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Numerical Modeling and Analysis of Turbine in Brayton Cycle for KALIMER-600

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1. Introduction

The supercritical fluid Brayton cycle has been adopted for power conversion in some of the Generation IV Nuclear Energy Systems, and also planned to be installed in the high efficiency power conversion cycles of fusion reactors. The reason for these welcomed applications is that the cycle can achieve overall energy conversion efficiency higher than 40%. Moreover economic competitiveness is increased due to the higher volumetric heat capacity of supercritical fluids. The supercritical carbon dioxide (SCO_2) turbine efficiency is one of the major parameters affecting the thermal efficiency. Thus, the optimal turbine design should contribute to economics of future nuclear energy systems.

2. Computational Analysis

Computational analysis of SCO_2 flow around a turbine blade was performed utilizing CFX® to check on the potential efficiency of turbine which determines such basic design values as the blade (or rotor) and nozzle (or stator) types, blade height, and the minimum and maximum radii of the hub and the tip. In this work a four stage turbine was designed and analyzed for performance testing.

2.1 Input of SCO_2 Properties to CFX®

CFX® need first be supplied with accurate SCO_2 thermophysical properties. The SCO_2 properties may be input to CFX® via such method as a lookup table or a user defined mode [1]. Since these methods are difficult to use, the real gas property (RGP) file provided by the Korea Atomic Energy Research Institute (KAERI) was adopted instead [2]. Simulation of the SCO_2 turbine with CFX® was found to produce proper results when compared against those obtained from the NIST code [3].

2.2 Three Dimensional Modeling

A three-dimensional (3D) configuration of the turbine was generated by ANSYS BladeGen™ on the basis of one-dimensional (1D) design properties provided by KAERI design code TURB1D [4]. Table I presents the basic design parameters for a four stage turbine [5-8].

Table I: Design parameters for four stages SCO_2 turbine in Brayton cycle for KALIMER-600

First Stage	Stator	Rotor
Hub Radius [cm]	53	
Shroud Radius [cm]	69~70	70~71
Depth [cm]	7.2	
Blade Angle [°]	60~65	
Second Stage	Stator	Rotor
Hub Radius [cm]	55	
Shroud Radius [cm]	72.5~73.5	73.5~74.5
Depth [cm]	7.8	
Blade Angle [°]	60~65	
Third Stage	Stator	Rotor
Hub Radius [cm]	57	
Shroud Radius [cm]	76~77	77~78
Depth [cm]	8.4	
Blade Angle [°]	60~65	
Fourth Stage	Stator	Rotor
Hub Radius [cm]	59	
Shroud Radius [cm]	79.5~80.5	80.5~81.5
Depth [cm]	9	
Blade Angle [°]	60~65	
Number of Blades	40	

2.3 Grids and Boundary Conditions

Optimal conditions were determined for the grid, time step, topology, and inflation through case studies. The boundary conditions were applied based on the secondary loop design. The inlet absolute pressure was 20 MPa at a total temperature of 787 K. The revolution of rotor was 60 per second. The average static pressure at the outlet was 7.6 MPa. Time scale in transient analysis was 8.3×10^{-4}. This value came from angular velocity of one grid. The result of the steady state was initial value for the transient analysis.

3. Results

The average pressure drop in each stage was 3.1 MPa, and the overall pressure drop 12.4 MPa. The total temperature difference was 100 K between the inlet and outlet. The 3D turbine shape and thermodynamic contours of working fluid which related with efficiency are described in Fig 1.

The mass flow rate was obtained as 8800 kg/s at 85 % of the turbine's static-to-static isentropic efficiency. The turbine efficiency was calculated as [9]

$$\frac{\dot{m}c_p(T_1-T_2)}{\dot{m}c_p(T_1-T_{2s})}=\frac{\dot{m}c_pT_1(1-\frac{T_2}{T_1})}{\dot{m}c_pT_1(1-\frac{T_{2s}}{T_1})}=\frac{1-(\frac{P_2}{P_1})^{\frac{\gamma-1}{\gamma}}}{1-\frac{T_{2s}}{T_1}} \quad (1)$$

The characteristic curves in Figs. 2 and 3 determined the operating point at the pressure ratio of 2.25. This indicates that the mass flow rate and efficiency are independent of the pressure ratio. Generally, the turbine is operated around this point. From the vector velocity and streamline, vortex and swirl were found at the trailing edge of the blade. This is the reason why the efficiency, which was calculated through the 3D CFX® analysis, is slightly lower than the design property. This means that the turbine shape need be modified to push the efficiency up to the optimal condition.

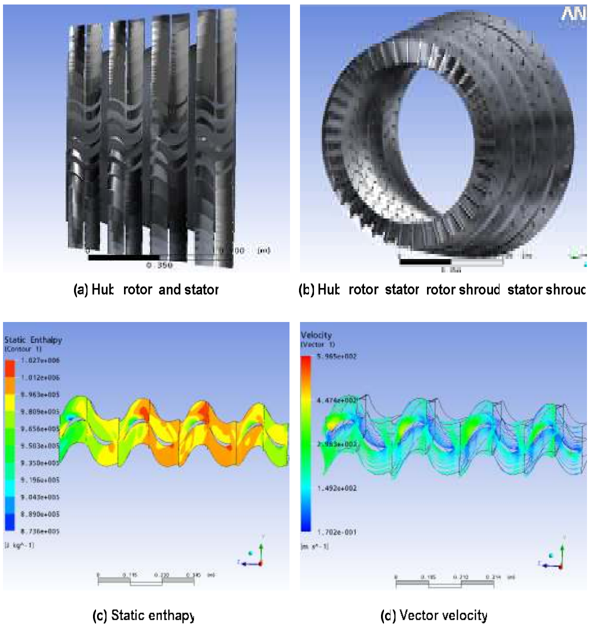

Fig. 1. 3D turbine shape and thermodynamic contours of working fluid around turbine blade: enthalpy, vector velocity.

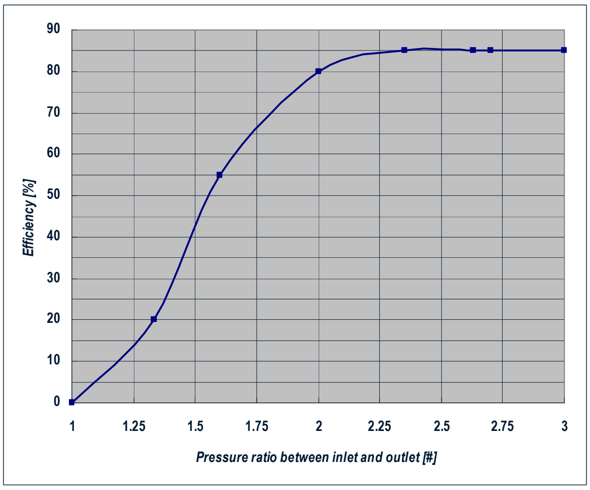

Fig. 2. Turbine performance curve: efficiency variation against pressure ratio between inlet and outlet.

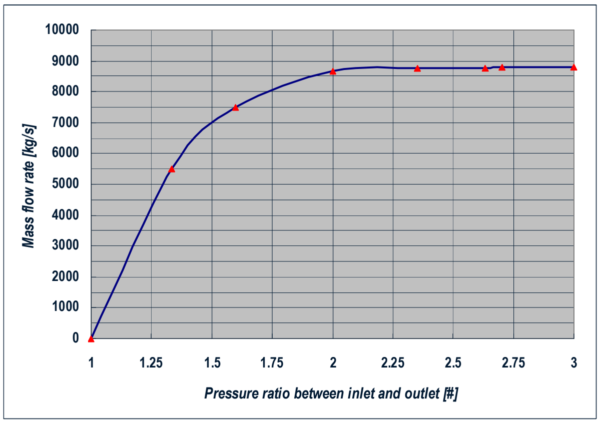

Fig. 3. Turbine performance curve: mass flow rate variation against pressure ratio.

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