Impact of S-CO2 Properties on Centrifugal Compressor Impeller: Comparison of Two Loss Models for Mean Line Analyses

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Outline of the Presentation

- Introduction
- Analysis Methodology
  - Power Cycle Chosen for Investigation
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  - Method **B** (Pressure and Work Based Losses)
- Results
  - Method **A**
  - Method **B**
  - Comparison between the two analyses
- Conclusions
- Future Work

Introduction

- Supercritical CO_2- potential to enhance cycle efficiency in turbomachinery
- The main advantages over conventional steam cycles
  - Reduction in capital cost[1] – mainly because of reduced size of turbomachinery
  - Lower required power for compression[2]
  - Favorable operational temperature range- applicable in multiple power generation environments
- Expanding area of research on cycle configurations and optimization

Introduction: Review of Past Work

- CATER past work:
  - Developed a thermodynamic analysis cycle optimization Genetic Algorithm code for S-CO_2[3]
    → Analyzed the effects of recompression, reheating, and intercooling on the thermodynamic performance of a recuperated S-CO_2 Brayton cycle
  - Performed a comparison of 1-D and 3-D aerodynamic analysis of a stage 1 vane for a S-CO_2 turbine[4]
- Outside influential work in the design of S-CO_2 compressors:
  - Sanghera[5] proved the applicability of work loss correlations to account for losses in S-CO_2 compressors
  - Similarly, Brenes[6] validated a combination of relative total pressure loss correlations and work loss correlation to account for losses in a S-CO_2 compressor
    - These loss correlation methods were utilized in the S-CO_2 compressor analyses performed for comparison in this study

Introduction: Review of Current Work

- Aim of Current Work:
  - Create a mean line analysis code for a S-CO_2 impeller
  - Compare two types of loss models through results for cycle efficiency, internal work losses, and parasitic losses
    - Method **A**: Impeller parasitic and internal losses accounted through work loss correlations
    - Method **B**: Relative total pressure loss correlations to account for internal losses and work loss correlations to account for parasitic losses.
  - Utilize this 1-D analysis as the starting point of the design process for S-CO_2 compressor

Introduction: Description of Loss Types

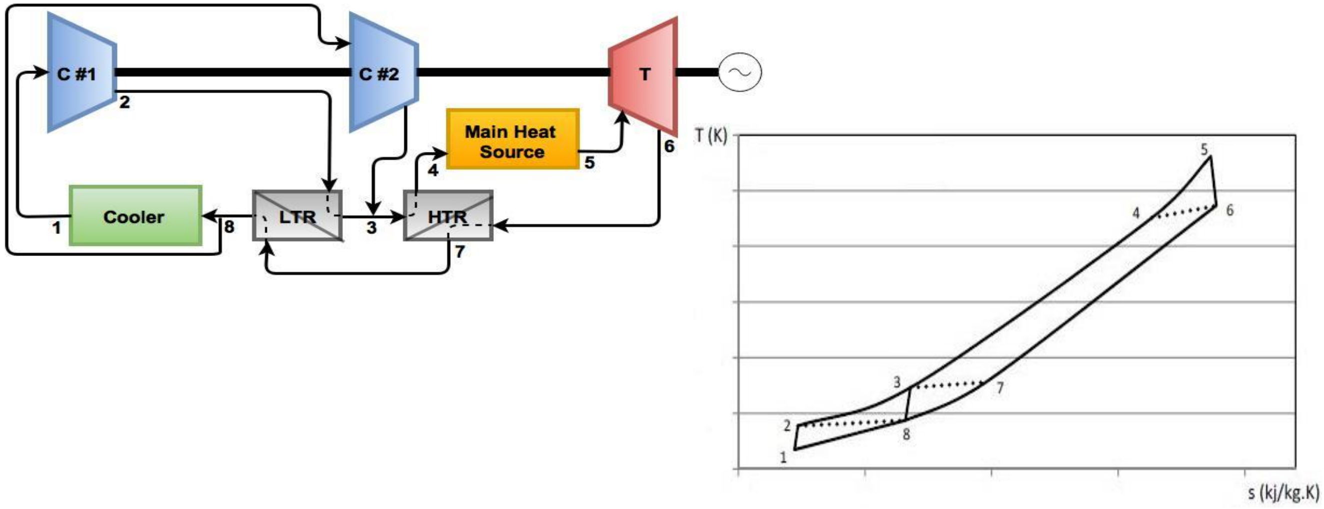
- Two types of losses occur in turbomachinery components
  - Internal Losses → originate due to non-ideal behavior of the flow
    - Comprised of incidence, aerodynamic loading, skin friction, tip clearance, and mixing losses
  - Parasitic Losses→ arise from mechanical deficiencies in the impeller
    - Consist of disk friction, recirculation, and seal leakage losses
- These Losses directly effect the aerodynamic and design efficiency of the turbomachinery component
- Aerodynamic and overall/design efficiency of the centrifugal compressor are defined as:
  - $\eta_{Aerodynamic} = \frac{\Delta h_{Euler} - \Delta h_{internal}}{\Delta h_{Euler}}$
  - $\eta_{Design} = \frac{\Delta h_{Euler} - \Delta h_{internal}}{\Delta h_{Euler} + \Delta h_{Parasitic}}$

Analysis Methodology: Power Cycle Definition

- Recuperated Recompression (RR) Brayton cycle utilized to carry out this study
- Net Output of 100 MW and an inlet turbine temperature (TIT) of 1350 K chosen as design criteria
- Thermodynamic cycle state points were obtained → source of inlet conditions of the S-CO_2 impeller

Tabulated Cycle States for the Specified RRC

State Points	Temperature (K)	Pressure (kPa)	Specific Enthalpy (kJ/kg)	Density (kg/m³)	Specific Entropy (kJ/kg-K)
1	320.0	9500	382.5	374.26	1.58
2	378.9	24000	420.5	544.16	1.60
3	487.9	23976	606.8	295.75	2.03
4	1154.4	23952	1455.6	103.75	3.13
5	1350.0	23904	1713	88.59	3.34
6	1196.6	9691	1511.8	41.91	3.36
7	498.2	9643	662.9	109.33	2.31
8	388.9	9595	529.7	162.65	2.00



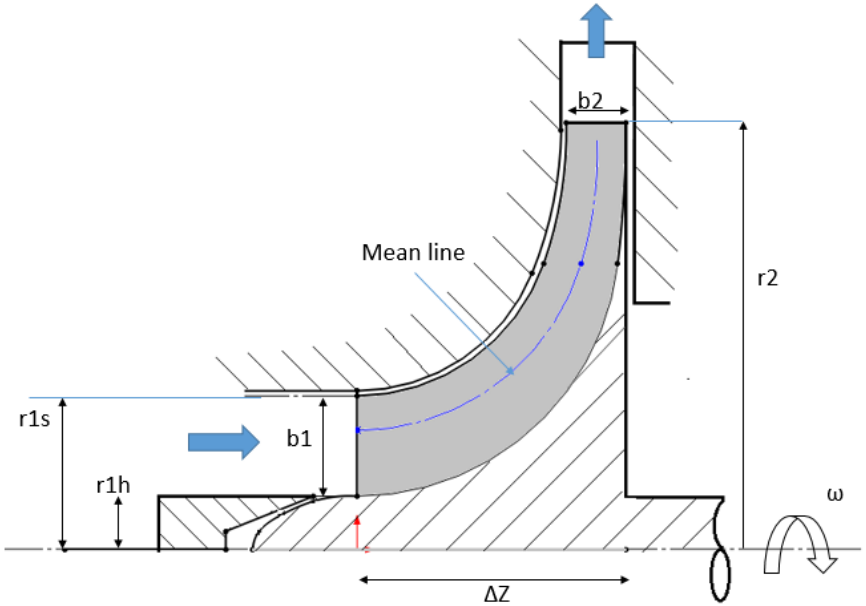
Recuperated Cycle Layout[5] and Corresponding T-S Diagram

Analysis Methodology: Mean Line Analysis Code

- Developed in MATLAB
- Based on law of conservation of mass, Euler turbine equation, and centrifugal compressor loss models found in literature
- Utilizes NIST REFPROP database to solve equation of state for specified points for S-CO_2
- Input Parameters:
  - P01, T01, mass flow rate, geometry parameters
- Input Variables:
  - Rotational Speed
- Output
  - Impeller exit conditions, converged efficiency, compressor impeller pressure ratio
- Iterative process for loss calculation is initialized using isentropic impeller exit conditions

Analysis Methodology: Impeller Geometry and Inputs

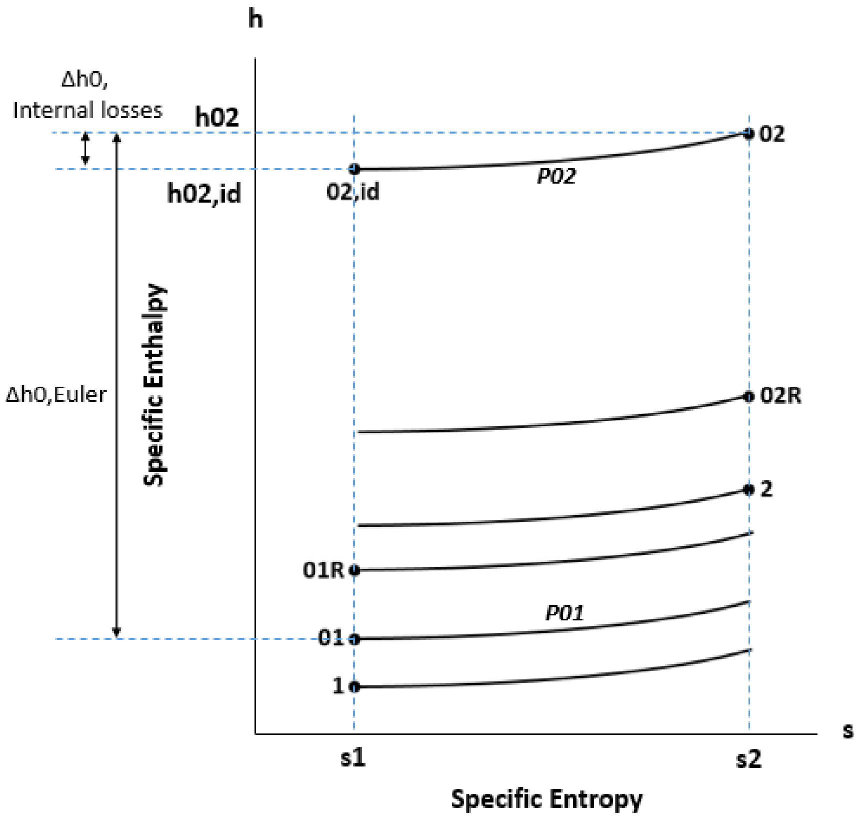
Input and Output Variable Tabulated Values for Mean Line Analysis Code	
Main Input Parameters and Variables	
Inlet Total Temperature, T01	320 K
Inlet Total Pressure, P01	9.5 MPa
Mass flow rate, $\dot{m}$	472.189 kg/s
Angular Speed, ω	6560 RPM
Main Geometrical Parameters	
Impeller Inlet Hub Radius, r_{1h}	0.1322 m
Impeller Inlet Shroud Radius, r_{1s}	0.1924 m
Impeller Exit Radius, r_2	0.2635 m
Axial Length of Impeller, ΔZ	0.144 m
Number of Blades in Impeller, Z	15
Blade Height, b_2	0.0231 m
Blade Thickness, t	5.7 mm



Schematic of the Centrifugal Compressor Impeller

Analysis Methodology- Method A

- Changes in enthalpy due to blade work, internal losses, and parasitic losses are defined as:
  - $\Delta h_{internal} = \Delta h_{ABL} + \Delta h_{SF} + \Delta h_{TCL} + \Delta h_{MIX}$
  - $\Delta h_{parasitic} = \Delta h_{DF} + \Delta h_{RC} + \Delta h_{LL}$
  - $\Delta h_{Euler} = C_{w2}U_2 - C_{w1}U_1$
- Further, the efficiencies can be determined:
  - $\eta_{Aerodynamic} = \frac{\Delta h_{Euler} - \Delta h_{internal}}{\Delta h_{Euler}}$
  - $\eta_{Design} = \frac{\Delta h_{Euler} - \Delta h_{internal}}{\Delta h_{Euler} + \Delta h_{Parasitic}}$

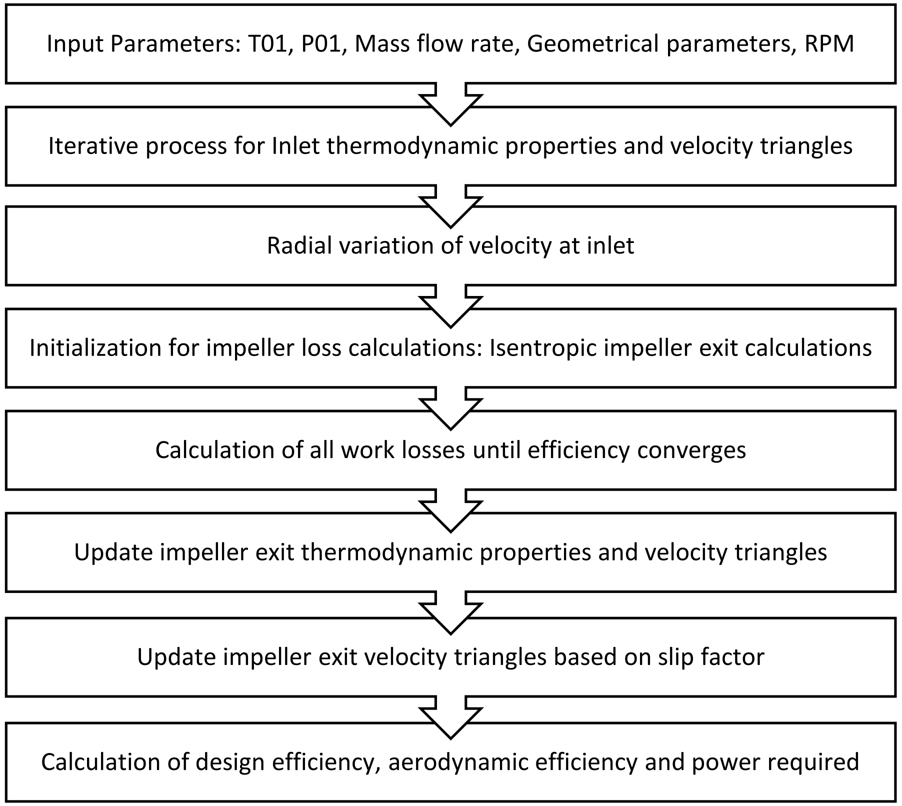


h-s Diagram Schematic for Method A

Analysis Methodology- Method **A**

- Axial inlet
- Constant blade angle along the span
- Isentropic exit calculation as initial point
    - Pressure ratio for the impeller is decided
- Calculation of losses as work losses
- Efficiency as convergence criterion
- Updated exit condition after convergence

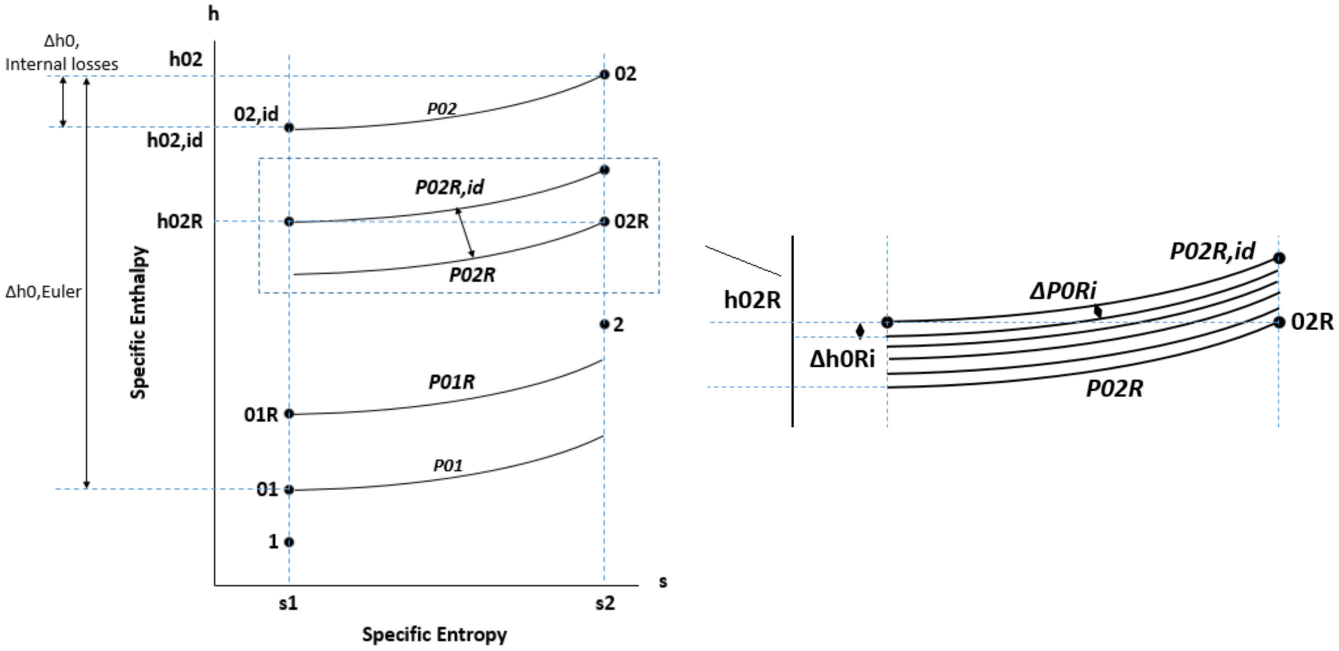
criterion is met



Algorithm for Mean-Line Analysis Code – Method A

Analysis Methodology- Method B

- Calculation of total change in enthalpy and relative pressure loss is as follows:
  - $\Delta h_{0,parasitic} = U_2^2 * (I_{DF} + I_{RC} + I_{LL})$
  - $\Delta h_{0,Euler} = I_B * U_2^2$
  - $P'_{02} = P'_{02,id} - f_c(P'_{01} - P_1)\sum_i \omega_i$
  - Where $\sum_i \omega_i = \omega_{ABL} + \omega_{SF} + \omega_{MIX} + \omega_{TCL}$
- Through the calculation of relative total pressure loss, the absolute total pressure loss is obtained
- Further, the ideal total enthalpy at the exit is then found and efficiencies are calculated

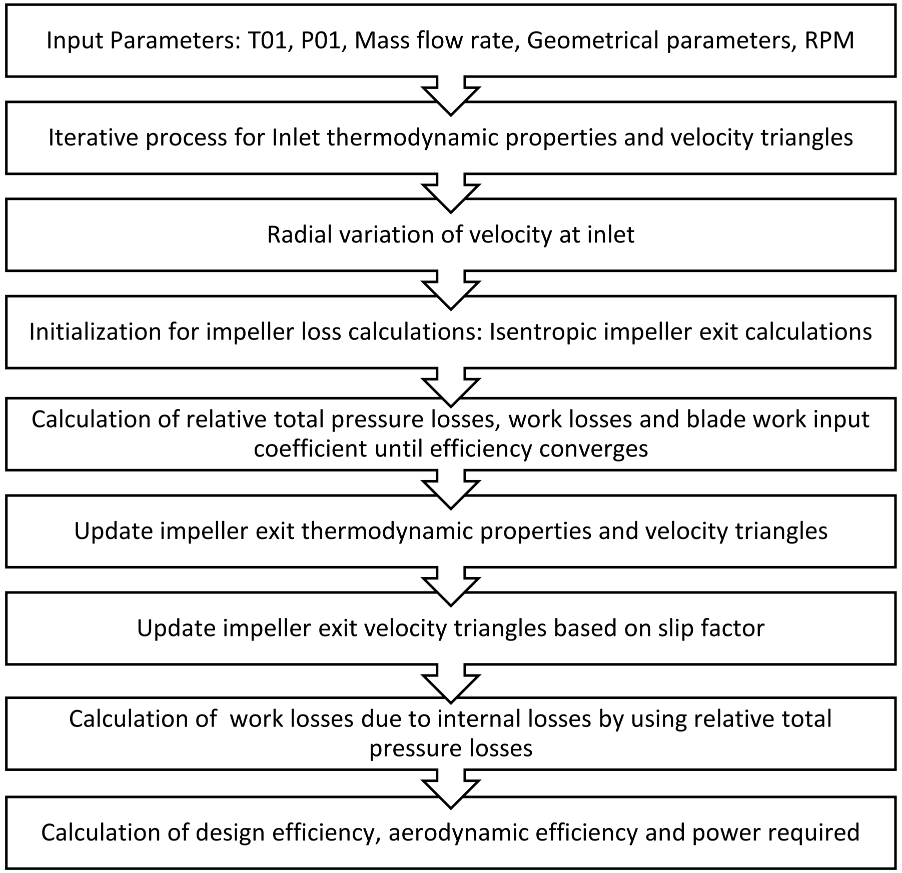


h-s Diagram Schematic for Method B

$$\eta_{Aerodynamic} = \frac{h_{02,id} - h_{01}}{\Delta h_{Euler}} \qquad \eta_{Design} = \frac{h_{02,id} - h_{01}}{\Delta h_{Euler} + \Delta h_{parasitic}}$$

Analysis Methodology- Method **B**

- Axial inlet
- Constant blade angle along the span
- Isentropic exit calculation as initial point
- Calculation of relative pressure loss coefficients – Pressure ratio updated in every loop
- Calculation of work losses
- Total enthalpy change from updated value of exit total pressure
- Efficiency as convergence criterion
- Internal losses are converted from pressure loss coefficients to enthalpy losses



Algorithm for Mean-Line Analysis Code – Method B

Results: Method A

- Impeller performance parameters and thermodynamic properties at the impeller exit were obtained
  - A resulting impeller total pressure ratio of 2.47 was observed

Slip Factor	0.87
Inlet Total Enthalpy, h_{01}	382.49 kJ/kg
$\Delta h_{0,Euler}$	31.36 kJ/kg
$\Delta h_{0,Internal}$	2.00 kJ/kg
Exit Ideal Total Enthalpy, $h_{02,id}$	411.84 kJ/kg
$\Delta h_{0,Parasitic}$	1.13 kJ/kg
Total-to-Total Efficiency (Aerodynamic)	93.60%
Total-to-Total Efficiency (Overall)	88.97%
Power required	15.34 MW

	Impeller Inlet conditions	**Impeller Exit conditions**
Total Pressure, P_0	9.5 MPa	23.42 MPa
Static Pressure P	9.5 MPa	17.44 MPa
Total Temperature, T_0	320 K	374.85 K
Static Temperature, T	319.49 K	357.15 K
Static Density, ρ	372.49 kg/m^3	297.91 kg/m^3
Static Enthalpy, h	382.28 kJ/kg	402.43 kJ/kg

Results: Method **B**

- Resulting pressure ratio of 2 is significantly lower than the pressure ratio calculated through thermodynamic cycle analysis

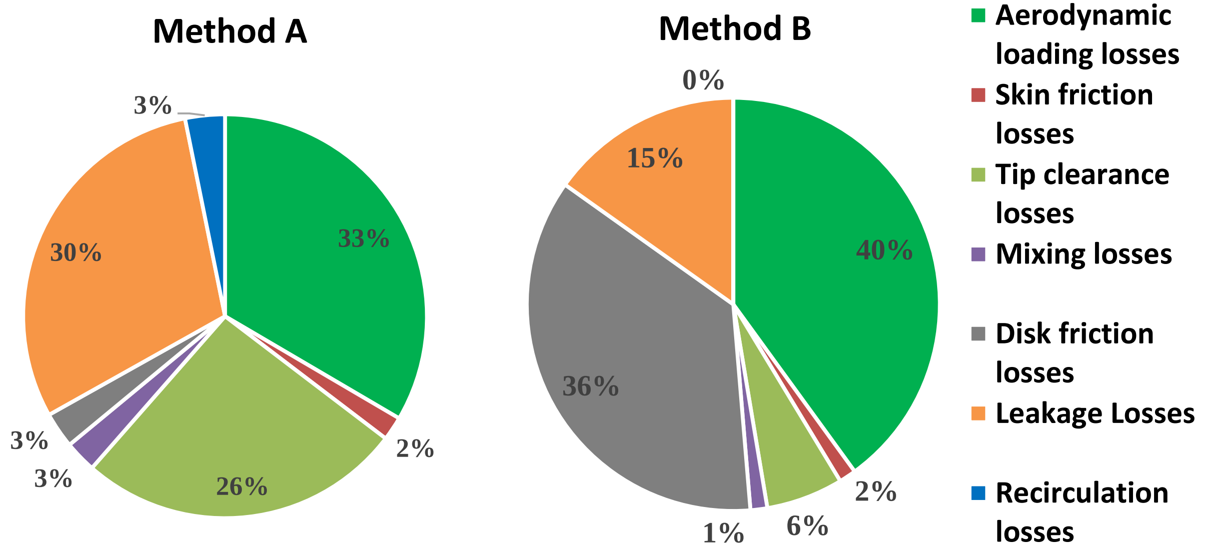
Slip Factor	0.87
Inlet Total Enthalpy, h_{01}	382.49 kJ/kg
$\Delta h_{0,Euler}$	24.00 kJ/kg
$\Delta h_{0,Internal}$	2.66 kJ/kg
Exit Ideal Total Enthalpy, $h_{02,id}$	403.83 kJ/kg
$\Delta h_{0,Parasitic}$	2.81 kJ/kg
Total-to-Total Efficiency (Aerodynamic)	88.92%
Total-to-Total Efficiency (Overall)	79.60%
Power required	12.66 MW

	Impeller Inlet conditions	**Impeller Exit conditions**
Total Pressure, P_0	9.5 MPa	19.08 MPa
Static Pressure P	9.5 MPa	12.97 MPa
Total Temperature, T_0	320 K	360.22 K
Static Temperature, T	319.49 K	359.90 K
Static Density, ρ	372.49 kg/m^3	250.95 kg/m^3
Static Enthalpy, h	382.28 kJ/kg	393.56 kJ/kg

Results: Comparison Between Method **A** & Method **B**

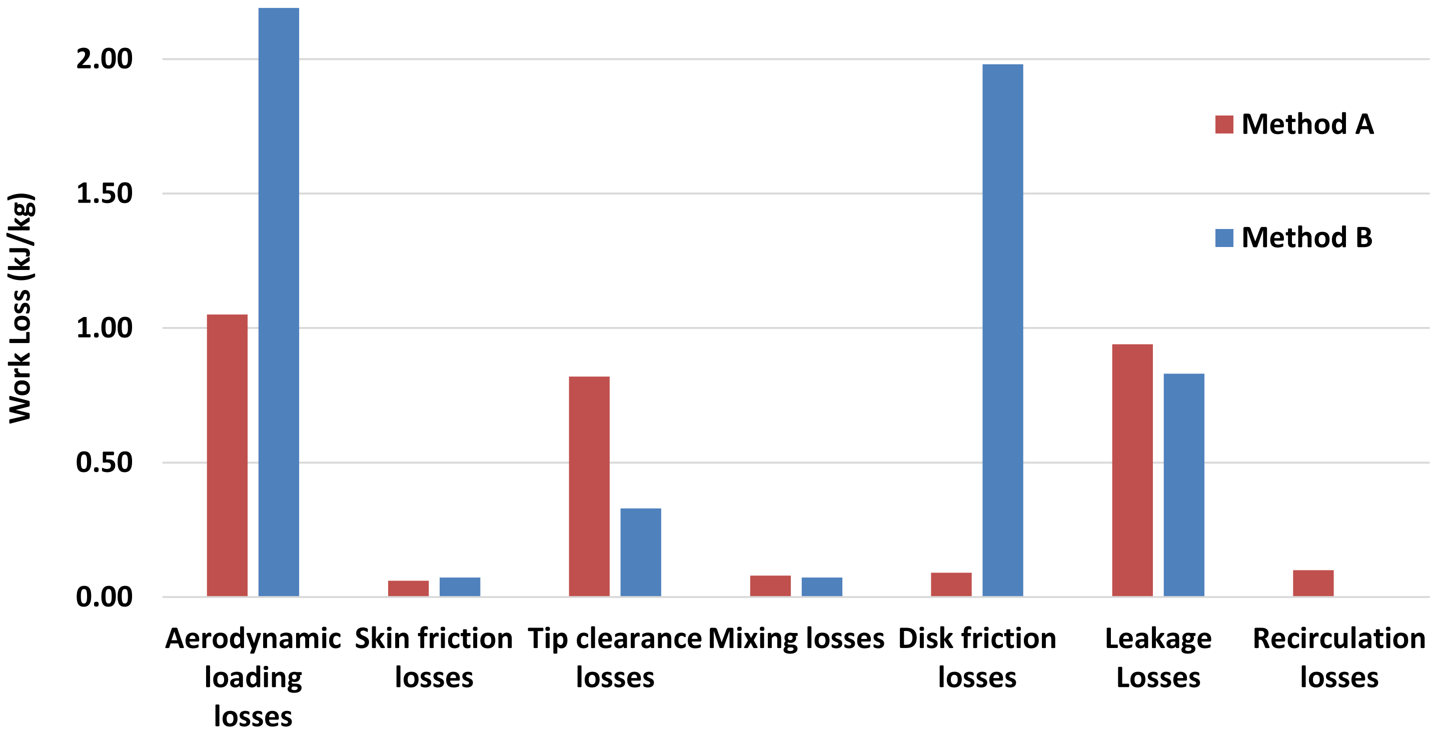
- For Method **B** ,individual relative pressure losses due to each internal loss were converted to change in relative total enthalpy for comparison purposes
- The sum of these converted losses is found out to be equal to $\Delta h_{internal}$ calculated using blade loading coefficient
  - $\Delta h_{02R,id\rightarrow 02R} = \Delta h_{0R,SF} + \Delta h_{0R,TCL} + \Delta h_{0R,ABL} + \Delta h_{0R,MIX} \approx \Delta h_{02,id\rightarrow 02} = \Delta h_{internal}$

Impeller Efficiency	Method A	Method B
Total-to-Total Efficiency (Aerodynamic)	92.55%	88.92%
Total-to-Total Efficiency (Overall)	**87.66%**	77.60%
Internal Work Losses		
Aerodynamic loading losses	1.05 kJ/kg	2.19 kJ/kg
Skin friction losses	0.06 kJ/kg	0.073 kJ/kg
Tip clearance losses	0.82 kJ/kg	0.33 kJ/kg
Mixing losses	0.08 kJ/kg	0.072 kJ/kg
Parasitic Losses		
Disk friction losses	0.09 kJ/kg	1.98 kJ/kg
Leakage Losses	0.94 kJ/kg	0.83 kJ/kg
Recirculation losses	0.10 kJ/kg	0 kJ/kg



Results: Comparison between Method **A** & Method **B**

- Main Difference: Results for Method **B** display a significant contribution from disk friction losses for the overall losses, where as Method **A** displays a relatively minor contribution due to disk friction losses
- Calculated leakage, mixing and skin friction losses are close in magnitude between the two methods
- Aerodynamic loading losses play a significant role in the overall calculated loss in both methods, but more so in Method **B**

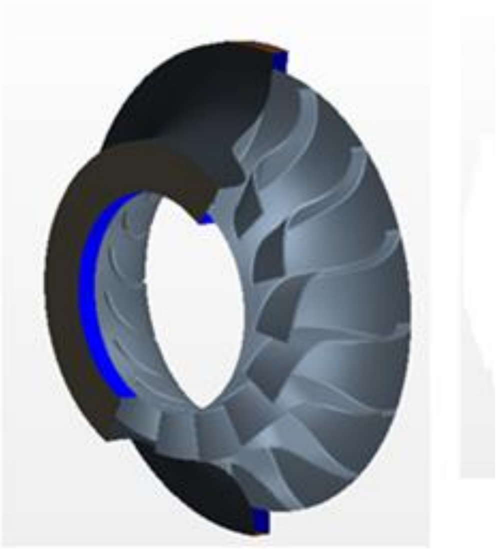


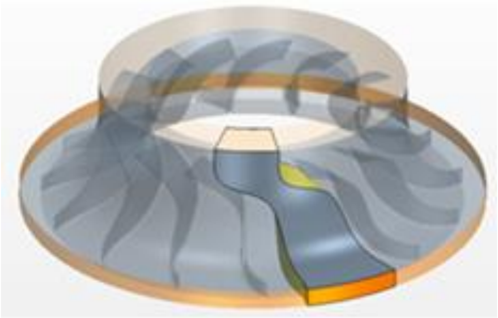
Conclusions

- Based on RPM, inlet conditions, and impeller geometry, mean line analysis codes deliver an estimate of:
  - Impeller efficiency
  - Pressure ratio
  - Resulting velocity triangles
  - Input power required
- This work serves as a comparison of two types of loss models used in centrifugal compressors for the application in $S\text{-}CO_2$ power cycles
- A starting point for further development into the 3-D design process and analysis of $S\text{-}CO_2$ centrifugal compressor

Future Work

- Inverse code to obtain the main geometrical parameters of impeller for the required inputs
- Streamline curvature method
- Develop 3-D geometry through CAD software
- Computational fluid dynamics (CFD) study of 3-D impeller model
  - Comparison of CFD analysis with mean-line analyses
    - Provide further insight on the significant differences in the results between each analysis method
  - Investigation of two phase regions in impeller through CFD





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[3] Mohagheghi, M., Kapat, J., "Thermodynamic Optimization of Recuperated S-CO_2 Brayton Cycles for Solar Tower Applications," *ASME Turbo Expo,* ASME, San Antonio, TX, 2013

[4] Joshua Schmitt, Rachel Willis, David Amos, Jayanta Kapat, and Chad Custer, "Study of a Supercritical CO2 Turbine With TIT of 1350 K for Brayton Cycle With 100 MW Output: Aerodynamic Analysis of Stage 1 Vane," Proceedings of the ASME Turbo Expo 2014, Paper No. GT2014-27214, June 16-20, Dusseldorf, Germany.

[5] Sanghera, S.S., "Centrifugal Compressor Mean line Design Using Real Gas Properties," DR4907-1213-PRFSS-001, Department of Mechanical and Aerospace Engineering, Carleton University, Ottawa, Ontario, Canada, 2013

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Background – Our Past Papers

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Thank you!
Questions??