Development of Throughflow Calculation Code for Axial Flow Compressors

Ji Hwan Kim, Hyeun Min Kim, and Hee Cheon NO

Korea Advanced Institute of Science and Technology, Guseong-Dong,Yuseong-Gu 373-1, Daejeon, Korea
Tel. +82-42-869-3857, Fax: +82-42-869-3810, Email: jhkim@nsys.kaist.ac.kr

1. Introduction

The power conversion systems of the current HTGRs are based on closed Brayton cycle and major concern is thermodynamic performance of the axial flow helium gas turbines. Particularly, the helium compressor has some unique design challenges compared to the air-breathing compressor such as high hub-to-tip ratios throughout the machine and a large number of stages due to the physical property of the helium and thermodynamic cycle. Therefore, it is necessary to develop a design and analysis code for helium compressor that can estimate the design-point and off-design performance accurately. KAIST nuclear system laboratory has developed a compressor design and analysis code by means of throughflow calculation and several loss models. This paper presents the outline of the development of a throughflow calculation code and its verification results.

2. Methods and Results

The flow in multi-stage compressor is inherently three-dimensional. It is necessary to simplify the flow as having an intermediate level of sophistication, because the real flow process in a multistage compressor is exceedingly complex. The flow is assumed to be inviscid and may be regarded as being obtained by circumferentially averaging all flow properties, and then the loss effects are added on to the throughflow solution.

From the assumption of axisymmetric and inviscid flow it is possible to define a series of meridional stream surfaces and there are surfaces of revolution along which fluid particles are assumed to move through the gas turbine. According to the description and approach given by J.D. Denton [1], the equation of motion along quasi-orthogonal line is

$$\frac{1}{2}\frac{d}{dq}V_m^{\ 2} = \frac{dh_0}{dq} - T\frac{ds}{dq} - \frac{1}{2r^2}\frac{d\left(r^2V_\theta^2\right)}{dq} + \frac{V_m^2}{r_c}\sin\alpha + V_m\frac{dV_m}{dm}\cos\alpha \qquad (1)$$

and the total mass flow rate across the quasi-orthogonal is given by

$$\dot{m} = \int_A^B 2\pi r\rho V_m \sin\alpha \ dq \qquad (2)$$

so that the meridional velocity and streamline curvature can be calculated from these equations.

Experiences of the open-cycle gas turbines have been sufficiently accumulated and steadily made to scale-up the capacity. In design of air compressors, loss estimation is very accurate based on the extensive experiences. Numerous empirical and semi-empirical relations have been provided to evaluate the losses of air compressors. However, there are very limited experiences with helium compressors, it is possible that loss estimation of helium compressor has some degree of error with existing loss relations.

The compressor losses are dependent on the total drag coefficient throughout the flow path. In devising the model, four kinds of primary loss sources were identified. Firstly, the profile loss is caused by the flow around the blade boundary layers and the wake of trailing edge. Secondly, the annulus loss is associated with the end-wall boundary layers. Thirdly, the secondary flow loss is due to the cascade vortices and the trailing vortices. Lastly, the tip clearance loss is due to the leakage through the tip clearances. In design of helium compressor, however, shock loss is not concerned because the compressor operates only at the low subsonic region. In other words, helium gas has the very high sonic velocity so that the formation of shocks on the blade does not occur.

The profile loss coefficient is given by the Koch-Smith [2] model.

$$\zeta_{profile} = \frac{2\times\dfrac{\delta_{momentum}\times\sigma}{\cos\beta_2}\left(\dfrac{\cos\beta_1}{\cos\beta_2}\right)^2\dfrac{2H}{3H-1}}{\left(1-\dfrac{\delta_{momentum}\times\sigma}{\cos\beta_2}H\right)^3} \qquad (3)$$

Applying the wetted-wall area ratio given by Stewart-Whitney, the annulus loss coefficient is presented as follows:

$$\zeta_{wall} = \zeta_{profile}\times\frac{l_m\times(r_{hub1}+r_{tip1}+r_{hub2}+r_{tip2})}{\sigma\times(r_1+r_2)\left(\dfrac{h_1+h_2}{2}\right)}\times\frac{\sin\dfrac{|\beta_1-\beta_2|}{2}}{|\beta_1-\beta_2|} \qquad (4)$$

The secondary flow loss is found from the Lakshiminarayana-Horlock [3] model.

$$\zeta_{secondary} = \frac{0.16}{\sigma^2}\times\frac{1}{AR}(\tan\beta_1-\tan\beta_2)^2\cos^2\beta + \frac{0.0423\times C_{lift,hub}^2\times\left(1-\dfrac{C_{lift,tip}}{C_{lift,hub}}\right)}{AR} \qquad (5)$$

The Lakshiminarayana [4] model gives the tip clearance loss as follows:

$$\Delta P_{tip} = \frac{\rho_1 (r_{tip1}\omega)^2}{2} \times \frac{0.7\times \psi_{tip1}^2}{\cos\beta_{tip}} \times \frac{c_{tip}}{(h_{blade1}+h_{blade2})/2} \times \left(1+10\sqrt{\frac{\phi_{tip1}}{\psi_{tip1}\times\cos\beta_{tip}} \times \frac{(h_{blade1}+h_{blade2})/2}{c_{tip}}}\right) \quad (6)$$

Finally, the total pressure loss for rotor and stator is calculated by following relations:

$$\Delta P_{rotor} = \int_{hub}^{tip} \left\{ \left(\zeta_{profile} + \zeta_{wall} + \zeta_{secondary}\right) \frac{\rho_1 W_1^2}{2} \right\} dm(r) + \Delta P_{tip} \quad (7)$$

$$\Delta P_{stator} = \int_{hub}^{tip} \left\{ \left(\zeta_{profile} + \zeta_{wall} + \zeta_{secondary}\right) \frac{\rho_2 V_2^2}{2} \right\} dm(r)$$

These loss models are widely accepted and used in major company such as GE and Concept-NREC.

The GTHTR300 compressor design of JAERI has been selected to verify the code results. Table 1 shows a summary of the design parameters. This compressor has the constant inner wall diameter and the unique design features such as the short blade height throughout the 20 stages. The performance comparisons of JAERI and KAIST are shown in Table 2 and Figure 1. There are some deviations at conditions far away from the design-point because of the applicability limits of the profile loss and annulus loss models. However, the code results of KAIST have generally good agreement, especially at the design-point, and the surge points are also well predicted.

Table 1. Design specifications of example

Mass flow rate (kg/s)	449.7
Inlet temperature (°C)	28
Inlet pressure (MPa)	3.52
Pressure ratio	2.0
Hub diameter (mm)	1500
Tip diameter (1st/20th stage, mm)	1704/1645
Hub-to-tip ratio (1st/20th stage, mm)	0.88/0.91
Number of stages	20
Rotational speed (rpm)	3600
Number of rotor/stator blades (1st stage)	72/94
Rotor/stator blade chord length (1st stage, mm)	78/60
Rotor/stator blade height (1st stage, mm)	102/101
Polytropic efficiency (%)	90.5

Table 2. Comparison of design-point performance

Parameters	JAERI	KAIST	Error (%)
Pressure ratio	1.9972	1.9988	0.008
Temperature ratio	1.3586	1.3533	-0.392
Polytropic efficiency (%)	90.50	91.51	1.116
Shaft Work (MW)	252.2	248.5	-1.467

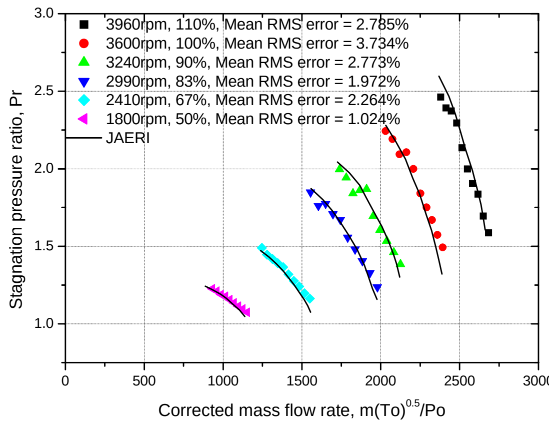

Figure 1. Pressure ratio vs. Mass flow characteristics

3. Conclusions

A throughflow calculation code has been developed for estimation of the performance of multistage axial flow compressors. The loss relations for air compressors are applied due to the lack of experiences with helium compressors. These throughflow solutions with the loss models those are adapted in this paper give good predictions around the design-point. The off-design results show also generally good agreement with the reference. However, the wide-range profile loss models are required to estimate the performance far away from the design-point.

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