DESIGN AND CFD ANALYSIS OF CENTRIFUGAL COMPRESSOR AND TURBINE FOR SUPERCRITICAL CO_2 POWER CYCLE

NABROS ENERGY
Dare To Dream Impossible

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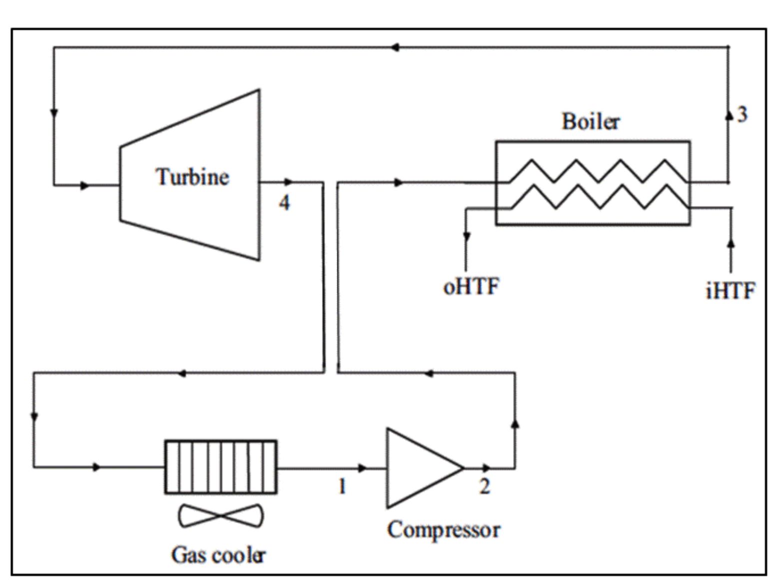
INTRODUCTION

- Supercritical CO_2 (S-CO_2) power cycles are the most promising technology for high temperature heat source namely concentrated solar thermal, nuclear, fossil fuel as well as low temperature heat source like waste heat recovery and geothermal applications.
- The main reason is lower compressibility factor of CO_2 near critical point (73.8 bar, 31.1 °C) .
- A design and CFD analysis of the compressor and turbine for supercritical CO_2 has been done in the present work .
- We have developed a in-house code for meanline design of compressor and turbine, considering Aungier's loss correlation in Engineering Equation Solver (EES) for net 10 kWe power.
- The operating range of compressor, temperature is 305 to 320 K and pressure is 75 to 110 bar. Turbine inlet temperature is 800 K.
- The fluid properties were implemented via property table in the computational analysis code to simulate nonlinear behavior near the critical point of CO_2.
- Overall results shows 80% isentropic efficiency of turbine and compressor, and compression work has also reduced by 50% as compared to ideal compression process.

THERMODYNAMIC ANALYSIS

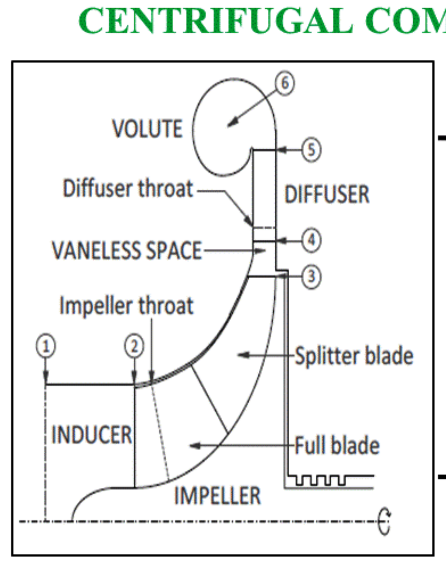


State Points	Pressure (bar)	Temperature (K)	Specific Enthalpy (kJ/kg)	Density (kg/m³)
1	75	308	396.74	274.97
2	105	331.9	409.27	335.06
3	101.3	800	1014.9	66.185
4	77.72	773.2	985.13	52.819

$$M_{wf} = \frac{30}{h_3 - h_4}$$

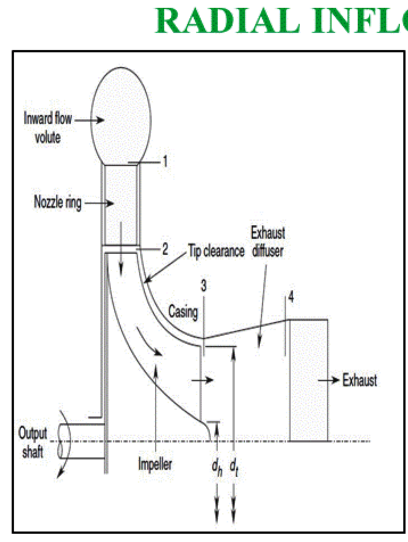
The calculated mass flow rate from above equation is 1.01 kg/s

CENTRIFUGAL COMPRESSOR



1-2	Inducer
2-3	Impeller
3-4	Vaneless space
4-5	Diffuser
5-6	Volute

RADIAL INFLOW TURBINE



0-1	Volute
1-2	Nozzle
2-3	Impeller
3-4	Diffuser

METHODOLOGY

Geometry design of compressor impeller and turbine by Meanline method

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Validation of obtained geometry with simulation software

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Geometry design of compressor impeller and turbine by Meanline method

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Three dimensional CAD model generation using ANSYS software

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CFD investigation on the flow characteristic of S-CO_2 compressor and turbine, considering ideal and real gas behavior

DESIGN

Turbine and Compressor Meanline Design and Meshing

Inlet and Exit velocities of Turbine

M_{abs}	0.396	M_{abs}	0.098
M_{rel}	0.114	M_{rel}	0.299
U_2 (m/s)	206.741	U_3 (m/s)	127.97
V_2(m/s)	181.683	V_3(m/s)	44.36
W_2(m/s)	52.526	W_3(m/s)	135.44
V_{w2}(m/s)	176.529	V_{w3}(m/s)	0
V_{r2}(m/s)	42.968	V_{r3}(m/s)	44.36
α_2	76.32	α_3	0
β_2	-35.112	β_3	-70.881

Impeller Inlet

$M_{rel,hub}$	0.193
$M_{rel,shr}$	0.316
$V_{m,hub}$(m/s)	47.11
$V_{w,hub}$(m/s)	0
β_{hub}	26.1
B_{shr}	56.8

Impeller Exit

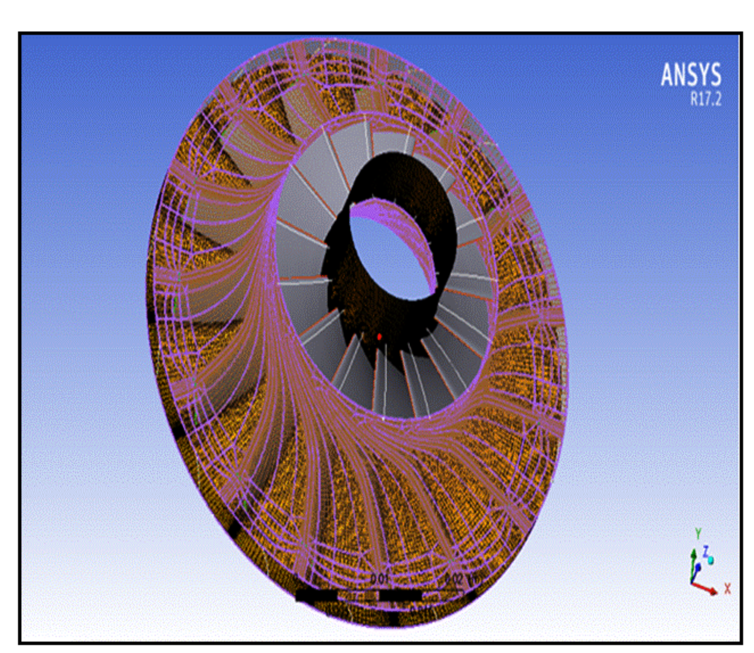
M_{rms}	0.595
M_U	0.639
V_{rms}(m/s)	167.54
W_{rms}(m/s)	34.39
β_{rms}	16.7
α_{rms}	78.7

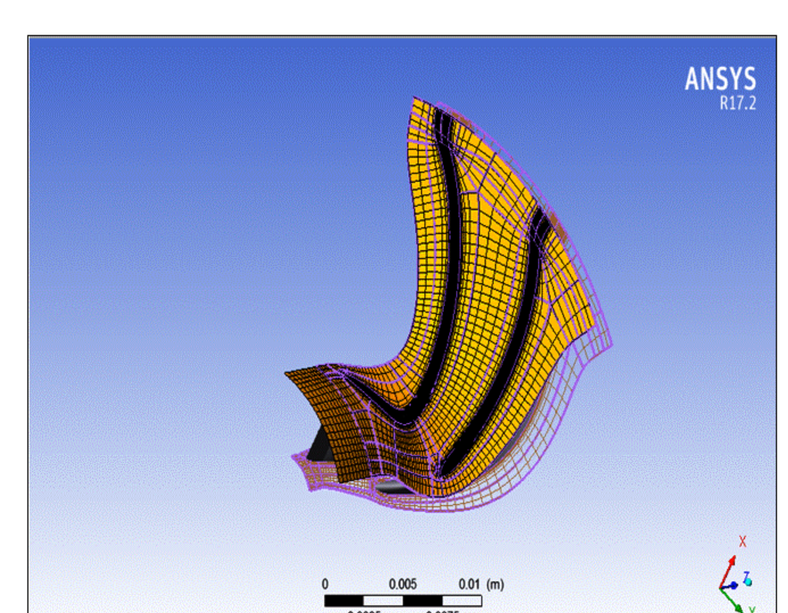
Geometry of Turbine

d_2 (mm)	Tip width (mm)	$d_{3,hub}$(mm)	$d_{3,shr}$(mm)	$d_2/d_{3,rms}$	$\beta_{3,rms}$	$\beta_{3,hub}$
45.47	3.109	12.732	28.146	2.082	-65.93	-52.536

Geometry of Compressor

$D_{hub,in}$ (mm)	$D_{shr,in}$ (mm)	A_{throat} (mm²)	$D_{imp,out}$ (mm)	Tip Width (mm)
5	15.61	95.2	37.79	1.823





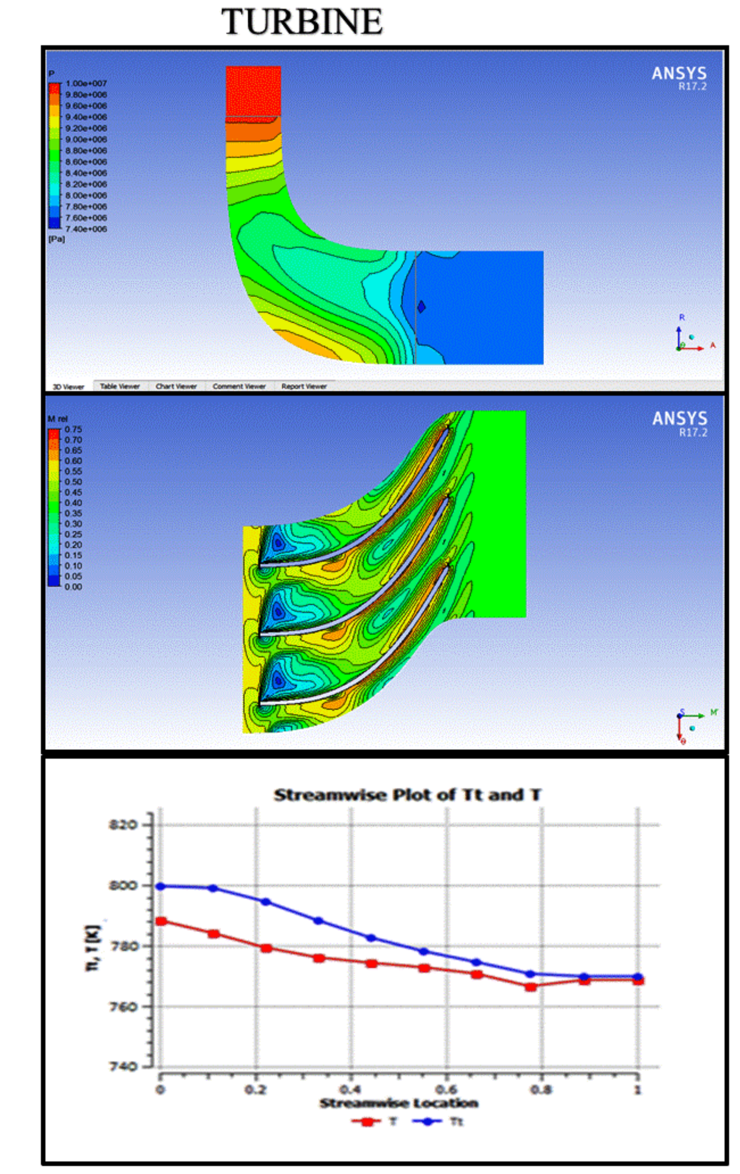
General Parameter	
Reynolds Number	$5*10^7$
Shroud Tip mesh Method	Match Expansion at Blade Tip
Topology Mesh Technique	ATM optimized
Boundary Layer Refinement Control Factor	Proportional to Mesh Size
Near Wall Treatment	Y+ Method
Turbulence Model	K-w SST model

Number nodes and Element

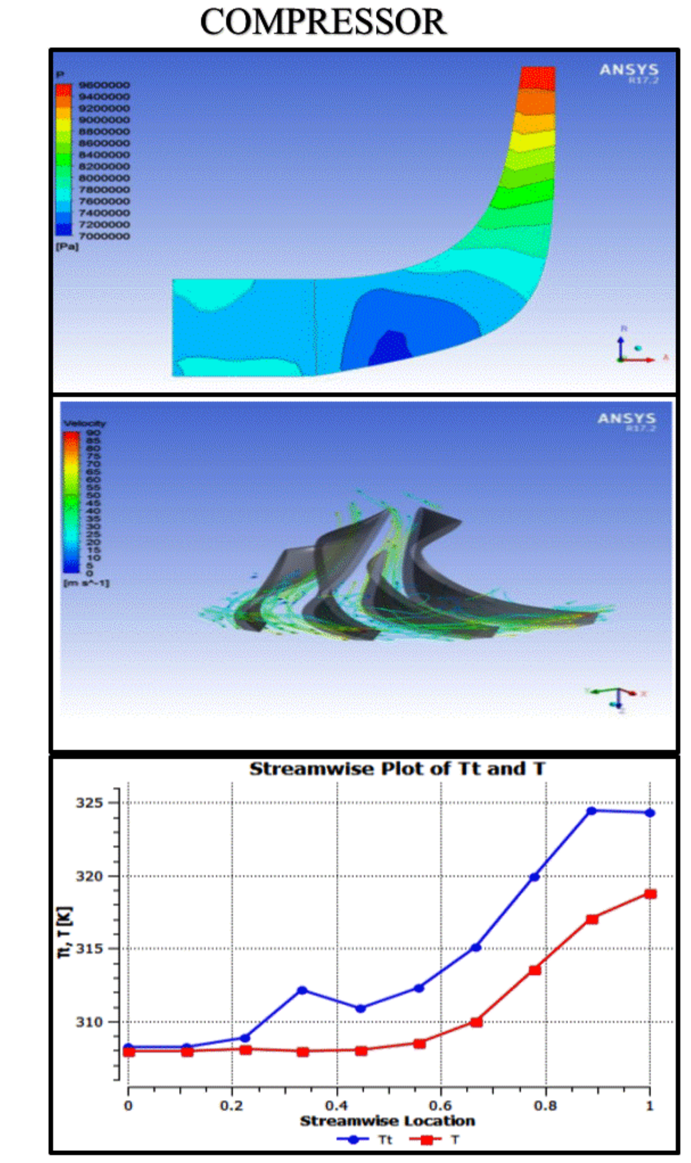
Turbine	Compressor
2,76,610 & 2,49,568	6,78,000 & 6,25,100

Analysis

TURBINE



COMPRESSOR



Comparison of thermodynamic property for Compressor for Ideal vs Real gas

Property	Location	Analysis Type: Meanline	Analysis Type: Real Gas (CFD)
T_{static} in K	In	307	307.2
	Out	326.7	318
T_{total} in K	In	308	308
	Out	338	324
P_{static} in bar	In	72	74
	Out	88	95
P_{total} in bar	In	75	75
	Out	105	105
Density in Kg/m³	In	129	271.14
	Out	146	418.5

CONCLUSION

1) Analysis for centrifugal compressor was carried out for real and ideal gas, The findings are, power for compression reduced by 50% for real gas compare to ideal gas compression under the effect of compressibility factor 0.3 near the liquid nature. Isentropic efficiency is 80% for both cases.
2) The incidence angle for turbine was optimized to -26.3 degree, head coefficient is 0.8927 and stage flow coefficient is 0.2384. We have achieved 80% isentropic efficiency of turbine, after considering all the losses