Transactions of the Korean Nuclear Society Autumn Meeting
Gyeongju, Korea, October 26-27, 2023

The effect of new windage loss model to KAIST-MMR turbomachinery performance prediction

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***Keywords : Windage loss model, KAIST-MMR, KAIST-TMD, Centrifugal compressor, Radial type turbine**

1. Introduction

Environmental regulations have made nuclear power a competitive option for many countries around the world due to its lower carbon emissions and lower cost of energy production than other green energy sources. Small modular nuclear power plants are particularly well suited to locations where water is scarce or far from the power grid system. The KAIST research team has developed a micro modular reactor (KAIST-MMR) with an electrical output of 10 MWe that fits these needs [1].

One of the key requirements for small modular nuclear power is to reduce the size of the power generation system. A power cycle configuration using supercritical carbon dioxide (sCO_2) can satisfy this need with the following advantages. First, the size of the rotor and heat exchanger can be reduced, especially for the compressor, which operates with inlet conditions near the critical point of 7.4 MPa and 31 degrees Celsius. At this inlet condition, the size of the compressor can be reduced due to the compressibility factor reaching 0.4, which is much smaller than the typical ideal gas value of unity. The second merit is that the cycle can be constructed with only a small number of components. KAIST research team has shown that even a simple recuperated cycle can achieve competitive cycle thermal efficiency. This small and simple cycle configuration has enabled the KAIST research team to develop a 10 MWe class micro modular reactor.

Many loss models for turbomachinery have been proposed based on ideal gas and steam. However, due to the special thermal properties of sCO_2, there is a need to develop suitable loss models. KAIST research team presented a set of loss models suitable for sCO_2 [2]. For compressors and turbines, windage losses dominate over other losses. Recently, Kim et al. presented a new windage loss model [3]. The new windage loss model is predicted to have about 50% smaller losses than the existing windage loss predicted with the model of Vrancik [4]. In this study, the authors applied the new windage loss model to the performance prediction code of compressor and turbine for KAIST-MMR conditions to check the changes in component performance and cycle efficiency. The concept diagram of KAIST-MMR is shown in Fig. 1.

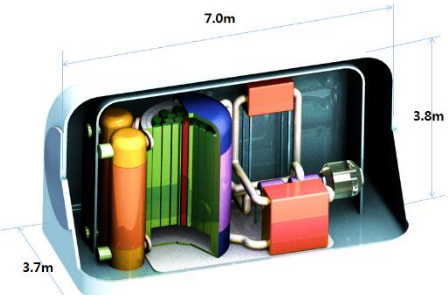

Fig. 1. Schematic of KAIST-MMR.

2. Windage loss model

Windage loss occurs in the gap between the rotor shaft, which transmits the turbomachinery's torque, and the casing surrounding the shaft as shown in Fig. 2. A tangential flow velocity difference occurs between the stationary casing and the rotating shaft. This phenomenon develops into Talyor vortex flow, which resists the torque of shaft and generates heat in the fluid [5]. Windage loss belongs to the external loss of the turbomachinery, so this loss does not affect the pressure ratio, but only the efficiency. Windage loss has higher contribution than other losses as shown in Fig. 3. Equation of specific enthalpy loss is shown in Eq. 1. The specific heat of sCO_2 near the critical point changes rapidly with temperature and pressure. Therefore, existing windage loss models such as the Vrancik and Mack models become inappropriate to apply under KAIST-MMR conditions. Kim et al. evaluated the existing windage loss models using a compressor section at the KAIST sCO_2 test facility. The compressor used in this experiment is shown in Fig. 4 [3]. Equations and regression results of sCO_2 condition for the Vrancik and Mack models are given in Table I [4, 6].

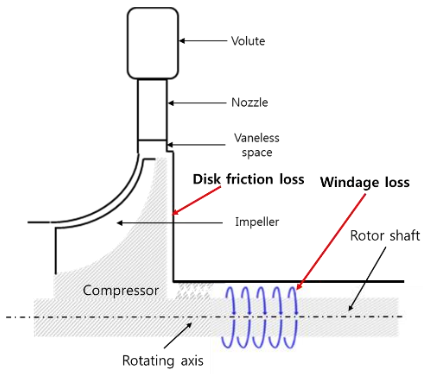

Fig. 2. Windage loss and Disk friction loss

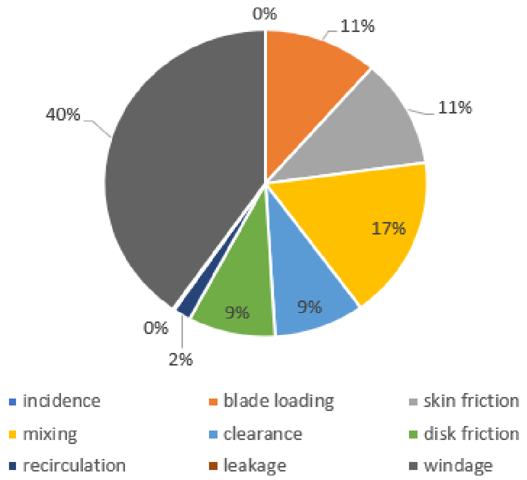

Fig. 3. Contribution of losses in compressor (Vrancik's model)

$$h_{wind,loss} = C_f \pi \rho \omega^3 r^4 l / \dot{m} \quad \text{(Eq .1)}$$

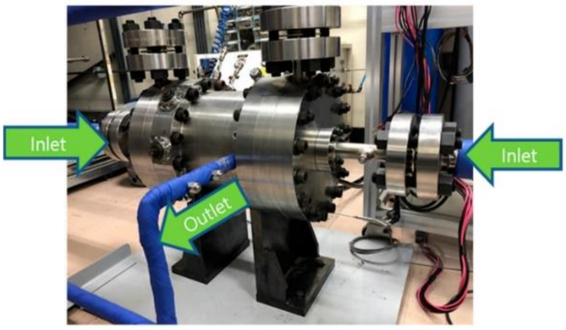

Fig. 4. KAIST sCO_2 rotating machinery test section

Table I: Existing windage loss models for Air and their regression results in sCO_2 condition [4, 6]

Author	Empirical Equation		R^2 (sCO_2 condition)
Vrancik	$C_f = \frac{K_{mack}}{Ta^{0.2}}$	$\sqrt{Ta} > 400$	0.7133
Mack	$C_f = \frac{1.8}{Re_r C_r^{0.25}} * \frac{(1+C_r)^2}{(1+C_r)^2 - 1}$	$\sqrt{Ta} \le 41.3$ (laminar)	0.8604
	$\frac{1}{\sqrt{C_f}} = 2.04 + 1.768 * \ln(Re_c \sqrt{C_f})$	$\sqrt{Ta} > 400$ (turbulent)	

Compared to other loss models, the Mack's model was evaluated to have the largest regression value [3]. Therefore, Kim et al. proposed a new model that can be applied to sCO_2 conditions by applying the similitude method by specific heat ratio (γ) of the Mack's model, which is shown in Eq. 2. $\boldsymbol{C_r}$ is the ratio of clearance and shaft radius, and the $\boldsymbol{Ta_{inlet}}$ is the Talyor number of sCO_2 condition. $\boldsymbol{Ta_{crit}}$ is a critical value that determines turbulent flow from laminar flow.

In this paper, this new windage loss model is applied to the performance prediction of compressor and turbine in KAIST-MMR and compared the results with the previous results obtained from Vrancik's model.

$$C_{f,new,s} = \frac{1.8}{Ta_{crit}^{0.4} c_r^{-1.25}} \frac{(1+c_r)^2}{(1+c_r)^2 - 1} Ta_{inlet}^{-0.1} \left(\frac{\gamma_{inlet}}{\gamma_{air}} \right)^{-0.4051} \quad \text{(Eq. 2)}$$

The major values of temperature, pressure, and mass flow rates of the KAIST-MMR and the specifications of the compressor and turbine in the system are shown in Table II. Both compressor and turbine are radial type.

Specification	Value	Unit
RPM	19300	rpm
Mass flow Rate	175	kg/s
Turbine inlet Temperature	550	°C
Turbine inlet Pressure	20	Mpa
Turbine inlet Radius	0.145	m
Compressor inlet Temperature	60	°C
Compressor inlet Pressure	8	Mpa
Compressor outlet Radius	0.13	m

Table II: Specifications of KAIST-MMR

3. Results

Fig. 5 shows the efficiency maps of the compressor with Kim's model and Vrancik's model. Efficiency of compressor is predicted to be higher when Kim's model is applied because Vrancik's and Mack's models predict enthalpy loss larger. The points shown are the mass flow rate and efficiency at the on-design state of the KAIST-MMR cycle. The other curves are efficiency curves for RPM from 60% to 120% of the design RPM and controlling the mass flow rate. Fig. 6 shows that when Kim's model is applied the contribution of total enthalpy loss due to windage loss is now reduced to 30%, which is different from Fig. 3.

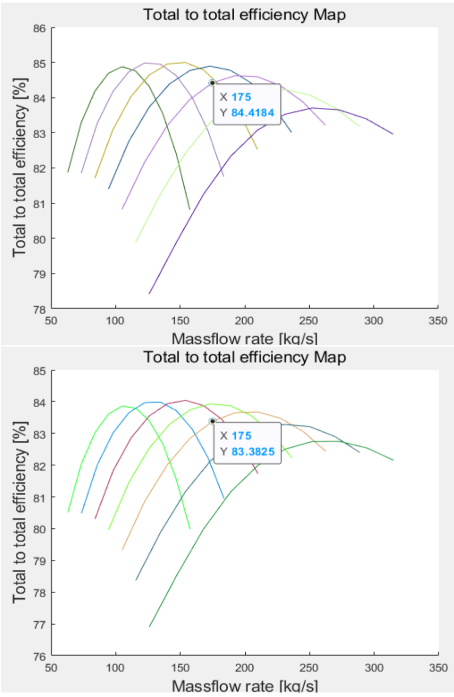

Fig. 5. Compressor efficiency map applying Kim's model (up) and Vrancik's model (down)

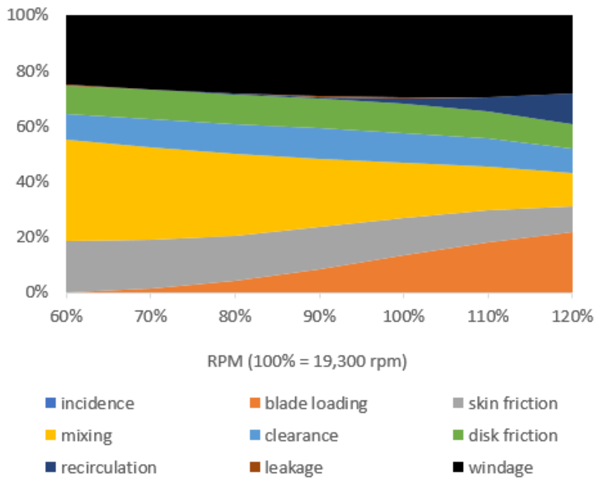

Fig. 6. Contribution of losses for different RPM conditions (Kim's model)

Fig. 7 shows the efficiency maps of turbine with Kim's model and Vrancik's model. Applying different windage loss models, compressor shows a 2% efficiency difference, while turbine shows only a 0.2% efficiency difference. Since the disk diameter of the turbine is about 50% larger than the disk diameter of the compressor, the disk friction loss has a larger value than windage loss as shown in Fig. 8. Also, since the density of the turbine side is 30% smaller than the density of the compressor side, the windage loss itself is calculated to be lower than that of compressor side. Therefore, even if the windage loss model is changed, the efficiency prediction of the turbine is only 0.2% different. Table III shows the summary of efficiency result.

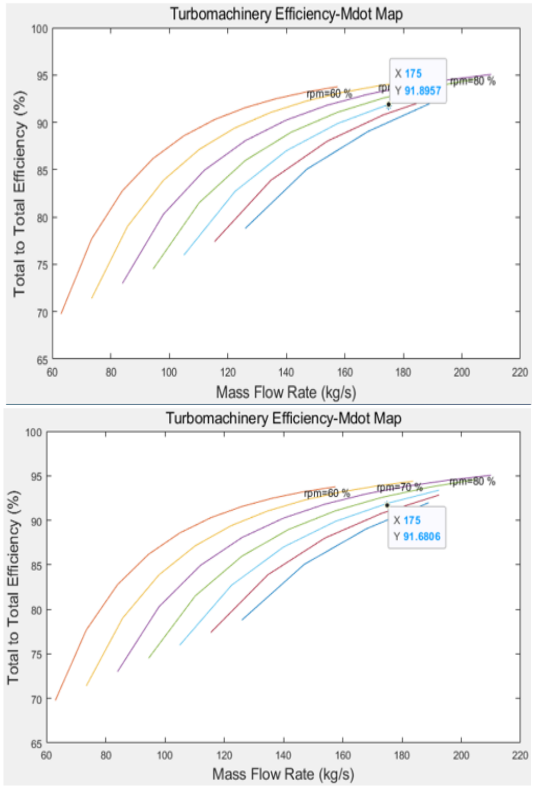

Fig. 7. Turbine efficiency maps applying Kim's model (up) and Vrancik's model (down)

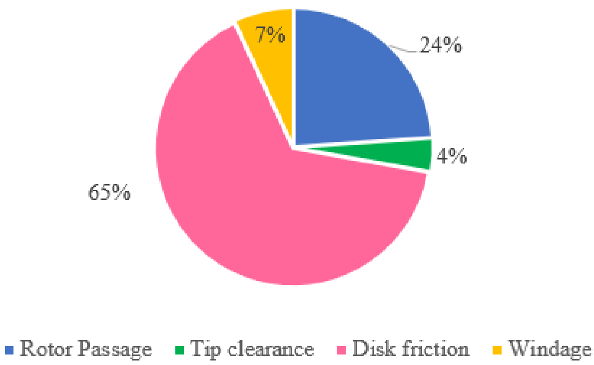

Fig. 8. Contribution of losses in turbine (Kim's model)

Table III: Efficiency Summary

Efficiency (%)	Kim	Vrancik	Difference
Compressor	84.42	83.38	1.04
Turbine	91.90	91.68	0.22

4. Summary

The KAIST research team has developed a new windage loss model that predicts 50% smaller losses than the the existing Vrancik model for turbomachinery. This model is applied to the performance prediction of compressor and turbine for KAIST-MMR to check the change in component efficiency. The new windage loss model is predicted to have smaller contribution to the total enthalpy loss. The efficiency of the compressor is predicted to be higher about 2% when Kim's model is applied, while the turbine shows only a 0.2% efficiency difference. The difference between compressor and turbine originates from the difference in the disk friction loss in turbine, which is larger than windage loss.

ACKNOWLEDGEMENT

This research was supported by the Challengeable Future Defense Technology Research and Development Program (No.912767601) of Agency for Defense Development in 2023.

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