Supercritical CO_2
Power Cycles
Symposium

S-CO_2 radial turbine testing data and performance map for thermodynamic conditions with strong real gas effect

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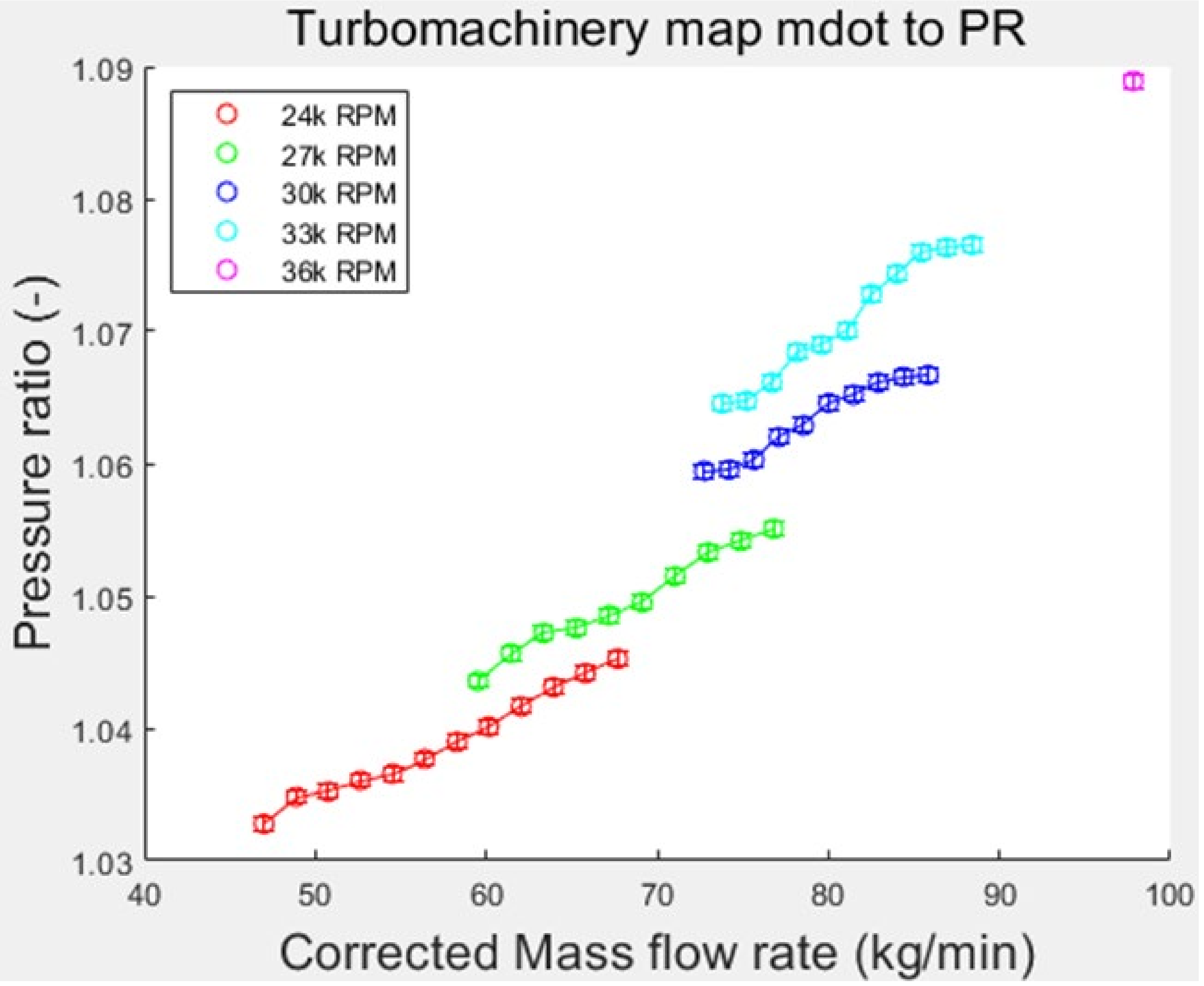
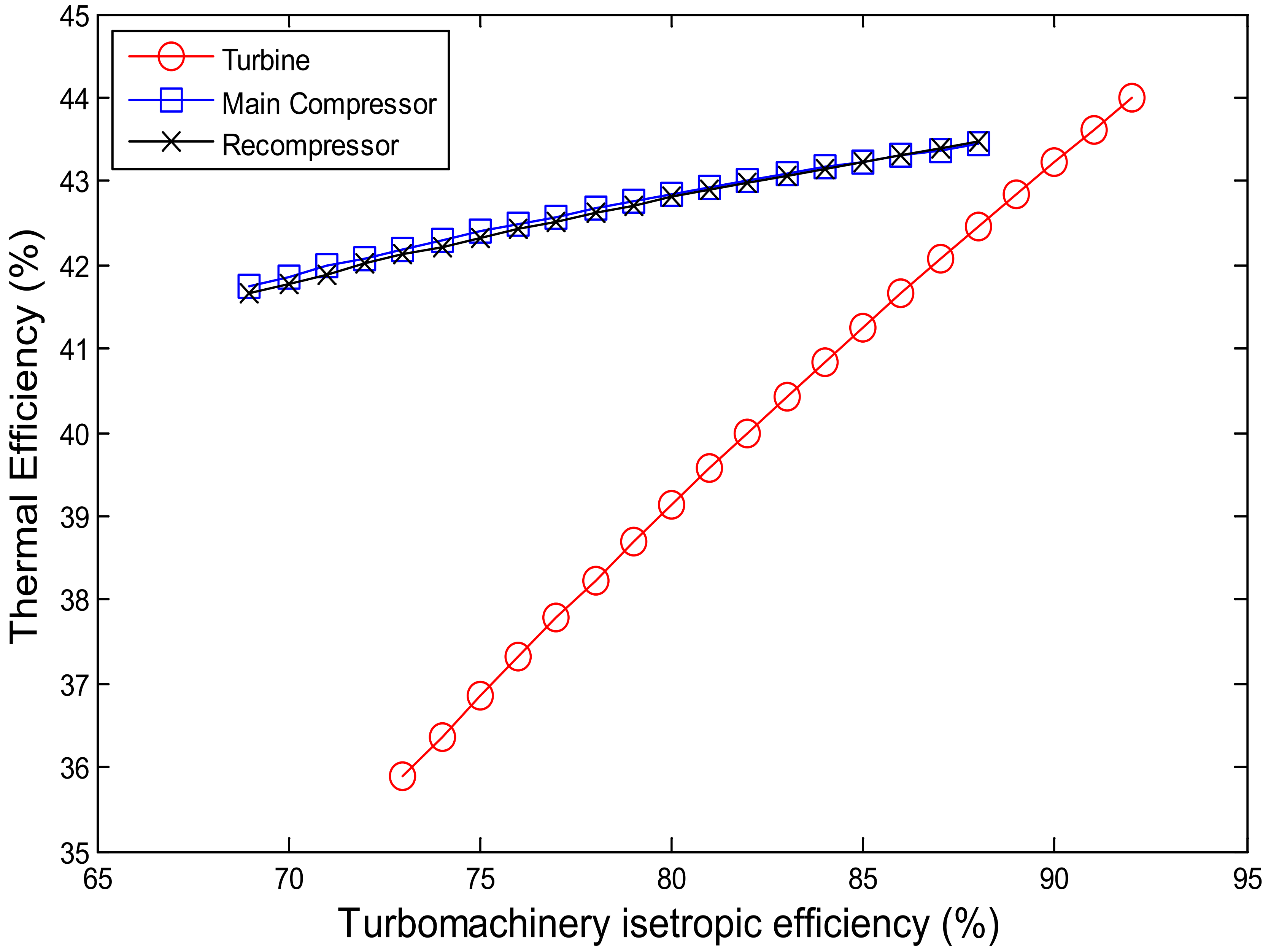
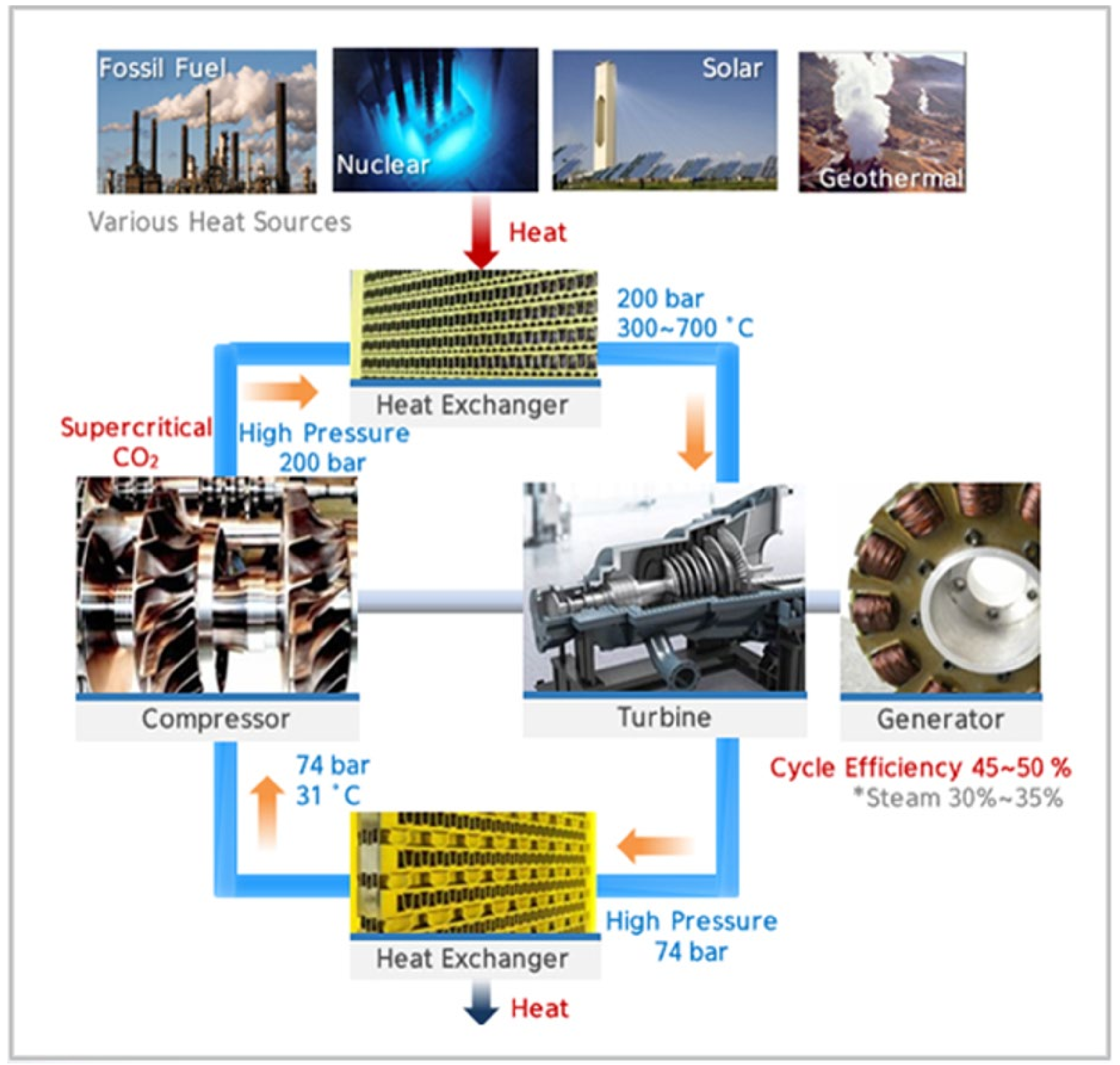
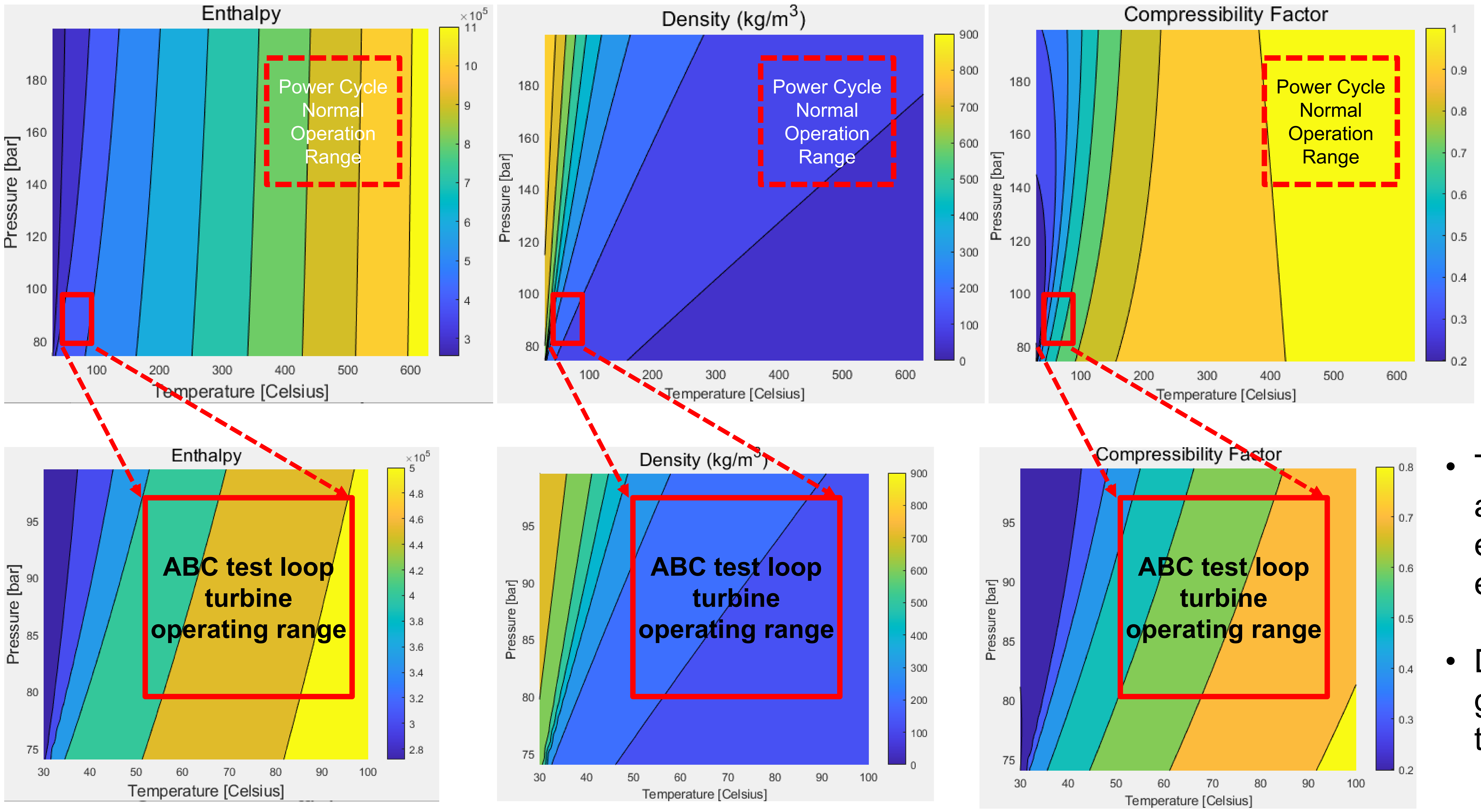


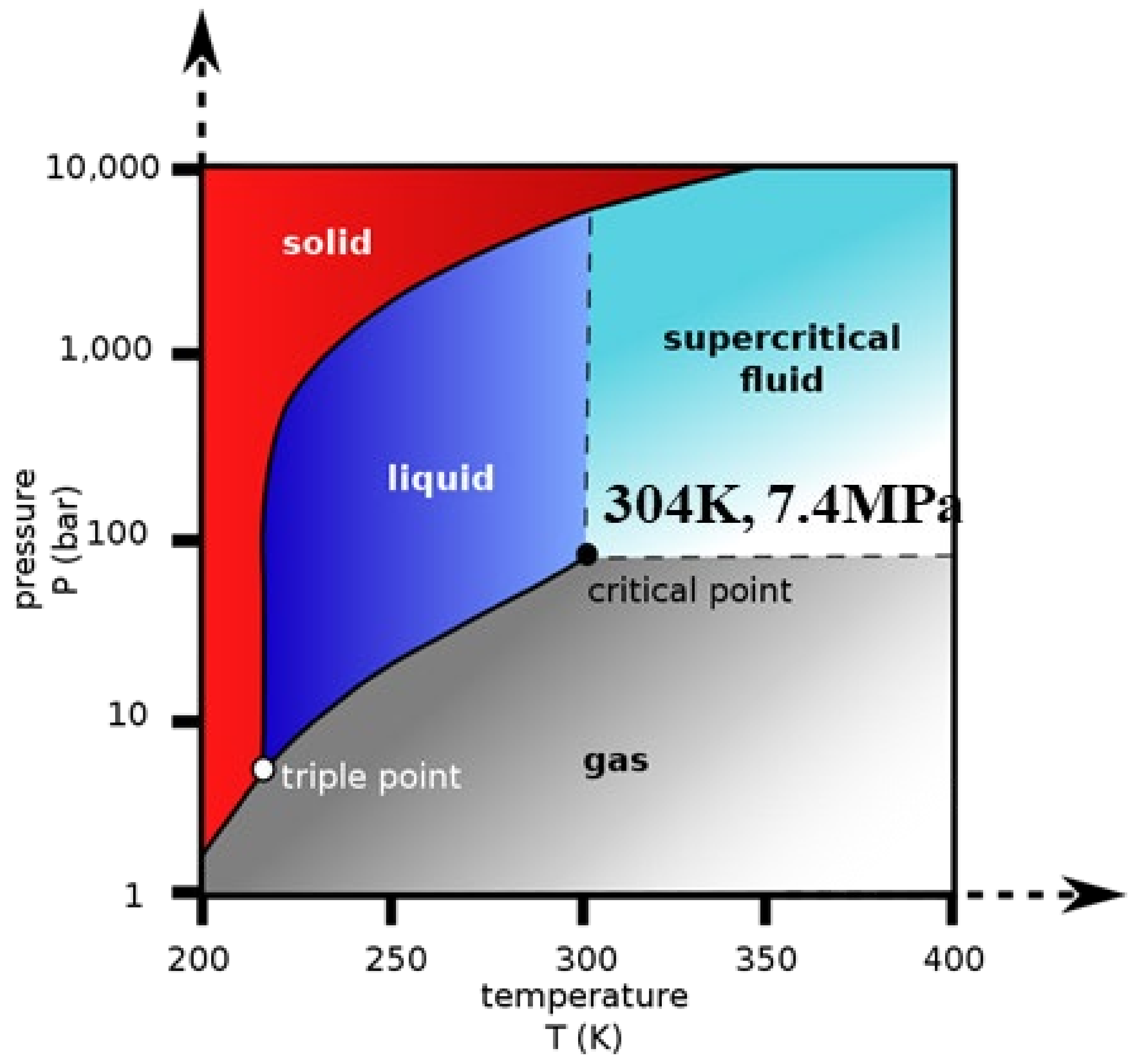




- In a typical power cycle, the behavior of a S-CO_2 turbine can be expected to be similar to an air-combustion turbine.
- However, the efficiency of the turbine has a significant impact on the overall efficiency of the power cycle.
- Therefore, when designing a turbine with a higher density of S-CO_2 as the working fluid, it is necessary to take into account the special properties of S-CO_2, such as the change in physical properties.
- Therefore, the experimental performance of turbine near the critical point can be used as a database for the precision design of turbines.

Thermal properties of S-CO_2

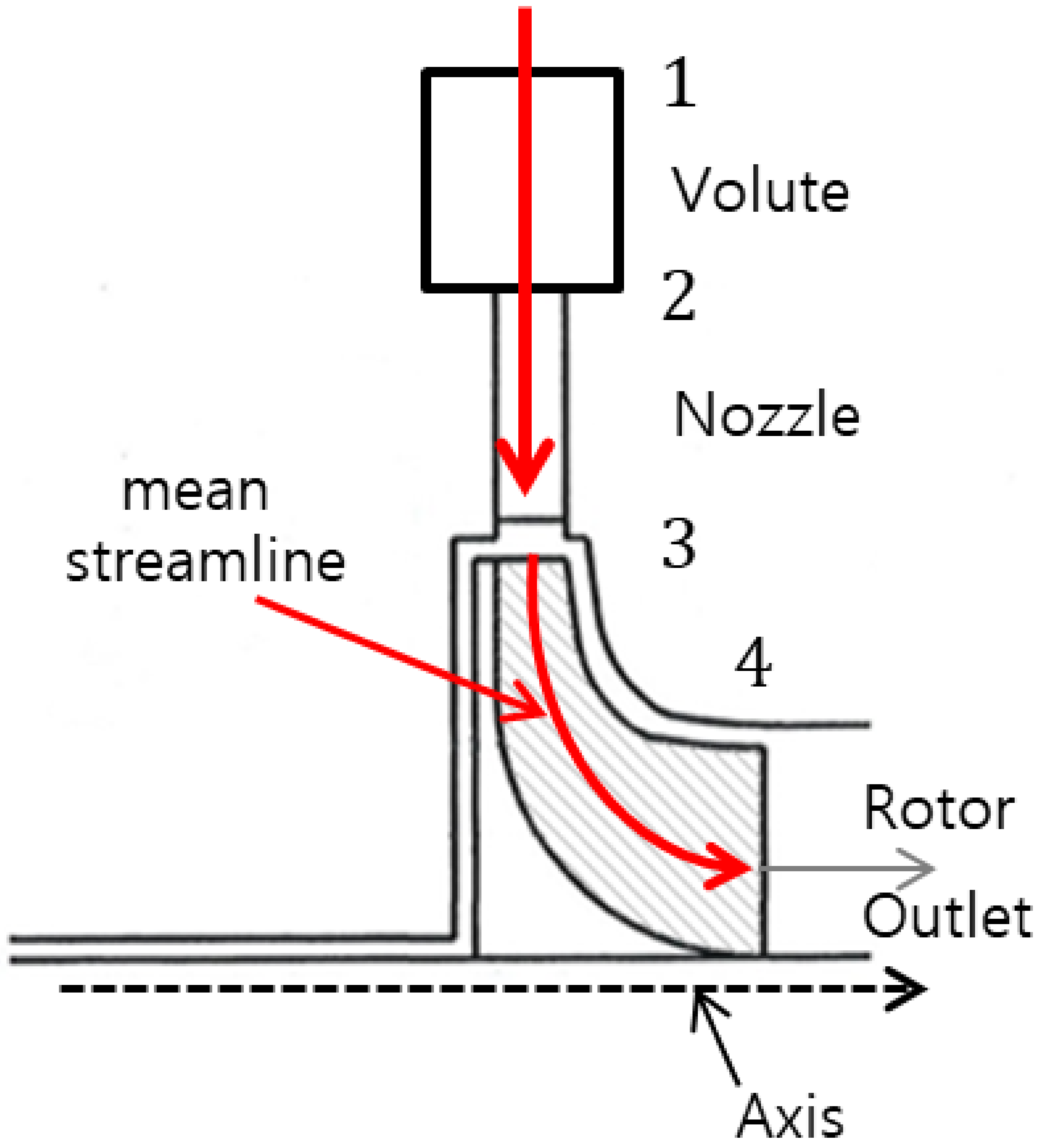




- The range of operating temperatures and pressures for the turbine in this experiment is where the real gas effect of S-CO_2 is significant.
- Density and compressibility factor greatly affect the performance of a turbine.

Radial turbine

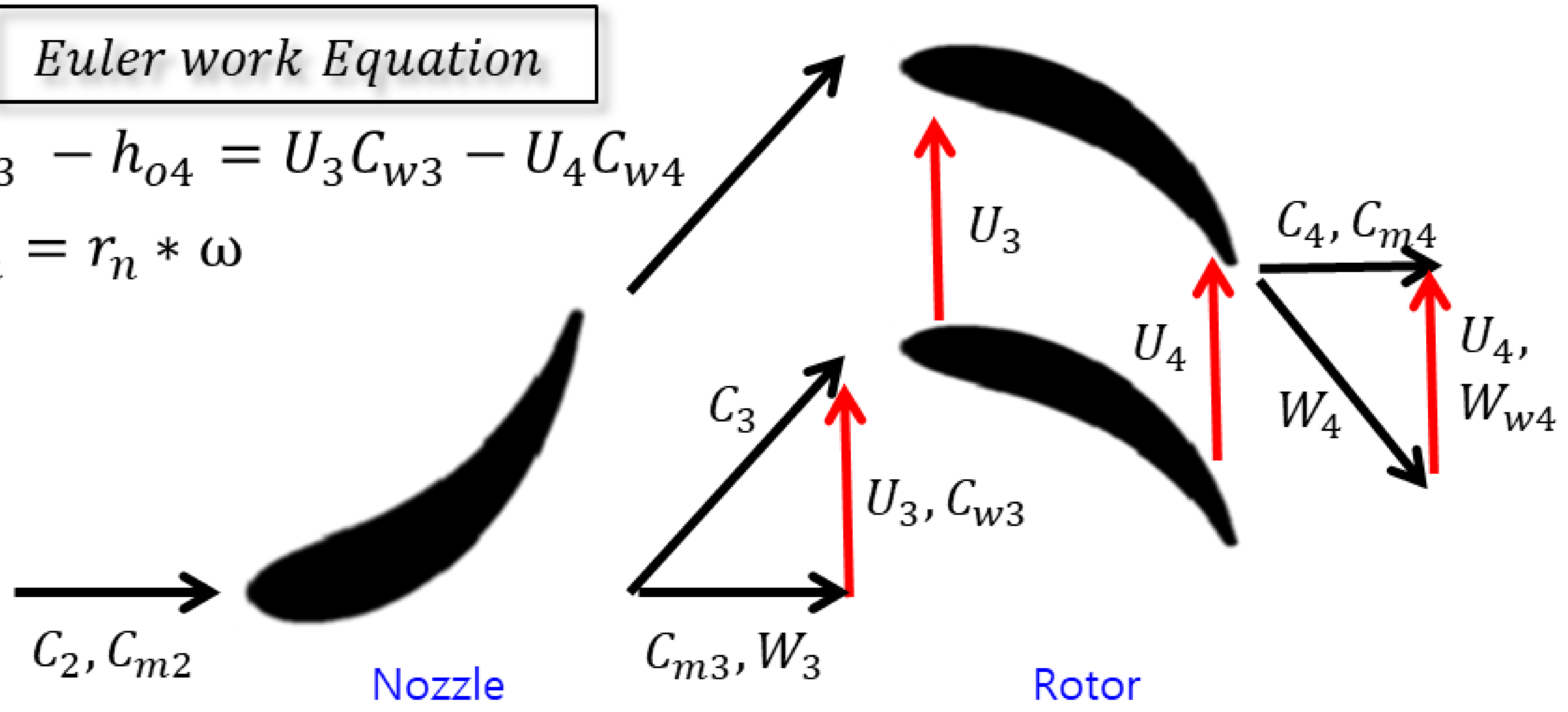
Nomenclature	
Absolute Velocity	C
Blade Velocity	U
Relative Velocity	W
Meridional direction	m
Tangential direction	w
Volute Inlet	1
Nozzle Inlet	2
Rotor Inlet	3
Rotor Outlet	4



Euler work Equation

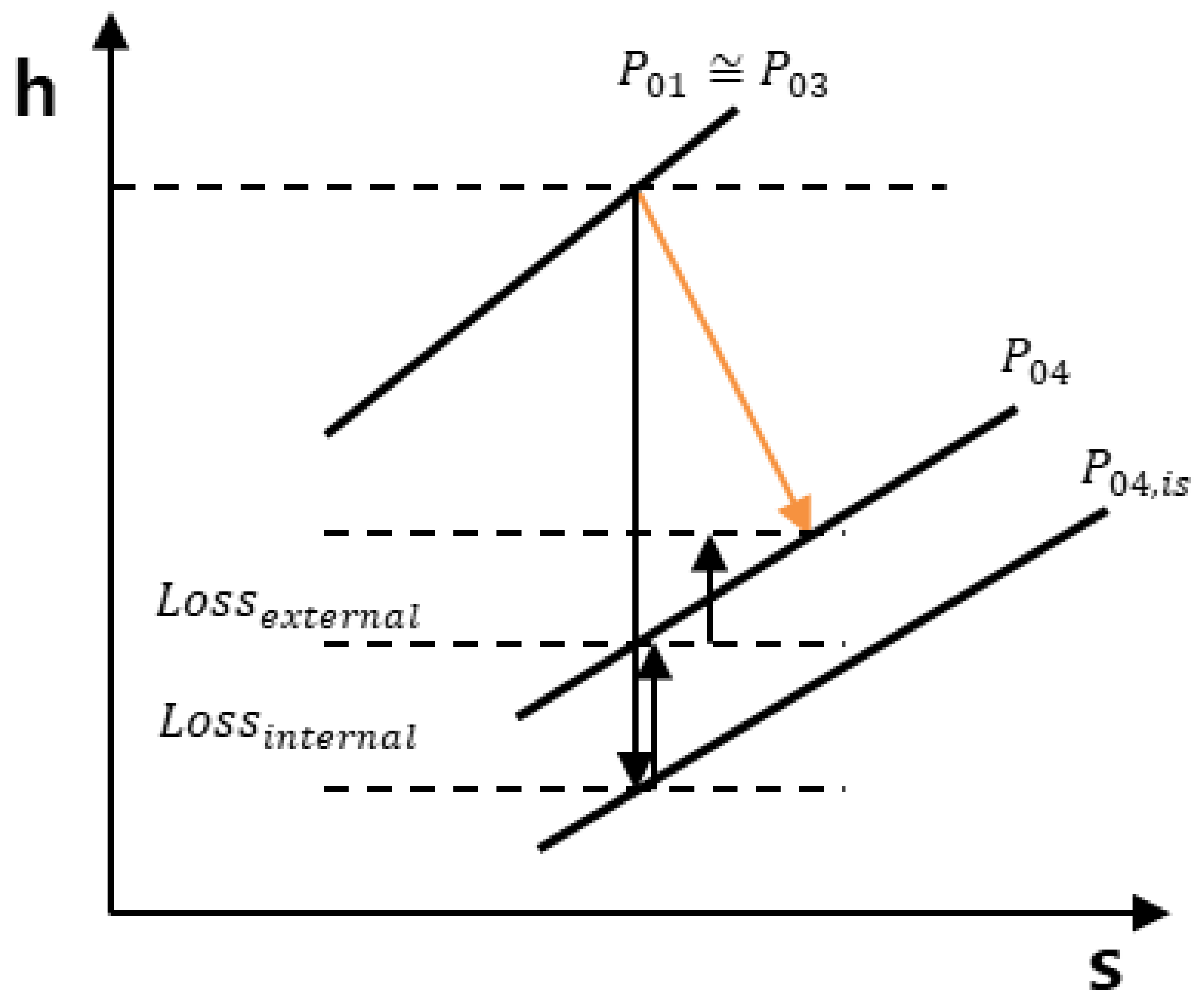
$$h_{o3} - h_{o4} = U_3 C_{w3} - U_4 C_{w4}$$

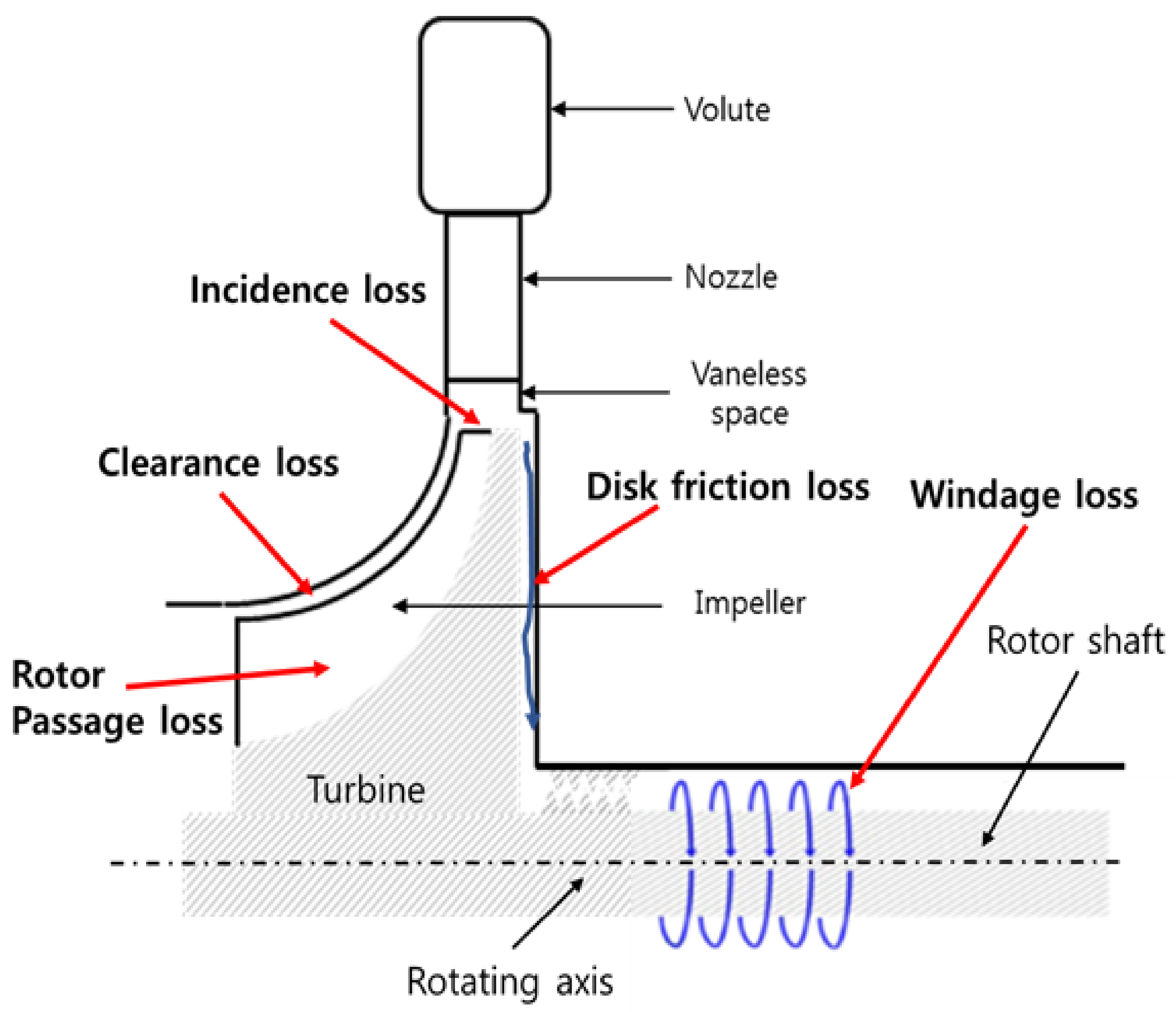
$$U_n = r_n * \omega$$



- Radial inflow turbine is more suitable than axial turbine for power cycle with a small power output of 10 MW electric or less.
- The working fluid passes through each part of the radial inflow turbine in the following order: volute, nozzle, and rotor in a radial inflow turbine.
- Euler Work Equation : The difference between the velocities of the blades (U) multiplied by the tangential absolute velocity (C_w) of the fluid becomes the stagnation enthalpy (h_o) difference at the turbine inlet and outlet.

Radial turbine losses



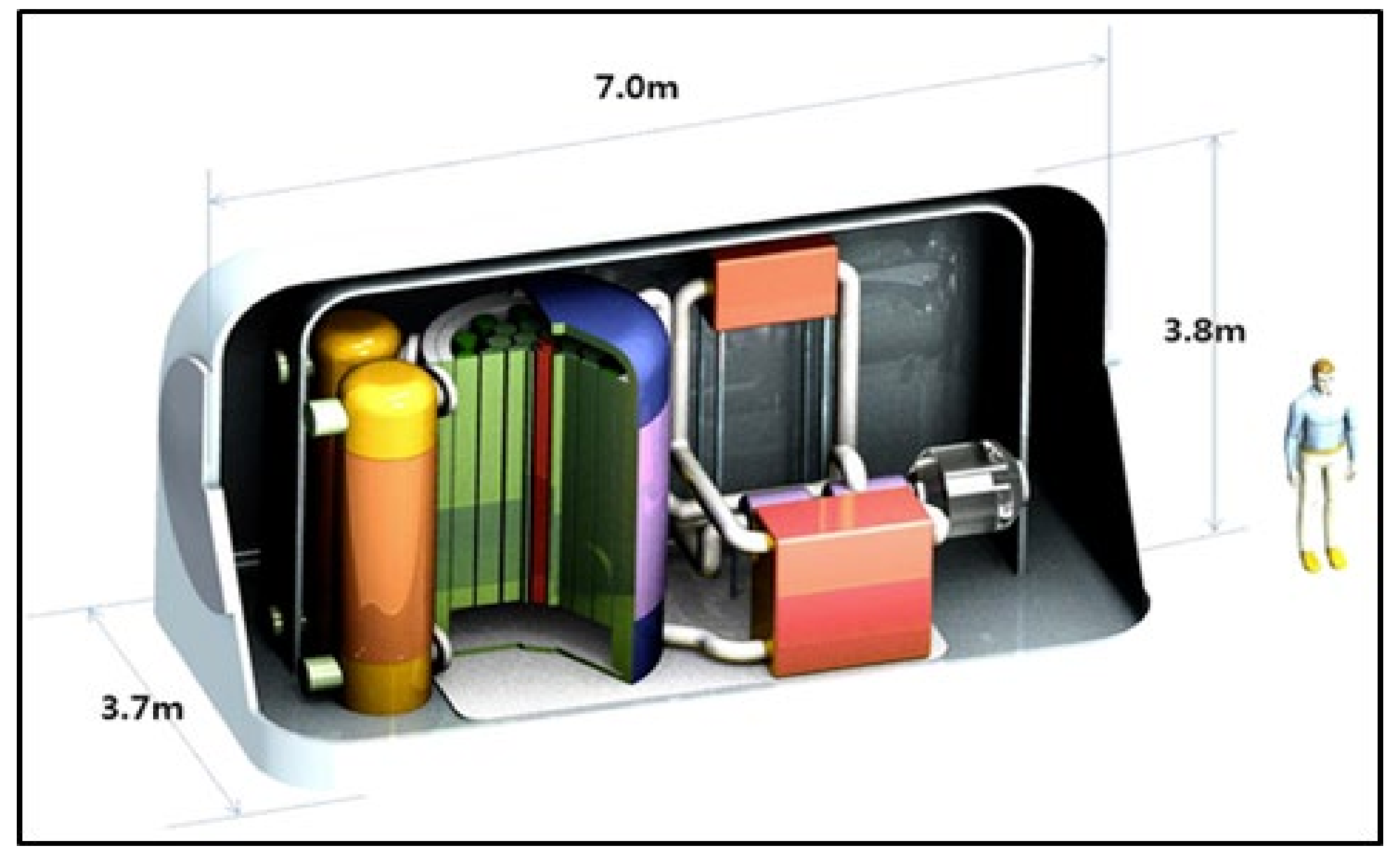


Loss category	Featured Model Proposer	Type
1. Nozzle loss	Hiett and Johnson	Internal
2. Incidence loss	Wasserbauer and Glassman	
3. Rotor Passage loss	Moustapha	
4. Clearance loss	(Absent in shrouded turbine)	
5. Disk friction loss	Daily and Nece	External
6. Windage loss	Kim and Lee	

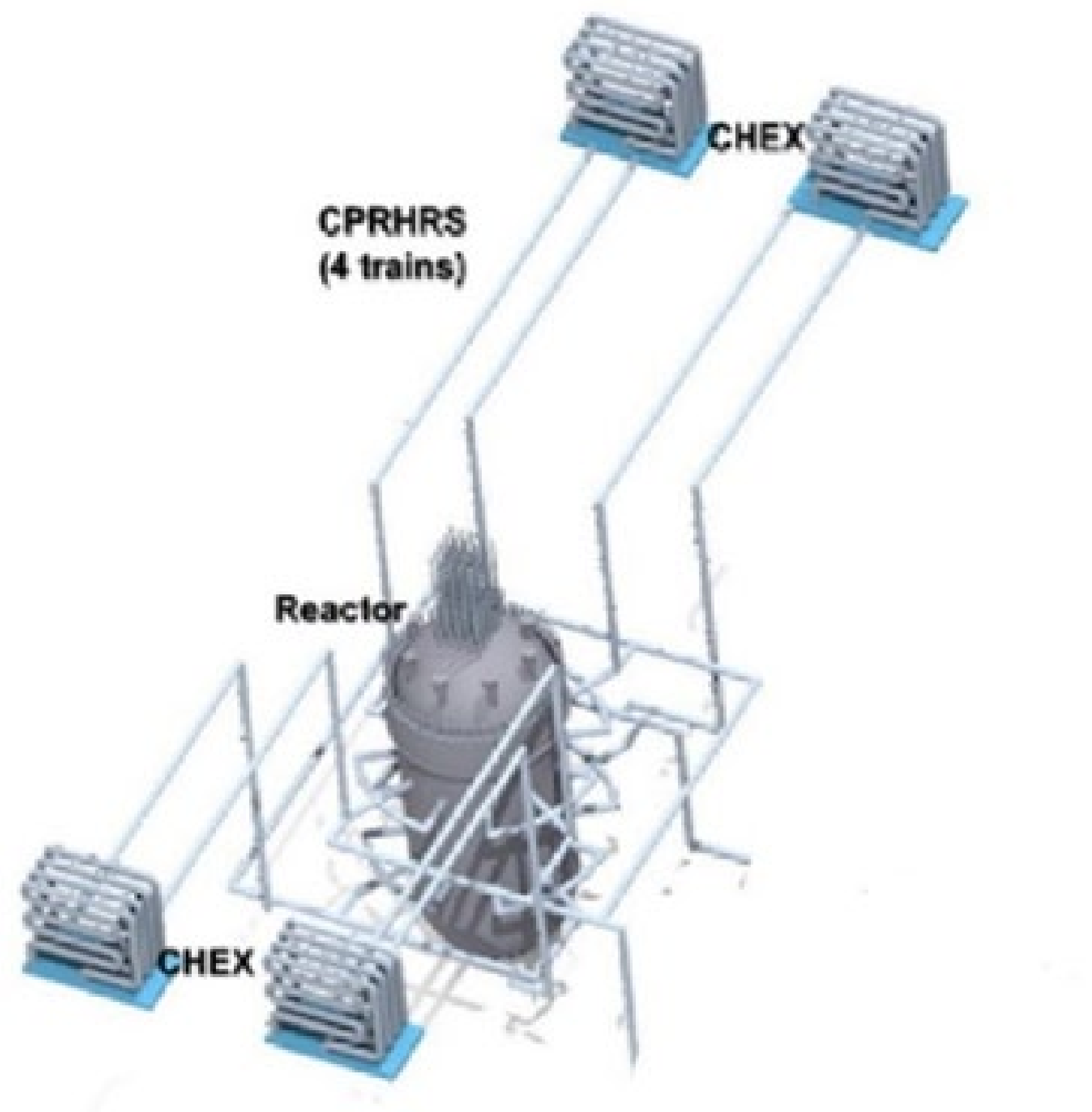
- There are two kinds of losses in a radial turbine; Internal losses and External losses.
- Internal losses affect both pressure ratio and efficiency, while external losses affect only efficiency.
- Loss models for S-CO_2 radial turbines have been validated in compressors, or are being developed and calibrated through recent experiments.
- It is noted that for a shrouded type rotor, there is a leakage flow that occurs between the shroud and stator. This loss is included in the external loss and does not affect the pressure ratio or pressure drop of the rotor performance.

Two methods for evaluate real gas effect

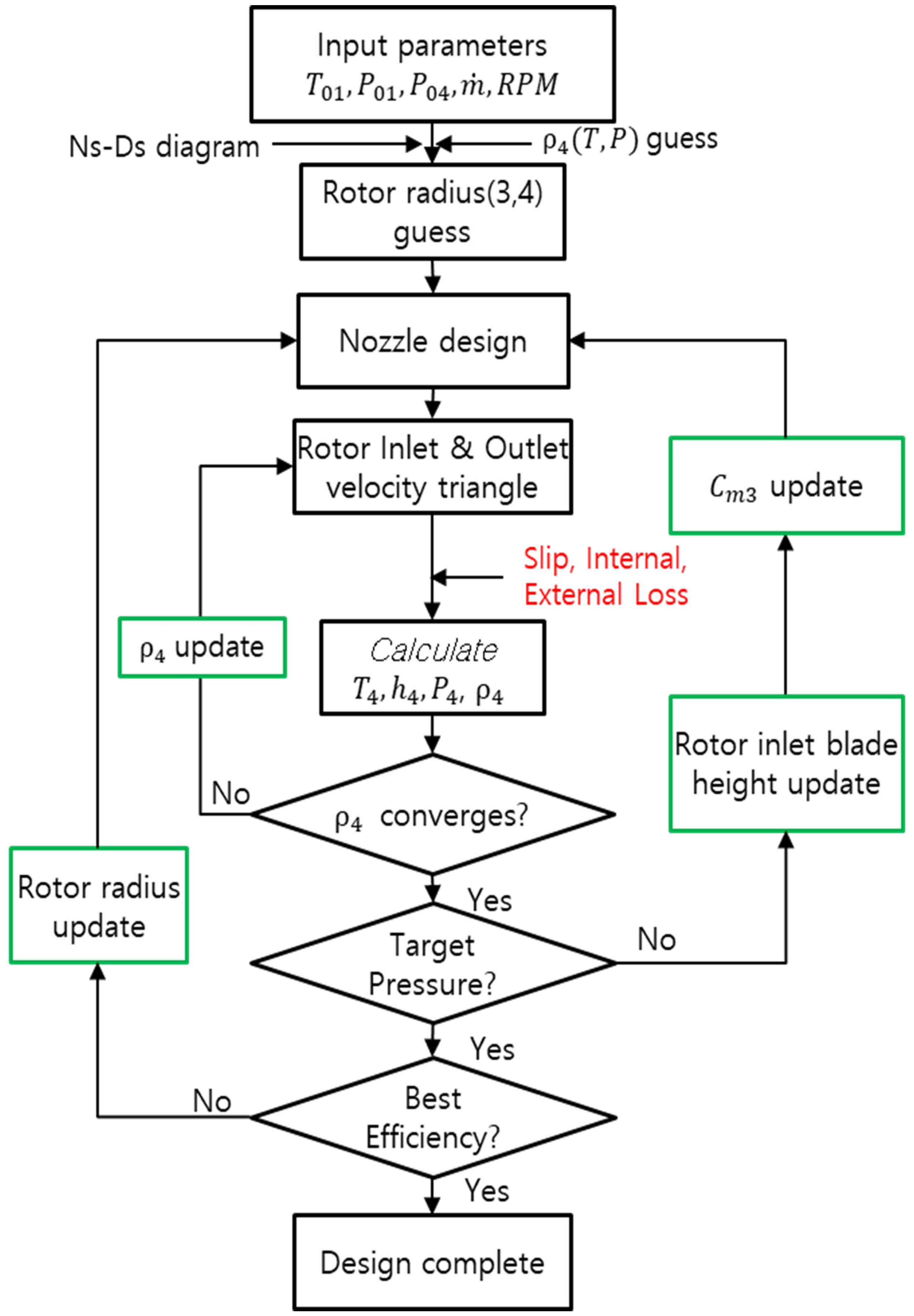
Code evaluation for large power cycle



KAIST-MMR



S-CO$_2$ PRHRS for SMR



Input parameters $T_{01}, P_{01}, P_{04}, \dot{m}, RPM$

Ns-Ds diagram

$\rho_4(T,P)$ guess

Rotor radius(3,4) guess

Nozzle design

Rotor Inlet & Outlet velocity triangle

Slip, Internal, External Loss

Calculate T_4, h_4, P_4, ρ_4

ρ_4 update

No

ρ_4 converges?

Yes

C_{m3} update

Rotor inlet blade height update

Target Pressure?

No

Rotor radius update

Yes

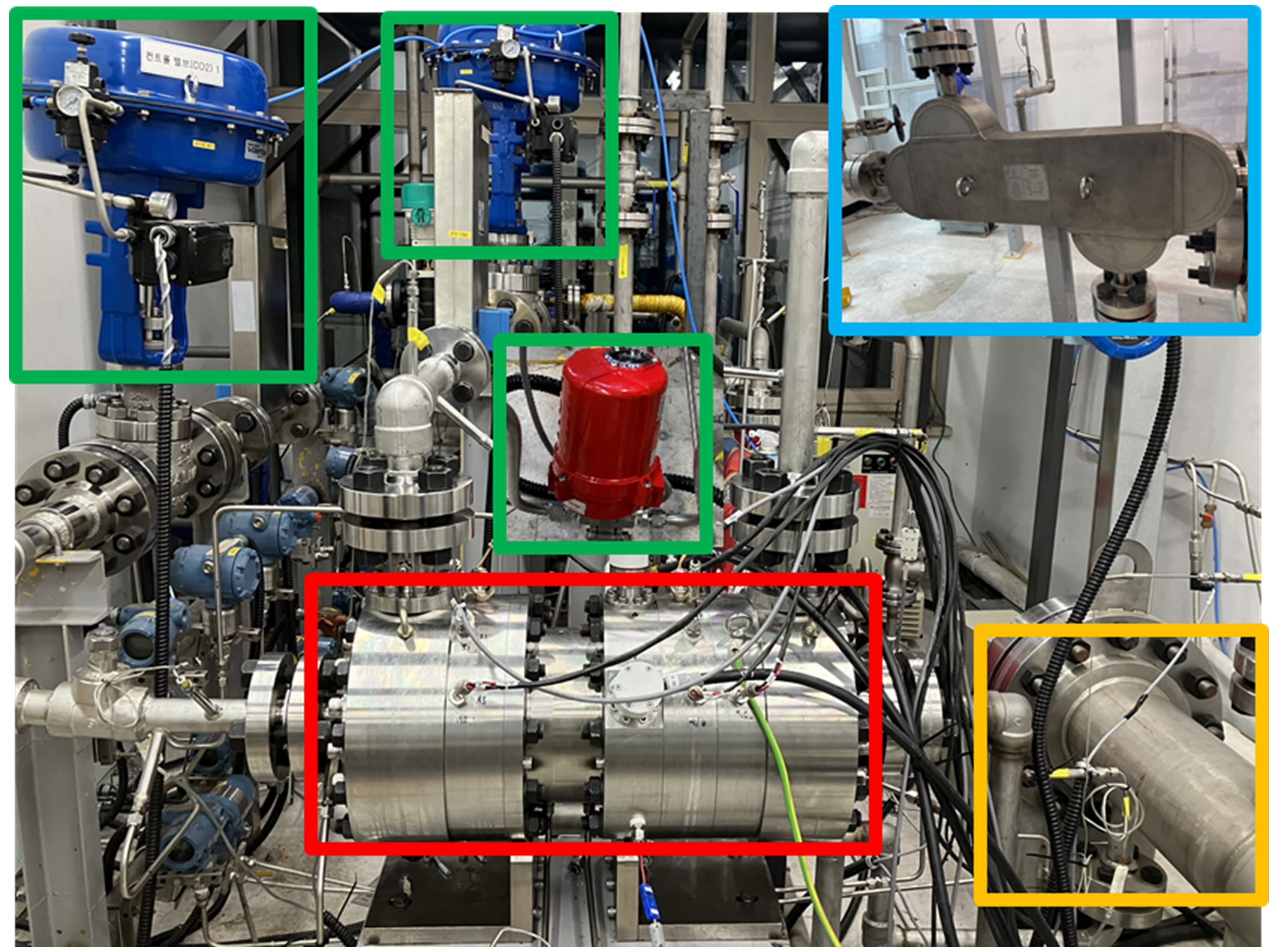
Best Efficiency?

No

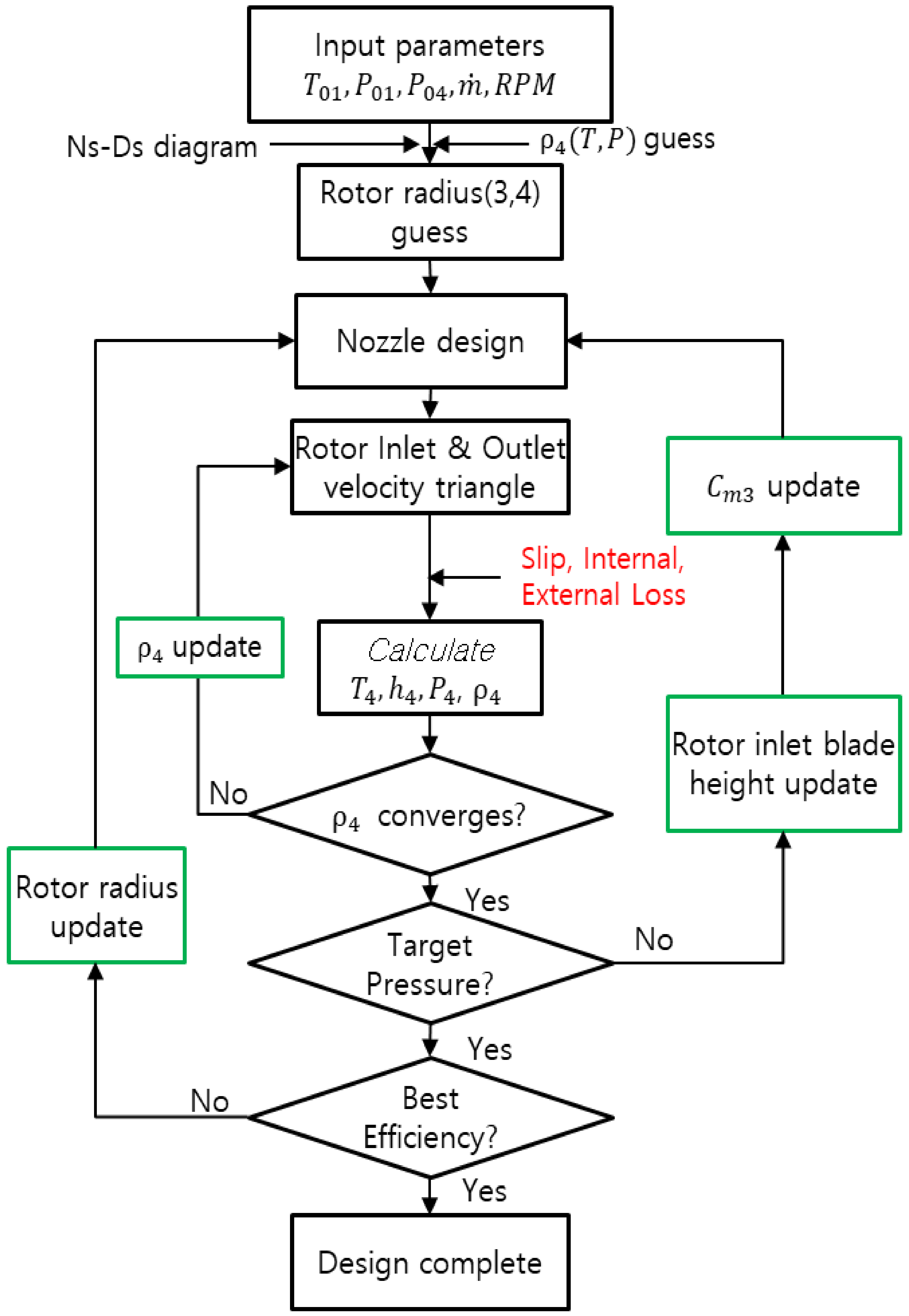
Yes

Design complete

Experimental data



The KAIST-TMD code and target systems



Input parameters
$T_{01}, P_{01}, P_{04}, \dot{m}, RPM$

Ns-Ds diagram

$\rho_4(T, P)$ guess

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Target Pressure?

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Best Efficiency?

No

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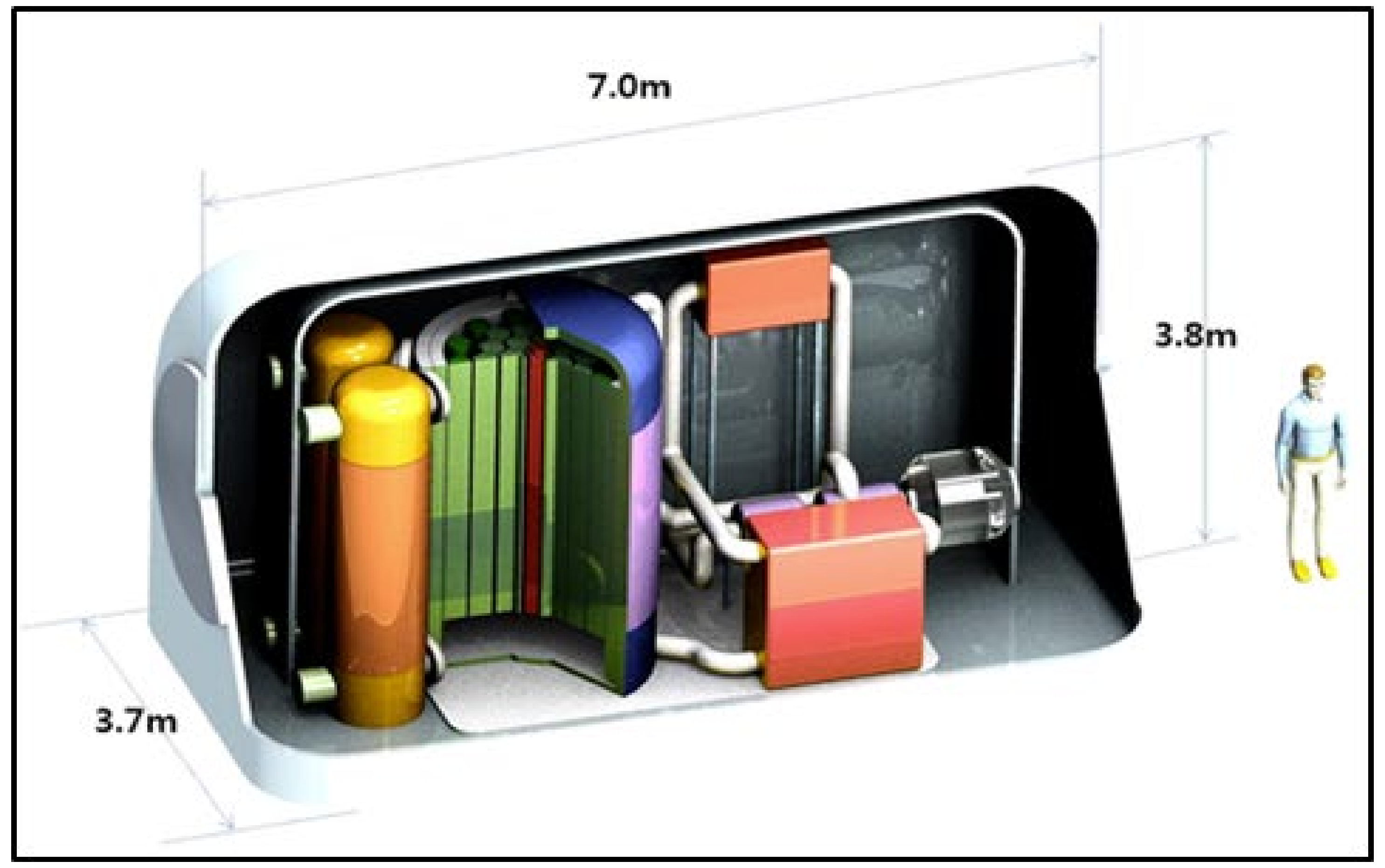
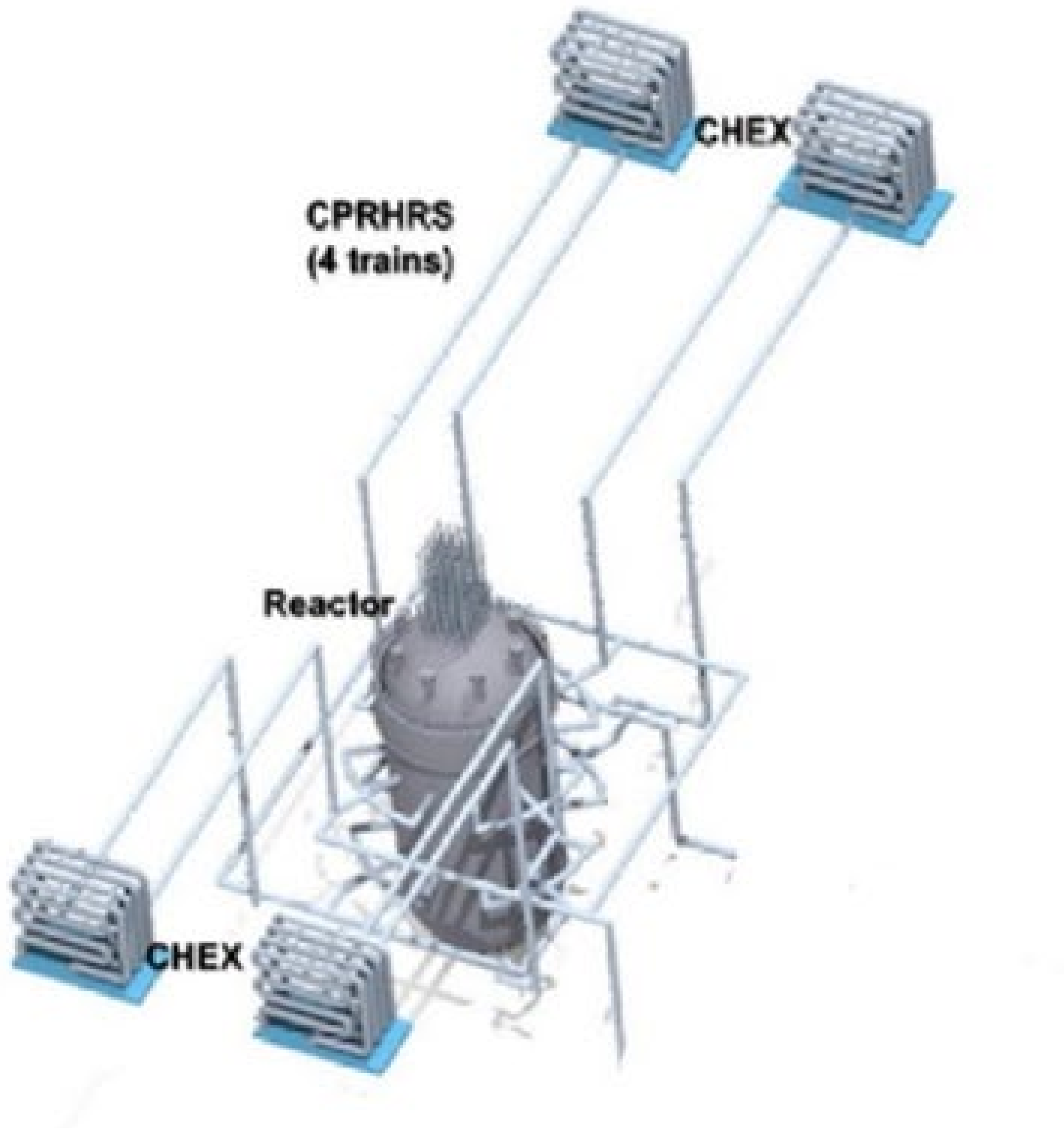
Design complete



KAIST-MMR



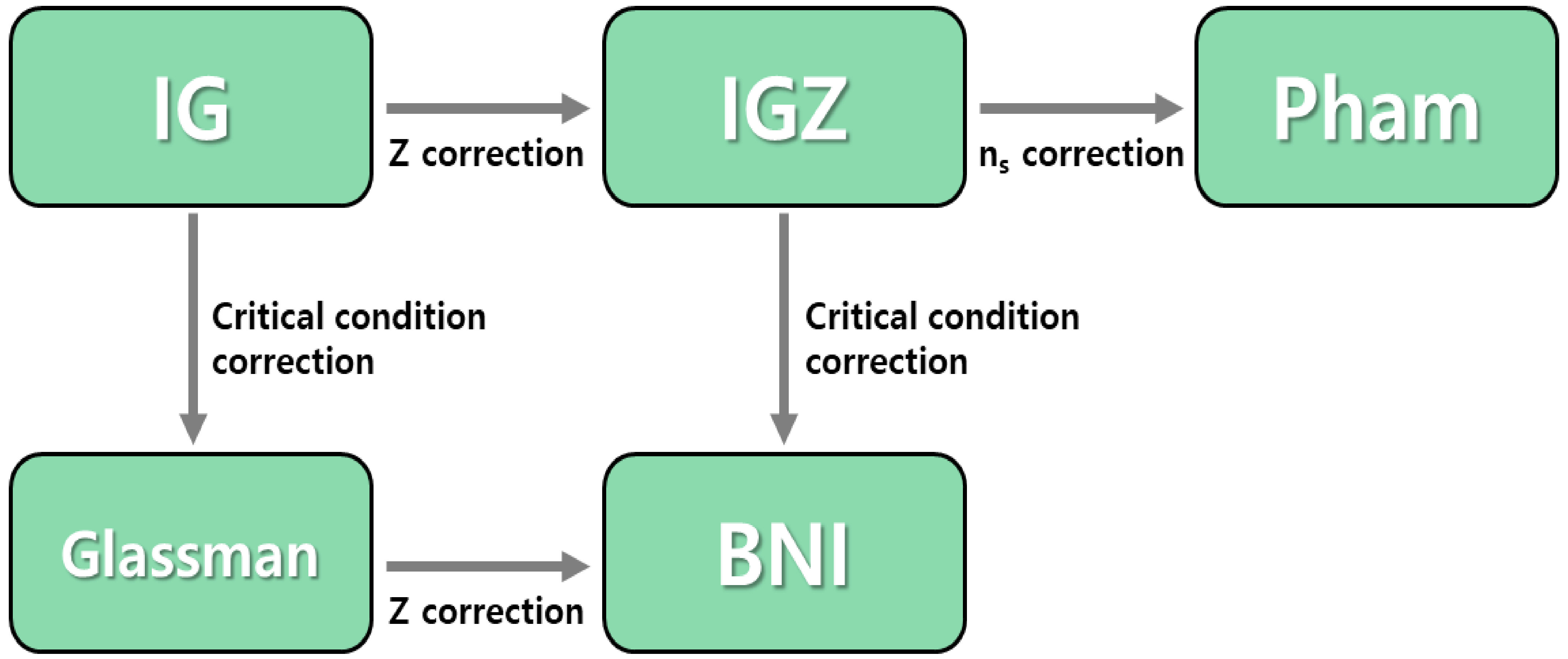
S-CO$_2$ PRHRS for SMR

- The KAIST-TMD code is designed for the design and analysis of S-CO$_2$ turbomachinery.
- It can design turbines and compressors to meet cycle requirements and predict behavior of off-design condition.
- In this study, two turbines are designed for the KAIST Micro-Modular reactor and the passive residual heat removal system of small nuclear reactor.

Five real gas models

Nomenclature	
Pressure	P
Temperature	T
Compressibility factor	Z
Specific heat ratio	γ
Specific enthalpy	h
Gas constant	R
Isentropic volume exponent	n_s
Entropy	s
Static property	_s
Stagnation property	_o
Critical property	_cr

	Flow parameter	Speed parameter	Head parameter	Pressure parameter	Efficiency parameter	
IG	$\frac{\dot{m}\sqrt{\gamma RT}}{\gamma P}$	$\frac{N}{\sqrt{\gamma RT}}$	$\frac{\Delta H}{\gamma RT}$	PR	η	
IGZ	$\frac{\dot{m}\sqrt{\gamma ZRT}}{\gamma P}$	$\frac{N}{\sqrt{\gamma ZRT}}$	$\frac{\Delta H}{\gamma ZRT}$			
Glassman	$\frac{\dot{m}\sqrt{\gamma RT_{cr}}}{\gamma P_{cr}}$	$\frac{N}{\sqrt{\gamma RT_{cr}}}$	$\frac{\Delta H}{\gamma RT_{cr}}$			$T_{cr} = T(\frac{2}{\gamma+1}),\ P_{cr} = P\left(\frac{2}{\gamma+1}\right)^{-\frac{\gamma}{\gamma+1}}$
BNI	$\frac{\dot{m}\sqrt{\gamma ZRT_{cr}}}{\gamma P_{cr}}$	$\frac{N}{\sqrt{\gamma ZRT_{cr}}}$	$\frac{\Delta H}{\gamma ZRT_{cr}}$			$T_{cr} = T(\frac{2}{\gamma+1}),\ P_{cr} = P\left(\frac{2}{\gamma+1}\right)^{-\frac{\gamma}{\gamma+1}}$
Pham	$\frac{\dot{m}\sqrt{n_s ZRT}}{n_s P}$	$\frac{N}{\sqrt{n_s ZRT}}$	$\frac{\Delta H}{n_s ZRT}$			$n_s = \gamma\frac{\rho}{P}(\frac{\partial P}{\partial \rho})_T$



IG

Z correction

IGZ

n_s correction

Pham

Critical condition correction

Critical condition correction

Glassman

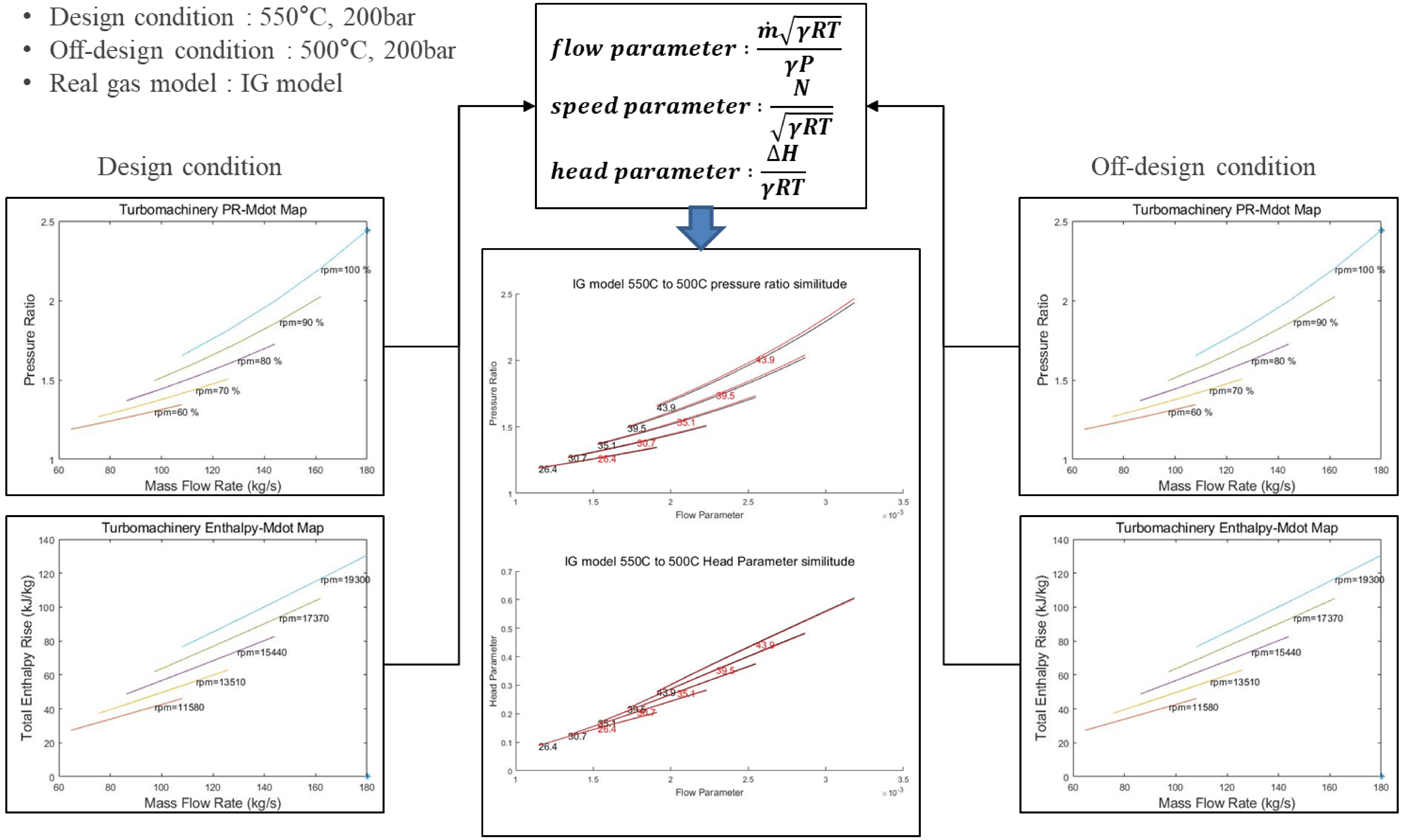
Z correction

BNI

- Real gas models convert measured data to non-dimensional parameters.
- There are five gas models, and previous studies have shown that the IGZ model performs best in compressors.
- Head parameter is more realistic for S-CO_2 turbomachinery similitude method than Pressure ratio.

Similitude method

- Example) KAIST-MMR turbine
- Design condition : 550°C, 200bar
- Off-design condition : 500°C, 200bar
- Real gas model : IG model



$$flow\ parameter: \frac{\dot{m}\sqrt{\gamma RT}}{\gamma P}$$

$$speed\ parameter: \frac{N}{\sqrt{\gamma RT}}$$

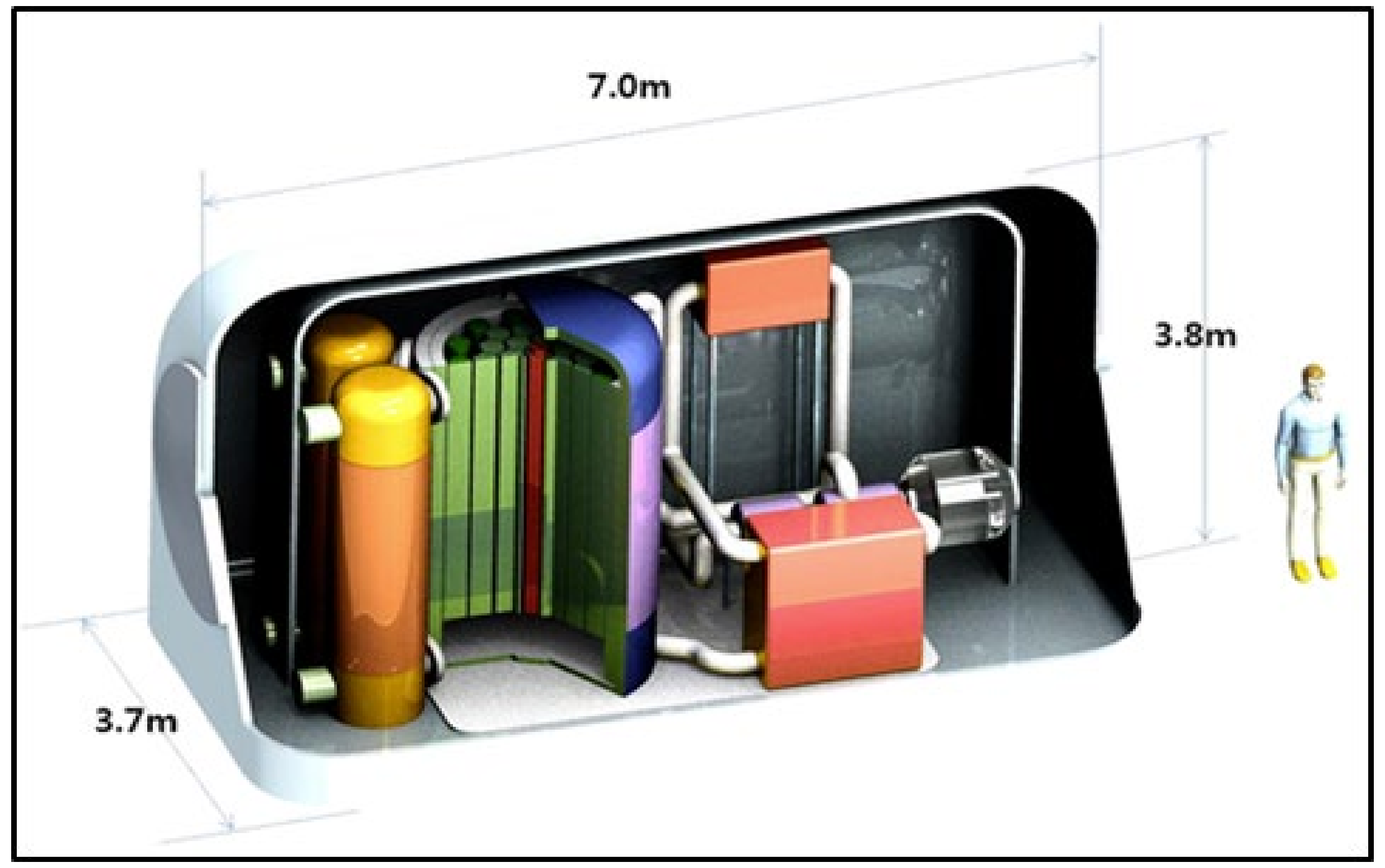
$$head\ parameter: \frac{\Delta H}{\gamma RT}$$

Design condition

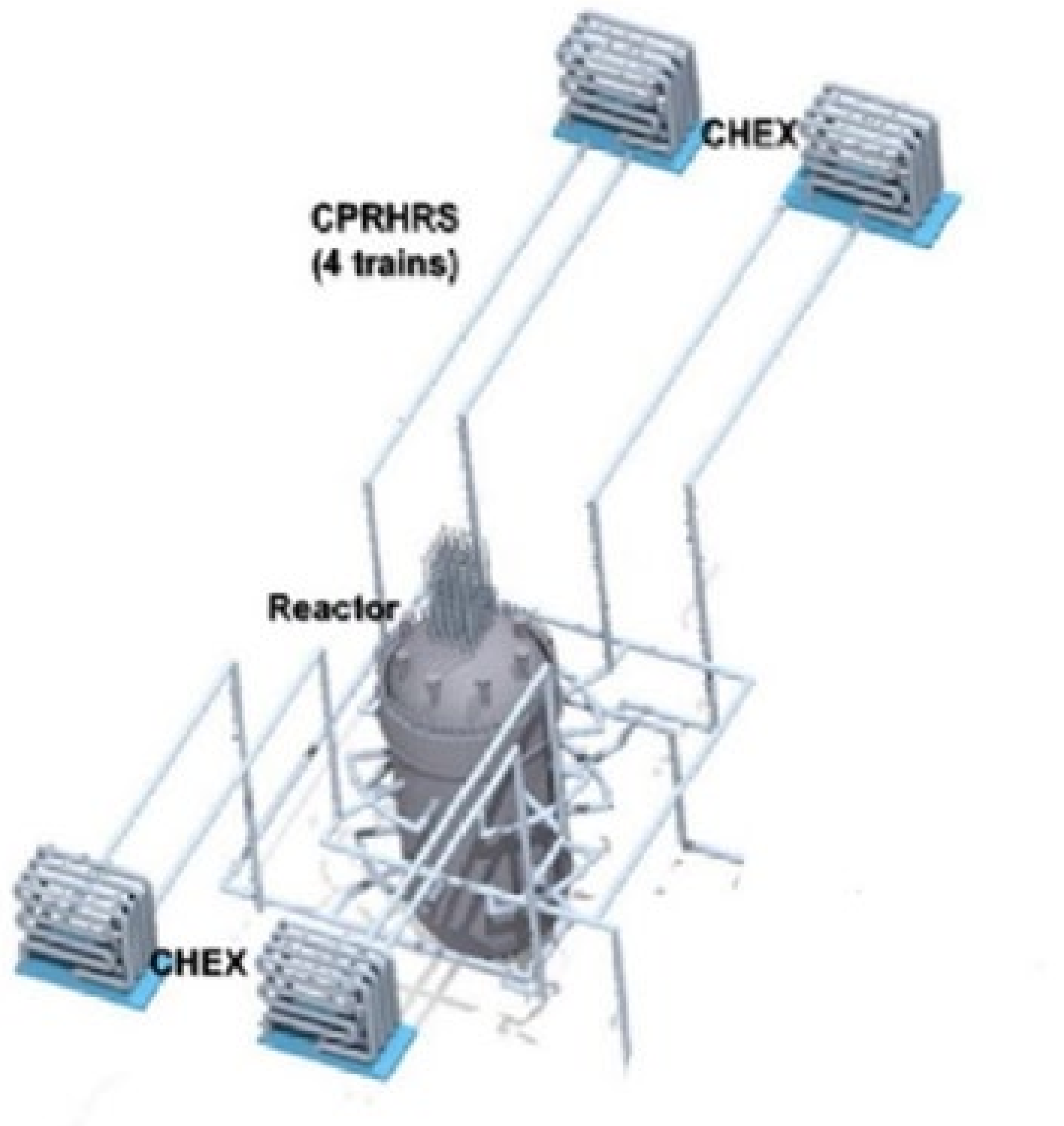
Off-design condition

1. Design a turbine with highest efficiency.
2. Plot design and off-design performance map.
3. Convert input and output parameters to non-dimensional parameters utilizing each real gas model.
4. Overlap the output parameters in one graph.
5. Compare the errors of five gas models for each RPM and mass flow rate.

Results of code evaluation



KAIST-MMR



S-CO_2 PRHRS for SMR

Pressure Ratio	IG	IGZ	Glass man	BNI	Pham	550°C, 200bar, Z:1.034
cond1	0.59	0.60	0.37	0.37	0.42	500°C, 200bar, Z:1.027
cond2	1.35	1.36	0.82	0.84	0.93	450°C, 200bar, Z:1.018
cond3	3.77	3.82	2.22	2.27	2.37	350°C, 200bar, Z:0.988
cond4	9.33	9.49	5.17	5.31	5.29	250°C, 200bar, Z:0.924
cond5	16.02	16.35	8.36	8.46	8.63	200°C, 200bar, Z:0.864
Average (%)	6.21	6.32	3.39	3.45	3.53	

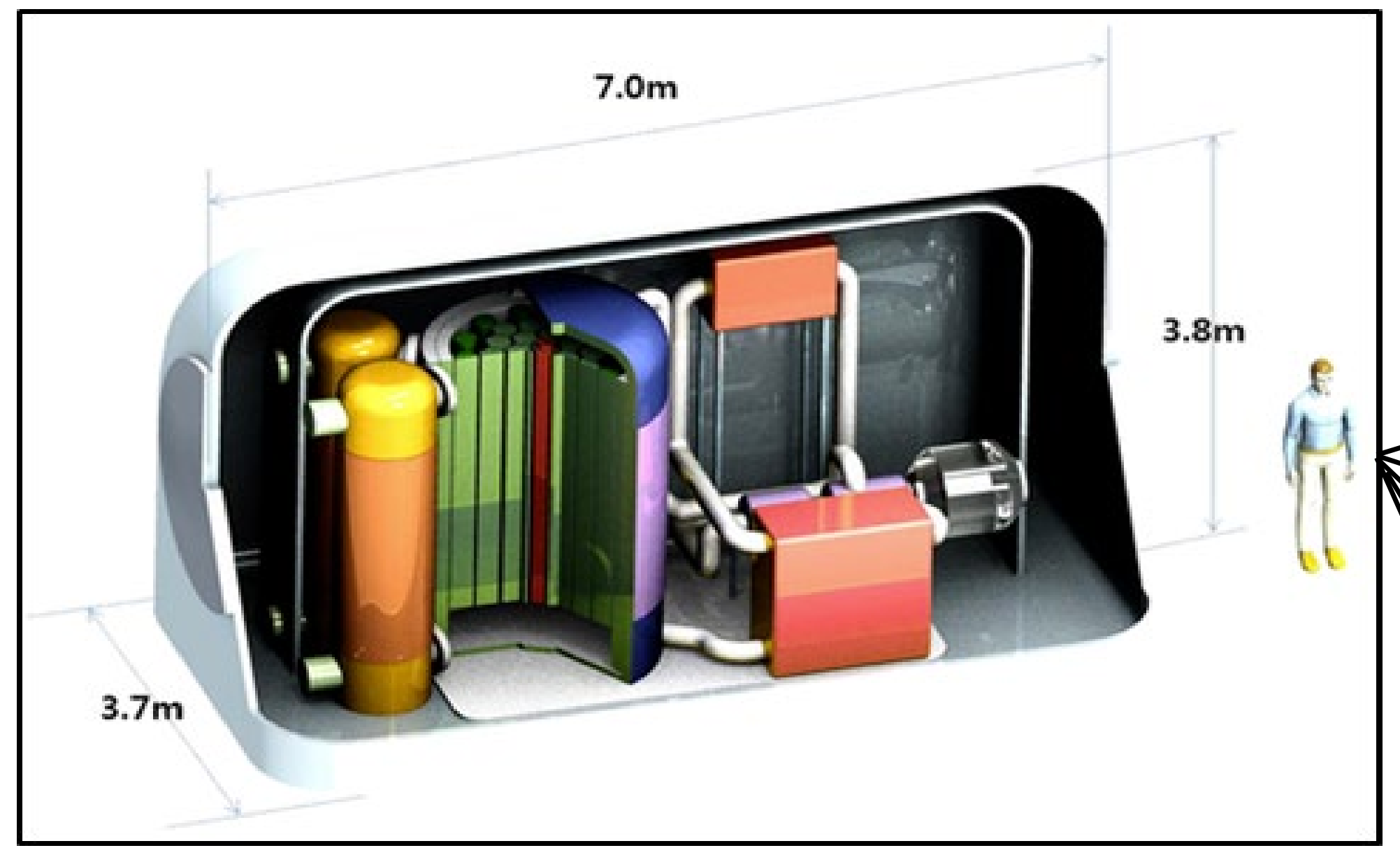
Head Parameter	IG	IGZ	Glass man	BNI	Pham	550°C, 200bar, Z:1.034
cond1	0.50	0.43	0.32	0.62	0.39	500°C, 200bar, Z:1.027
cond2	1.34	0.88	0.92	1.31	0.81	450°C, 200bar, Z:1.018
cond3	4.58	1.97	3.50	3.09	1.75	350°C, 200bar, Z:0.988
cond4	12.31	3.41	10.00	5.98	2.87	250°C, 200bar, Z:0.924
cond5	19.97	4.42	16.57	8.60	3.55	200°C, 200bar, Z:0.864
Average (%)	7.74	2.22	6.26	3.92	1.87	

Pressure Ratio	IG	IGZ	Glass man	BNI	Pham	200°C, 115bar, Z:0.906
cond1	0.63	0.65	0.32	0.34	0.06	130°C, 100bar, Z:0.825
cond2	1.51	1.55	0.73	0.77	0.11	90°C, 90bar, Z:0.748
cond3	3.07	3.16	1.34	1.41	0.16	60°C, 80bar, Z:0.663
Average (%)	1.74	1.78	0.80	0.84	0.11	

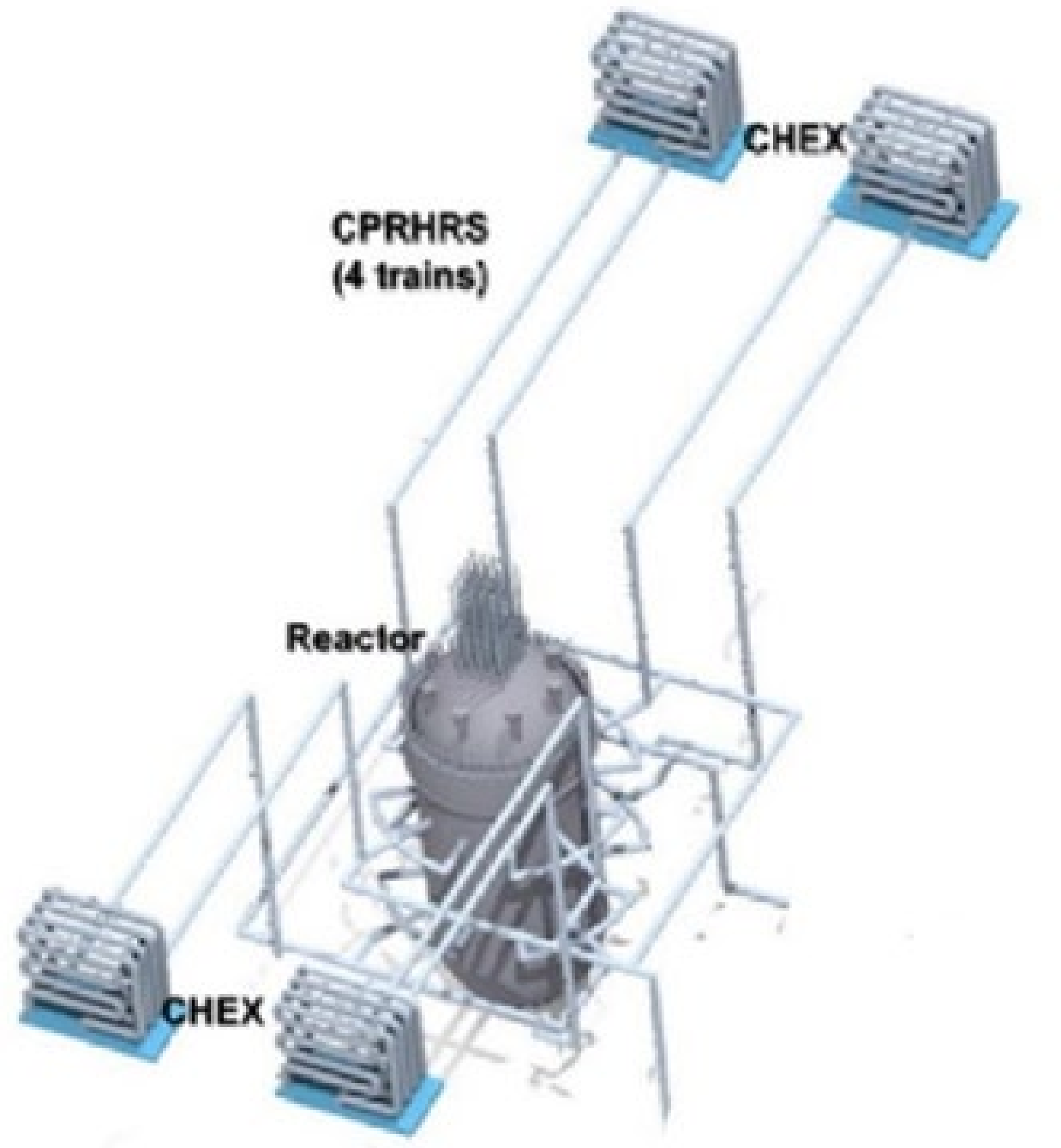
Head Parameter	IG	IGZ	Glass man	BNI	Pham	200°C, 115bar, Z:0.906
cond1	9.13	0.10	7.69	1.67	0.00	130°C, 100bar, Z:0.825
cond2	18.03	0.23	15.04	3.76	0.01	90°C, 90bar, Z:0.748
cond3	27.88	0.46	22.90	6.99	0.01	60°C, 80bar, Z:0.663
Average (%)	18.35	0.27	15.21	4.14	0.01	

Outline of code evaluation

2 Systems
2 Turbines



KAIST-MMR



S-CO_2 PRHRS for SMR

5 Gas models

- IG
- IGZ
- Glassman
- BNI
- Pham

2 Output parameters

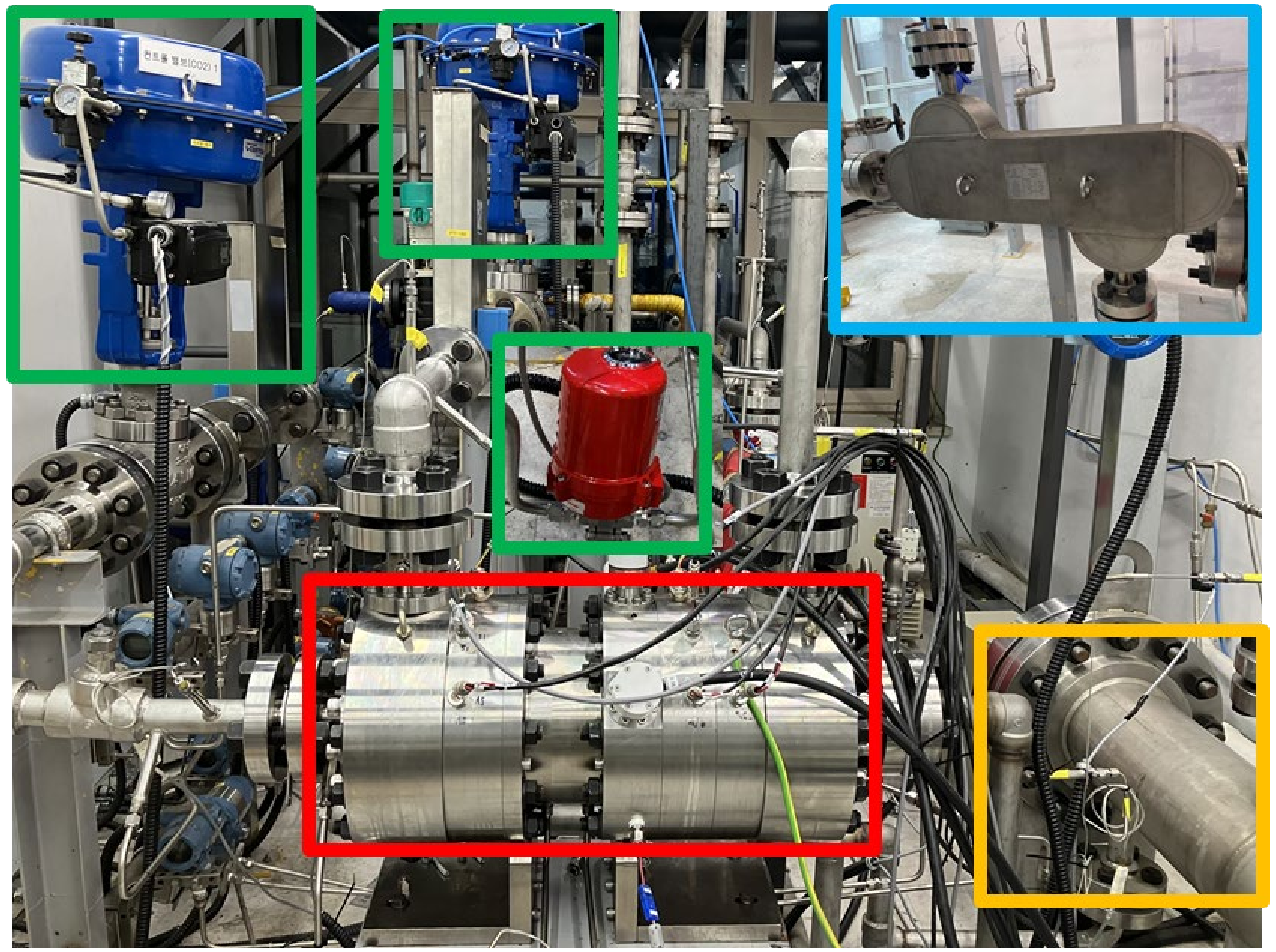
- Pressure ratio
- Head parameter

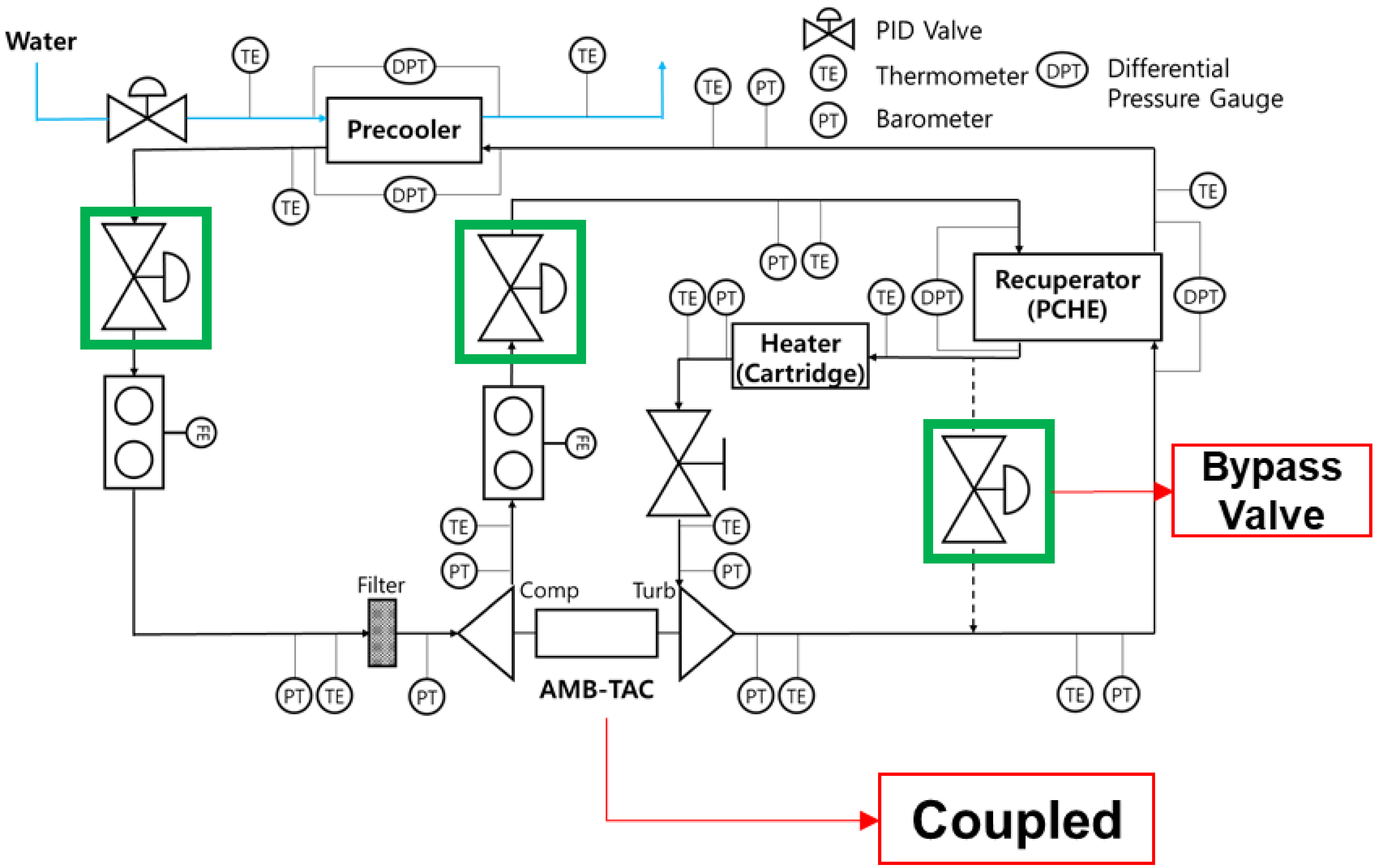
Select best gas model

Pham model

*Only by code evaluation,
Not by experimental results

Experimental Facility : ABC test loop





Bypass Valve

Coupled

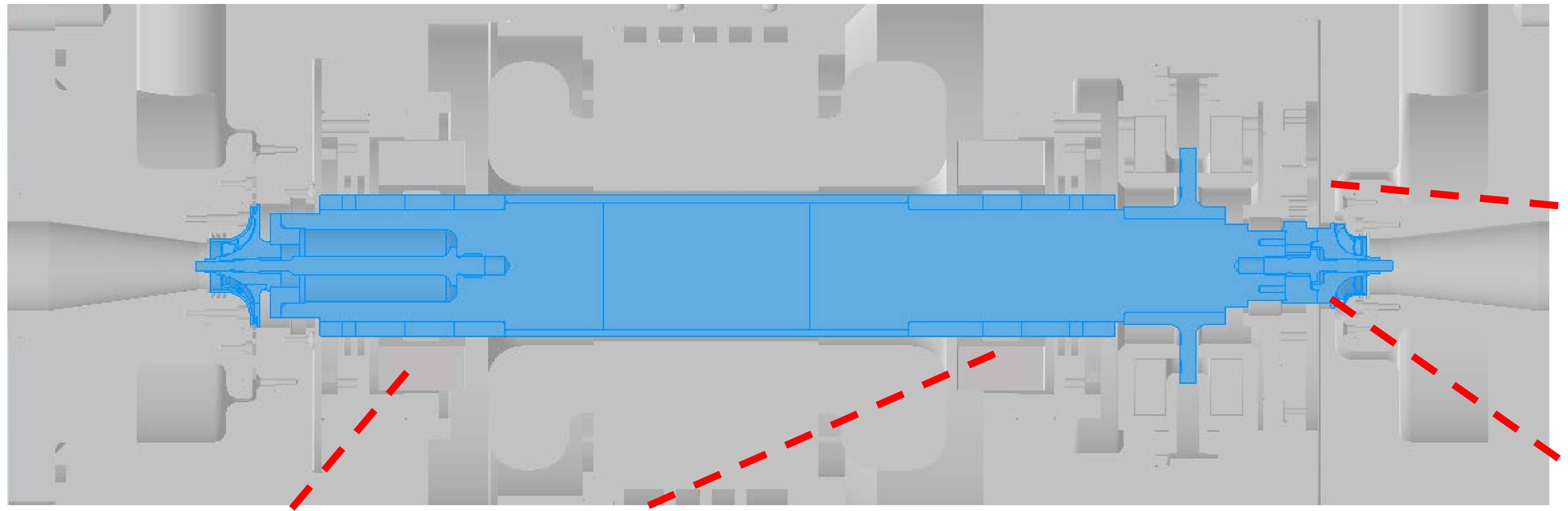
1 AMB-TAC (140kW)

3 Control Valves (2 Pneumatic, 1 Electric)

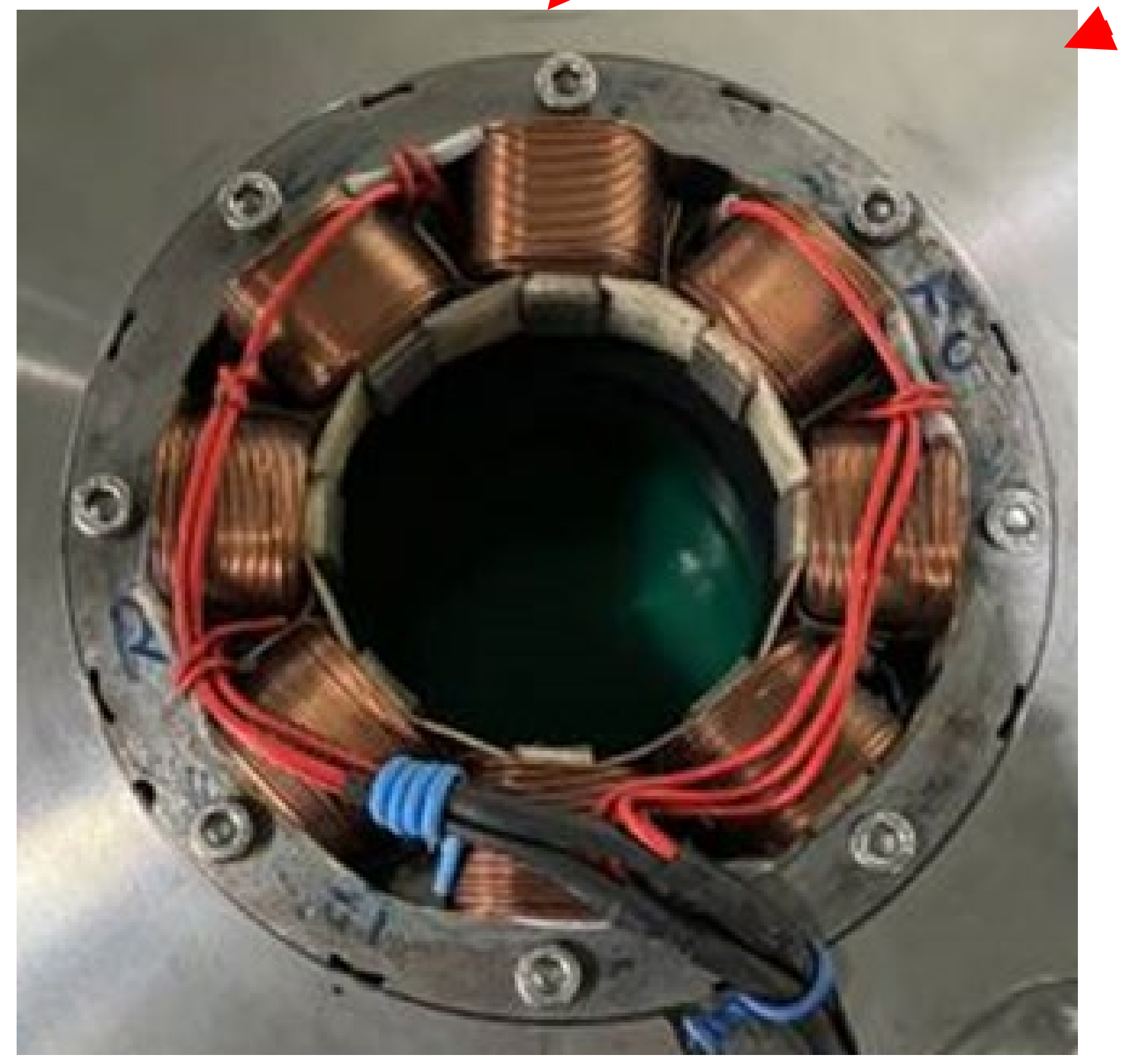
3 Heat Exchangers (1 Recuperator, 2 Pre-cooler)

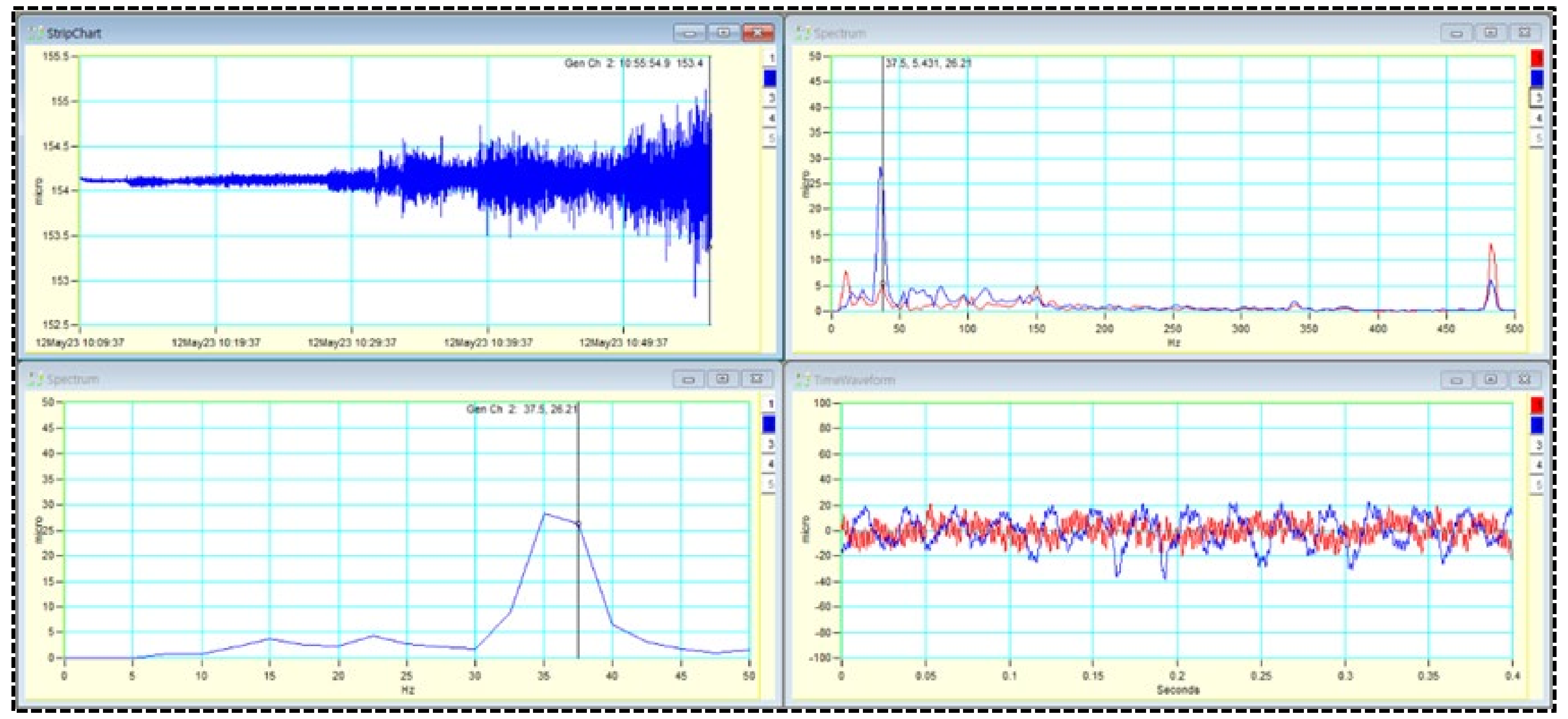
1 Heater (Cartridge, 50kW)

Experimental Facility : AMB-TAC



Turbine Nozzle	Dimension	Unit
Inlet blade angle	35	degree
Outlet blade angle	72	degree
Inlet radius	22.82	mm
Outlet radius	19.1	mm
Blade height	3.91	mm
Number of blades	20	ea



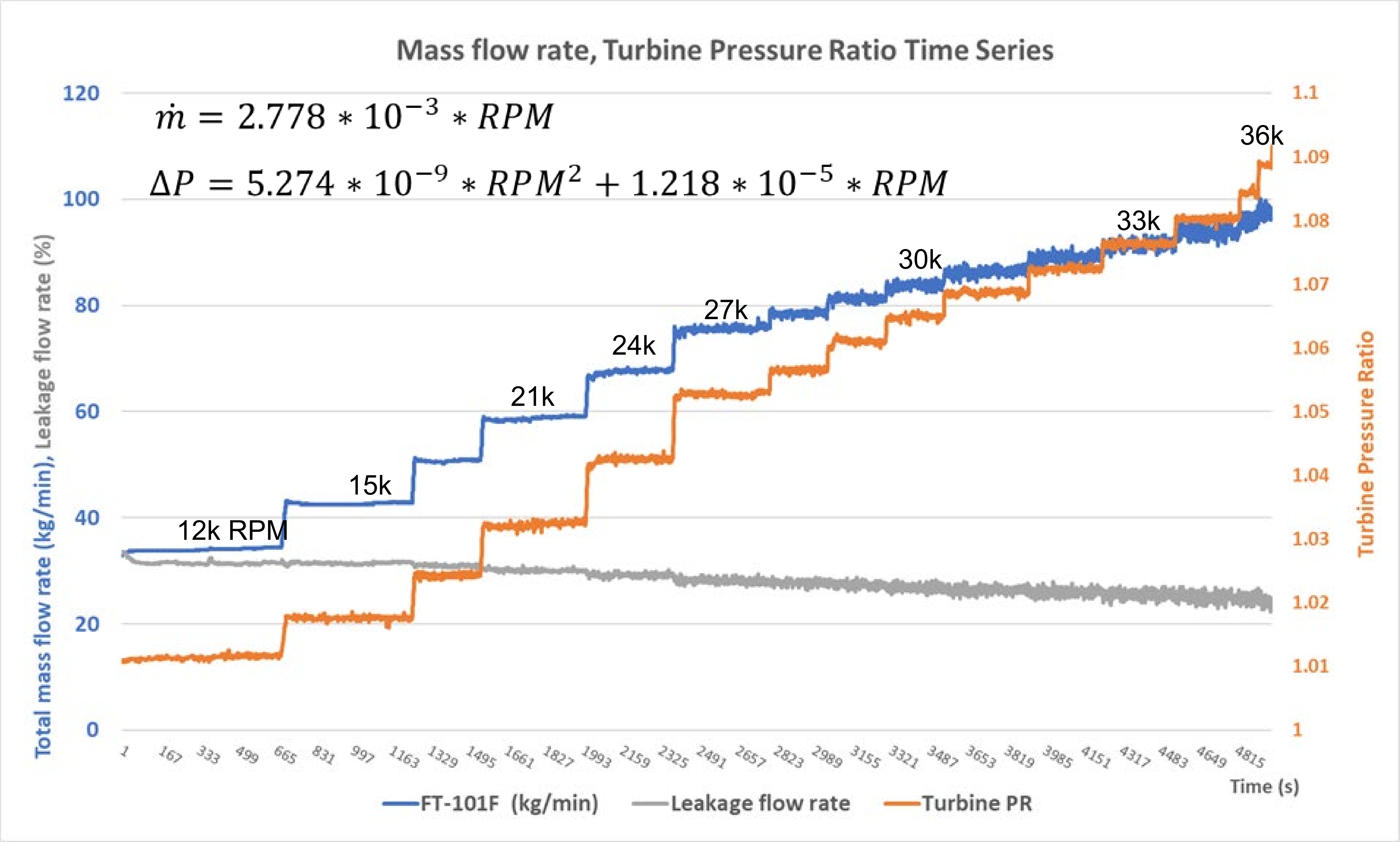


29,000 RPM, DN = 1,740,000

Turbine Rotor	Dimension	Unit
Inlet blade angle	0	degree
Outlet blade angle (hub)	40	degree
Outlet blade angle (tip)	46	degree
Inlet radius	17.75	mm
Outlet radius (hub)	19.1	mm
Outlet radius (tip)	4.19	mm
Inlet blade height	3.81	mm
Axial length	9.907	mm
Number of blades	10	ea

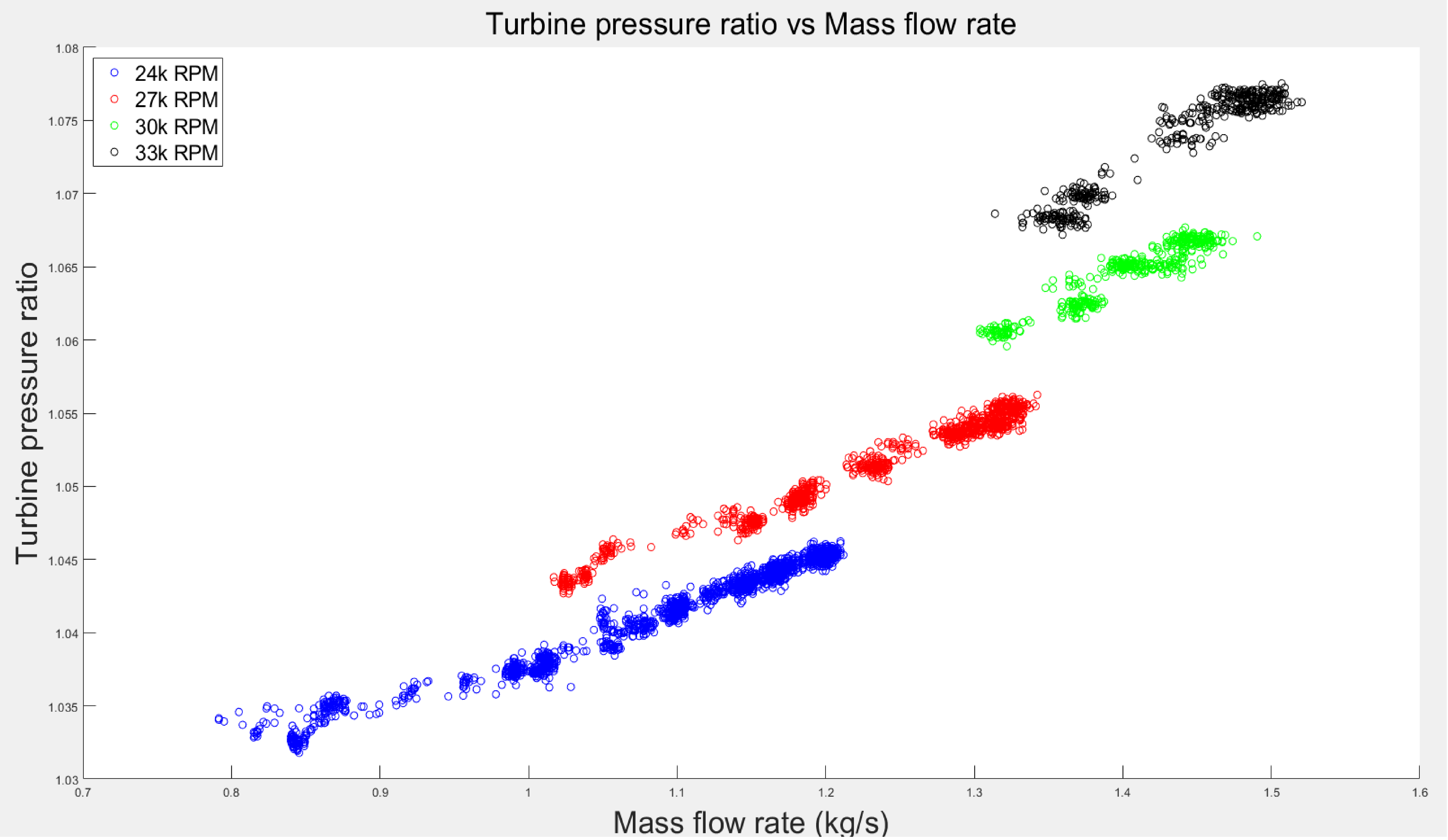
- The compressor and turbine in the ABC test loop are both radial turbomachinery and coupled by one shaft.
- The advantage of a TAC is that the work done by the turbine reduces the work consumed by the compressor.
- The rotor assembly of the TAC is supported by an active magnetic bearing because conventional oil bearings would cause the oil to dissolve in the S-CO_2.
- There are two radial bearings and one axial bearing supporting the TAC.

Measured data of turbine test


Mass flow rate, Turbine Pressure Ratio Time Series

$$\dot{m} = 2.778 * 10^{-3} * RPM$$

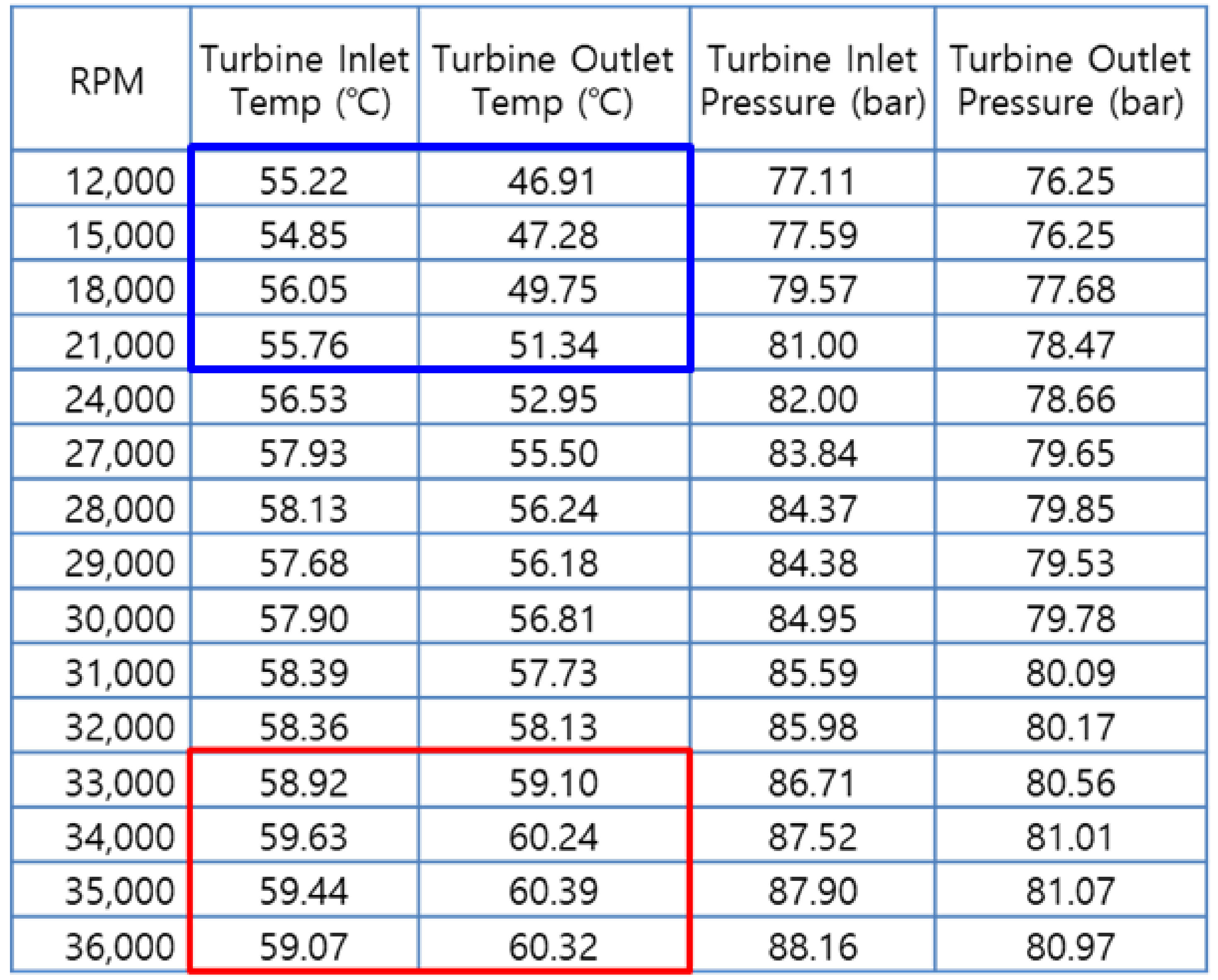
$$\Delta P = 5.274 * 10^{-9} * RPM^2 + 1.218 * 10^{-5} * RPM$$


Turbine pressure ratio vs Mass flow rate

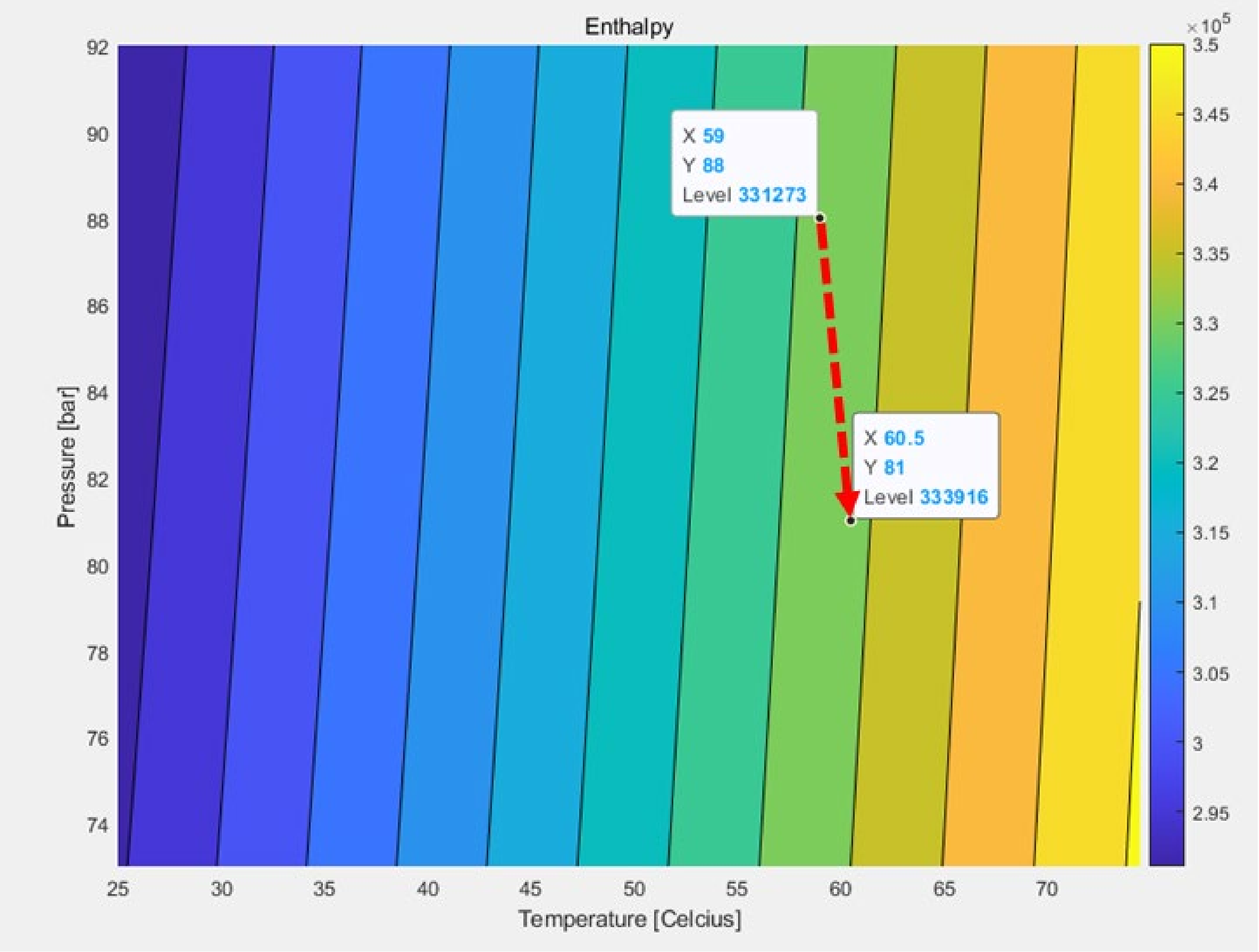
- As the RPM increases, mass flow rate increases linearly to RPM. Pressure difference also form a quadratic equation to RPM.
- As mass flow rate decreases in a fixed RPM by adjusting control valves, pressure ratio also decreases.
- Due to the characteristics of magnetic bearings, the vibration of the compressor caused fluctuations in the mass flow rate.
- The higher the RPM, the more the compressor and shaft vibrate, so at 27,000 RPM and above, the mass flow rate fluctuation is severe.

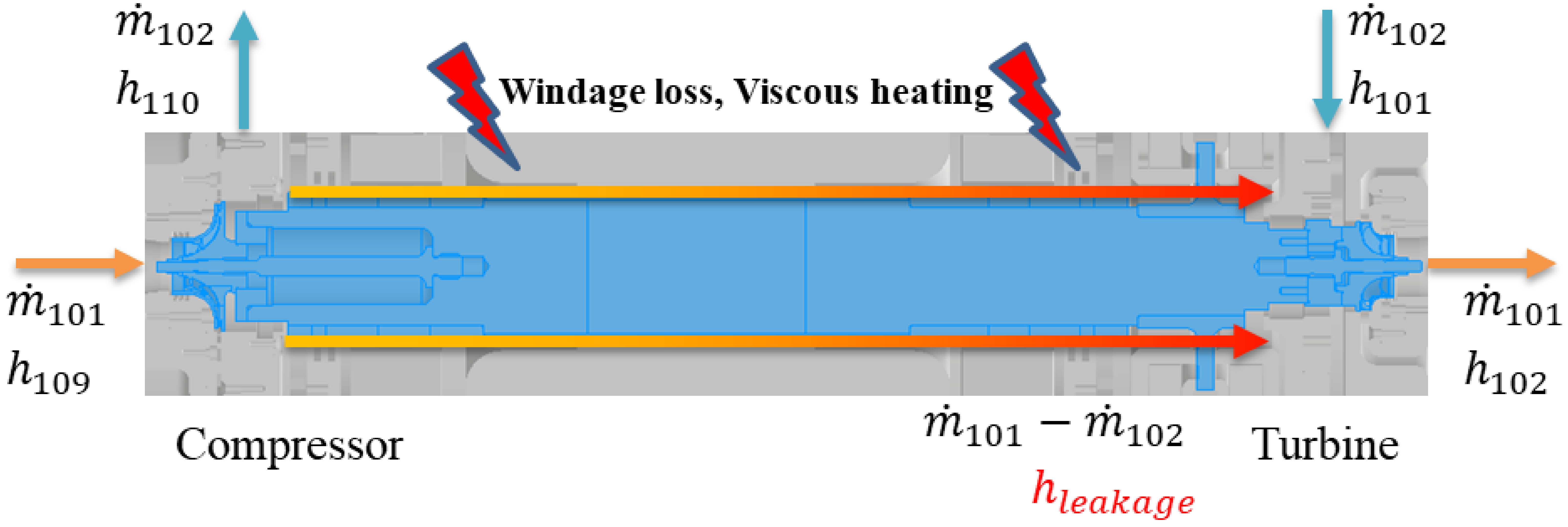
Post-processing of measured data

- It is shown in the table that for high RPM cases, the turbine inlet temperature becomes lower than the turbine outlet temperature.
- Because the secondary flow coming from compressor outlet to turbine inlet experiences a large viscous heating.
- Thus, the turbine inlet temperature is not correctly representing the actual turbine inlet temperature since the measurement is performed before the secondary flow merges with the main flow.
- It isn't possible to measure thermal properties in small clearance secondary flow path.
- Therefore, the enthalpy from the secondary flow had to be corrected with the newly developed windage loss model.

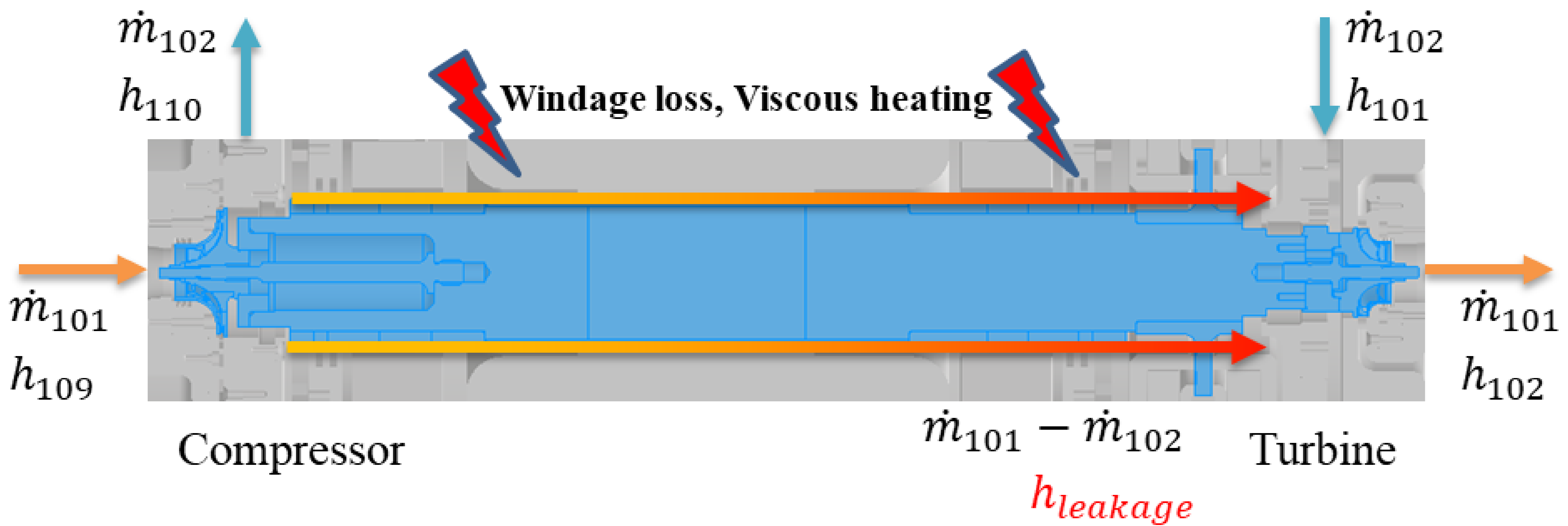


RPM	Turbine Inlet Temp (℃)	Turbine Outlet Temp (℃)	Turbine Inlet Pressure (bar)	Turbine Outlet Pressure (bar)
12,000	55.22	46.91	77.11	76.25
15,000	54.85	47.28	77.59	76.25
18,000	56.05	49.75	79.57	77.68
21,000	55.76	51.34	81.00	78.47
24,000	56.53	52.95	82.00	78.66
27,000	57.93	55.50	83.84	79.65
28,000	58.13	56.24	84.37	79.85
29,000	57.68	56.18	84.38	79.53
30,000	57.90	56.81	84.95	79.78
31,000	58.39	57.73	85.59	80.09
32,000	58.36	58.13	85.98	80.17
33,000	58.92	59.10	86.71	80.56
34,000	59.63	60.24	87.52	81.01
35,000	59.44	60.39	87.90	81.07
36,000	59.07	60.32	88.16	80.97





Post-processing of measured data

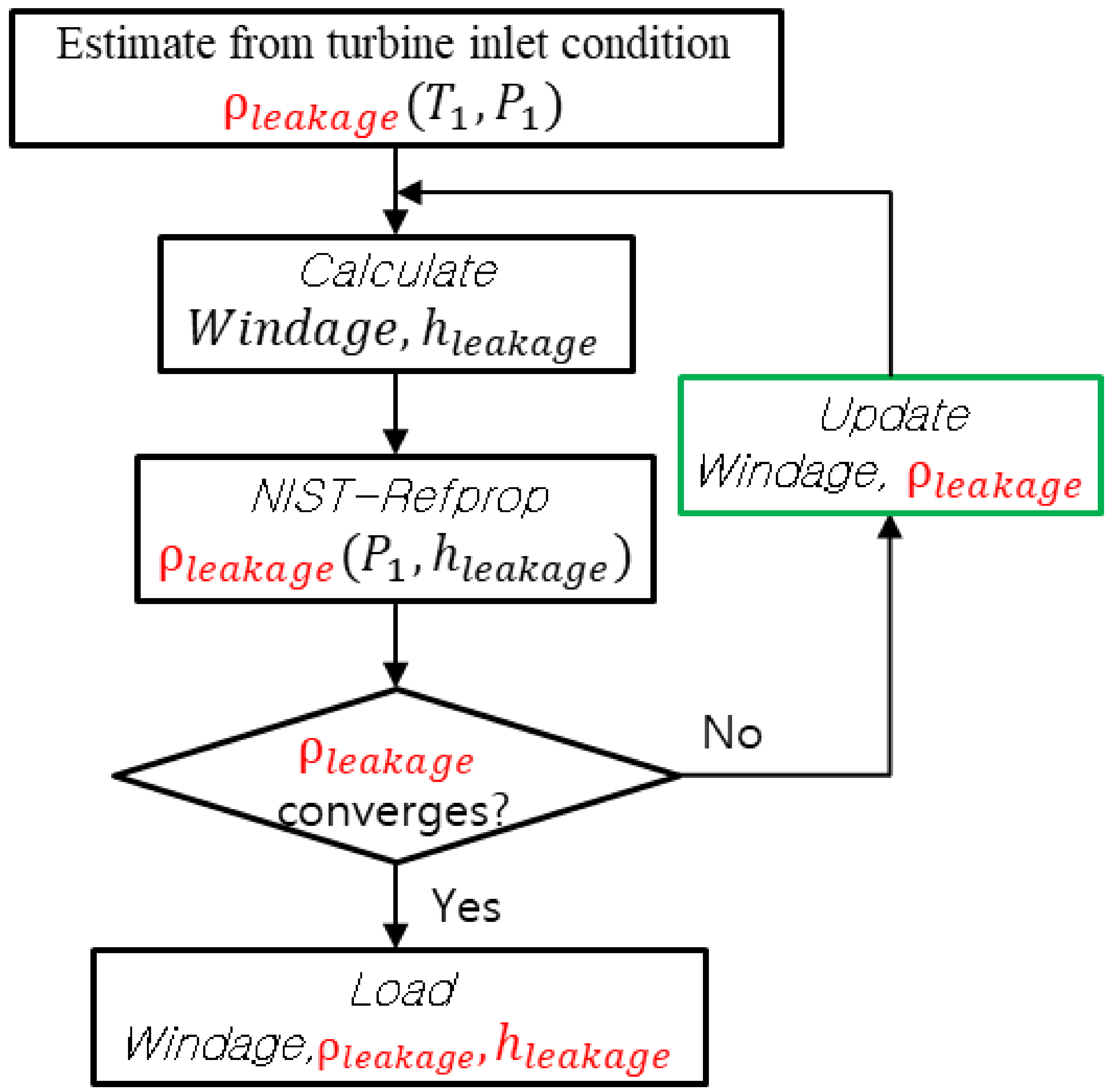


$$W_{windage} = \pi * C_{f,corr} * \frac{\rho_{inlet} + \rho_{outlet}}{2} * r_{shaft}^{4} * \left(\frac{RPM * 2\pi}{60}\right)^{3} * L_{shaft}$$

$$C_{f,corr} = \frac{1.8}{Ta_{crit}^{0.4} * C_r^{-1.25}} * \frac{(1 + c_r)^2}{(1 + c_r)^2 - 1} * Ta_{inlet}^{-0.1} * \left(\frac{\gamma_{inlet}}{\gamma_{air}}\right)^{-0.4051}$$

$$Ta_{crit}^{0.4} = \left(\frac{RPM * 2\pi}{60}\right)^{2} * r_{rotor} * \frac{c^3}{v_{inlet}^2}$$

$$W_{windage} - Q_{cooling} = (\dot{m}_{101} - \dot{m}_{102})(h_{leakage} - h_{110})$$



Estimate from turbine inlet condition $\rho_{leakage}(T_1, P_1)$

Calculate $Windage, h_{leakage}$

NIST–Refprop $\rho_{leakage}(P_1, h_{leakage})$

$\rho_{leakage}$ converges?

No

Update Windage, $\rho_{leakage}$

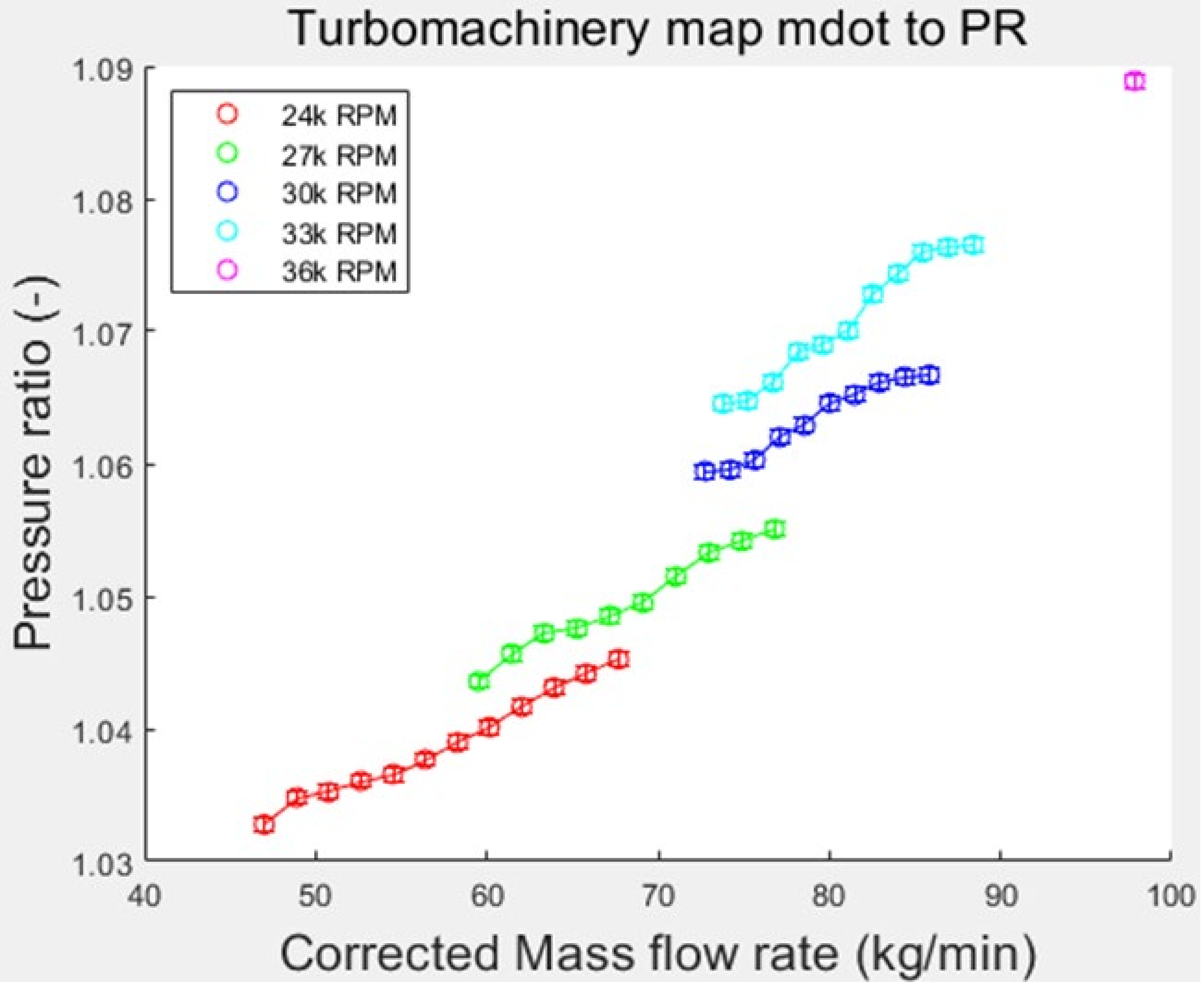
Yes

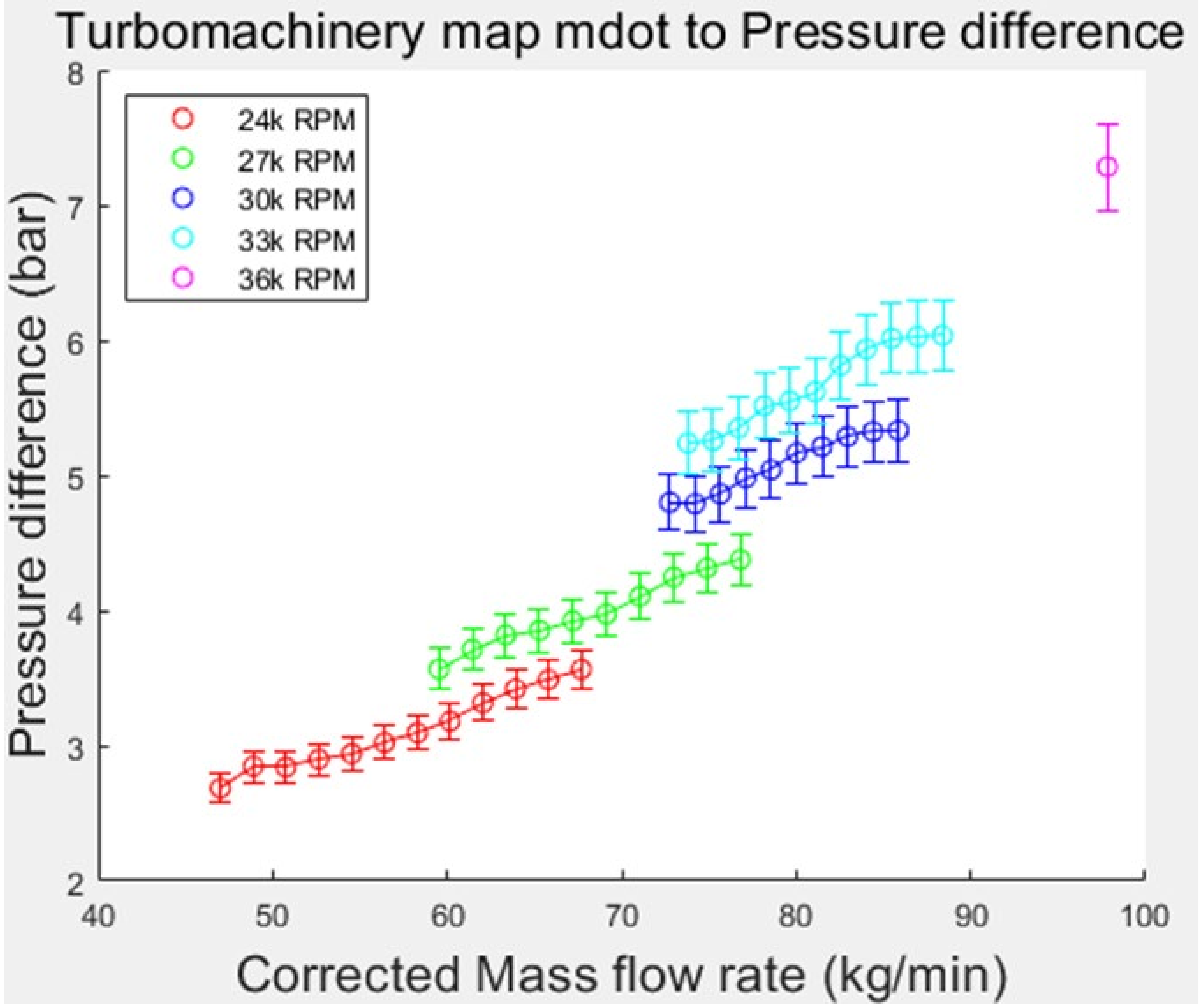
Load Windage, $\rho_{leakage}$, $h_{leakage}$

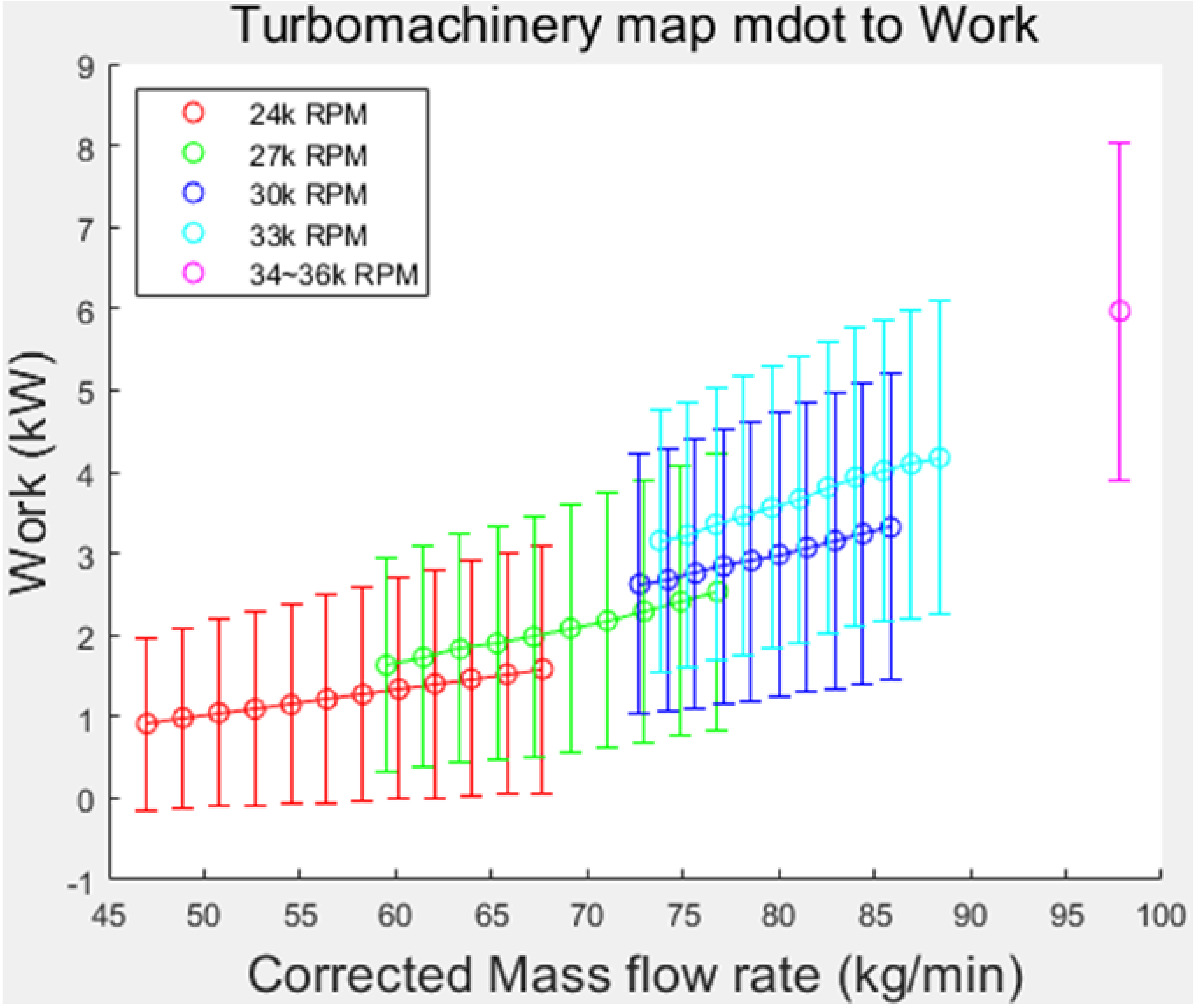
RPM	Turbine Inlet Temp (℃)	Corrected Turbine inlet Temp (℃)	Turbine Outlet Temp (℃)	Turbine Inlet Pressure (bar)	Turbine Outlet Pressure (bar)
12,000	55.22	47.71	46.91	77.11	76.25
15,000	54.85	48.77	47.28	77.59	76.25
18,000	56.05	51.51	49.75	79.57	77.68
21,000	55.76	53.91	51.34	81.00	78.47
24,000	56.53	56.50	52.95	82.00	78.66
27,000	57.93	60.17	55.50	83.84	79.65
28,000	58.13	60.57	56.24	84.37	79.85
29,000	57.68	60.68	56.18	84.38	79.53
30,000	57.90	62.11	56.81	84.95	79.78
31,000	58.39	63.19	57.73	85.59	80.09
32,000	58.36	64.19	58.13	85.98	80.17
33,000	58.92	65.86	59.10	86.71	80.56
34,000	59.63	67.40	60.24	87.52	81.01
35,000	59.44	68.02	60.39	87.90	81.07
36,000	59.07	68.52	60.32	88.16	80.97

- Turbine work is calculated by the difference between inlet enthalpy and outlet enthalpy.
- Inlet enthalpy is calculated by mass flow rate averaged mixing of the leakage flow enthalpy and main stream enthalpy.
- Leakage flow enthalpy is calculated by adding the compressor outlet enthalpy and viscous heating power (i.e. windage loss).
- Due to the leakage outlet state (pressure or density) is unknown, iterating calculation was done.

Performance map of radial turbine


Turbomachinery map mdot to PR


Turbomachinery map mdot to Pressure difference


Turbomachinery map mdot to Work

- Using the similitude analysis described earlier, the measured mass flow rate can be converted to the corrected mass flow rate.
- The mass flow rate was converted to corrected mass flow rate utilizing the Pham model.
- Since the enthalpy of S-CO_2 changes significantly with a small temperature difference, it can be seen that the uncertainties in the work of the turbine is larger than the uncertainties in the pressure ratio or pressure difference.

Future works

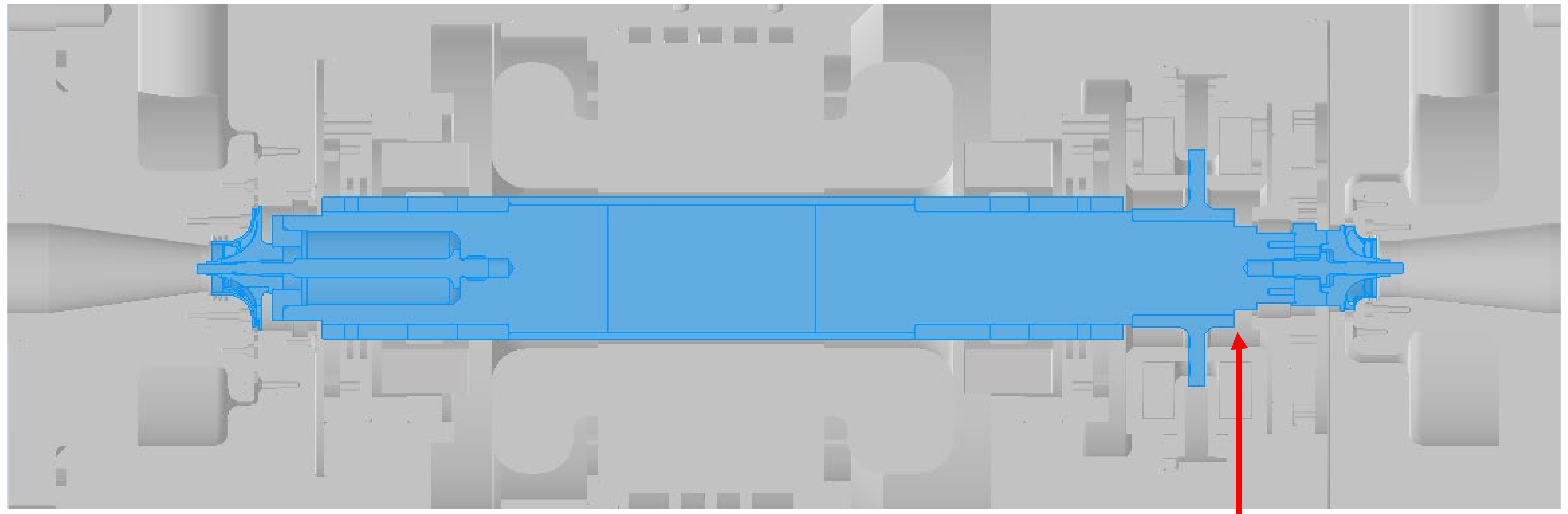
✓ The approach used to obtain the performance map of the radial turbine is indirect way.

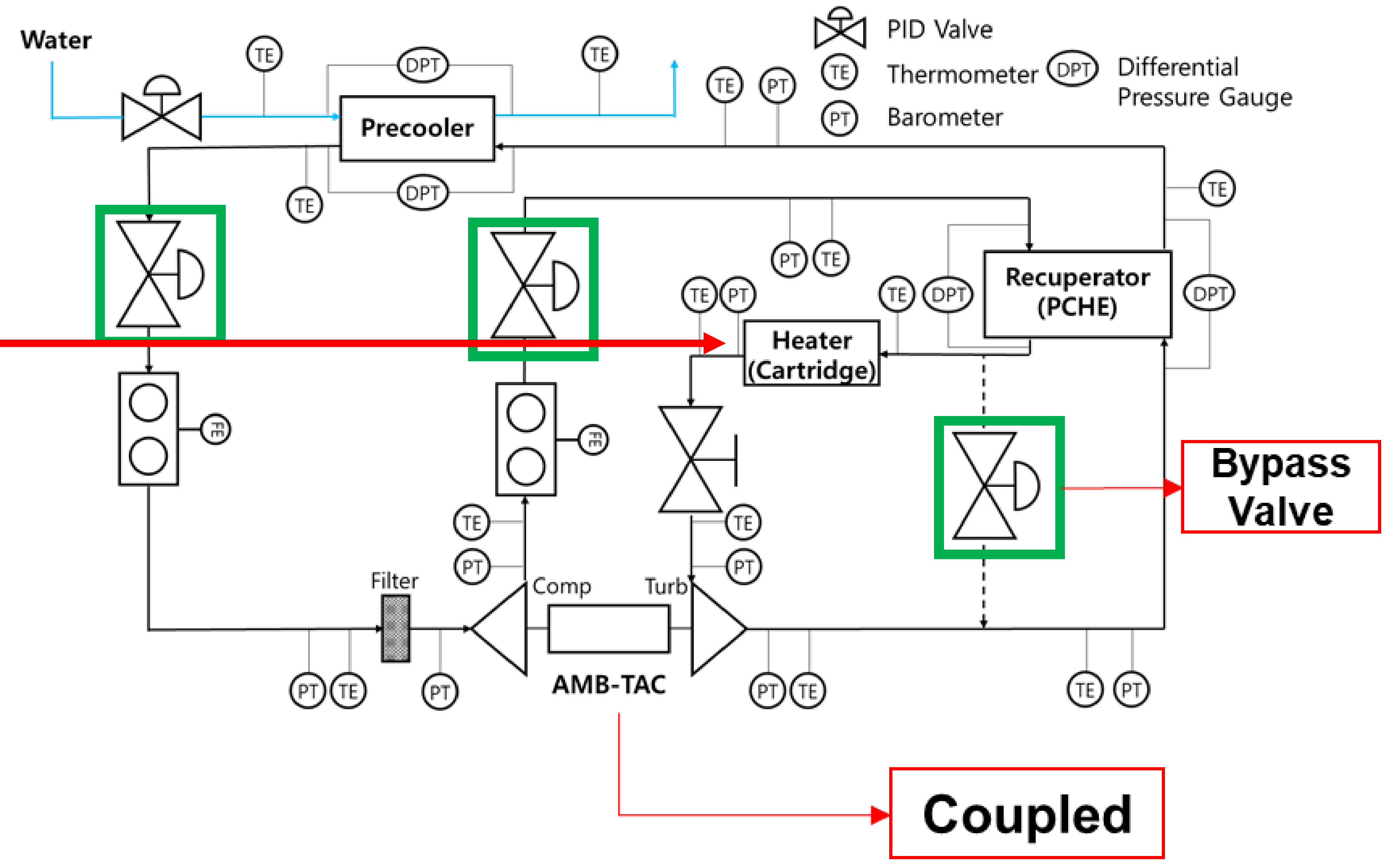
✓ By reducing the leakage flow rate, it should be possible to improve the accuracy of the turbine power, although in an indirect way.

✓ We are also looking for a way to add a thermometer to the small clearance inside the TAC shaft and stator casing.

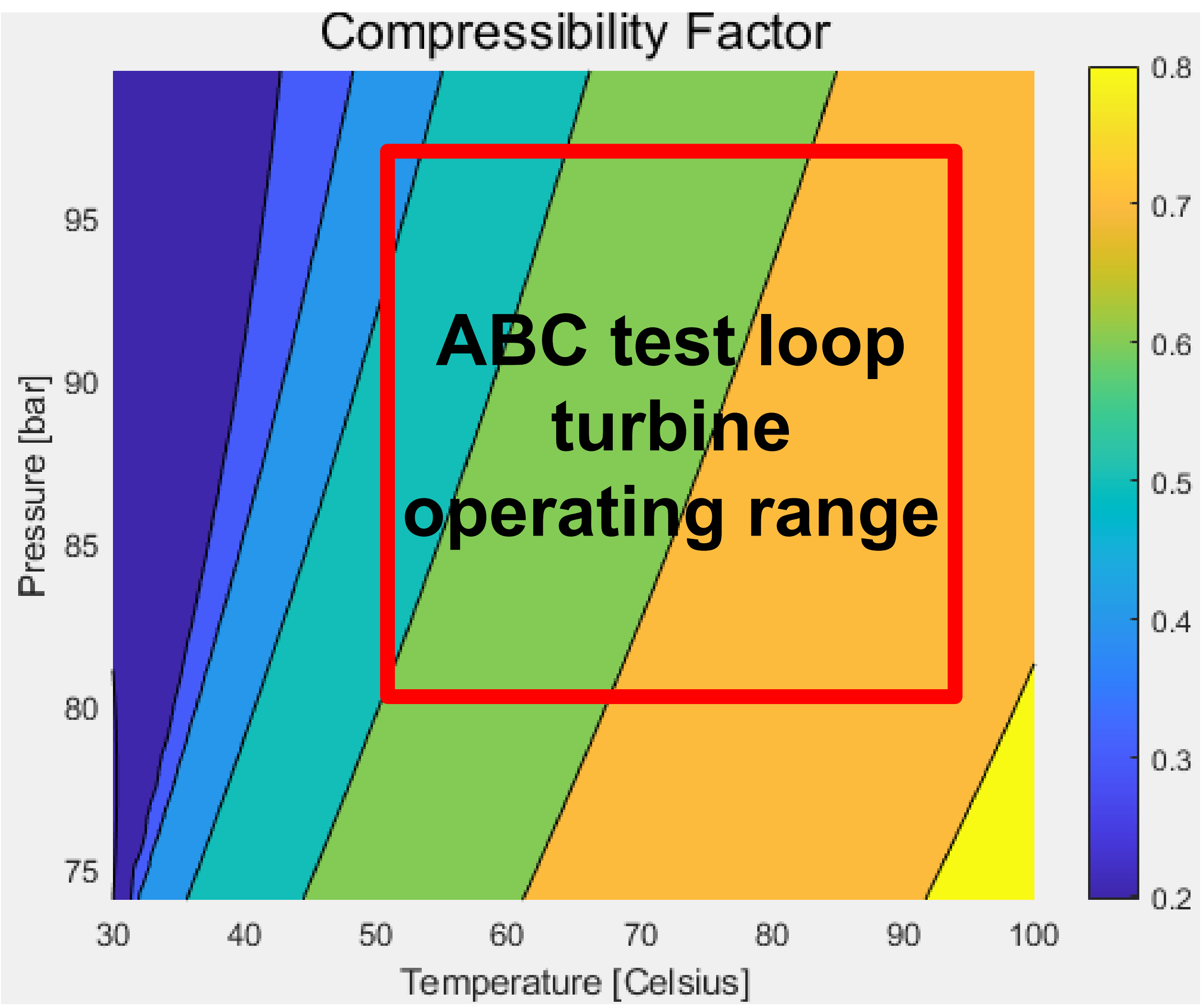
✓ The heat generated by the viscous heating was greater than the capacity of the heater, so a wide temperature range for the turbine was not possible.

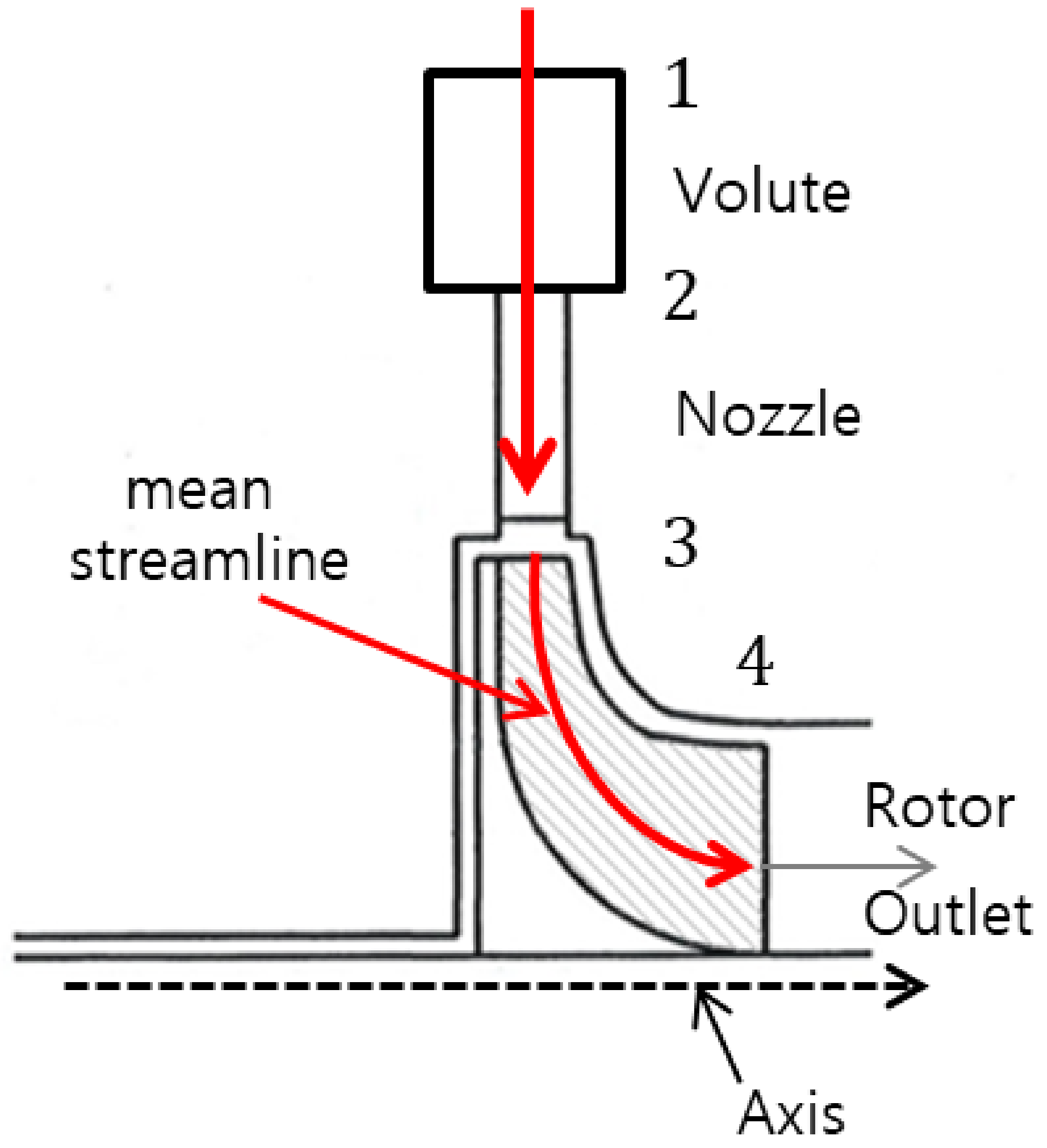
✓ So, we aim to increase the heater power to raise the turbine inlet temperature and the effect of the real gas effect and best real gas model will be found out experimentally.

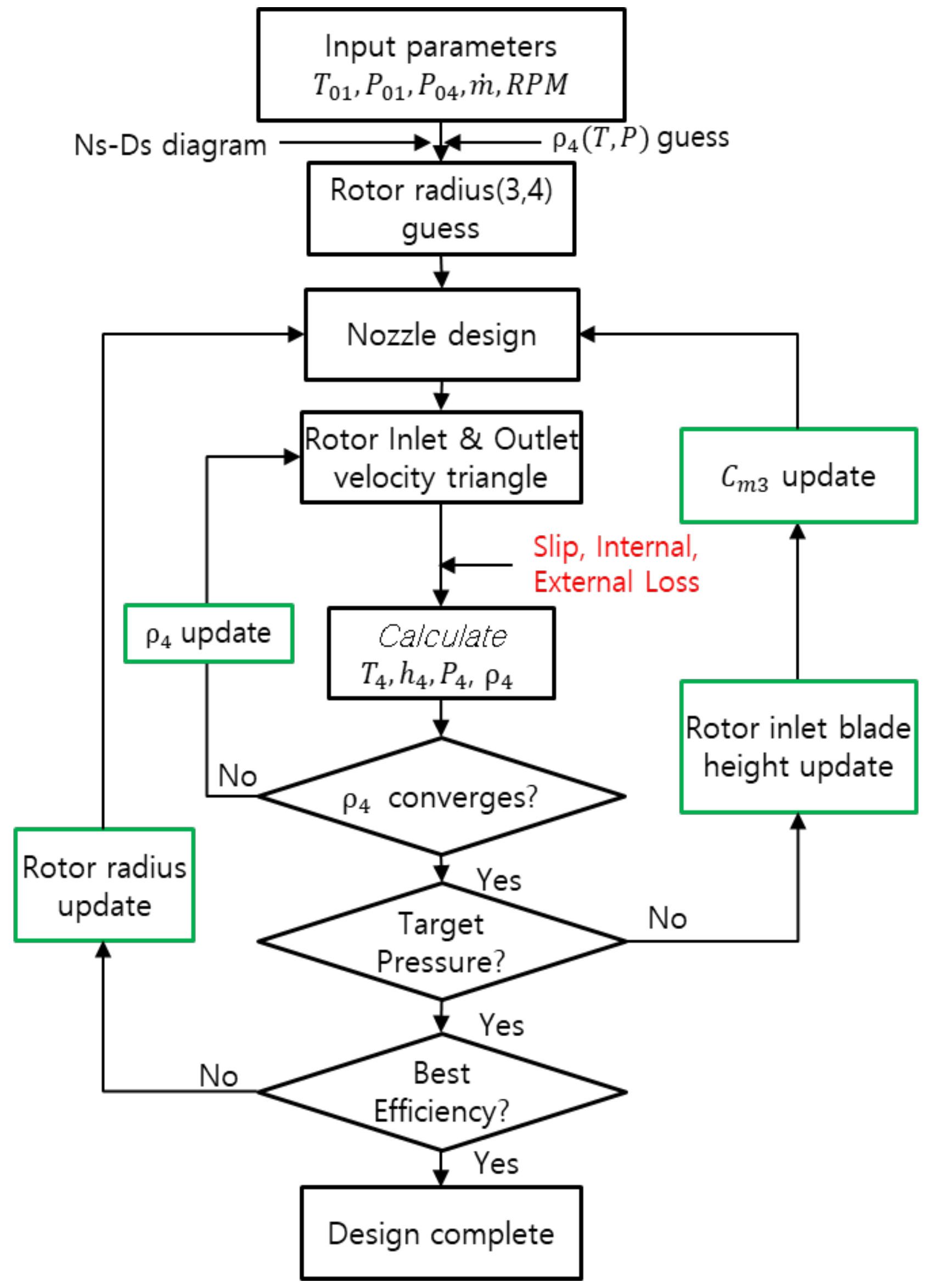


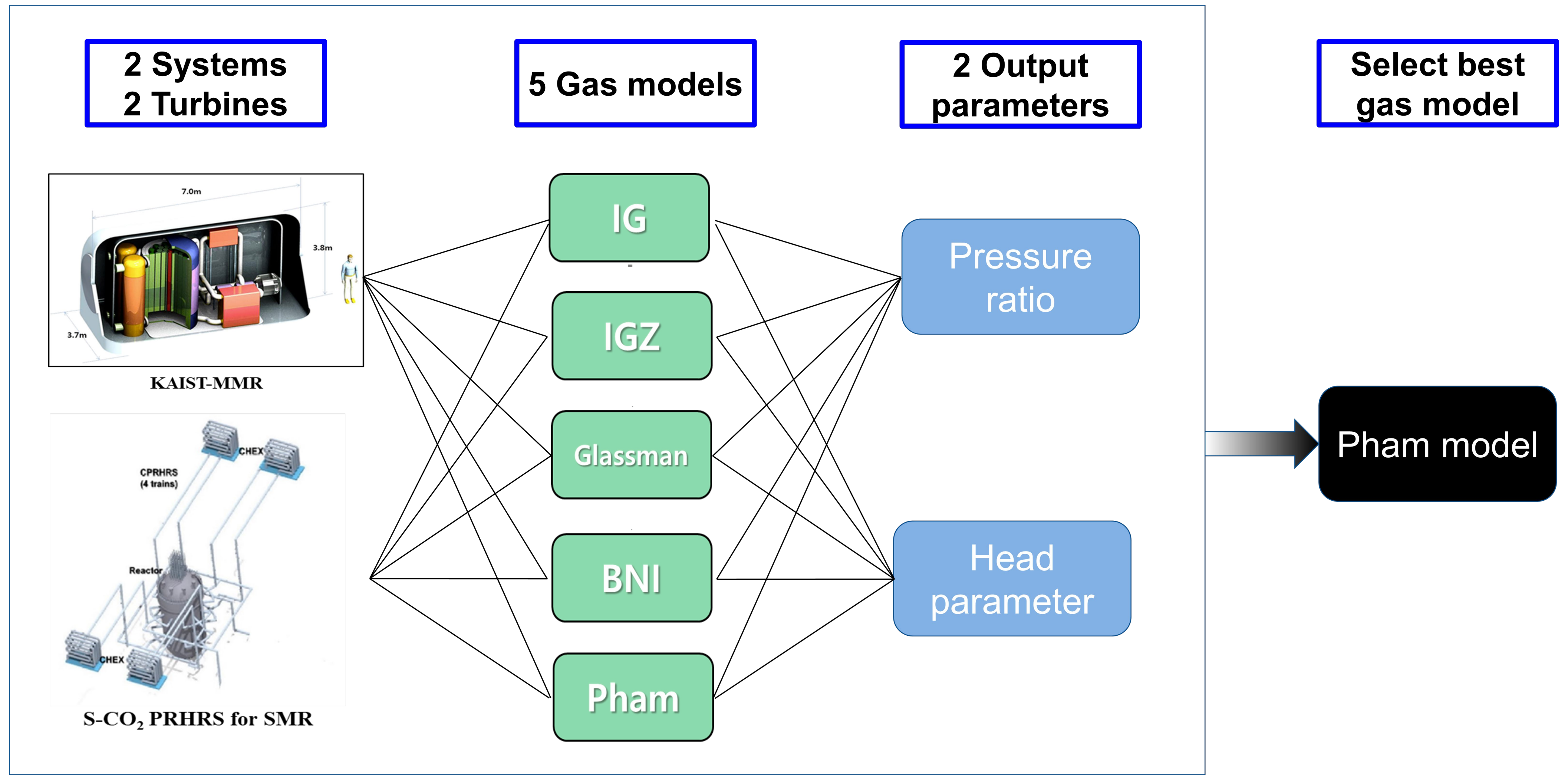


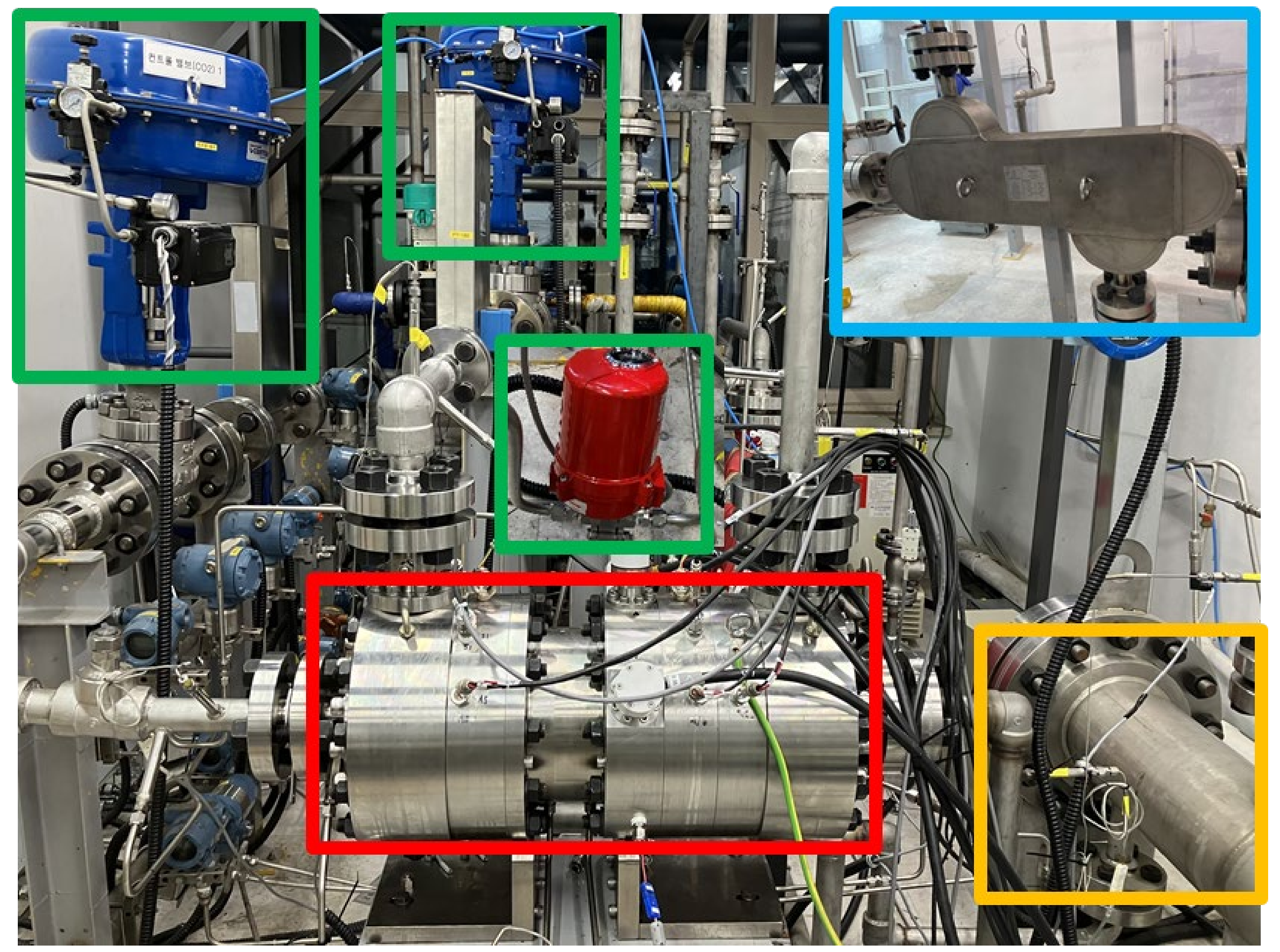
Summary

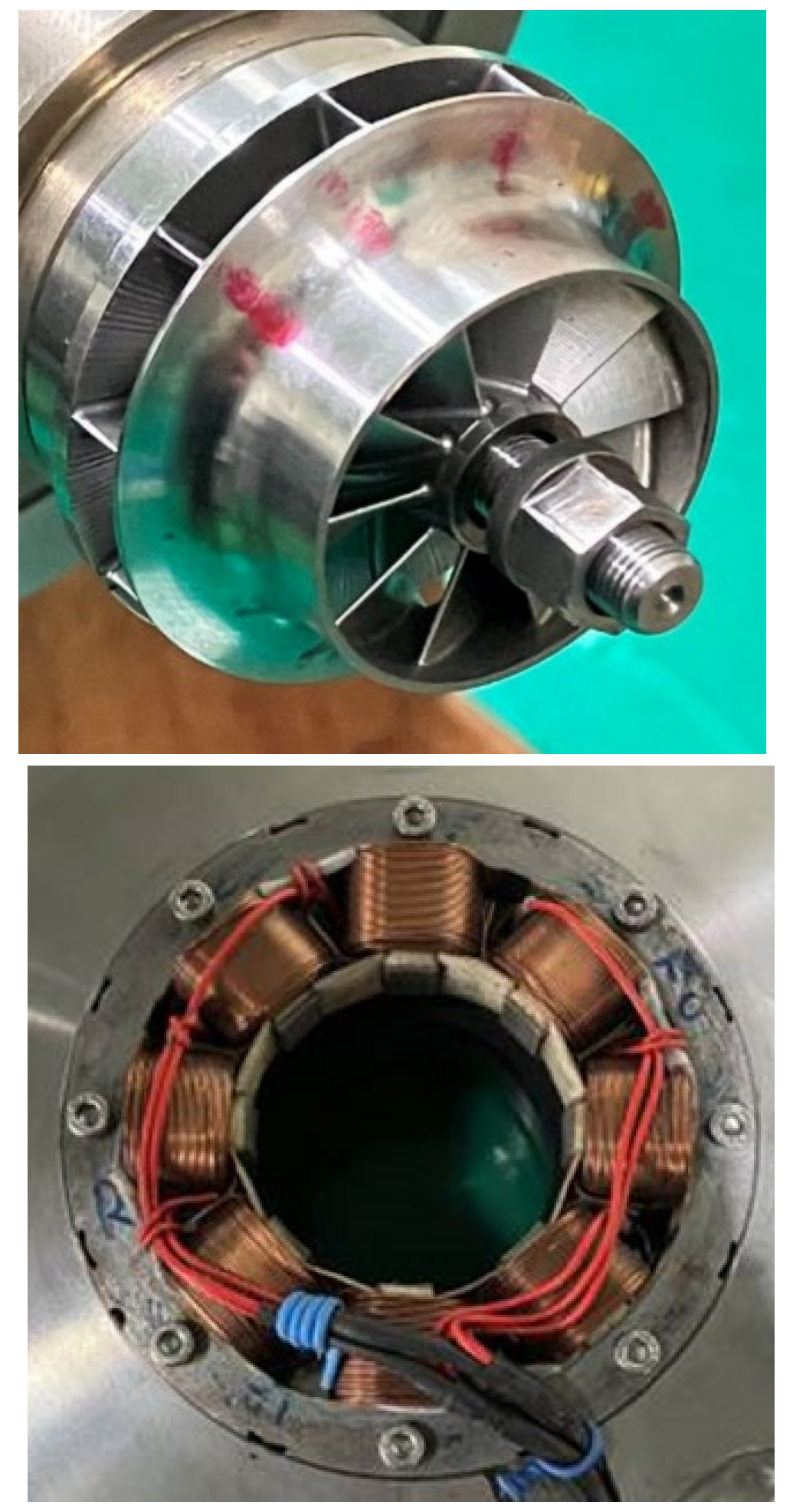

Compressibility Factor
ABC test loop turbine operating range
Pressure [bar]
95
90
85
80
75
30
40
50
60
70
80
90
100
Temperature [Celsius]
0.8
0.7
0.6
0.5
0.4
0.3
0.2

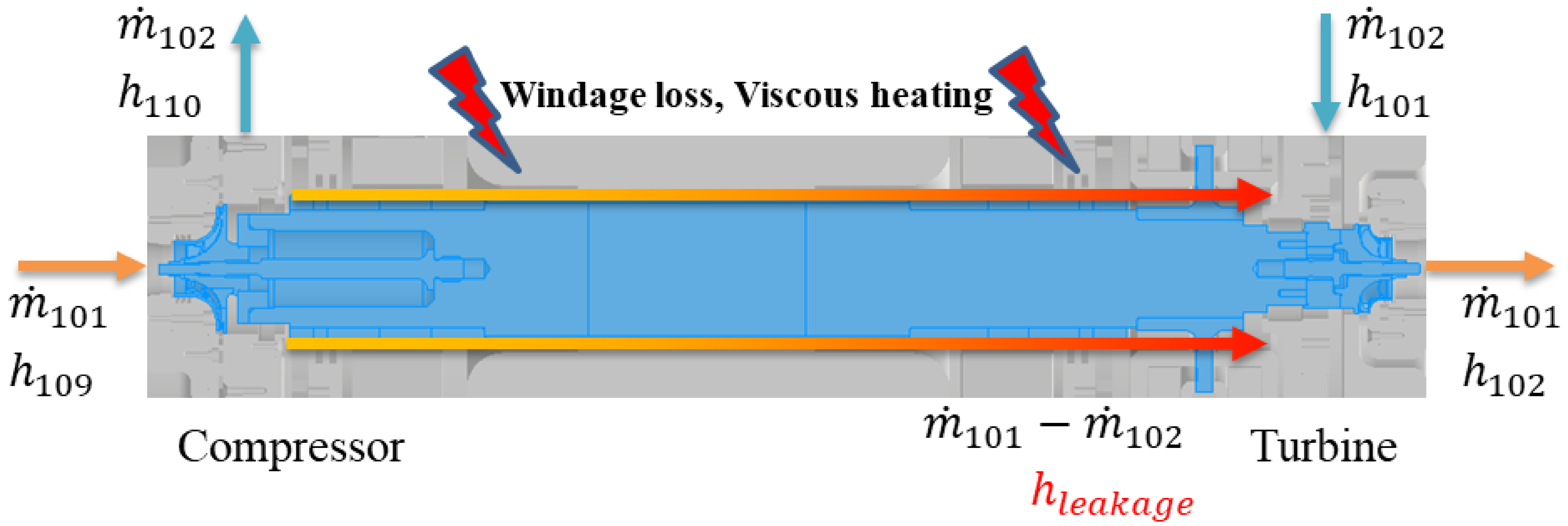

1
Volute
2
Nozzle
3
4
mean streamline
Rotor
Outlet
Axis

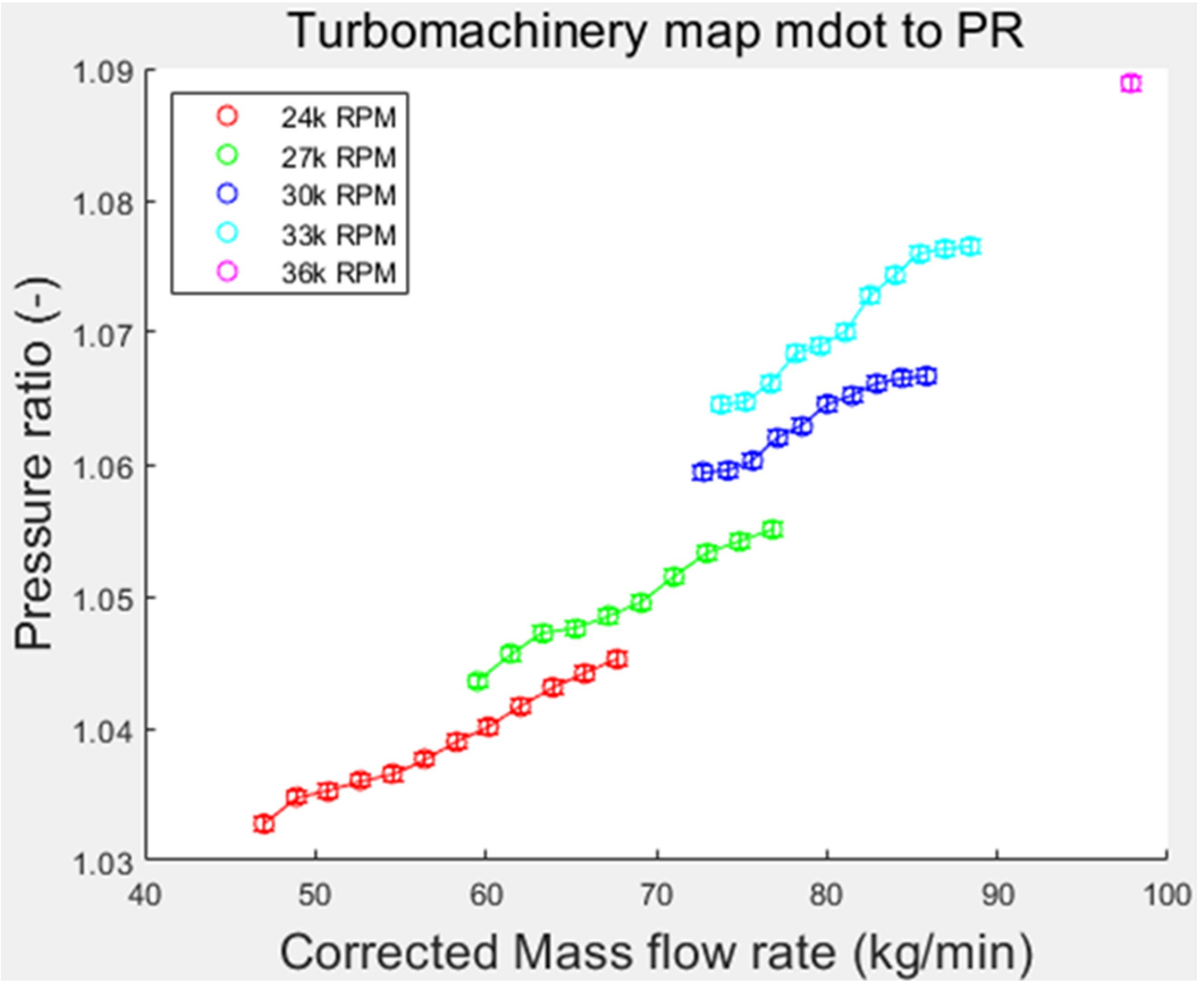

Input parameters $T_{01}, P_{01}, P_{04}, \dot{m}, RPM$
Ns-Ds diagram
$\rho_4(T,P)$ guess
Rotor radius(3,4) guess
Nozzle design
Rotor Inlet & Outlet velocity triangle
C_{m3} update
Slip, Internal, External Loss
ρ_4 update
Calculate T_4, h_4, P_4, ρ_4
Rotor inlet blade height update
No
ρ_4 converges?
Yes
Rotor radius update
Target Pressure?
No
Yes
No
Best Efficiency?
Yes
Design complete


2 Systems
2 Turbines
5 Gas models
2 Output parameters
Select best gas model
KAIST-MMR
S-CO₂ PRHRS for SMR
IG
IGZ
Glassman
BNI
Pham
Pressure ratio
Head parameter
Pham model






$\dot{m}_{102}$
h_{110}
Windage loss, Viscous heating
$\dot{m}_{102}$
h_{101}
$\dot{m}_{101}$
h_{109}
$\dot{m}_{101}$
h_{102}
Compressor
$\dot{m}_{101} - \dot{m}_{102}$
$h_{leakage}$
Turbine


Turbomachinery map mdot to PR
24k RPM
27k RPM
30k RPM
33k RPM
36k RPM
Pressure ratio (-)
1.09
1.08
1.07
1.06
1.05
1.04
1.03
40
50
60
70
80
90
100
Corrected Mass flow rate (kg/min)

Thanks