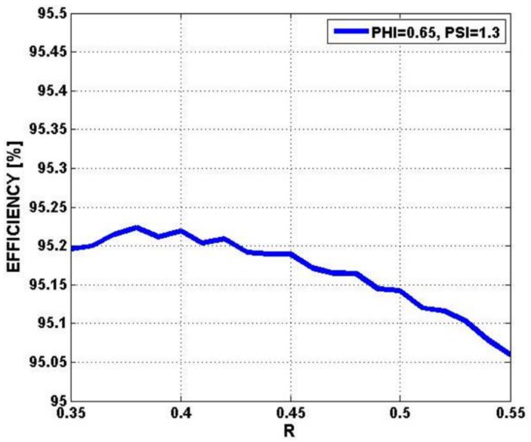

Fig. 10. Effect of degree of reaction on turbine efficiency

Turbine efficiency of Fig. 6 cycle is 93% and newly designed turbine has 95.22% efficiency. From the analysis about turbomachinery components efficiencies effect on cycle efficiency by Olumayegun et al. [1], it is expected to be able to get more than 0.5% point increased cycle thermal efficiency by replacing 93% efficiency turbine with 95.22% efficiency turbine.

Table 2. Results of an unshrouded axial-type turbine design

Parameters	Results
Shaft rotational speed (rpm)	3600
Mean radius (m)	0.6
Mass flow rate (kg/s)	4956.07
Turbine pressure (inlet/outlet, MPa)	18.0/9.73
Turbine temp. (inlet/outlet, ℃)	503.1/389.7
Chord length (stator/rotor, cm)	26.4/28.4
Blade span (min/max, cm)	12.30/16.90
Solidity (IGV/rotor/EGV)	1.26/1.13/0.98
Hub-tip ratio (min/max)	0.753/0.814
Aspect ratio (IGV/rotor/EGV)	0.466/0.433/0.640
Number of vane/blade (IGV/rotor/EGV)	18/15/14
Stagger angle (IGV/rotor/EGV, deg)	46.2/27.1/16.7
Number of repeating stage	1
Isentropic efficiency (%)	95.22

4. Conclusions

The optimum design parameters for the maximum turbine efficiency were found by one-dimensional analysis code (SCCOT-T) using the mean-line prediction method. The obtained turbine performance distributions and design results will be utilized for turbine designs through CFD.

ACKNOWLEDGEMENT

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REFERENCES

[1] O. Olumayegun, M. Wang and G. Kelsall, "Thermodynamic analysis and preliminary design of closed Brayton cycle using nitrogen as working fluid and coupled to small modular Sodium-cooled fast reactor (SM-SFR)", Applied Energy, Vol. 191, pp. 436-453, 2017.

[2] J. H. Lee, C. G. Park, J. B. Kim and G. H. Koo, "Structural Concept Design of KALIMER-600 Sodium Cooled Fast Reactor", Transactions of the Korean Society of Mechanical Engineers Spring Meeting, pp. 37-42, May, 2007.

[3] J. H. Park, J. Yoon, J. H. Eoh, H. M. Kim and M. H. Kim, "Optimization and sensitivity analysis of the nitrogen Brayton cycle as a power conversion system for sodium cooled fast reactor", Nuclear Engineering and Design, Under review.

[4] R. I. Lewis, *Turbomachinery performance analysis*, Butterworth-Heinemann, 1996.

[5] J.-M. Tournier and M. S. El-Genk, "Axial flow, multi-stage turbine and compressor models", Energy Conversion and Management, Vol. 51, pp. 16-29, 2010.