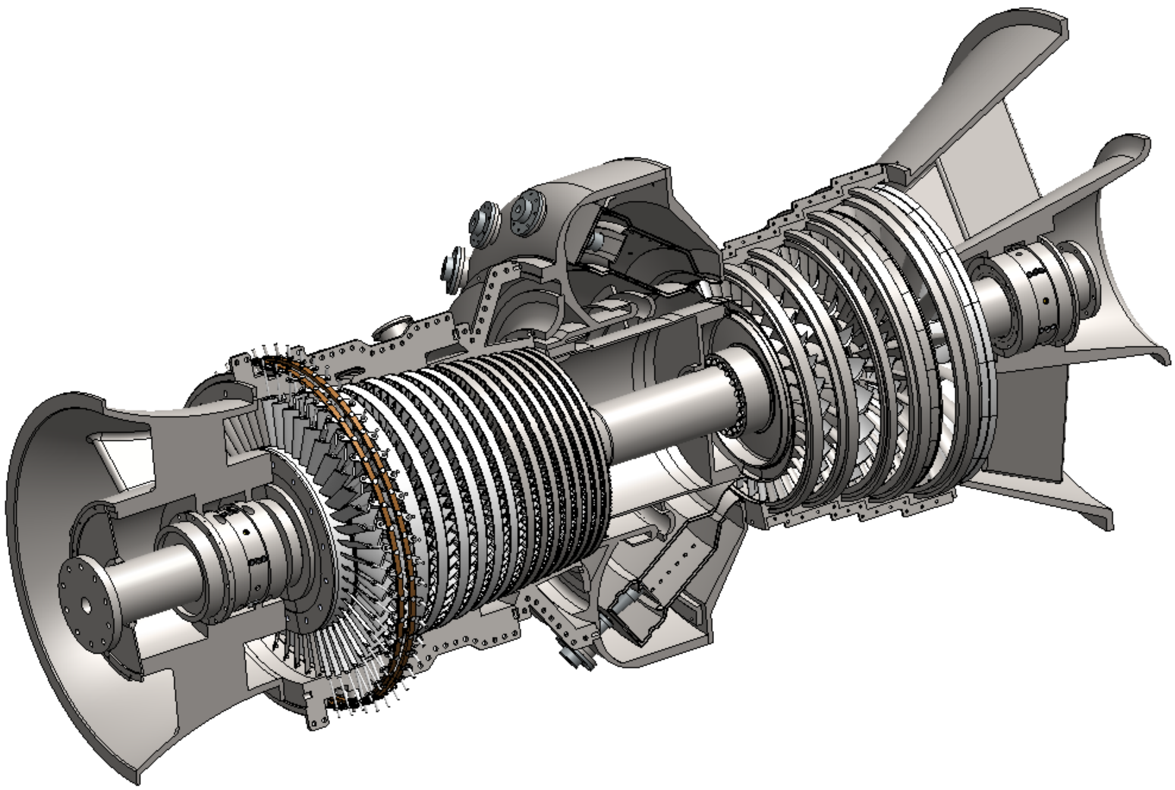
MEEN 646 – Aerothermodynamics of Turbomachinery



Final Project:

Aero-Thermo-Mechanical Design of a Power Generation Gas Turbine Engine

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Nomenclature

D	Diameter	Ma	Mach number
r	Radius	V_{ax}	Axial velocity
h_b	Blade height	U	Circumferential velocity
c	Chord length	r	Degree of reaction
s	Blade spacing	V	Absolute velocity
N or ω	Rotational speed	W	Relative velocity
ρ	Gas density	λ	Stage load coefficient
$\dot{m}$	Mass flow rate	ϕ	Flow coefficient
T	Static temperature	η_s	Isentropic Efficiency
P	Static pressure	δ_{1y}	Boundary layer displacement thickness
H	Enthalpy	C_D	Drag coefficient
T_0	Total temperature	ϵ	Drag lift ratio ($\frac{C_D}{C_L}$)
P_0	Total pressure	C_L	Lift coefficient
H_0	Total enthalpy	δ_{2y}	Momentum deficiency thickness
C_p	Specific heat at constant pressure	b	Trailing edge diameter
C_v	Specific heat at constant volume	l	Specific stage mechanical energy
R	Universal Gas Constant	μ	Ratio of inlet meridional velocity to outlet meridional velocity $\frac{V_{m2}}{V_{m3}}$
k	Specific Heat Ratio	ν	Ratio of inlet radius to outlet radius $\frac{r_2}{r_3}$
c	Sound speed	σ	solidity
α	Absolute velocity angle	β	Relative velocity angle

Subscript	
1	Station blade inlet condition
2	Station blade outlet or rotating blade inlet condition
3	Rotating blade outlet condition
ax	Axial Component
m	At mean diameter
t	At tip diameter
h	At hub diamter
s	isentropic
p	polytropic
S	stator
R	rotor

Introduction

The project is completed within the course MEEN-646 Aerothermodynamics of Turbomachinery. The report presents a solid aerodynamic and mechanical design of an industrial gas turbine with the given thermos-parameters. The design includes an axial compressor, a combustion chamber and an axial turbine.

Part 1: Aero-Thermo Analysis and Calculations

The first part of this report presents a primary study and calculation for a gas turbine. The calculation includes the blades dimension design for the LP/IP/HP compressors and the turbine, the losses calculation, the nozzle and diffuser calculation, and the combustion study. The data will be used for a mechanical design, which is the second part of this report.

A. Compressor calculation

Given conditions

LP-Compressor

Mass flow	$\dot{m}$	150.0	kg/s
Inlet static pressure	p_{in}	98.61	kPa
Pressure ratio	Π_{LP}	1.8048	
Inlet total temperature	T_{0in}	288.21	K
Exit total temperature	T_{0out}	347.2	K
Inlet mean diameter	D_{m_in}	1.2043	m
Exit mean diameter	D_{m_out}	1.1253	m
Angular velocity	ω	469.35	rad/s

IP-Compressor

Mass flow	$\dot{m}$	150.0	kg/s
Inlet static pressure	p_{in}	177.97	kPa
Pressure ratio	Π_{IP}	1.6739	
Inlet total temperature	T_{0in}	347.02	K
Exit total temperature	T_{0out}	407.51	K
Inlet mean diameter	D_{m_in}	1.1253	m
Exit mean diameter	D_{m_out}	1.0809	m
Angular velocity	ω	469.35	rad/s

HP-Compressor

Mass flow	$\dot{m}$	150.0	kg/s
Inlet static pressure	p_{in}	297.01	kPa
Pressure ratio	π_{HP}	3.0629	
Inlet total temperature	T_{0in}	407.51	K
Exit total temperature	T_{0out}	576.89	K
Inlet mean diameter	D_{m_in}	1.0809	m
Exit mean diameter	D_{m_out}	1.0130	m
Angular velocity	ω	469.35	rad/s

Velocity and H-S diagram for each stage

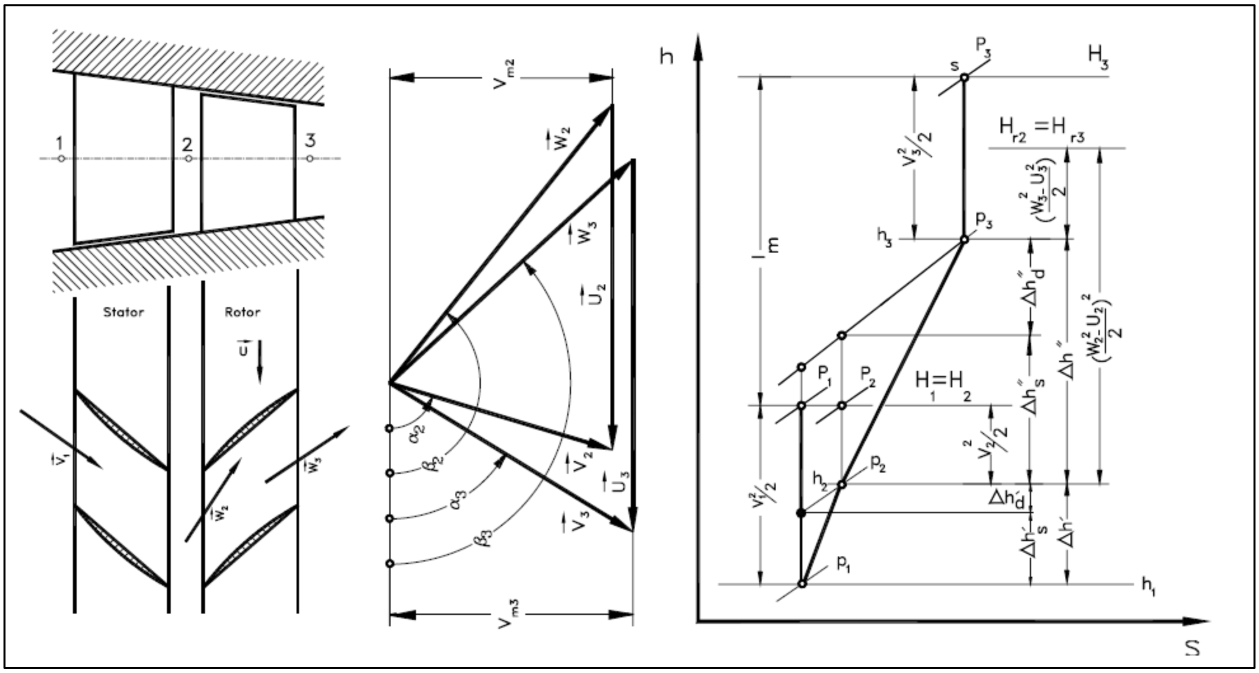

Figure A 1: Compressor stage (left) with the velocity diagram (middle) and the compression h-s diagram (right). P: total pressure, p: static pressure

General Assumptions

1. Compressor working fluid is assumed as ideal gas air.
2. Potential energy is neglected.

Calculation procedures

Step 1: Estimation number of stages for each compressor (LP, IP, and HP)

1. For LP compressor

Assume (only for this estimation)

- Axial velocity (V_{ax}) and density (ρ) are constant through compressor
- At 1st stg rotor, $\alpha_2 = 90\ deg$
- Mean diameter is constant in each stage
- Isentropic efficiency $\eta_s = 0.9$
- Use R, k, C_p as average of inlet and outlet pressure and temperature condition

   a. *Determine the r_{hub}/r_{tip} at the mean diameter for the 1st stage, by selecting r_{hub}/r_{tip} ratio which gives the lowest Mach No. at the tip of rotating blade.*

Select r_hub/r_tip for 1st stg

R	287	J/kg.K
k	1.4	
Cp	1005	J/kg.K

T0_in (degK)	p1 (Pa)	Estimated rho (kg/m3)	Dm_in	r_hub/r_tip	h_b	r_tip	N (rad/sec)	N (rpm)	V2ax (or Vax)	U2_tip	W2_tip	M2_tip
288.21	98610	1.1921	1.2043	0.4	0.5161	0.86021429	469.35	4484.24	64.47	403.74	408.86	1.20
				0.45	0.4568	0.83055172			72.84	389.82	396.57	1.17
				0.5	0.4014	0.80286667			82.89	376.83	385.83	1.13
				0.55	0.3496	0.77696774			95.17	364.67	376.88	1.11
				0.6	0.3011	0.7526875			110.52	353.27	370.16	1.09
				0.65	0.2555	0.72987879			130.25	342.57	366.49	1.08
				0.7	0.2125	0.70841176			156.56	332.49	367.51	1.08
				0.75	0.1720	0.68817143			193.40	322.99	376.47	1.11
				0.8	0.1338	0.66905556			248.66	314.02	400.55	1.18

	r_hub/r_tip	V2 (=V2ax=Vax)	U2 (=U)	W2
At Rm	0.65	130.2501168	282.6191	311.189

Figure A 2: Calculation of r_hub/r_tip for the 1st stage of the LP compressor

List of equations:

(1) Density

$$\rho = \frac{P}{RT}$$

(2) Blade height

$$\mathrm{h}_b = \mathrm{D}_m \frac{1 - \frac{\mathrm{r}_h}{\mathrm{r}_t}}{1 + \frac{\mathrm{r}_h}{\mathrm{r}_t}}$$

(3) Tip radius

$$\mathrm{r}_t = \frac{\mathrm{D}_m}{2} + \frac{\mathrm{h}_b}{2}$$

(4) Axial velocity

$$\mathrm{V}_{ax} = \frac{\dot{m}}{\pi \rho \mathrm{D}_m \mathrm{h}_b}$$

(5) Tip tangential velocity

$$U_t = \omega \mathrm{r}_t$$

(6) Tip relative velocity

$$W_t = \sqrt{{U_t}^2 + {V_{ax}}^2}$$

(7) Mach no

$$Ma = \frac{c}{\sqrt{kRT}}$$

b. Determine the approximate total temperature across the 1st stage

W2	W3 (eq2)	beta2 (eq1)	beta3 (eq3)	alpha3 (eq4)	deltaT0_23 (eq5)	alpha1 (=alpha3)	V1 (eq6)	V3
311.18909	224.0561451	155.2896325	144.4558694	52.40767	28.261329	52.40766605	164.43888	164.4389

List of equations:

(1) At station 2, $\beta_2 = 90 + \tan^{-1}\left(\frac{U_2}{V_2}\right)$

(2) Assume $W_3 = 0.72W_2$ (deHatller)

(3) At station 3, $V_{ax} = W_3 \cos(\beta_2 - 90)$

(4) At station 3, $\tan(\alpha_3) = \frac{V_{ax}}{U_3 - W_3 sin(\beta_3 - 90)}$

(5) Stage total temperature increment

$$\Delta T_{0-23} = \frac{\lambda U V_{ax}}{C_p}[\tan(\beta_2 - 90) - \tan(\beta_3 - 90)]$$

where $\lambda = 1$ for this estimation

(6) Assume $\alpha_1 = \alpha_3$, so $V_1 = \frac{V_{ax}}{\sin \alpha_1}$

c. Stages estimation

	no of stg	=	deltaT0_comp/deltaT0_stg			
		=	2.0873045			
Select	no of stg	=	3		lamda (work load factor)	
Set	deltaT0_stg	=	17	K	0.98	
	deltaT0_stg	=	21	K	0.93	
	deltaT0_stg	=	21	K	0.88	

Figure A 3: Temp distribution for the LP compressor

2. For LP and HP compressors

Assume:

- Axial velocity (V_{ax}) and density (ρ) are constant through compressor
- Mean diameter is constant in each stage
- Isentropic efficiency $\eta_s = 0.9$
- Use R, k, C_p as average of inlet and outlet pressure and temperature condition

a. *Select degree of reaction "r" and a reasonable "α_2" for the 1st stage of the compressor. In this exercise, the following value are chosen*

Compressor	α_2	r
IP-1st stg	55 deg	0.5
HP-1st stg	57 deg	0.5

b. *Solve for $\alpha_3, \beta_2, \beta_3$ using the following relationships, given V_{ax}, U, r, α_2*

$$\frac{U}{V_{ax}} = \tan(90 - \alpha_2) + \tan(\beta_2 - 90)$$

$$\frac{U}{V_{ax}} = \tan(90 - \alpha_3) + \tan(\beta_3 - 90)$$

$$r = \frac{V_x(\tan(\beta_2 - 90) + \tan(\beta_3 - 90))}{2U}$$

c. *Determine the approximate temperature increment in the 1st stage using*

$$\Delta T_{0-stage} = \frac{\lambda \mathrm{V}_{ax} U}{C_p}[\tan(\beta_2 - 90) - \tan(\beta_3 - 90)]$$

d. *Stage temperature distribution for IP and HP compressors:*

No of stage estimation for IP compressor					
	no of stg	=	deltaT0_comp/deltaT0_stg		
		=	2.828383		
Select	no of stg	=	4		
	ave deltaT0	=	15.1225		
					lamda (work load factor)
Set	deltaT0_stg1	=	15	K	0.85
	deltaT0_stg2	=	15	K	0.85
	deltaT0_stg3	=	15	K	0.85
	deltaT0_stg4	=	15.49	K	0.85

Figure A 4: Temp distribution for the IP compressor

No of stage estimation for HP compressor					
	no of stg	=	deltaT0_comp/deltaT0_stg		
		=	8.13		
Select	no of stg	=	9		
	ave deltaT0	=	18.82		
					lamda (work load factor)
Set	deltaT0_stg1	=	19	K	0.83
	deltaT0_stg2	=	19	K	0.83
	deltaT0_stg3	=	19	K	0.83
	deltaT0_stg4	=	19	K	0.83
	deltaT0_stg5	=	19	K	0.83
	deltaT0_stg6	=	19	K	0.83
	deltaT0_stg7	=	19	K	0.83
	deltaT0_stg8	=	19	K	0.83
	deltaT0_stg9	=	17.38	K	0.83

Figure A 5: Temp distribution for the HP compressor

Step 2: Calculate the stage parameters (applicable for all IP, LP, and HP compressor)

Known parameters					
D_m	Stage mean diameter	V_{ax}	Axial velocity	ΔT_0	Stage total temp increment
N	Rotational speed	U	Circumferential velocity	T_{01}	Station 1 total temperature
		r	Degree of reaction	T_{03}	Station 3 total temperature
				P_1	Station 1 static pressure
Unknown parameters					
ρ_1	Station 1 density	α_2	Station 2 relative angle	T_1	Station 1 static temperature
ρ_2	Station 2 density	β_2	Station 2 absolute angle	T_3	Station 3 static temperature
ρ_3	Station 3 density	β_3	Station 3 absolute angle	P_{01}	Station 1 total pressure
		α_3	Station 3 relative angle	P_{03}	Station 3 total pressure
		α_1	Station 1 relative angle	P_3	Station 3 static pressure
		V_2	Station 2 abs. vel.	T_{02}	Station 2 total temperature
		W_2	Station 2 rel. vel.	T_2	Station 2 static temperature
		V_3	Station 3 abs. vel.	P_2	Station 2 static pressure
		W_3	Station 3 rel. vel.	P_{02}	Station 2 total pressure
		V_1	Station 1 abs. vel.		

Note:

- For only the 1st stage of the LP compressor, the degree of reaction will be calculated from stage angle as calculated given $\alpha_2 = 90\ deg$. For the rest of stages, the degree of reaction is given for the stage angle calculation.

List of equations:

(1) Solve for β_2 and β_3 using the relationship

$$\Delta T_{0-stage} = \frac{\lambda V_{ax} U}{C_p}[\tan(\beta_2 - 90) - \tan(\beta_3 - 90)]$$

$$r = \frac{V_{ax}}{2U}[\tan(\beta_2 - 90) + \tan(\beta_3 - 90)]$$

(2) Solve for α_2 and α_3

$$\frac{U}{V_{ax}} = \tan(90 - \alpha_2) + \tan(\beta_2 - 90)$$

$$\frac{U}{V_{ax}} = \tan(90 - \alpha_3) + \tan(\beta_3 - 90)$$

(3) Assume $\alpha_1 = \alpha_3$ (of the previous stage)

(4) Calculate the velocity triangle

$$V_1 = V_{ax}/\cos(90 - \alpha_1)$$
$$V_2 = V_{ax}/\cos(90 - \alpha_2)$$
$$W_2 = V_{ax}/\cos(\beta_2 - 90)$$
$$V_3 = V_{ax}/\cos(90 - \alpha_3)$$
$$W_3 = V_{ax}/\cos(\beta_3 - 90)$$

(5) Static temp at station 1, $T_1 = T_{01} - \frac{1}{2C_p} {V_1}^2$

(6) Static temp at station 3, $T_3 = T_{03} - \frac{1}{2C_p} {V_3}^2$

(7) Gas density at station 1, $\rho_1 = \frac{P_1}{RT_1}$

(8) Total pressure at station 1, $P_{01} = P_1 + \frac{1}{2}\rho_1 {V_1}^2$

(9) Total pressure at station 3,

$$\left(\frac{P_{03}}{P_{01}}\right) = \left(1 + \frac{\eta_s * \Delta T_{OS}}{T_{01}}\right)^{k/(k-1)}$$

(10) Static pressure at station 3

$$P_3 = P_{03}\left[\frac{T_3}{T_{03}}\right]^{k/(k-1)}$$

(11) Gas density at station 3, $\rho_3 = \frac{P_3}{RT_3}$

(12) Total temp at station 2, $T_{02} = T_{01}$

(13) Static temp at station 2, $T_2 = T_{02} - \frac{1}{2C_p} {V_2}^2$

(14) Static pressure at station 2,

$$P_2 = \frac{P_{01}}{\left(\frac{T_{01}}{T_2}\right)^{\frac{k}{k-1}}}$$

(15) Gas density at station 2, $\rho_2 = \frac{P_2}{RT_2}$

(16) Total pressure at station 2, $P_{02} = P_2 + \frac{1}{2}\rho_2 \mathrm{V_2}^2$

Step 3 Iterate the data (dimension and isentropic efficiency, based on losses)

Since the initial guess for η_s is 0.9, the losses are calculated and the blade dimension is iterated 3 times to achieve a converged results. Refer to the losses section to find out the losses calculation.

The final result are presented as below.

The representative velocity triangle for LP, IP and HP compressor stage is listed in the following figures:

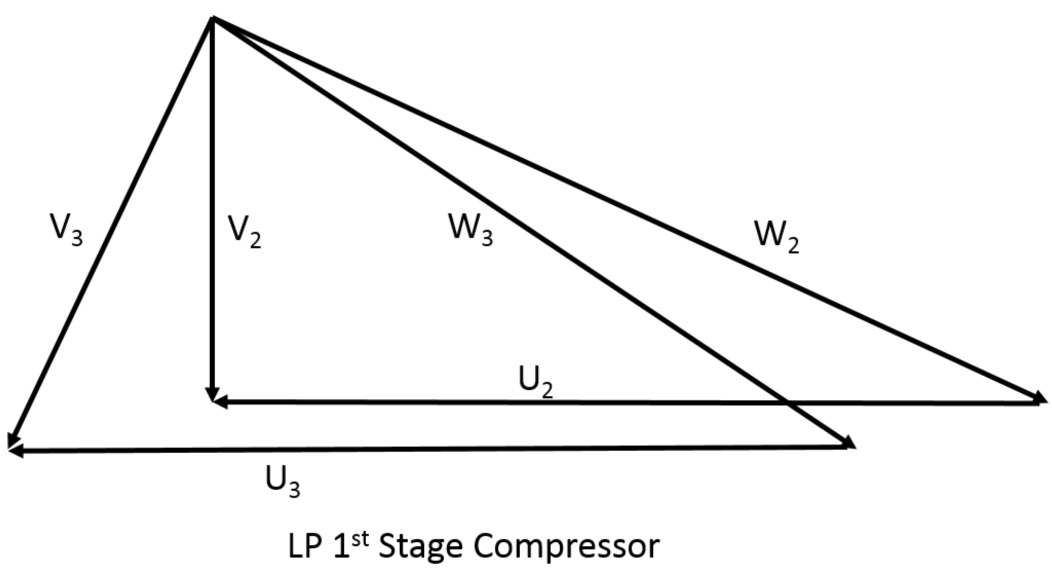

Figure A 6 Velocity Triangle for LP compressor

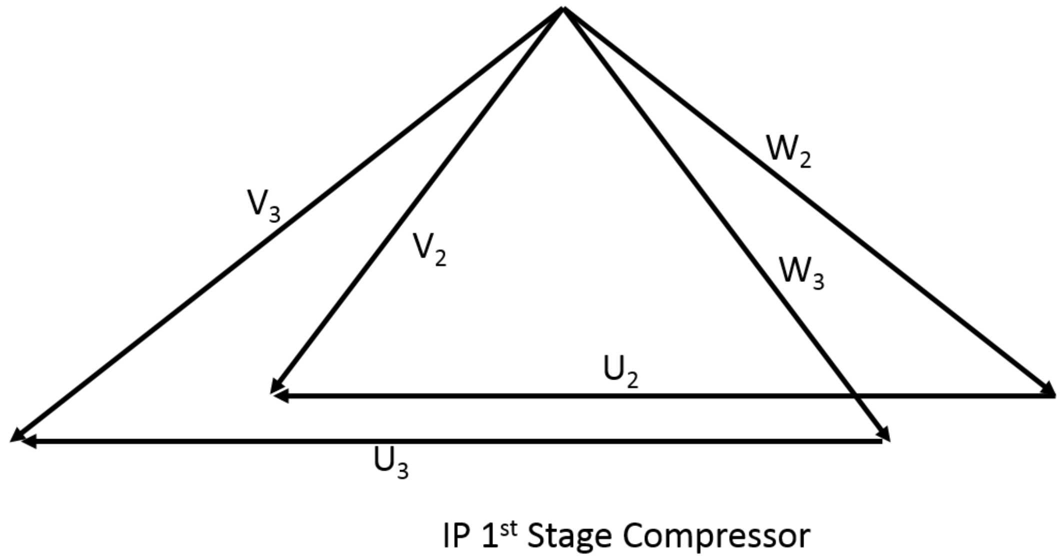

Figure A 7 Velocity Triangle for IP compressor

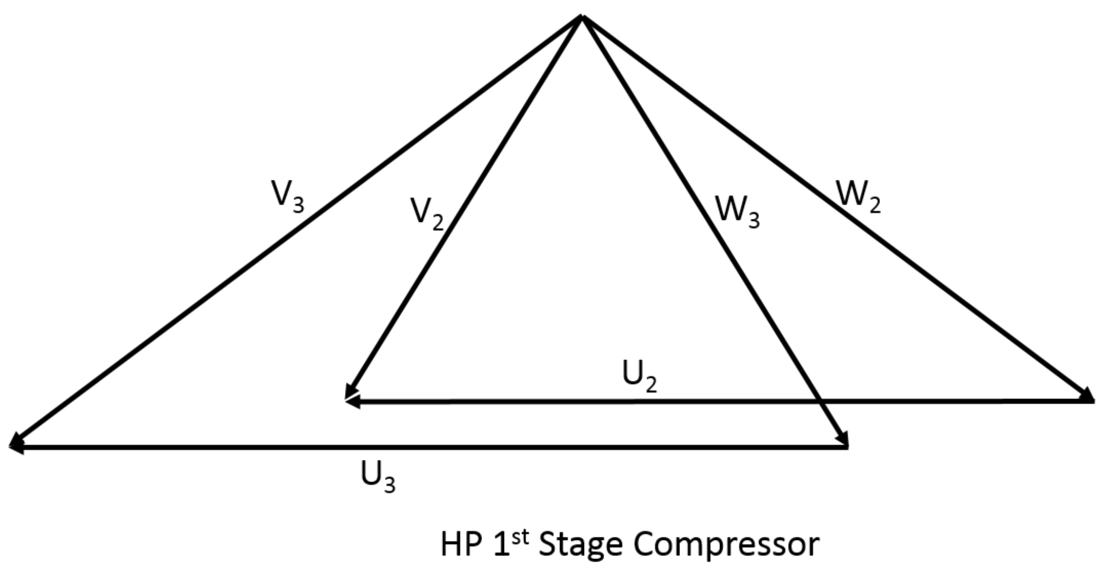

Figure A 8 Velocity Triangle for HP compressor

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Aerodynamic	Mass flow (kg/s)	150	150	150	150	150	150	150	150	150	150	150	150	150	150	150	150
	lamda (work done factor)	0.98	0.93	0.88	0.85	0.85	0.85	0.85	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83
	Dm (m)	1.2043	1.1648	1.1253	1.1253	1.1105	1.0957	1.0809	1.0809	1.0724	1.0639	1.0554	1.0470	1.0385	1.0300	1.0215	1.0130
	N (rad/s)	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35	469.35
	Vax (m/s)	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25	130.25
	U (m/s)	282.62	273.35	264.08	264.08	260.61	257.13	253.66	253.66	251.67	249.68	247.68	245.69	243.70	241.71	239.72	237.73
For mean diameter	alpha1 (deg)	64.65	64.65	49.84	49.43	38.13	38.37	38.59	44.49	36.90	37.18	37.28	37.39	37.49	37.59	37.69	37.79
	alpha2 (deg)	90.00	78.36	81.00	52.94	53.51	54.14	65.15	58.31	58.38	58.82	59.26	59.71	60.17	60.63	61.10	60.18
	beta2 (deg)	155.29	152.18	151.89	141.87	141.63	141.41	146.07	143.10	142.82	142.72	142.61	142.51	142.41	142.31	142.21	141.42
	alpha3 (deg)	64.65	49.84	49.43	38.13	38.37	38.59	44.49	36.90	37.18	37.28	37.39	37.49	37.59	37.69	37.79	38.58
	beta3 (deg)	149.51	141.48	139.55	127.06	126.49	125.86	132.95	121.69	121.62	121.18	120.74	120.29	119.83	119.37	118.90	119.82
	V1 (m/s)	144.12	144.12	170.39	171.41	210.83	209.69	208.69	185.79	216.80	215.39	214.88	214.39	213.90	213.41	212.93	212.45
	V2 (m/s)	130.25	132.98	131.87	163.18	161.97	160.67	143.53	153.04	152.92	152.21	151.51	150.82	150.13	149.44	148.76	150.10
	W2 (m/s)	311.19	278.82	276.12	210.83	209.69	208.69	233.15	216.80	215.39	214.88	214.39	213.90	213.41	212.93	212.45	208.75
	V3 (m/s)	144.12	170.39	171.41	210.83	209.69	208.69	185.79	216.80	215.39	214.88	214.39	213.90	213.41	212.93	212.45	208.75
	W3 (m/s)	256.47	209.04	200.67	163.18	161.97	160.67	177.90	153.04	152.92	152.21	151.51	150.82	150.13	149.44	148.76	150.10
	Deflection in rotor blades (deg)	5.78	10.70	12.33	14.81	15.14	15.55	13.11	21.41	21.20	21.54	21.88	22.23	22.58	22.94	23.31	21.60
	Diffusion	0.824	0.750	0.727	0.774	0.772	0.770	0.763	0.706	0.710	0.708	0.707	0.705	0.703	0.702	0.700	0.719
	Degree of reaction	0.89	0.75	0.75	0.50	0.50	0.50	0.62	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
	Delta whirl velocity (m/s)	61.69	83.02	90.82	67.49	68.05	68.97	72.20	92.95	91.41	92.14	92.88	93.64	94.40	95.18	95.97	88.52
	CL_mean_stator	-1.00	-0.58	-1.47	-0.18	-0.99	-1.01	-1.65	-0.76	-1.43	-1.43	-1.44	-1.46	-1.48	-1.50	-1.52	-1.43
	CL_mean_rotor	-0.79	-0.79	-0.89	-0.51	-0.56	-0.57	-0.69	-0.66	-0.69	-0.70	-0.71	-0.71	-0.72	-0.72	-0.73	-0.70
	CL_tip_stator	-0.85	-0.51	-1.29	-0.20	-0.92	-0.95	-1.51	-0.73	-1.34	-1.35	-1.37	-1.39	-1.41	-1.44	-1.45	-1.34
	CL_tip_rotor	-0.44	-0.59	-0.82	-0.57	-0.65	-0.66	-0.81	-0.90	-0.96	-0.99	-1.01	-1.03	-1.05	-1.07	-1.09	-1.07
	CL_hub_stator	-1.29	-0.74	-1.88	-0.22	-1.17	-1.18	-1.94	-0.86	-1.62	-1.60	-1.60	-1.61	-1.62	-1.63	-1.63	-1.52
	CL_hub_rotor	-1.00	-1.15	-1.48	-0.93	-1.03	-1.01	-1.16	-1.23	-1.28	-1.27	-1.27	-1.26	-1.26	-1.26	-1.27	-1.23
	Flow coefficient	0.51	0.62	0.65	0.80	0.80	0.81	0.73	0.85	0.86	0.86	0.87	0.87	0.88	0.88	0.89	0.88
	Stage load eoefficient	0.21	0.28	0.30	0.22	0.22	0.23	0.25	0.30	0.31	0.32	0.32	0.33	0.33	0.34	0.35	0.33
	Im	17048.31	21139.48	21227.68	15216.06	15261.06	15306.06	15853.29	19511.14	19583.34	19655.54	19727.74	19799.94	19872.14	19944.34	20016.54	18373.09

Figure A 9: Aerodynamic result of the compressors

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Thermodynamic	deltaT0_stg	17	21	21	15	15	15	15.49	19	19	19	19	19	19	19	19	17.38
	T01 (K)	288.21	305.21	326.21	347.02	362.02	377.02	392.02	407.51	426.51	445.51	464.51	483.51	502.51	521.51	540.51	559.51
	T02 (K)	288.21	305.21	326.21	347.02	362.02	377.02	392.02	407.51	426.51	445.51	464.51	483.51	502.51	521.51	540.51	559.51
	T03 (K)	305.21	326.21	347.21	362.02	377.02	392.02	407.51	426.51	445.51	464.51	483.51	502.51	521.51	540.51	559.51	576.89
	T1 (K)	277.88	294.88	311.77	332.40	340.01	355.25	370.46	390.34	403.69	422.99	442.09	461.20	480.30	499.40	518.50	537.60
	T2 (K)	279.77	296.41	317.56	333.84	349.03	364.24	381.82	396.14	415.16	434.26	453.37	472.47	491.57	510.67	529.77	548.57
	T3 (K)	294.88	311.77	332.59	339.91	355.25	370.46	390.42	403.13	422.99	442.09	461.20	480.30	499.40	518.50	537.60	555.74
	P01 (Pa)	111451	131518	163393	205376	237076	270949	308054	342769	400240	462462	531194	606748	689495	779802	878046	984607
	P02 (Pa)	111050	131060	162911	204014	235665	269500	307053	341805	399226	461421	530128	605660	688386	778675	876904	983376
	P03 (Pa)	131518	163393	199311	237076	270949	308054	348437	400240	462462	531194	606748	689495	779802	878046	984607	1089086
	P1 (Pa)	98610	116582	139443	177970	189763	219623	252269	297010	327374	384402	445346	512721	586897	668241	757130	853941
	P2 (Pa)	100439	118720	148722	179124	208379	239880	280672	309886	363550	422180	487155	558794	637468	723549	817414	917712
	P3 (Pa)	116582	139443	177970	189763	219623	252269	297010	327374	384402	445346	512721	586897	668241	757130	853941	909712
	P_ratio	1.1801	1.2424	1.2198	1.1544	1.1429	1.1369	1.1311	1.1677	1.1555	1.1486	1.1422	1.1364	1.1310	1.1260	1.1214	1.1061
	rho1 (kg/m3)	1.236	1.378	1.558	1.866	1.945	2.154	2.373	2.651	2.826	3.166	3.510	3.874	4.258	4.662	5.088	5.535
	rho2 (kg/m3)	1.251	1.396	1.632	1.870	2.080	2.295	2.561	2.726	3.051	3.387	3.744	4.121	4.518	4.937	5.376	5.829
	rho3 (kg/m3)	1.378	1.558	1.864	1.945	2.154	2.373	2.651	2.830	3.166	3.510	3.874	4.258	4.662	5.088	5.535	5.704
	R (J/kgK)	287	287	287	287	287	287	287	287	287	287	287	287	287	287	287	287
	k	1.4	1.4	1.4	1.395	1.395	1.395	1.395	1.39	1.39	1.39	1.39	1.39	1.39	1.39	1.39	1.39
	Cp (kJ/kgK)	1.0028	1.0066	1.0108	1.0144	1.0174	1.0204	1.0235	1.0269	1.0307	1.0345	1.0383	1.0421	1.0459	1.0497	1.0535	1.0571
	(%)	0.821	0.930	0.907	0.960	0.930	0.930	0.898	0.953	0.929	0.930	0.929	0.929	0.929	0.929	0.929	0.924

Figure A 10: Thermodynamics result of the compressors

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Blade dimension	h_b1 (m)	0.24630	0.22857	0.20914	0.17471	0.16984	0.15539	0.14301	0.12798	0.12104	0.10887	0.09900	0.09044	0.08295	0.07638	0.07057	0.06542
	h_b2 (m)	0.24346	0.22562	0.19973	0.17433	0.15877	0.14587	0.13248	0.12449	0.11209	0.10177	0.09281	0.08501	0.07816	0.07213	0.06678	0.06211
	h_b3 (m)	0.22108	0.20204	0.17481	0.16755	0.15332	0.14108	0.12801	0.11992	0.10801	0.09821	0.08971	0.08228	0.07575	0.06999	0.06487	0.06348
	r_h1 (m)	0.47900	0.46811	0.45808	0.47530	0.47033	0.47015	0.46895	0.47646	0.47569	0.47753	0.47822	0.47826	0.47776	0.47680	0.47546	0.47379
	r_h2 (m)	0.48042	0.46959	0.46278	0.47548	0.47587	0.47491	0.47421	0.47821	0.48016	0.48108	0.48131	0.48097	0.48015	0.47892	0.47735	0.47544
	r_h3 (m)	0.49161	0.48138	0.47525	0.47888	0.47859	0.47731	0.47645	0.48049	0.48220	0.48286	0.48286	0.48234	0.48136	0.47999	0.47831	0.47476
	r_t1 (m)	0.72530	0.69669	0.66722	0.65000	0.64017	0.62555	0.61195	0.60444	0.59672	0.58640	0.57722	0.56869	0.56071	0.55318	0.54603	0.53921
	r_t2 (m)	0.72388	0.69521	0.66252	0.64982	0.63463	0.62079	0.60669	0.60269	0.59225	0.58285	0.57413	0.56598	0.55831	0.55105	0.54414	0.53756
	r_t3 (m)	0.71269	0.68342	0.65005	0.64642	0.63191	0.61839	0.60445	0.60041	0.59021	0.58107	0.57257	0.56461	0.55711	0.54998	0.54318	0.53824
	r_h1/r_t1	0.660	0.672	0.687	0.731	0.735	0.752	0.766	0.788	0.797	0.814	0.828	0.841	0.852	0.862	0.871	0.879
	r_h2/r_t2	0.664	0.675	0.699	0.732	0.750	0.765	0.782	0.793	0.811	0.825	0.838	0.850	0.860	0.869	0.877	0.884
	r_h3/r_t3	0.690	0.704	0.731	0.741	0.757	0.772	0.788	0.800	0.817	0.831	0.843	0.854	0.864	0.873	0.881	0.882
	Stagger_mean_stator	88.59	98.35	96.16	124.83	115.30	114.69	105.00	114.01	109.50	109.22	108.92	108.51	108.11	107.82	107.51	108.51
	Stagger_mean_rotor	130.11	120.78	115.15	111.66	110.89	110.06	113.55	102.09	102.12	101.48	100.94	100.29	99.73	99.17	98.50	99.72
	Stagger_tip_stator	89.58	100.32	98.31	129.37	117.56	116.88	106.57	115.99	110.91	110.52	110.07	109.55	109.16	108.70	108.25	109.12
	Stagger_tip_rotor	133.15	127.66	131.88	121.15	118.74	118.43	119.95	115.60	114.32	114.05	113.83	113.63	113.36	113.11	112.77	112.48
	Stagger_hub_stator	67.09	85.69	67.85	118.26	99.29	89.01	93.01	99.55	87.92	87.50	86.75	86.16	85.36	84.73	83.98	85.86
	Stagger_hub_rotor	131.72	119.45	112.43	106.42	104.02	103.43	107.07	94.36	93.32	92.58	91.82	91.14	90.52	89.79	89.14	90.24
	c/h	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
	c1 (m)	0.0821	0.0762	0.0697	0.0582	0.0566	0.0518	0.0477	0.0427	0.0403	0.0363	0.0330	0.0301	0.0277	0.0255	0.0235	0.0218
	c2 (m)	0.0812	0.0752	0.0666	0.0581	0.0529	0.0486	0.0442	0.0415	0.0374	0.0339	0.0309	0.0283	0.0261	0.0240	0.0223	0.0207
	c/s	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
	s1 (m)	0.0821	0.0762	0.0697	0.0582	0.0566	0.0518	0.0477	0.0427	0.0403	0.0363	0.0330	0.0301	0.0277	0.0255	0.0235	0.0218
	s2 (m)	0.0812	0.0752	0.0666	0.0581	0.0529	0.0486	0.0442	0.0415	0.0374	0.0339	0.0309	0.0283	0.0261	0.0240	0.0223	0.0207
	Number of stator baldes	46	48	51	61	62	66	71	80	84	92	100	109	118	127	136	146
	Number of rotary blades	47	49	53	61	66	71	77	82	90	99	107	116	125	135	144	154

Figure A 11: Blade dimension result of the compressors

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Power	dT0 (K)	17.00	21.00	21.00	15.00	15.00	15.00	15.49	19.00	19.00	19.00	19.00	19.00	19.00	19.00	19.00	17.38
	Im	17048.31	21139.48	21227.68	15216.06	15261.06	15306.06	15853.29	19511.14	19583.34	19655.54	19727.74	19799.94	19872.14	19944.34	20016.54	18373.09
	Power (W)	2557247	3170922	3184152	2282409	2289159	2295909	2377993	2926671	2937501	2948331	2959161	2969991	2980821	2991651	3002481	2755964
	Total required power (MW)	44.63036															

Figure A 12: Power result of the compressors

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Degree of reaction at tip	r_h1 (m)	0.47900	0.46811	0.45808	0.47530	0.47033	0.47015	0.46895	0.47646	0.47569	0.47753	0.47822	0.47826	0.47776	0.47680	0.47546	0.47379
	r_h2 (m)	0.48042	0.46959	0.46278	0.47548	0.47587	0.47491	0.47421	0.47821	0.48016	0.48108	0.48131	0.48097	0.48015	0.47892	0.47735	0.47544
	r_h3 (m)	0.49161	0.48138	0.47525	0.47888	0.47859	0.47731	0.47645	0.48049	0.48220	0.48286	0.48286	0.48234	0.48136	0.47999	0.47831	0.47476
	r_m1 (m)	0.60215	0.5824	0.56265	0.56265	0.55525	0.54785	0.54045	0.54045	0.536206	0.531963	0.527719	0.523475	0.519231	0.514988	0.510744	0.5065
	r_m2 (m)	0.60215	0.5824	0.56265	0.56265	0.55525	0.54785	0.54045	0.54045	0.536206	0.531963	0.527719	0.523475	0.519231	0.514988	0.510744	0.5065
	r_m3 (m)	0.60215	0.5824	0.56265	0.56265	0.55525	0.54785	0.54045	0.54045	0.536206	0.531963	0.527719	0.523475	0.519231	0.514988	0.510744	0.5065
	r_t1 (m)	0.72530	0.69669	0.66722	0.65000	0.64017	0.62555	0.61195	0.60444	0.59672	0.58640	0.57722	0.56869	0.56071	0.55318	0.54603	0.53921
	r_t2 (m)	0.72388	0.69521	0.66252	0.64982	0.63463	0.62079	0.60669	0.60269	0.59225	0.58285	0.57413	0.56598	0.55831	0.55105	0.54414	0.53756
	r_t3 (m)	0.71269	0.68342	0.65005	0.64642	0.63191	0.61839	0.60445	0.60041	0.59021	0.58107	0.57257	0.56461	0.55711	0.54998	0.54318	0.53824
	Degree of reaction_h	0.8275	0.6130	0.6228	0.2993	0.3031	0.3211	0.4953	0.3567	0.3647	0.3795	0.3911	0.4010	0.4094	0.4167	0.4230	0.4286

Figure A 13: Degree of reaction at compressor blades

Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Zp	0.0242	0.0146	0.0203	0.0111	0.0236	0.0234	0.0278	0.0146	0.0218	0.0215	0.0213	0.0212	0.0211	0.0210	0.0209	0.0220
Zs	0.1349	0.0379	0.0552	0.0066	0.0239	0.0243	0.0536	0.0130	0.0304	0.0302	0.0305	0.0309	0.0312	0.0315	0.0319	0.0330
Zt	0.0121	0.0073	0.0068	0.0087	0.0086	0.0084	0.0082	0.0060	0.0060	0.0059	0.0058	0.0057	0.0057	0.0056	0.0055	0.0061
Z_leak	0.0082	0.0106	0.0103	0.0138	0.0137	0.0136	0.0120	0.0130	0.0130	0.0129	0.0129	0.0128	0.0127	0.0127	0.0126	0.0127
Z_exit																0.0022
Z	0.1794	0.0703	0.0926	0.0403	0.0699	0.0698	0.1017	0.0466	0.0712	0.0705	0.0706	0.0706	0.0707	0.0708	0.0709	0.0738
isen_eff	0.8206	0.9297	0.9074	0.9597	0.9301	0.9302	0.8983	0.9534	0.9288	0.9295	0.9294	0.9294	0.9293	0.9292	0.9291	0.9240
Stg_avg_isen_eff			0.8859				0.9296									0.9313

Figure A 14: Compressor Losses and Isentropic Efficiency

Polytropic_Eff	
LP compressor	0.8920
IP compressor	0.9332
HP compressor	0.9399
Complete compressor	0.9409

Figure A 15: Compressor Polytropic efficiency

B. Turbine calculation

Assumptions

1. Turbine working fluid is assumed as ideal gas air. R = 287 J/kg K;
2. Cp = 1148 J/kg K for combustion gas specific heat constant. k = 1.333 for isentropic constant.
3. Potential energy is neglected.

Given conditions

Mass flow rate, $\dot{m}$ = 152.97 kg/s

Angular velocity, $\omega = 469.35$ rad/s

The inlet and exit thermodynamics conditions are listed in the below table

Table B 1: Given thermodynamic conditions

	Inlet	Exit
P static pressure (Pa)	873350	102200
T static temperature (K)	1222.7	806.77
Mean line diameter (m)	1.062	1.12

List of equations

The velocity triangles of blades are determined by the following equations:

$$\cot\alpha_2 = \frac{1}{\phi}\left(1 - r + \frac{\lambda}{2}\right)$$

$$\cot\alpha_3 = \frac{1}{\phi}\left(1 - r - \frac{\lambda}{2}\right)$$

$$\cot\beta_2 = \frac{1}{\phi}\left(-r + \frac{\lambda}{2}\right)$$

$$\cot\beta_3 = -\frac{1}{\phi}\left(r + \frac{\lambda}{2}\right)$$

For the turbine blades, we designed a five stages turbine, based on the stage load coefficient λ, degree of reaction r and stage flow coefficient ϕ. For the first stage of the turbine, the inlet velocity angle is $\alpha_1 = 90^{\circ}$. And for the last stage of the turbine, $\alpha_3 = 90^{\circ}$ to minimize the exit loss. For the other stages, $\alpha_1 = \alpha_3$.

The degree of reaction r = 0.5 and stage flow coefficient $\phi = 0.7$ for first four stages based on the usual multistage axial gas turbine for power plant application. As we have fixed the inlet and exit condition, we can estimate the stage load of coefficient λ for the first four stages is 1.55, and for the last stage, we can calculated the $\lambda = 0.8$, based on r = 0.6 and $\alpha_3 = 90^{\circ}$ for last stage.

Based on the above design parameters, we can solve the velocity triangle for each stage. The subscript 1, 2 and 3 represents the inlet of stator, the inlet of rotor and exit of rotor respectively, the detailed values for each stage are attached in Appendix.

The circumferential velocity of the rotor is:

$$U = \omega * D_m/2$$

Dm is the mean diameter of the blade.

V is the absolute velocity:

$$V_3 = U * \phi$$

V_{ax} is the axial velocity is:

$$V_{ax} = V_3 * \sin(\alpha_3)$$

The absolute velocity at station 2 is:

$$V_2 = V_{ax}/\sin(\alpha_2)$$

Because the r=0.5, the velocity triangle is symmetric. Thus we have: $W_2 = V_3$ and $W_3 = V_2$

The isentropic temperature of a turbine is defined as:

$$T_{exit\_s} = T_{in} * (\frac{P_{exit}}{P_{in}})^{\frac{k-1}{k}}$$

The isentropic efficiency is:

$$\eta_s = \frac{T_{in} - T_{exit}}{T_{in} - T_{exit\_s}}$$

The total temperature at station 1 is:

$$T_{01} = T_1 + \frac{{V_1}^2}{2c_p}$$

The total enthalpy is:

$$H_{01} = c_p T_{01}$$

The total pressure at station 1 is:

$$P_{01} = P_1(\frac{T_{01}}{T_1})^{\frac{k}{k-1}}$$

There is no total enthalpy change in the stator, thus, $H_{01} = H_{02}$ and $T_{01} = T_{02}$. The static temperature at station 2 is:

$$T_2 = T_{02} - \frac{{V_2}^2}{2c_p}$$

The Mach number at station 2 is:

$$Ma_2 = \frac{V_2}{\sqrt{kRT_2}}$$

Assume the isentropic efficiency at the stator η_S is 0.9, then the isentropic temperature at station 2 is:

$$T_{2s} = T_{02}(1 - \frac{1 - \frac{T_2}{T_{02}}}{\eta_S})$$

Then the static pressure at station 2 is:

$$P_2 = \frac{P_{01}}{\left(\frac{T_{01}}{T_2}\right)^{\frac{k}{k-1}}}$$

The total pressure at station 2 is:

$$P_{02} = P_2 + \frac{\rho_2 {V_2}^2}{2}$$

The total temperature at station 3 is:

$$T_{03} = T_{01} - \frac{\lambda {U_3}^2}{c_p}$$

The static temperature at station 3 is:

$$T_3 = T_{03} - \frac{{V_3}^2}{2c_p}$$

The isentropic efficiency of the whole stage is assumed as the same as the whole turbine's isentropic efficiency for initial calculation. Based on the previous calculation, $\eta = 0.8199$. The isentropic temperature at station 3 is:

$$T_{3s} = T_1 - \frac{T_1 - T_3}{\eta_S}$$

The static pressure at 3 is:

$$P_3 = P_1(\frac{T_{3s}}{T_1})^{\frac{k}{k-1}}$$

The total pressure at 3 is:

$$P_{03} = P_3 + \frac{\rho_3 {V_3}^2}{2}$$

The gas density at each station is calculated based on ideal gas law:

$$\rho = \frac{P}{RT}$$

The stage annulus flow area is:

$$A = \frac{m}{\rho V_{ax}}$$

The blade height at each station is approximated by the mean diameter and flow area:

$$h_{blade} = \frac{A}{\pi * D_m}$$

The hub diameter and tip diameter can be calculated:

$$D_{hub} = D_m - h_{blade}$$

$$D_{tip} = D_m + h_{blade}$$

Results

Based on the above equations, the velocity triangles for each stage are solved, and the magnitude of the velocity values and angles are listed in below.

Table B 2: Velocity values of all stages

	stage 1			stage 2			stage 3			stage 4			stage 5		
Station	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
V	174.46	378.99	195.05	195.05	380.6	195.88	195.88	382.21	196.7	196.7	383.81	197.53	197.53	269.42	177.41
W	NA	195.05	378.99	NA	195.88	380.6	NA	196.7	382.21	NA	197.53	383.81	NA	184.51	309.37
U	NA	249.22	249.22	NA	250.28	250.28	NA	251.34	251.34	NA	252.39	252.39	NA	253.45	253.45
α	90	27.41	116.57	116.57	27.41	116.57	116.57	27.41	116.57	116.57	27.41	116.57	116.57	41.19	90
β	NA	63.43	152.59	NA	63.43	152.59	NA	63.43	152.59	NA	63.43	152.59	NA	105.95	145.01

For the solidity, chord length, spacing and blade numbers are listed in Table B3.

Table B 3: Geometry parameters for turbine

	Stage 1		Stage 2		Stage 3		Stage 4		Stage 5	
	Stator	**Rotor**	**Stator**	**Rotor**	**Stator**	**Rotor**	**Stator**	**Rotor**	**Stator**	**Rotor**
Solidity c/s	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
chord length c	0.0941	0.1096	0.1269	0.1408	0.0702	0.0832	0.0987	0.1192	0.1352	0.1525
spacing s	0.0855	0.0996	0.1154	0.1280	0.0638	0.0756	0.0898	0.1083	0.1229	0.1386
Blade number N	35.00	30.00	25.00	22.00	43.00	34.00	28.00	21.00	18.00	15.00
Aspect ratio h/c	1.2	1.2	1.2	1.2	3	3	3	3	3	3
Lift coefficient Cl	1.14	4.34	1.26	4.34	1.26	4.34	1.26	4.34	1.5	2.63

The velocity triangles are plotted in Figure B1 and Figure B2.

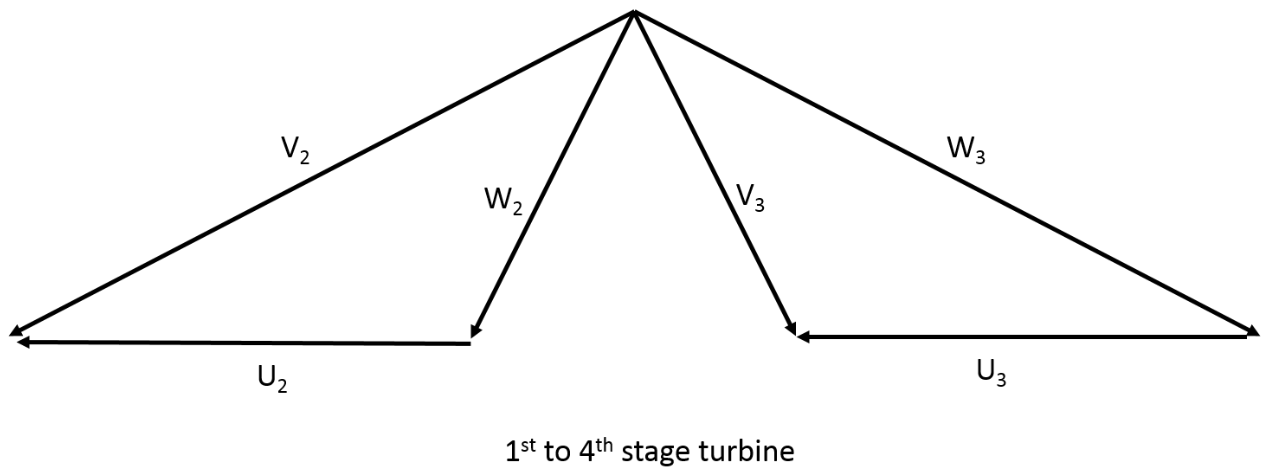

Figure B 1: Velocity triangle of first four stages

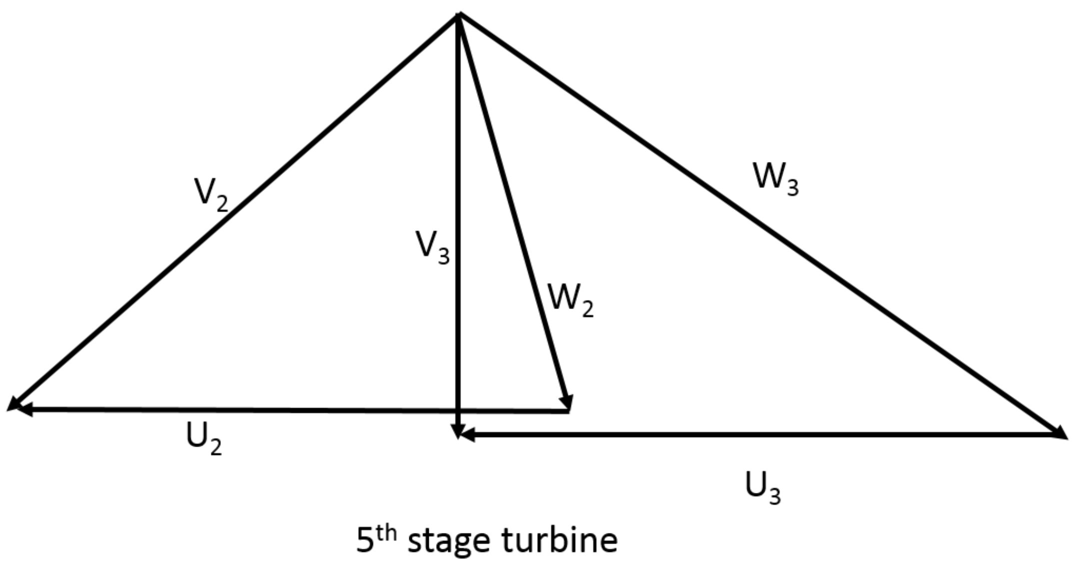

Figure B 2: Velocity triangle of last stage

By now, we have solved the velocity triangle for stator and rotor. Then the thermodynamics properties in one stage from the station 1 to station 3 are solved.

The thermodynamics properties for each stage are listed in Table B3.

Table B 4: Thermodynamics properties of all stages

	stage 1			stage 2			stage 3		
Section NO.	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
Density (kg/m3)	2.49	2.14	1.86	1.86	1.59	1.37	1.37	1.15	0.97

Temperature (K)	1222.70	1173.40	1127.41	1127.41	1080.89	1034.51	1034.51	987.59	940.82
Static Pressure (Pa)	8.734E+05	7.213E+05	6.017E+05	6.017E+05	4.938E+05	4.055E+05	4.055E+05	3.261E+05	2.627E+05
Total Pressure (Pa)	9.119E+05	8.752E+05	6.371E+05	6.379E+05	6.091E+05	4.317E+05	4.324E+05	4.102E+05	2.815E+05
Total Temperature	1235.96	1235.96	1143.98	1143.98	1143.98	1051.22	1051.22	1051.22	957.67
dT0 (K)	91.98			92.76			93.54		
Im (J/kg)	1.056E+05			1.065E+05			1.074E+05		
Power (W)	1.598E+07			1.611E+07			1.625E+07		

	stage 4			stage 5		
Section NO.	**1**	**2**	**3**	**1**	**2**	**3**
Density (kg/m3)	0.97	0.80	0.67	0.67	0.61	0.53
Temperature (K)	940.82	893.51	846.35	846.35	831.72	804.87
Static Pressure (Pa)	2.627E+05	2.062E+05	1.621E+05	1.621E+05	1.455E+05	1.216E+05
Total Pressure (Pa)	2.820E+05	2.654E+05	1.751E+05	1.755E+05	1.676E+05	1.299E+05
Total Temperature	957.67	957.67	863.34	863.34	863.34	818.57
dT0 (K)	94.33			44.76		
Im (J/kg)	1.083E+05			5.139E+04		
Power (W)	1.638E+07			7.775E+06		

The based on the above calculations, the geometry of the turbine hub diameter and tip diameter are calculated. These information are listed in Table B4 and B5.

Table B 5: Geometry information for first 3 stages

	stage 1			stage 2			stage 3		
	1	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
D_hub (m)	0.9576	0.9406	0.9222	0.9279	0.9046	0.8977	0.8839	0.8489	0.8083
D_tip(m)	1.1664	1.1834	1.2018	1.2051	1.2284	1.2353	1.2581	1.2931	1.3337
Annulus area (m2)	0.3485	0.4049	0.4664	0.4644	0.5426	0.5656	0.6297	0.7474	0.8841
Blade height(m)	0.1044	0.1214	0.1398	0.1386	0.1619	0.1688	0.1871	0.2221	0.2627
D_mean (m)	1.0620	1.0620	1.0620	1.0665	1.0665	1.0665	1.0710	1.0710	1.0710

Table B 6: Geometry information for last 2 stages

	stage 4			stage 5		
	1	**2**	**3**	**1**	**2**	**3**
D_hub	0.8149	0.7602	0.6956	0.7033	0.6675	0.6025
D_tip	1.3361	1.3908	1.4554	1.4567	1.4925	1.5575
Annulus area	0.8804	1.0652	1.2836	1.2783	1.3994	1.6201
Blade height	0.2606	0.3153	0.3799	0.3767	0.4125	0.4775
D_mean	1.0755	1.0755	1.0755	1.0800	1.0800	1.0800

Table B 7: Design parameters for all stages

	stage 1	stage 2	stage 3	stage 4	stage 5
Stage flow coefficient ϕ	0.7	0.7	0.7	0.7	0.7
Stage load coefficient λ	1.7	1.7	1.7	1.7	0.8
Degree of reaction r	0.5	0.5	0.5	0.5	0.6

Also, to calculate the losses, we also assumed the following parameters for initial estimation.

The aspect ratio is assumed to be 3, aka, h/c=3, the solidity c/s=1.1.

For turbine losses, we considered five types of losses happened in the turbine:

1. Primary loss.
2. Endwall friction and secondary loss.
3. Trailing edge loss.
4. Exit loss for last stage.
5. Shrouded blade clearance leakage loss.

The detailed losses equations are as the same as for the compressor. Thus, the equations will not be listed here. The final losses and turbine efficiency are listed in Table 5.

Table B 8: Stage losses and efficiency for the turbine

	Stage 1	Stage 2	Stage 3	Stage 4	Stage 5
Z_P	0.0648	0.0669	0.0669	0.0669	0.0546
Z_S	0.0824	0.0862	0.0407	0.0402	0.0300
Z_{LC}	0.0182	0.0135	0.0098	0.0069	0.0042
Z_T	0.0146	0.0145	0.0166	0.0165	0.0170
Z_E					0.30625
Z	0.18	0.1811	0.1340	0.1305	0.4121
Z_s	0.1526	0.1533	0.1181	0.1155	0.2918
η_S	0.8644	0.8564	0.8564	0.8564	0.6491
η_R	0.8245	0.8311	0.9017	0.9065	0.7380
η_{stage}	0.8474	0.8467	0.8819	0.8845	0.7082

After 5 times of iteration of the efficiency for each stage, we obtained the overall isentropic efficiency, the total power output and the polytropic efficiency for this 5 stages turbine.

$$P = 72.5\ MW,\ \eta_s=0.8786,\ \eta_p = 0.849.$$

C. Loss Calculations

The losses considered were:

- Primary loss
- Trailing edge loss
- Endwall friction and secondary loss
- Shrouded blade clearance leakage loss
- Exit loss for last stage

Primary losses

Profile loss coefficient is calculated using (Eq 7.1) [1]

$$\zeta_p = \left(\epsilon_{opt}\right)_{single}\left[1 + K\left(\frac{c}{s}\right)^3\right] C_L \frac{c}{s}\frac{1}{\sin(\alpha_\infty)}$$

Where,

$\left(\epsilon_{opt}\right)_{single} = 0.0115$ for (t_{max}/c) = 0.15 for turbine and (t_{max}/c) = 0.1 for compressor and $K = 0.11$.
Lift coefficient can be calculated using (Eq 6.59) and for the rotor using (Eq. 6.69) [1]

$$C_L \frac{c}{s} = \frac{\sin^2 \alpha_2}{\sin \alpha_\infty}\,(1+\mu)(cot\alpha_2 - \mu v cot\alpha_1) \text{ (For stator)}$$

$$C_L \frac{c}{s} = \frac{\sin^2 \alpha_2}{\sin \alpha_\infty}\,(1+\mu)\left(cot\beta_3 - \mu v cot\beta_2 + \frac{v^2-1}{\phi}\right) \text{ (For rotor)}$$

Trailing edge loss

Trailing edge loss for both compressor and turbine can be calculated using (Eq 7.53) [1]

$$\zeta_t = \frac{G_1^2 - 2G_2 + 1}{G_1^2} - \cos^2(\alpha_3)\left\{\frac{2G_1^2 - 2G_2 + 1}{G_1^2} - \frac{G_1^2}{G_2^2}\right\}$$

Where $G_1 = 1 - D - \Delta_1$, $G_2 = 1 - D - \Delta_1 - \Delta_2$

And $\Delta_1 = \frac{\delta_{1y}}{s}$, $\Delta_2 = \frac{\delta_{2y}}{s}$, $D = \frac{d}{s} = \frac{b}{s * sin\alpha_2}$

Endwall friction and secondary flow loss

Endwall friction and secondary flow loss for both compressor and turbine is calculated using (Eq 7.83) [1]

$$\zeta_{sf}\frac{h}{c} = 2c_f\left[K_1 + K_2\left(C_L\frac{c}{s}\right)^2 \frac{sin\alpha_\infty}{sin\alpha_2}\right]$$

Where $K_1 = 4.65$ and $K_2 = 0.675$.

$c_{f_{FP}} = 2\left(\frac{\delta_2}{c}\right)_{FP} = 0.074 Re_C^{-\frac{1}{5}}$, $c_f = c_{f_{FP}} f^{0.8}$, $f = 0.2\sum_{n=0}^{4}\left(\frac{V_1}{V_2}\right)^n$

Flow loss in shrouded blades

Mass leakage for stator ($\frac{\dot{m}'}{m}$) and rotor ($\frac{\dot{m}''}{m}$) for both compressor and turbine is calculated as follows (Eqs 7.102, 7.103) [1]:

$$\frac{\dot{m}'}{m} = \frac{\alpha' D' C'}{D_m h} \sqrt{\frac{2r\lambda}{n'\phi^2}}$$

$$\frac{\dot{m}''}{m} = \frac{\alpha'' D'' C''}{D_m h} \sqrt{\frac{2r\lambda}{n''\phi^2}}$$

The total loss due mass flow and mixing is:

$$Z_{LC} = \frac{\dot{m}'}{m}\frac{\phi^2}{\lambda}\frac{1}{sin(\alpha_2^2)} + \frac{\dot{m}''}{m}\frac{\phi^2}{\lambda}\frac{1}{sin\beta_3^2} + \frac{\dot{m}''}{m}$$

Exit loss

Exit loss for final stage can be calculated using (Eq 7.106) [1]

$$Z_E = \frac{V_3^2}{2\lambda U_3^2}$$

Total stage loss coefficient

After calculating the individual loss for stator and rotor, they are summed as follows

$$\zeta' = \zeta_p' + \zeta_t' + \zeta_{sf}'$$

$$\zeta'' = \zeta_p'' + \zeta_t'' + \zeta_{sf}''$$

Where $'$ represent stator and $''$ represents the rotor. The total stage loss coefficient is (Eq 7.34) [1]:

$$Z = \zeta' \frac{V_2^2}{2l} + \zeta'' \frac{W_3^2}{2l}$$

The isentropic stage efficiency of turbine is (Eq 7.131) [1]

$$\eta_s = \frac{1}{1+Z}$$

Isentropic stage efficiency of compressor is (Eq 7.133) [1]

$$\eta_s = 1 - Z$$

Polytropic Efficiency calculation

Polytropic efficiency for a compressor can be calculated using the following equations (Eqs 8.33, 8.34, 8.35) [1]:

$$\overline{\eta_s} = \overline{\eta_p}\frac{1}{1+f_{\infty_C}}$$

$$1+f_{\infty_C} = \frac{\overline{\eta_p}\left(\left(\frac{p_2}{p_1}\right)^{\frac{1}{\overline{\eta_p}}\frac{\overline{\kappa}-1}{\overline{\kappa}}}-1\right)}{\left(\left(\frac{p_2}{p_1}\right)^{\frac{\overline{\kappa}-1}{\overline{\kappa}}}-1\right)}$$

$$f_c = \frac{f_{\infty_C}(N-1)}{N}$$

Where, N=number of stages, $\overline{\eta_s}$=average isentropic efficiency of the stages, $\overline{\eta_p}$ = average polytropic efficiency.

Polytropic efficiency for a turbine can be calculated using (Eqs 8.25, 8.29, 8.30) [1]:

$$\overline{\eta_s} = \overline{\eta_p}(1+f_{\infty_T})$$

$$1+f_{\infty_T} = \frac{1}{\overline{\eta_p}}\frac{\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{1}{\overline{\eta_p}}\frac{\overline{\kappa}-1}{\overline{\kappa}}}\right)}{\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{\overline{\kappa}-1}{\overline{\kappa}}}\right)}$$

$$f_T = \frac{f_{\infty_T}(N-1)}{N}$$

D. Nozzle and Diffuser calculation and Combustion selection

Introduction

Inlet nozzle is assembled before compressor and used to transport mass and accelerate the mass flow. Exit diffuser is assembled after the last stage of turbine and used to decelerate mass flow and reduce flow kinetic energy. As for the compressor chamber, which is installed between compressor and turbine and make combustion happen. In this module, the design of inlet nozzle, exit diffuser and combustion chamber are performed. The primary design include the calculation of basic thermal parameters, basic components dimensions.

Assumptions and given conditions

1. Dry air is regarded as working medium, ideal gas law is applicable.
2. Potential energy is neglected.
3. Cp and k is constant. Cp=1 kJ/kgK, k=1.4.
4. Flow is inviscid through nozzle and diffuser. So Bernoulli equation is applicable.
5. Air is incompressible in nozzle.
6. There is a 5% pressure loss in nozzle passage and the total pressure loss coefficient along the diffuser K = 2.5.

List of Equations

(1) Ideal gas equation:

$$\mathrm{P} = \rho \mathrm{RT}$$

(2) Mass balance:

$$\rho_0 V_0 A_0 = \rho_1 V_1 A_1$$

(3) For compressible flow:

$$\frac{T}{T_0} = \left(\frac{P}{P_0}\right)^{\frac{k-1}{k}}$$

(4) Speed of sound at a certain temperature:

$$c = \sqrt{kRT}$$

(5) Total Temperature:

$$T_0 = T + \frac{V^2}{2C_P}$$

(6) Total Pressure:

$$P_0 = P + \frac{1}{2}\rho V^2$$

(7) Total pressure drop coefficient:

$$K = \frac{P_{t,1} - P_{t,2}}{P_{t,1} - p_1}$$

Assume K = 2.5, then the total pressure ($P_{t,2}$) at the diffuser can be obtain.

Results and Discussion

Based on the above equations, the velocity and thermodynamics conditions are calculated for the inlet of the nozzle and the outlet of the diffuser. Thus, the area changing across the nozzle and diffuser can be designed. This will help for the future mechanical design of the gas turbine engine.

Figure D 1 Inlet nozzle design

Result at the inlet nozzle are listed in Table D1.

Table D 1: Calculated result for nozzle inlet

p_0	101325pa
v_0	67.03m/s
T_0	296.14K
Ma_0	0.197
A_0	1.877m^2
ρ_0	1.19kg/m3
Length	0.8015m
$D_{0,i}$	0.958
$D_{0,o}$	1.8187

p_1	98610pa
v_1	135.02m/s
T_1	288.21K
Ma_1	0.3968
A_1	0.932m^2
ρ_1	1.192kg/m3
Length	0.8015m
$D_{1,i}$	0.958m
$D_{1,o}$	1.4506m

Result at the exit of diffuser are listed in Table D2.

Table D 2: Calculated result for diffuser exit

p_2	118.3Kpa
v_2	178.12m/s
T_2	804.87K
Ma_2	0.3185
$Area_2$	1.6655m^2
ρ_2	0.51kg/m3
$\dot{m}$	151.3kg/s
$D_{2,o}$	1.775m
$D_{2,i}$	0.958m
Length	1m

p_3	102.2Kpa
v_3	124.67/s
T_3	775.03K
Ma_3	0.22
$Area_3$	2.6412m^2
ρ_3	0.4595kg/m3

$D_{3,o}$	2.3m
$D_{3,i}$	0.958m
Length	1m

So far, thermal parameters at the exit of diffuser have been figured out and listed in Table D2. It shows that air exit diffuser with temperature at 758K (485C) and velocity at 62.38m/s, which are close to real condition. Compared with nozzle length, diffuser length (Dl) is a critical parameter to consider, which has an obvious effect on flow separation.

For power generation gas turbine, the fuel is CH_4. It has great storage and great heating value. Tubular combustor is selected and shown as below. Based on the given conditions, the pressure loss coefficient of combustion chamber is calculated as:

$$\zeta_{CC} = \frac{\text{Pccin} - \text{Pccout}}{\text{Pccin}}$$

Inserted in numbers we have $\zeta_{CC} = 0.0397$.

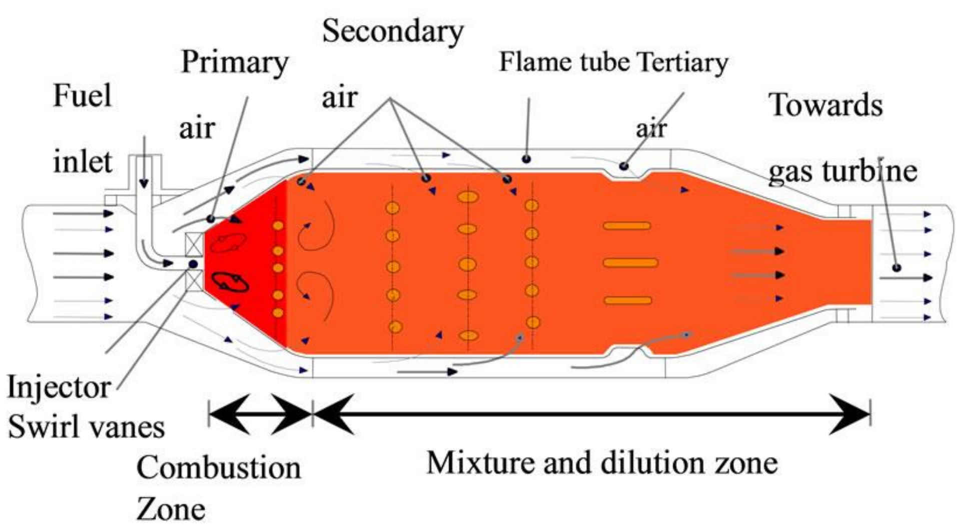

Figure D 2 Tubular combustor

Based on energy equation of the combustion chamber, we have:

$$m_{air} c_p T_{ccin} + m_{fuel} H = (m_{air} + m_{fuel}) c_p T_{ccout} + Q_{loss}$$

Where H is the high heating value of the fuel. Here for CH_4, the HHV is 55.5 MJ/kg. From the above, we can calculate the combustion chamber efficiency:

$$\eta_{cc} = \frac{(m_{air} + m_{fuel}) c_p T_{ccout}}{m_{air} c_p T_{ccin} + m_{fuel} H}$$

Inserted in the numbers, we have $\eta_{cc} = 0.9793$.

Combustion chamber design mainly focus on configuration design and dimension design, which rely on the resources on public domain (internet).

In this part, calculation of inlet nozzle, exit diffuser and combustion chamber are performed. The calculation results are continues and reasonable. For optimization, CFD will be employed to determine configuration and analysis flow field.

Part 2: Solid – Mechanical and CFD design

The second part of the report presents the mechanical design of a gas turbine. By employing Solidworks software, a complete and actual 3D model of gas turbine is generated which mainly consists of a 16-stages compressor, tubular combustion chamber and a 5-stage turbine, inlet nozzle and diffuser. The Configuration of compressor casing, combustion chamber and turbine casing are referred to the design of SIMENS-SGT750 and SIMENS-SGT800. However, geometry dimensions and of all blades and relative matching parameters exactly follow the calculation results of aero-thermal analysis.

In addition, CFD simulation of compressor and turbine blades are performed to obtain a better understanding of flow field in flow passage. Rotor blade stress analysis is done to select suitable material for the rotor blades.

E. Compressor Mechanical Design

This 16-stage compressor consists of 3-stage low-pressure compressor, 4-stage intermediate-pressure compressor and 9-stage high pressure compressor. The blades in the first four stage are twisted and follow radial equilibrium due to large blade height.

E.1 Compressor blade design

Table E 1 gives important dimensions of stator and rotor includes: blade height, blade chord length, blade spacing and blade radius height etc. Based on these given data, the 3D model of stator and rotor are generated. The actual dimensions in the Solidworks file might vary slightly due to accounting for tolerances and clearances. Figure E 1 displays the rotor and stator of compressor stage 5. The rotor blades were made detachable from the rotor shaft. This is was done by placing the blade on a fir tree feature. Figure E 1 shows the fir tree feature for rotor stage 5.

Table E 1: Blade dimensions

	Comp	LP	LP	LP	IP	IP	IP	IP	HP	HP	HP	HP	HP	HP	HP	HP	HP
	Stg	1	2	3	1	2	3	4	1	2	3	4	5	6	7	8	9
Blade dimension	h_b1 (m)	0.24630	0.22857	0.20914	0.17471	0.16984	0.15539	0.14301	0.12798	0.12104	0.10887	0.09900	0.09044	0.08295	0.07638	0.07057	0.06542
	h_b2 (m)	0.24346	0.22562	0.19973	0.17433	0.15877	0.14587	0.13248	0.12449	0.11209	0.10177	0.09281	0.08501	0.07816	0.07213	0.06678	0.06211
	h_b3 (m)	0.22108	0.20204	0.17481	0.16755	0.15332	0.14108	0.12801	0.11992	0.10801	0.09821	0.08971	0.08228	0.07575	0.06999	0.06487	0.06348
	r_h1 (m)	0.47900	0.46811	0.45808	0.47530	0.47033	0.47015	0.46895	0.47646	0.47569	0.47753	0.47822	0.47826	0.47776	0.47680	0.47546	0.47379
	r_h2 (m)	0.48042	0.46959	0.46278	0.47548	0.47587	0.47491	0.47421	0.47821	0.48016	0.48108	0.48131	0.48097	0.48015	0.47892	0.47735	0.47544
	r_h3 (m)	0.49161	0.48138	0.47525	0.47888	0.47859	0.47731	0.47645	0.48049	0.48220	0.48286	0.48286	0.48234	0.48136	0.47999	0.47831	0.47476
	r_t1 (m)	0.72530	0.69669	0.66722	0.65000	0.64017	0.62555	0.61195	0.60444	0.59672	0.58640	0.57722	0.56869	0.56071	0.55318	0.54603	0.53921
	r_t2 (m)	0.72388	0.69521	0.66252	0.64982	0.63463	0.62079	0.60669	0.60269	0.59225	0.58285	0.57413	0.56598	0.55831	0.55105	0.54414	0.53756
	r_t3 (m)	0.71269	0.68342	0.65005	0.64642	0.63191	0.61839	0.60445	0.60041	0.59021	0.58107	0.57257	0.56461	0.55711	0.54998	0.54318	0.53824
	r_h1/r_t1	0.660	0.672	0.687	0.731	0.735	0.752	0.766	0.788	0.797	0.814	0.828	0.841	0.852	0.862	0.871	0.879
	r_h2/r_t2	0.664	0.675	0.699	0.732	0.750	0.765	0.782	0.793	0.811	0.825	0.838	0.850	0.860	0.869	0.877	0.884
	r_h3/r_t3	0.690	0.704	0.731	0.741	0.757	0.772	0.788	0.800	0.817	0.831	0.843	0.854	0.864	0.873	0.881	0.882
	c/h	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33	0.33
	c1 (m)	0.0821	0.0762	0.0697	0.0582	0.0566	0.0518	0.0477	0.0427	0.0403	0.0363	0.0330	0.0301	0.0277	0.0255	0.0235	0.0218
	c2 (m)	0.0812	0.0752	0.0666	0.0581	0.0529	0.0486	0.0442	0.0415	0.0374	0.0339	0.0309	0.0283	0.0261	0.0240	0.0223	0.0207
	c/s	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
	s1 (m)	0.0821	0.0762	0.0697	0.0582	0.0566	0.0518	0.0477	0.0427	0.0403	0.0363	0.0330	0.0301	0.0277	0.0255	0.0235	0.0218
	s2 (m)	0.0812	0.0752	0.0666	0.0581	0.0529	0.0486	0.0442	0.0415	0.0374	0.0339	0.0309	0.0283	0.0261	0.0240	0.0223	0.0207
	Number of stator baldes	46	48	51	61	62	66	71	80	84	92	100	109	118	127	136	146
	Number of rotary blades	47	49	53	61	66	71	77	82	90	99	107	116	125	135	144	154

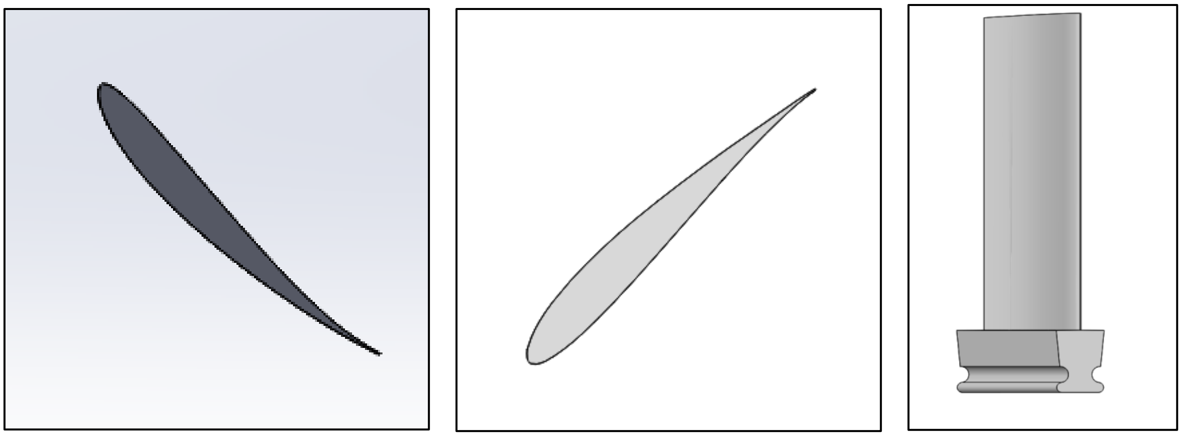

Figure E 1: Rotor blade (on left), stator blade (middle), and blade with fir tree feature (on right)

Compressor blades of stages 1 until 4 were twisted due to large blade height. Figure E 2 shows one example of the twist and it shows the rotor blade of stage 3.

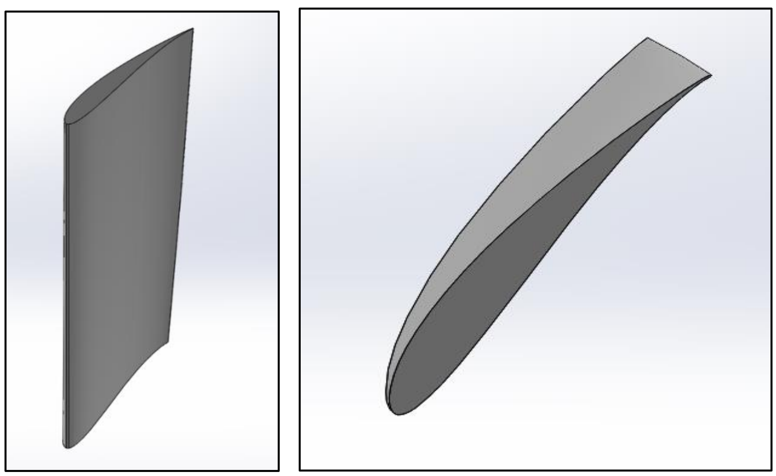

Figure E 2: Twisted rotor blade of stage 3

E.2 Compressor rotor assembly

Figure E 5 shows the compressor rotor assembly after fitting all the fir tree of the rotor blades with the shaft. The fir tree features present on the blades were extrude cut on the shaft to fit the rotor blades on the shaft. With a configuration of shell and wheel design, the compressor shaft is stiff while being light.

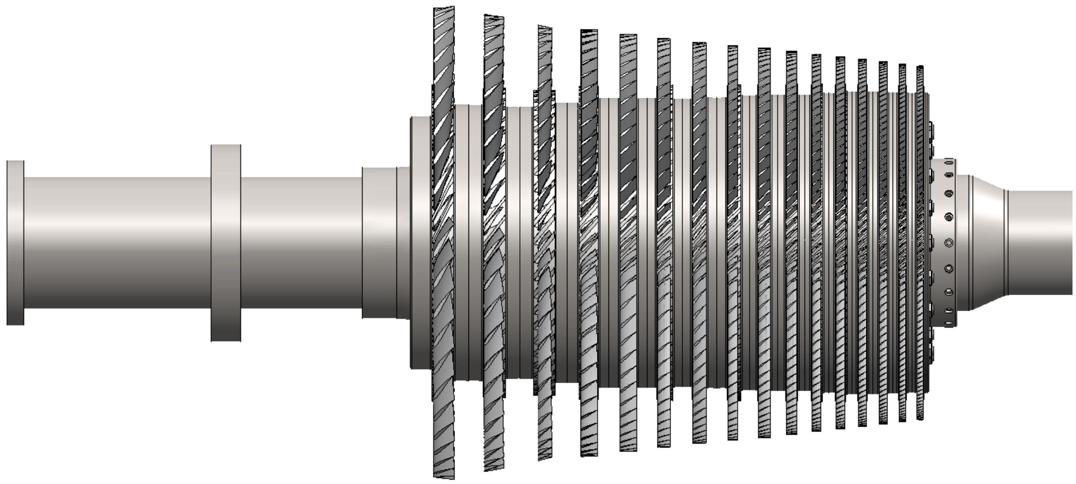

Figure E 3: Compressor rotor

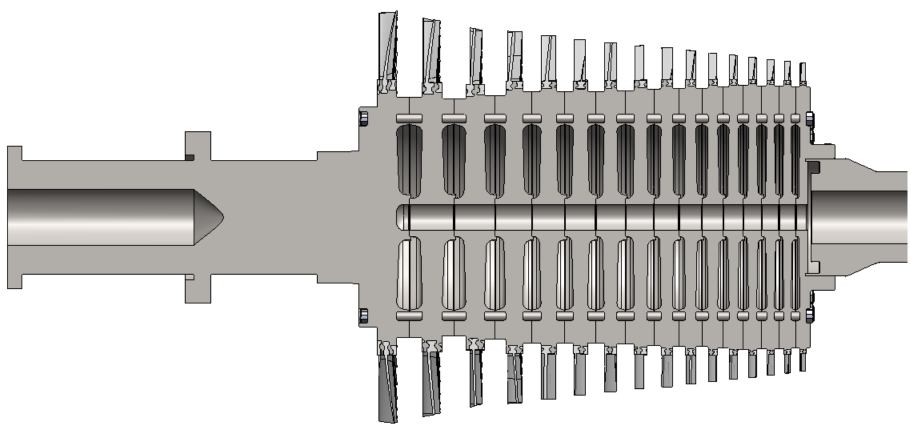

Figure E 4: Compressor cross sectional view

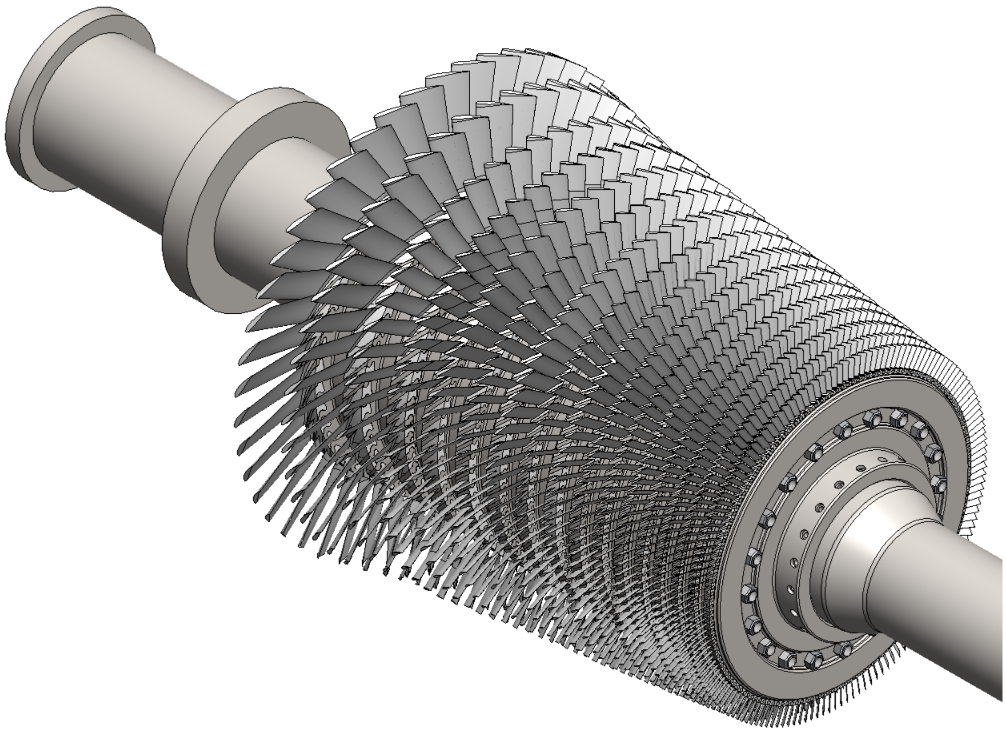

Figure E 5: Compressor shaft assembly with blades

E.3 Compressor casing design

The dimensions of casing are determined by the stator data in. The casing design of this 16-stage compressor are divided in two part: LPIP casing and HP casing. The LPIP casing has two vents for diverting some of the working medium. This is necessary during startup and this air can also be used for other processes. For ease of maintenance and disassembly, the compressor casing is horizontally split. The LPIP compressor casing also contains IGV (Inlet Guide Vane) at the first and second stages. This mechanism can rotate the stator blades that are fixed to the casing and thus change the stagger angle. Doing this, can improve performance at off design conditions and help avoid rotating stall and surge. Figure E 6 shows the LPIP casing.

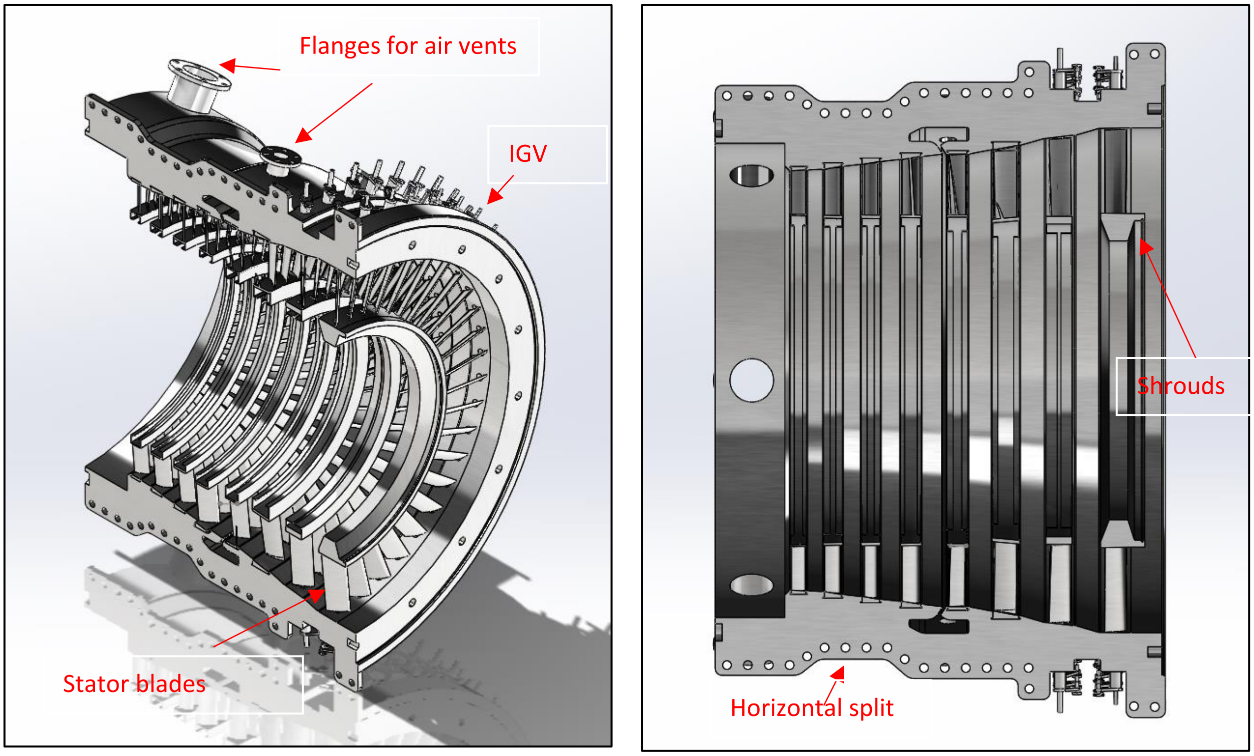

Figure E 6: LP-IP compressor casing

The HP compressor casing is also horizontal split. The combustion is chamber is placed significantly higher the compressor blades and is at angle. Due to this, structural support is required and it is provided by the HP compressor casing. HP compressor casing is shown in Figure E 7.

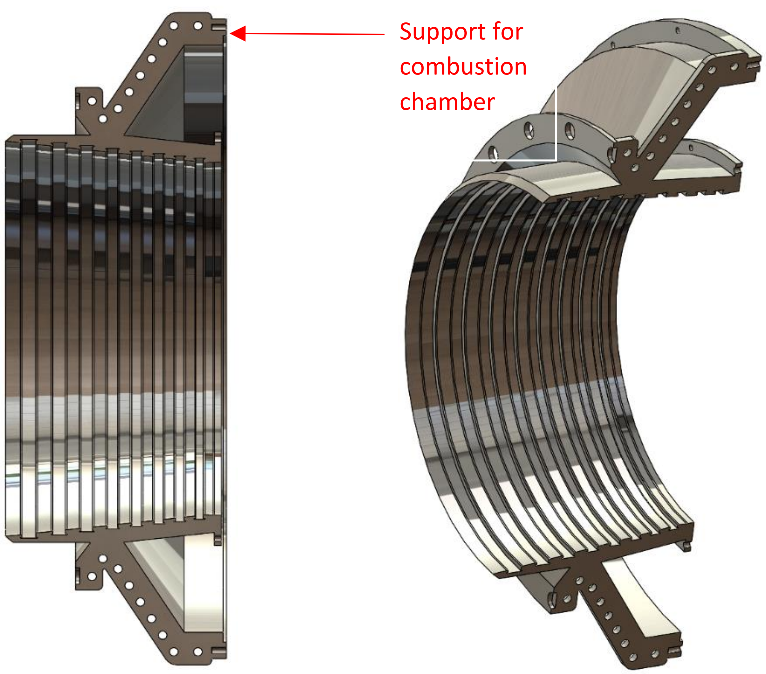

Figure E 7: HP compressor casing

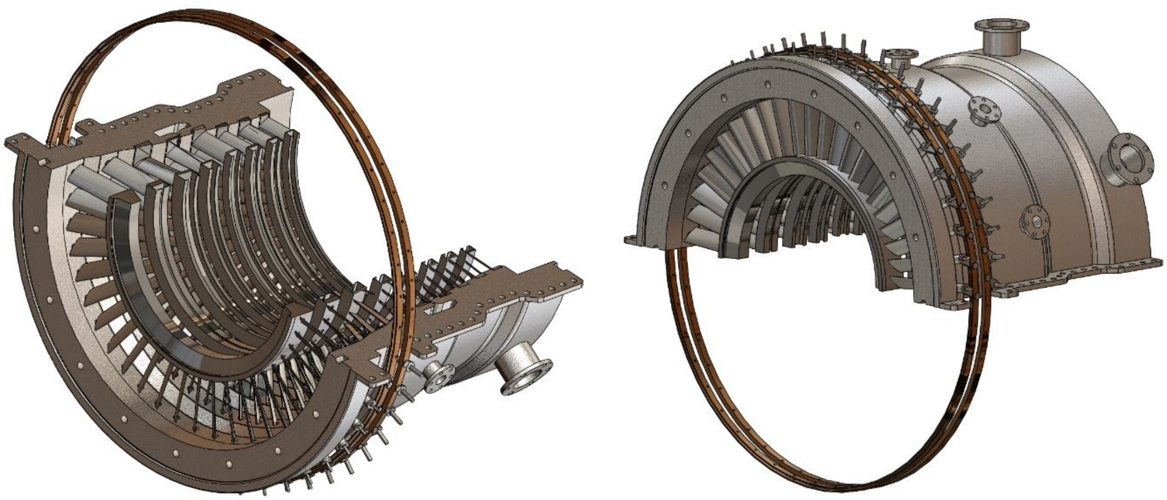

Figure E 8: LP-IP Compressor casing with IGV

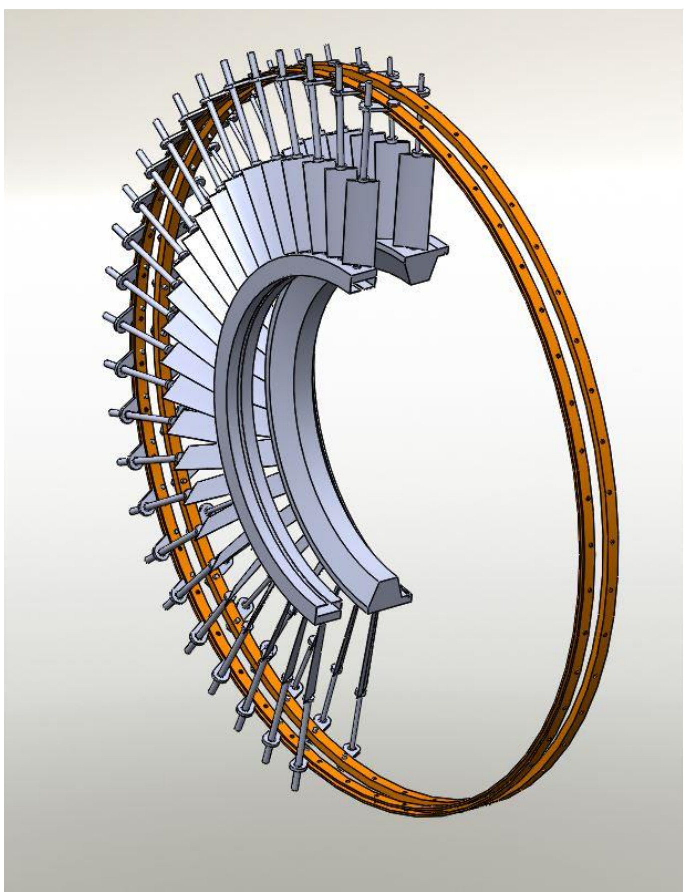

Figure E 9: IGV on 1st and 2nd stages of the LP compressor

Figure E 10: Full compressor casing (LP-IP-HP)

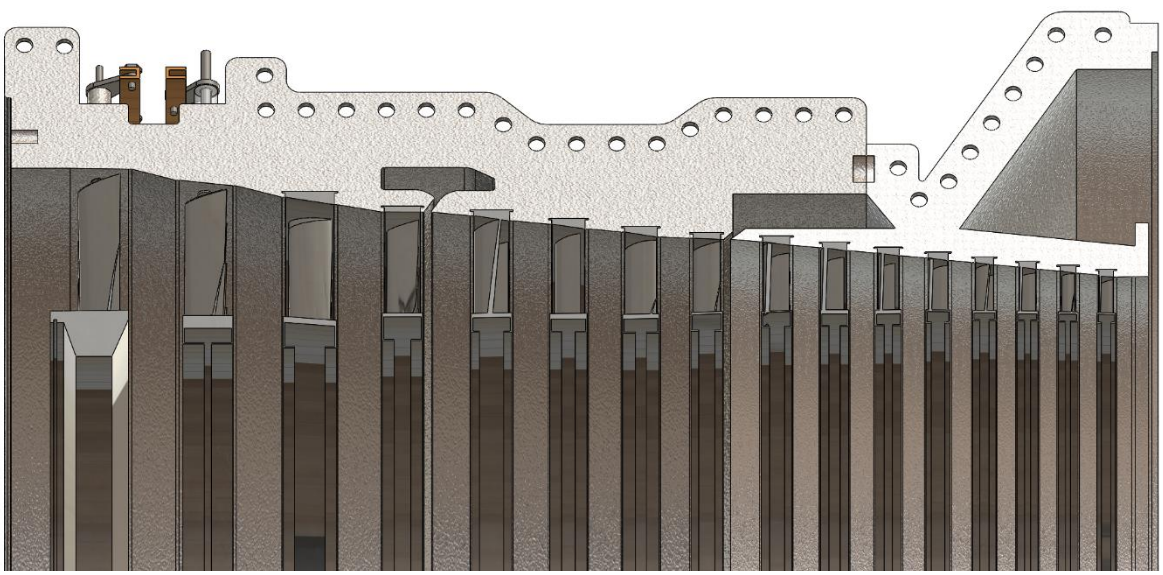

Figure E 11: Detail view of Compressor casing

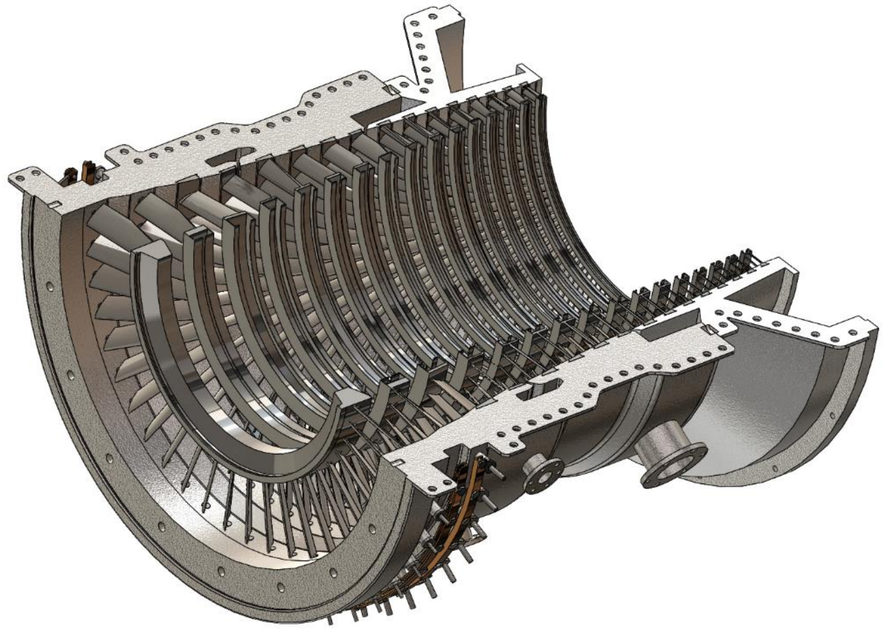

Figure E 12: Cross sectional view of compressor casing

F. Turbine Mechanical Design

F.1 Turbine Blade design

The turbine blade design is based on the previous methodology we learned from class. The Bezier function and base profile of airfoil is superimposed together to get a complete blade cross section. Also, we designed the shrouded blade seals for the turbine stator blade and rotor blade to reduce the stage leakage. Fir tree features are applied on the rotor blades for installation, while the stator blade are installed on the inner casing of the turbine with a circumferential insertion feature.

The turbine has 5 stages. For radial equilibrium reason, we twisted the last two stages blades because of the lower hub tip ratio. The velocity angle at the hub and tip are determined based on the following equation:

$$V_u R = constant$$

$$\frac{R}{R_m} = \frac{\cot(\alpha_m)}{\cot(\alpha)}$$

The twisted angles for stage 4 and 5 are listed in Table F 1. The velocity triangle at hub and tip are plotted respectively.

Table F 1: Twisted angles for stage 4 and 5

	stage 4			stage 5		
	1	2	3	1	2	3
α at hub	123.6628	19.90803	128.1373	127.9406	27.98742	90
α at mean	116.57	27.41	116.57	116.57	41.19	90.00
α at tip	111.8137	34.01477	110.1425	110.2041	50.62658	90
β at hub		29.55571	149.4591		44.59931	127.9286
β at mean		63.43	152.59		105.95	145.01
β at tip		110.6884	156.63		139.4522	154.3002

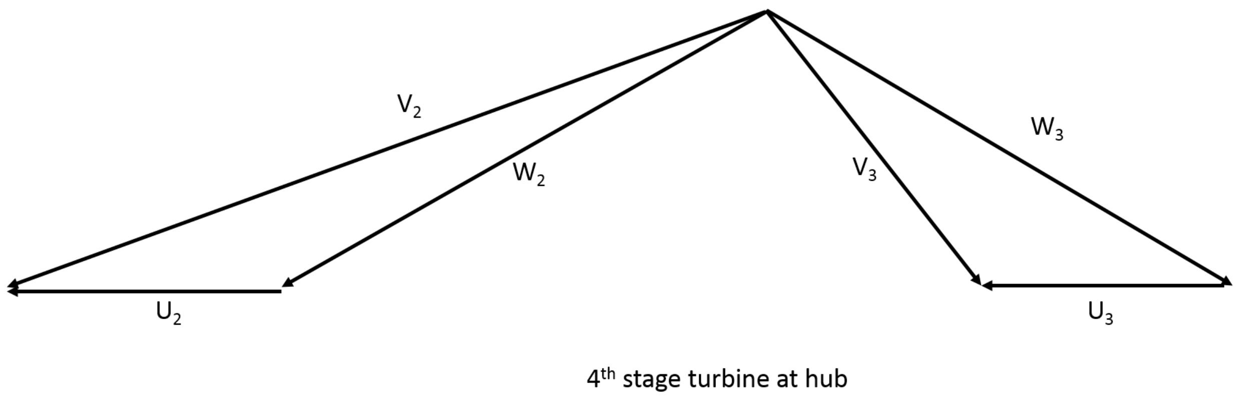

Figure F 1: Velocity triangle at stage 4 hub

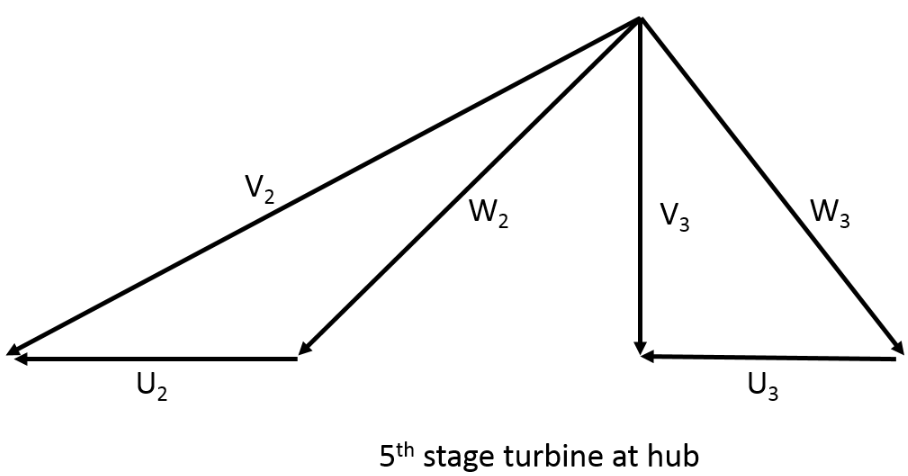

Figure F 2: Velocity triangle at stage 5 hub

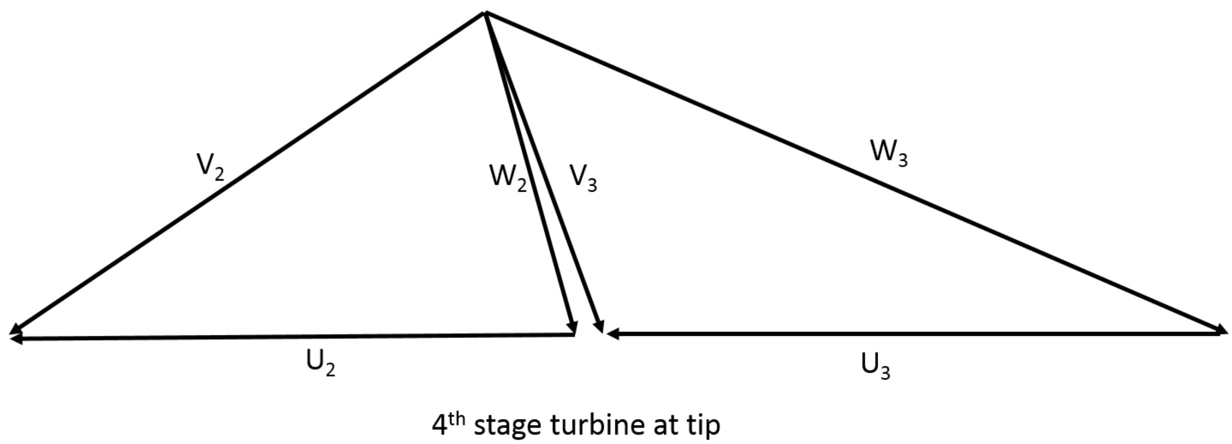

Figure F 3: Velocity triangle at stage 4 tip

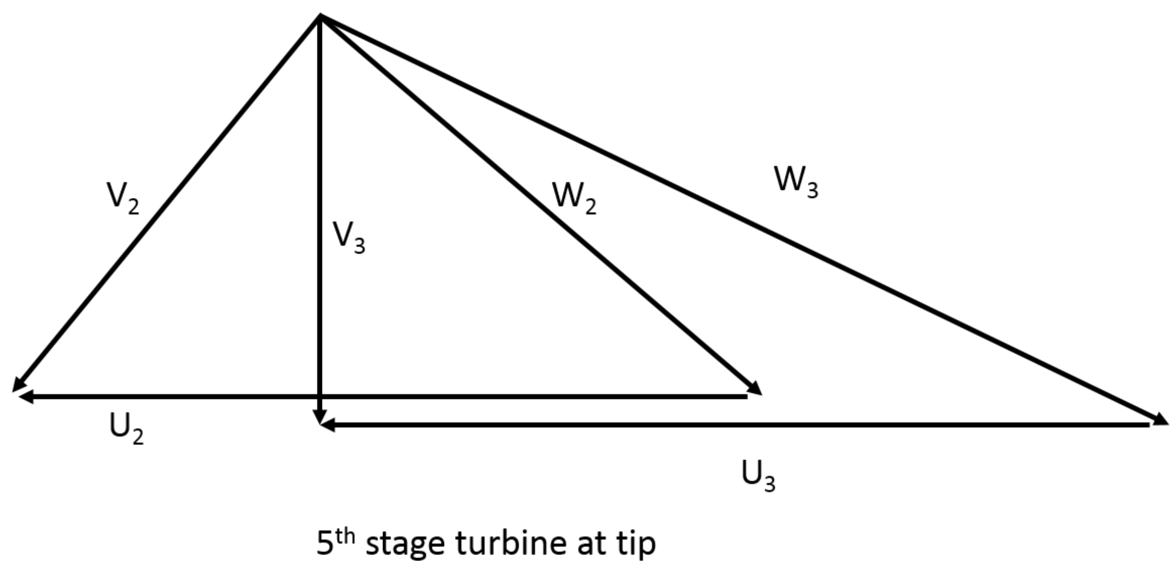

Figure F 4: Velocity triangle at stage 5 tip

According to the above data, the 3D model of turbine blades are created in Solidworks. The first 3 stages are just cylinder blades, while the last two are twisted. The 2nd stage stator ring and rotor are shown in **Error! Reference source not found.**. And the 4th stage stator ring and rotor are shown in Figure F 5.

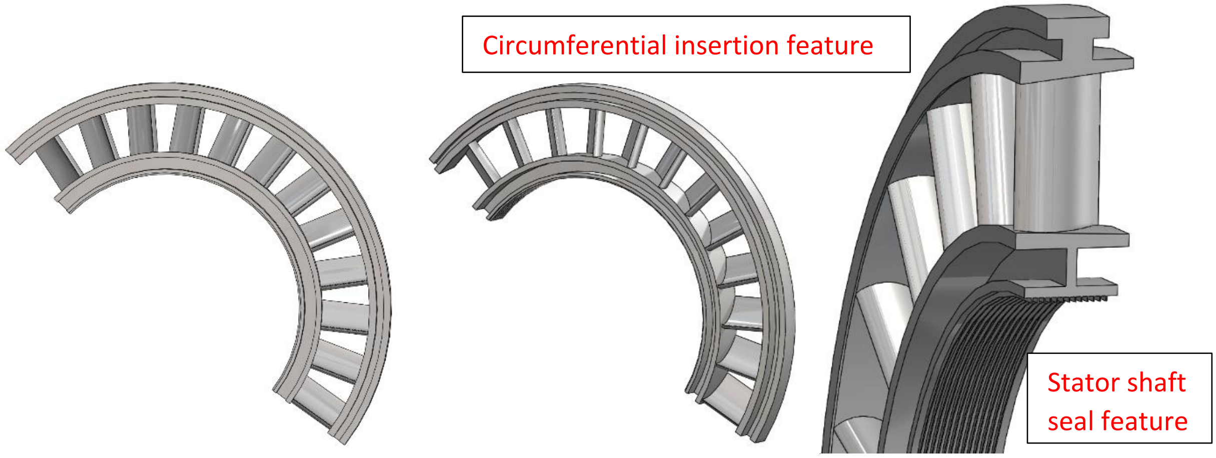

Figure F 5: Stator ring and seal feature of 2nd stage

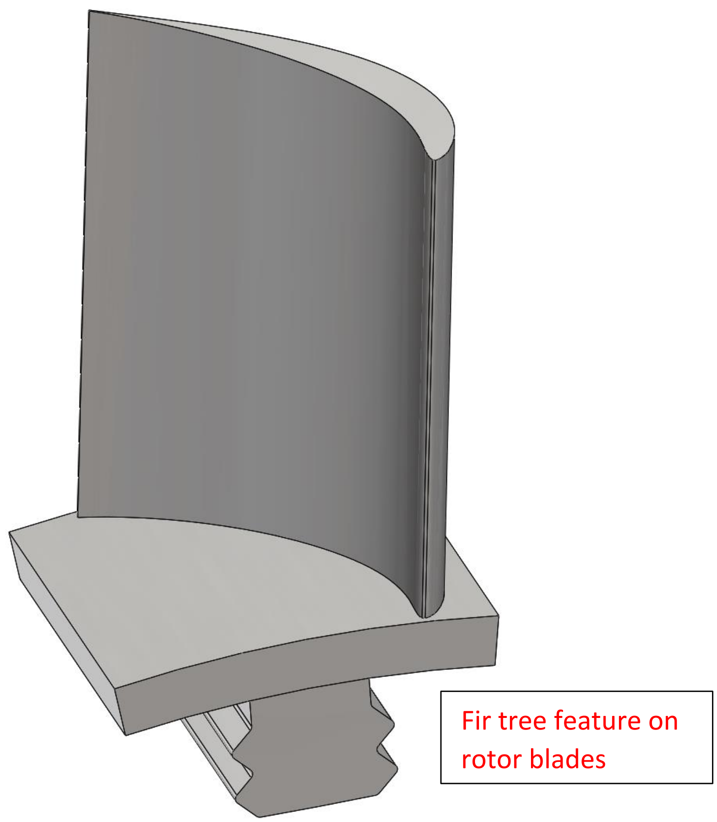

Figure F 6: 2nd stage rotor fir tree feature

Figure F 7: 4th stage twisted stator

Figure F 8: 4th stage twisted rotor

After the blades are built, we assemble them on the turbine casing and turbine rotor wheel. The rotor wheel and casing design will be introduced in the following sections.

F.2 Turbine main shaft design

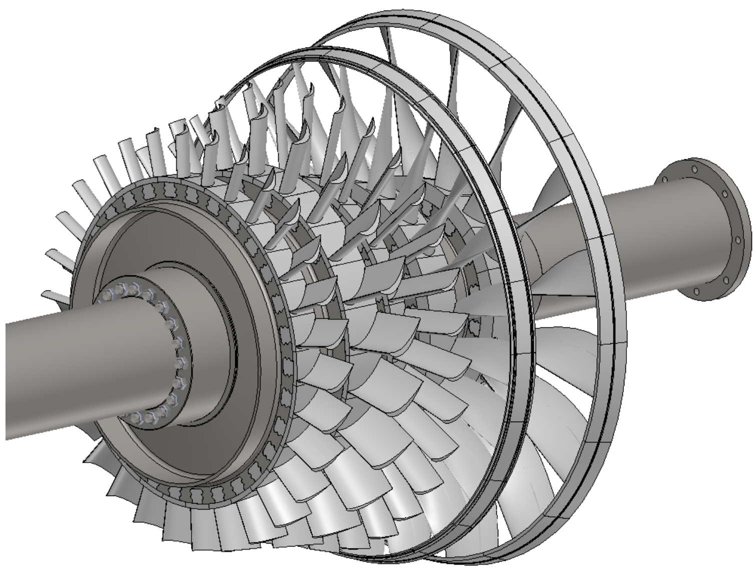

Figure F 9: Turbine rotor

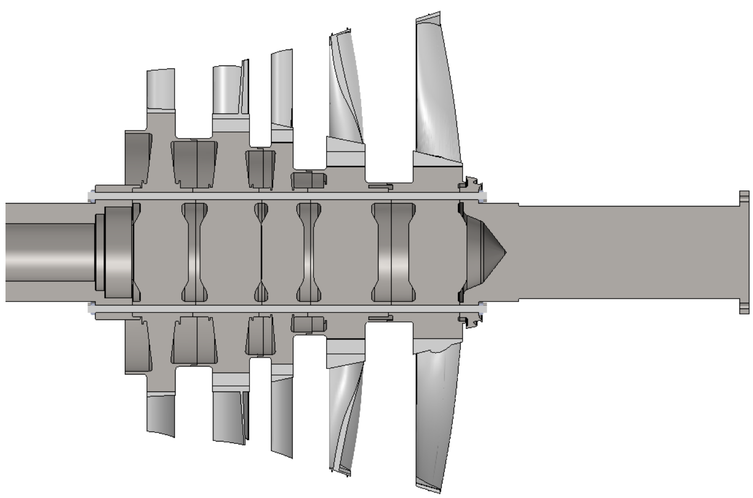

Figure F 10: Turbine rotor cross sectional view

F.3 Turbine Casing Design

Different from compressor casing design, the design of turbine casing includes: inner casing design and outer casing design. Stators are inserted into inner casing and outer casing is created to support the weight of inner casing and stators. The inner casing are shown in Figure F 11 and Figure F 12. As shown in Figure F 12, the inner casing has the installation feature for the stator, and also has the seal feature for the shrouded rotor.

For installation reason, we also design the inner casing as a horizontal split design. In this way, the casing can be disassembled by taking it off from the top.

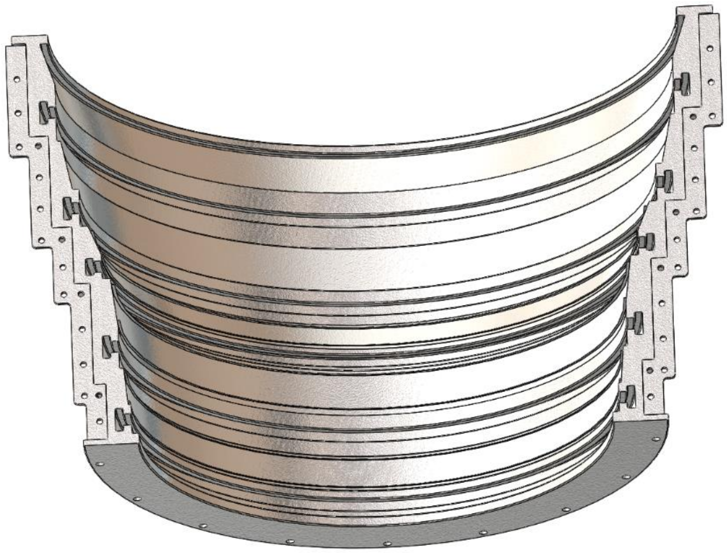

Figure F 11: Inner casing of turbine

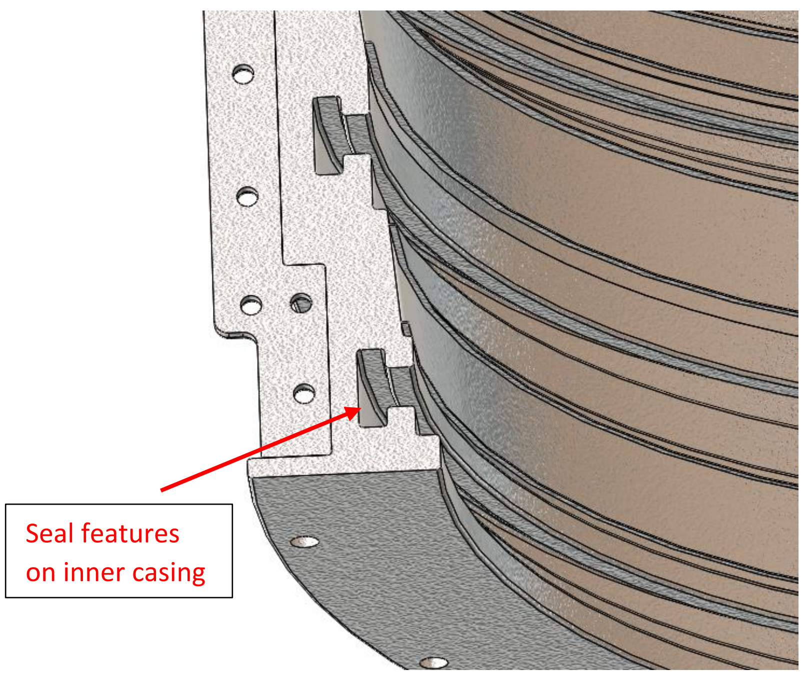

Figure F 12: Inner casing details

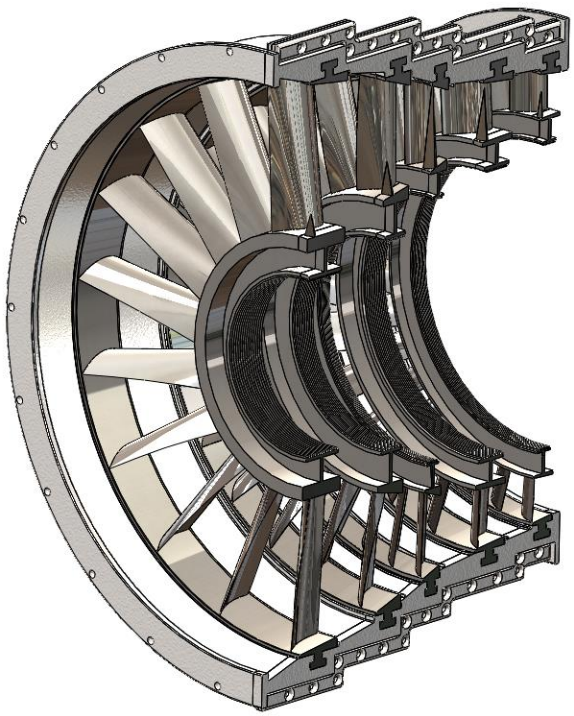

Figure F 13: Inner casing assembled with stator blades

Now, we have the inner casing assembled with the stator blades ring. Next step is to assemble them onto the turbine rotor wheel.

F.4 Rotor Blades Stress Analysis

At the last stage of blade design, we checked the stress analysis and material selection for the turbine rotor blades and compressor rotor blades. The main stress exerted on a rotor blade is the centrifugal tensile stress. The maximum value of this stress occurs at the root and is given by:

$$\sigma_{max} = 2\pi N^2 \rho_{blade} A$$

Where A is the annulus area, N is the rotational speed in rev/s. Thus, for our turbine design, the stress exerted on each stages rotor is calculated and listed in the following table:

Table F 2: Stress analysis for turbine rotor blades

	stage 1	stage 2	stage 3	stage 4	stage 5
N rev/s	74.69937254	74.69937	74.69937	74.69937	74.69937
blade density	8000	8000	8000	8000	8000
A (m2)	0.4388	0.5660	0.8393	1.2078	1.5521
stress (MPa)	123.0821185	158.7658	235.4207	338.7645	435.3465

Safety factor	2	2	2	2	2
allowable stress (MPa)	246.1642371	317.5315	470.8414	677.5291	870.693

In the above, the nickel based super alloy IN-100 is chosen for the turbine blades to satisfy the stress need.

For the compressor the same equation is used here. The difference is for LP and IP part, the material is chosen as aluminum alloy, while for the HP part, the material is chosen as nickel alloy.

Table F 3: Stress analysis for compressor rotor blades

compressor	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
N rev/s	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70	74.70
blade density kg/m3	2800.00	2800.00	2800.00	2800.00	2800.00	2800.00	2800.00	7700.00	7700.00	7700.00	7700.00	7700.00	7700.00	7700.00	7700.00	7700.00
A (m2)	0.87	0.77	0.66	0.61	0.55	0.50	0.45	0.42	0.37	0.34	0.31	0.28	0.26	0.24	0.22	0.20
stress (MPa)	85.63	75.87	64.69	59.58	54.03	49.22	43.80	112.50	101.16	91.59	83.28	76.02	69.65	64.02	59.04	54.94
allowable stress	171.26	151.73	129.37	119.16	108.05	98.43	87.60	224.99	202.33	183.18	166.56	152.04	139.30	128.05	118.07	109.88

G. Combustion Chamber, Nozzle and Diffuser Mechanical Design

G.1 Combustion chamber

This gas turbine has a can annular combustion chamber. There are 16 combustion nozzles fitted to the combustion chamber. Combustion chamber consists of 5 separate parts and it was designed such that it is completely disassemble. Figure G 1 shows picture of the combustion nozzle, which injects fuel into the combustion chamber.

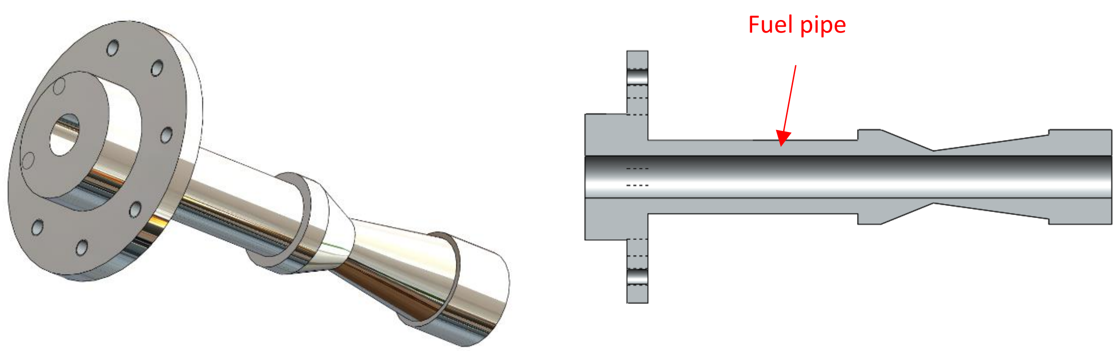

Figure G 1: Combustor nozzle

Figure G 2 shows the outer casing of the combustion chamber. As can be seen from the figure, there are 16 flanges, which are placed to connect with the 16 combustion nozzles. The inlet of the combustion chamber consists of diffuser guide vane. The function of the diffuser guide vane is to straighten the air flow entering the combustion chamber.

Figure G 4 shows the part that connects combustion chamber and turbine and shows the part that connects combustion chamber with compressor.

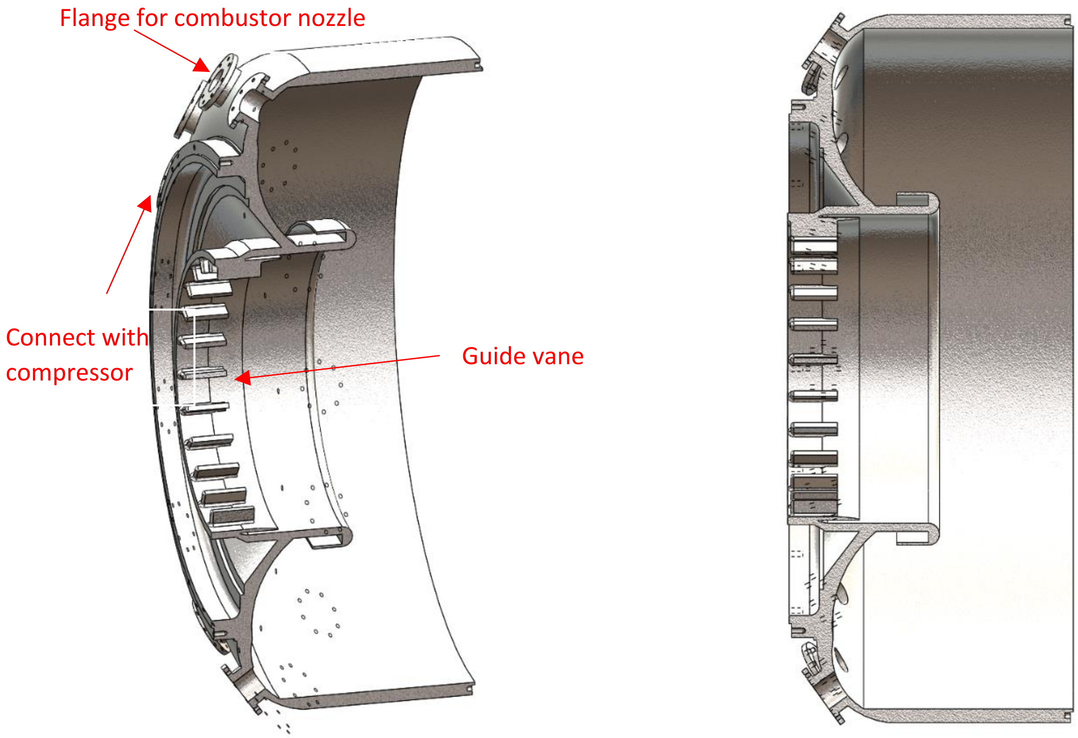

Figure G 2: Outer casing of combustion chamber

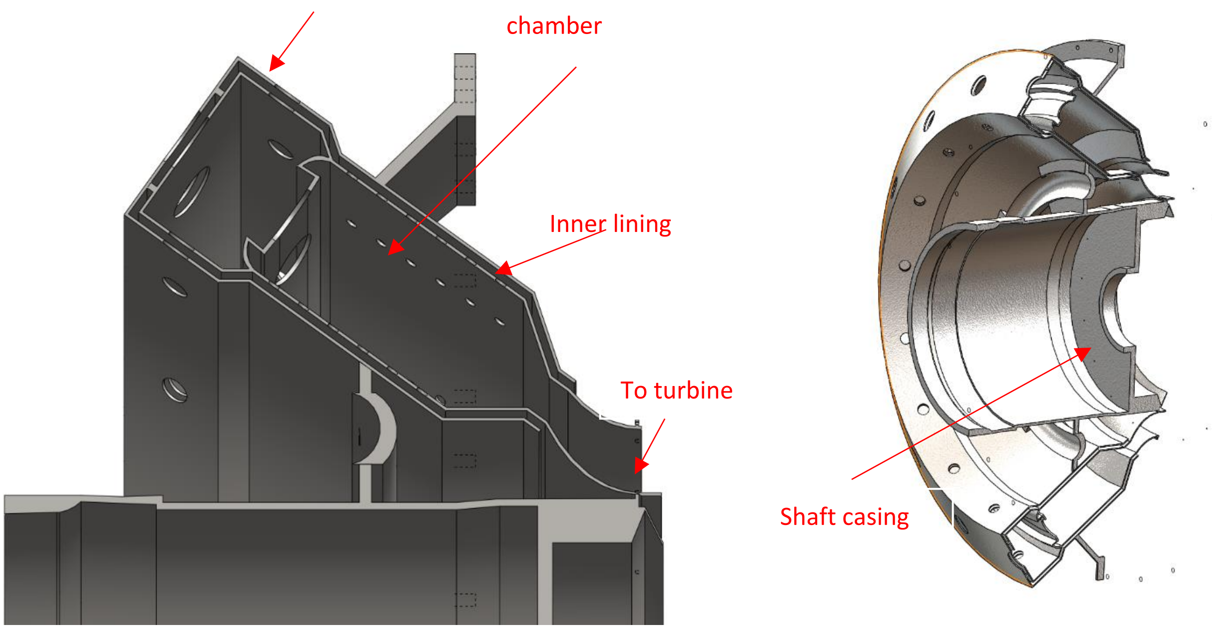

Figure G 3: Primary combustion chamber

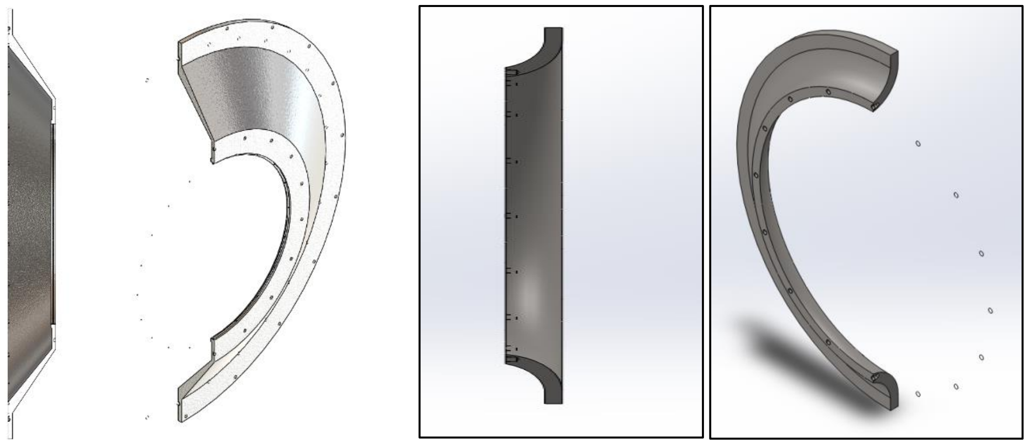

Figure G 4: Connects combustion chamber with turbine (left) connects combustion chamber with compressor (right)

Complete assembly of the combustion chamber is shown in Figure G 5. It also shows the air flow in the combustion chamber using red arrows.

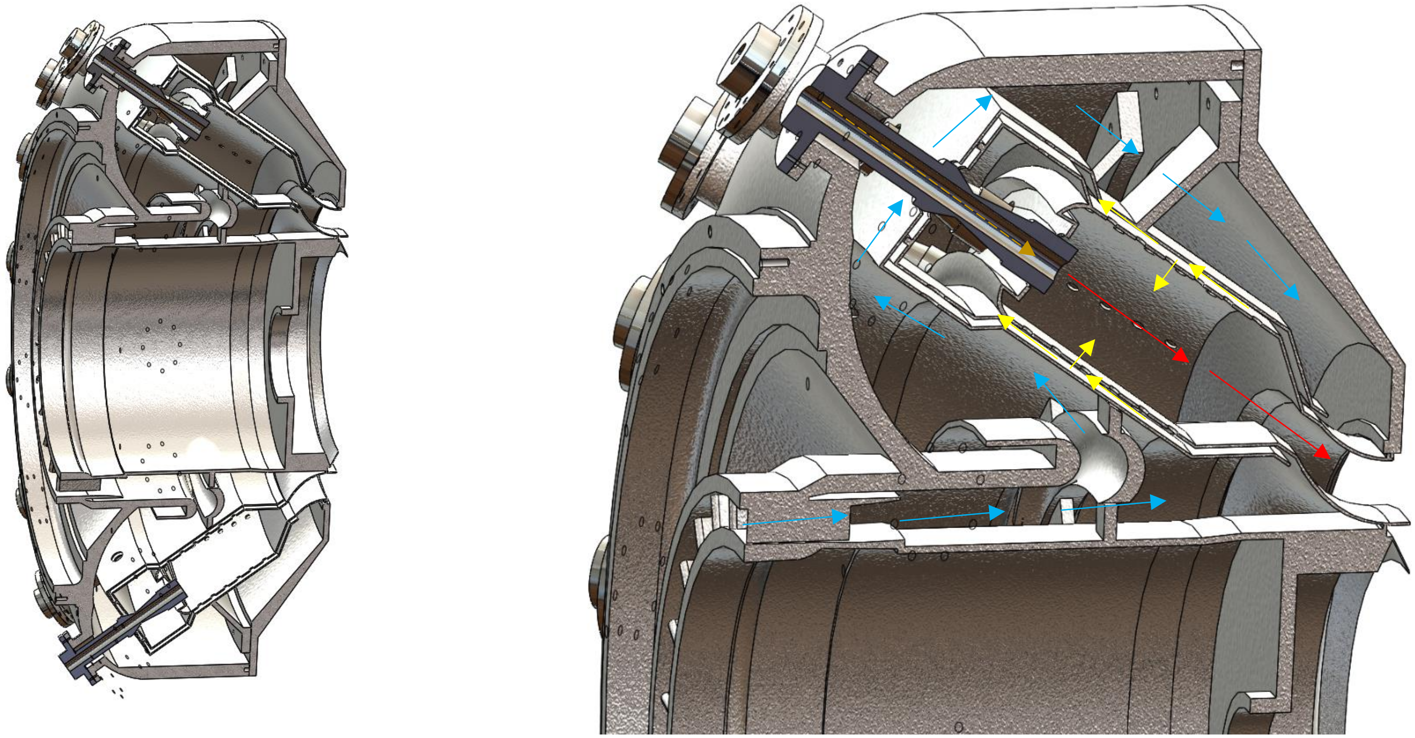

Figure G 5: Complete assembly of combustion chamber

G.2 Nozzle Design

The nozzle is designed to accelerate the flow that enters the compressor. The guide vanes in the nozzle, guide the flow into the compressor. The nozzle is shown in Figure G 6. As can be seen, the nozzle has labyrinth seals to prevent air from leaking in or out of the shaft.

p

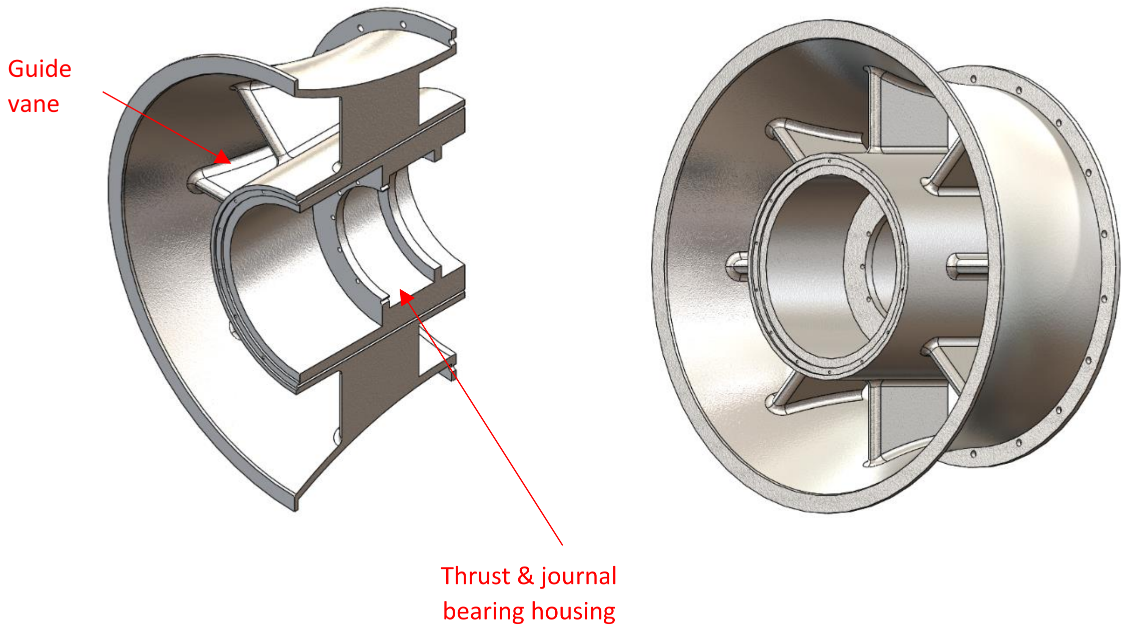

Figure G 6: Nozzle design

G.3 Diffuser Design

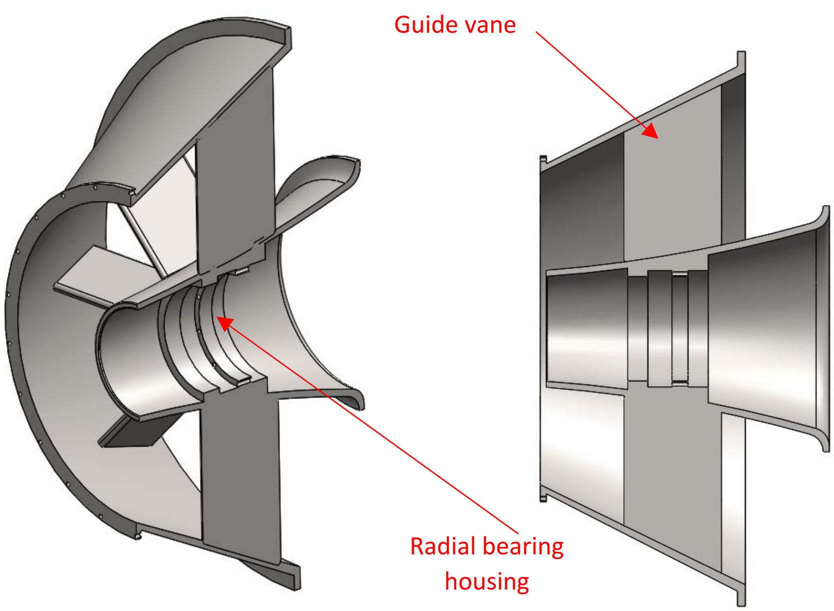

Figure G 7: Diffuser design

H. Bearing arrangement

The single shaft is supported by a journal bearing at the turbine end and a set of a thrust bearing and journal bearing at the compressor end. The design used spherical tilting pad bearing for journal bearing (the design is refer to the bearing catalog from John Crane)

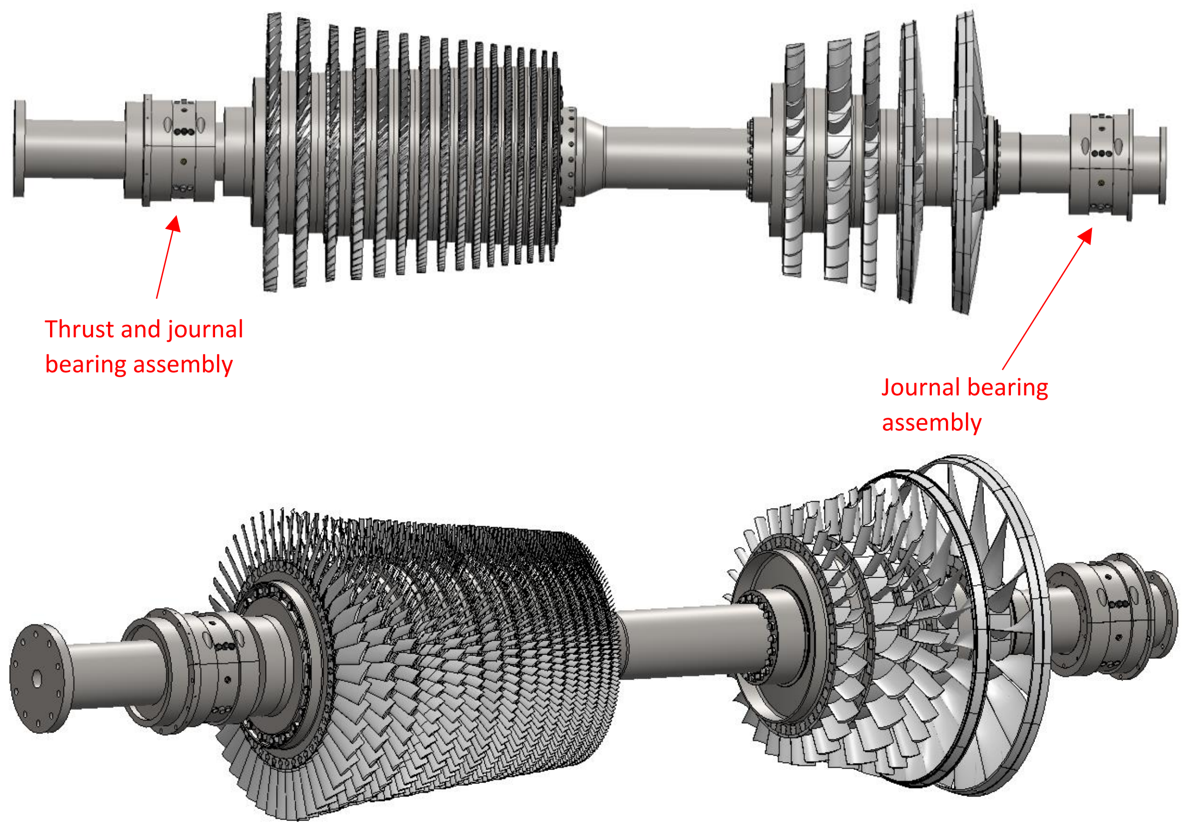

Figure H 1 Bearing arrangement

H1. Thrust and journal bearing assembly at compressor side

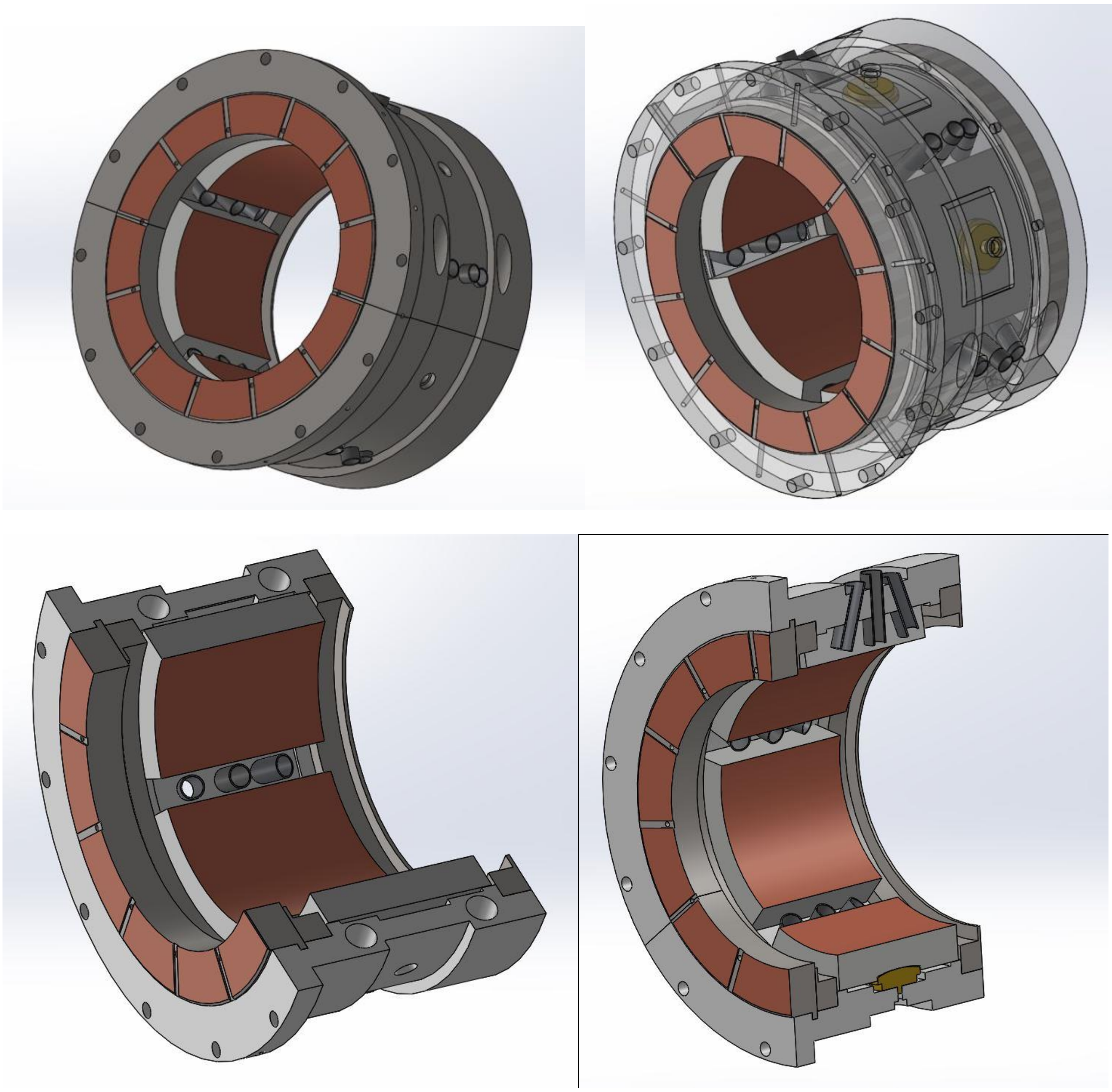

Figure H 2: Thrust and journal bearing assembly at compressor side

H2. Journal bearing at turbine side

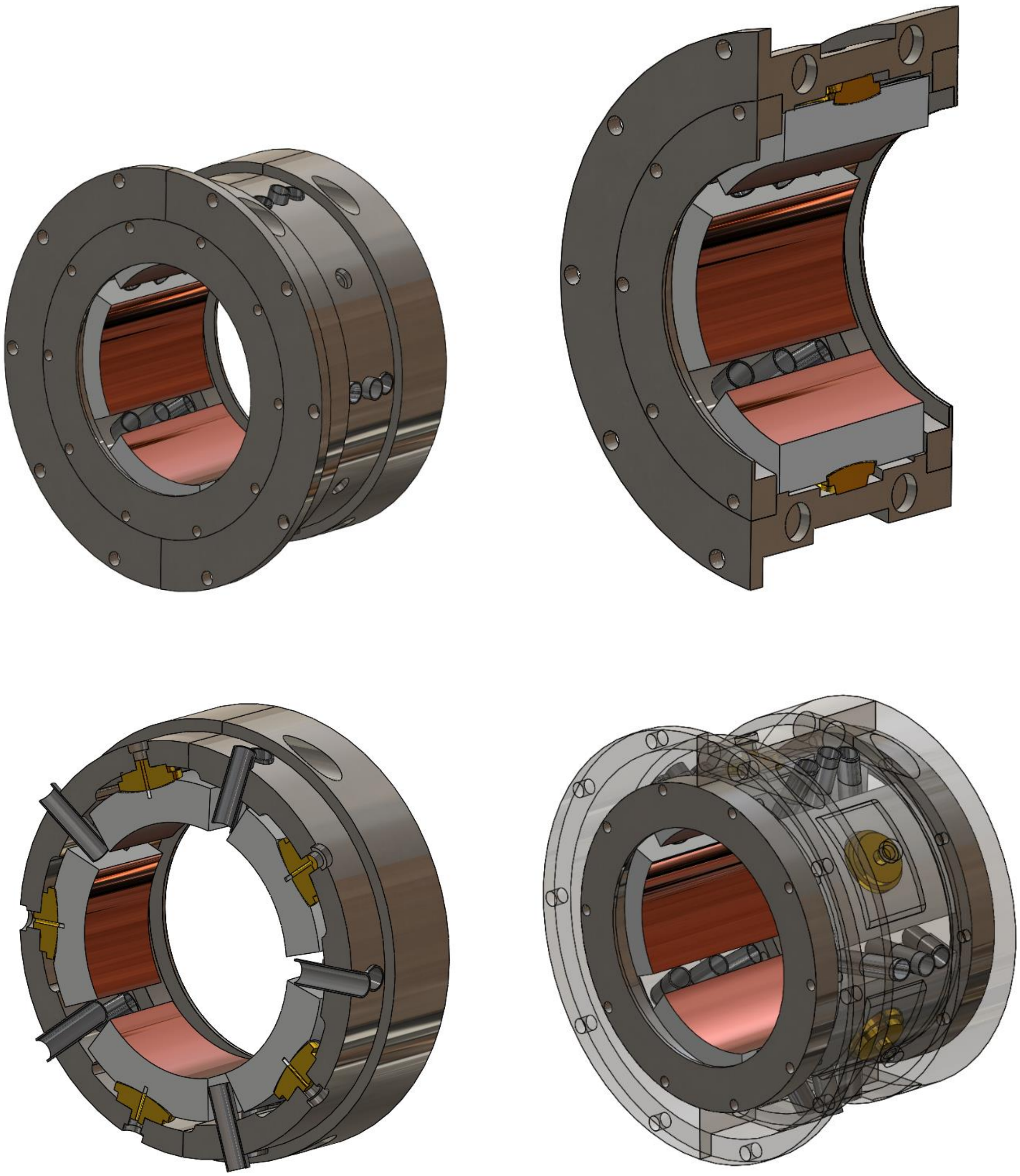

Figure H 3: Journal bearing at the turbine side

I. Full Gas Turbine Assembly

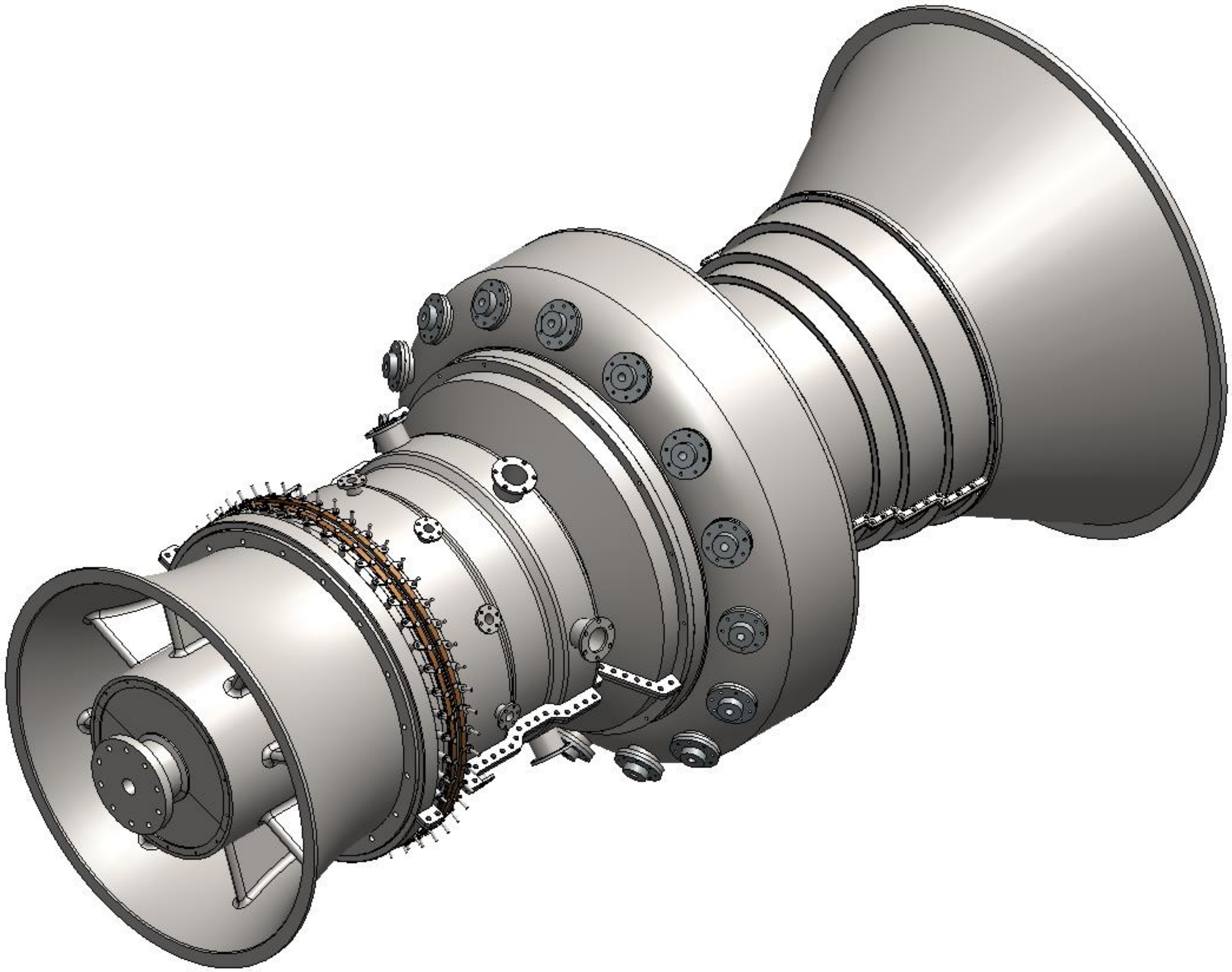

Figure I 1: Full assembly isometric view

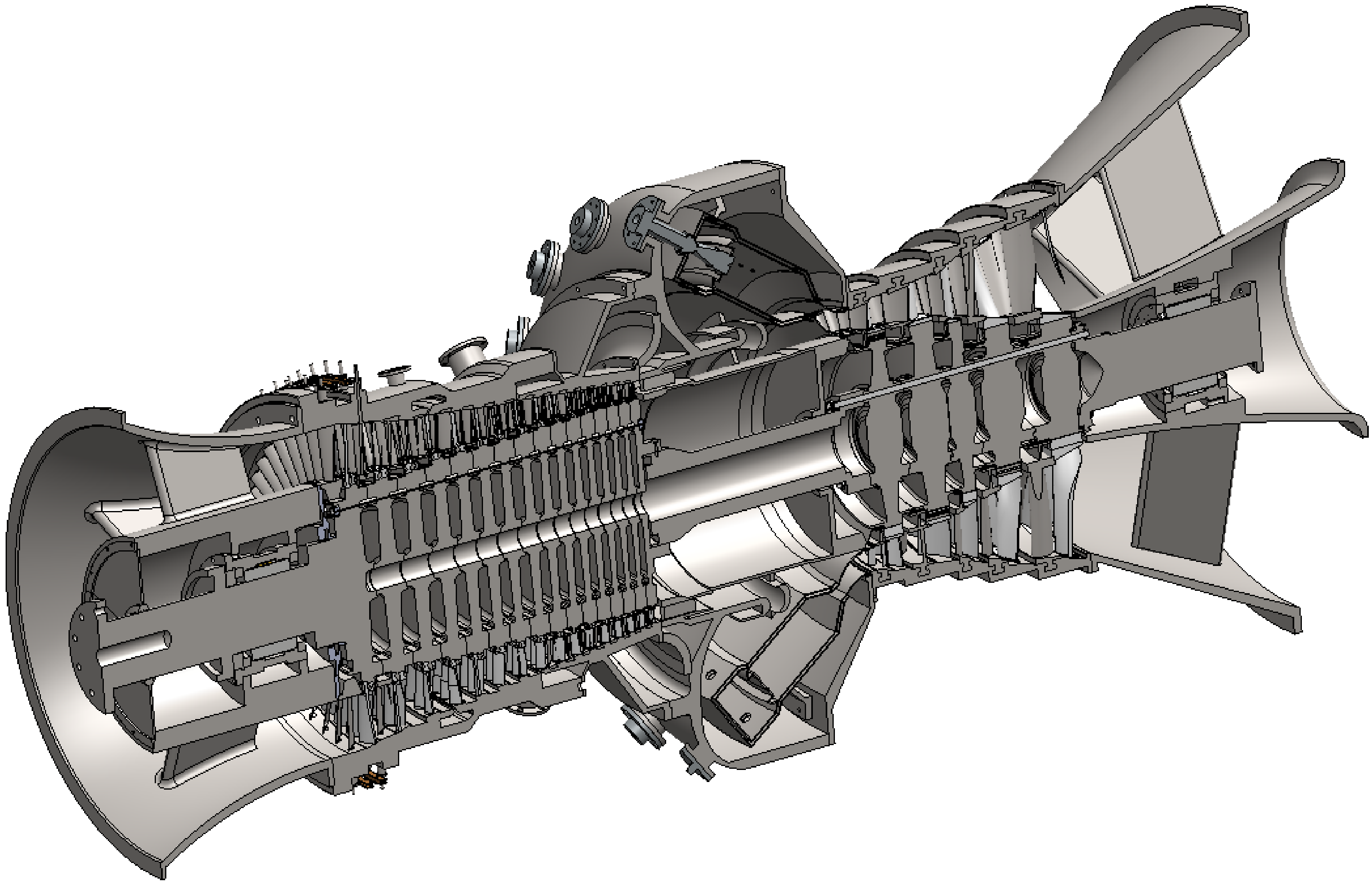

Figure I 2: Full assembly isometric cross sectional view

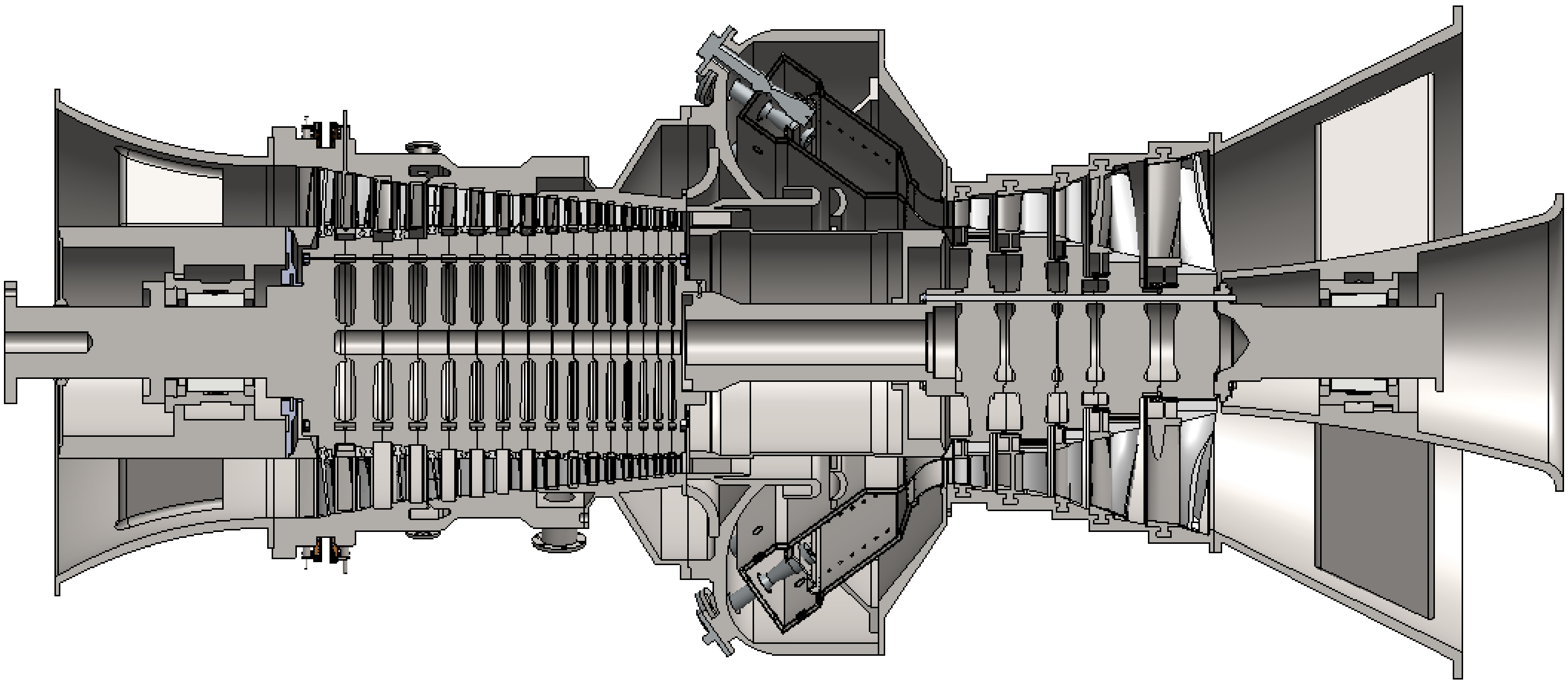

Figure I 3: Full assembly front cross sectional view

J. CFD simulation Analysis

J.1 Introduction

In order to understand the flow movement inside compressor channel and turbine channel, 3 CFD cases are selected which including: the 5th stage stator of compressor, the 10th stage stator of compressor and the 2th stage stator of turbine. The simulation geometry are referred to the above gas turbine mechanical design model.

The software ANSYS-Workbench is applied to generate mesh file of fluid region and the mesh quality is about 0.4 to 0.5. Then FLUENT 14.0 is employed to calculate the pressure distribution and velocity distribution of turbine cascade and compressor cascade respectively. k-epsilon standard model is selected and converged criteria is set as 10e-5.

Table J 1: Fluid properties selection in Fluent

Air density	Ideal Gas Model
Air specific heat	1006.43 J/kg.K
Thermal conductivity	0.0242 W/m.K
Viscosity	1.78e-5 kg/m.s

For the gas turbine application, the fluid is nearly compressible flow. Thus, we choose mass flow rate inlet as the inlet boundary condition, and the pressure outlet as the outlet boundary condition. All the boundary values are based on previous thermodynamics calculations.

Table J 2: Mesh numbers and boundary condition

Boundary Conditions	Compressor 5th Stator	Compressor 10th Stator	Turbine 2nd Stator
Inlet mass flow rate	28.48 kg/s	14.975 kg/s	318.6 kg/s/m^2
Inlet gauge pressure (Pa)	88438	283077	500375
Inlet X of flow direction	-1	-1	-1
Inlet Y of flow direction	-0.7849	-0.75869	-1.999
Outlet gauge pressure (Pa)	107054	320855	385785
Grid number	18907	17896	36485

In the ANSYS Workbench, Meshing are used to generate the 2D meshes. Here we set the Y+ value in the range of 11~300. In this case, the standard wall function combined with k-epsilon model is selected to solve the flow near the wall.

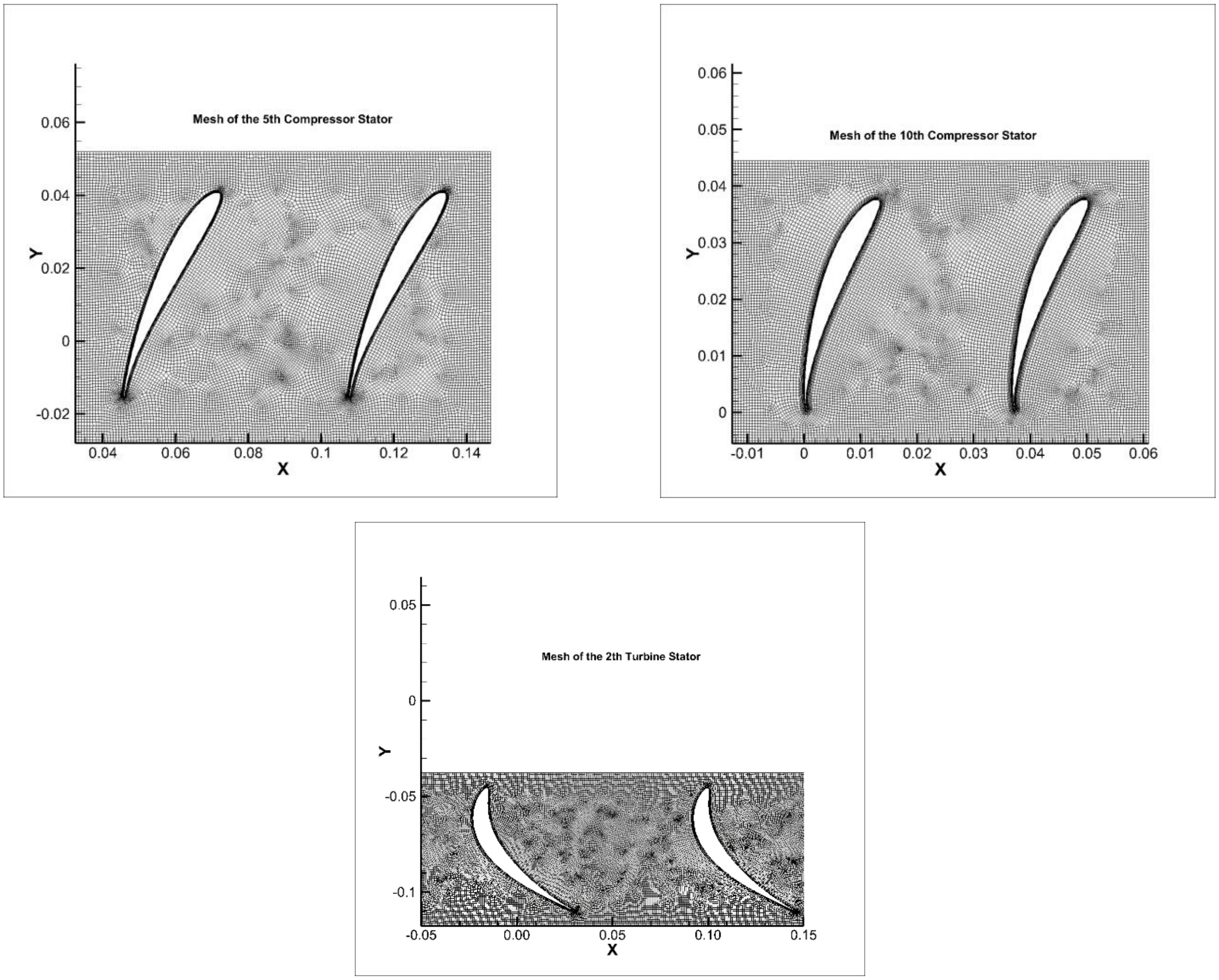

Figure J 1: Mesh illustration of the three cases

J.2 CFD Results and Discussion

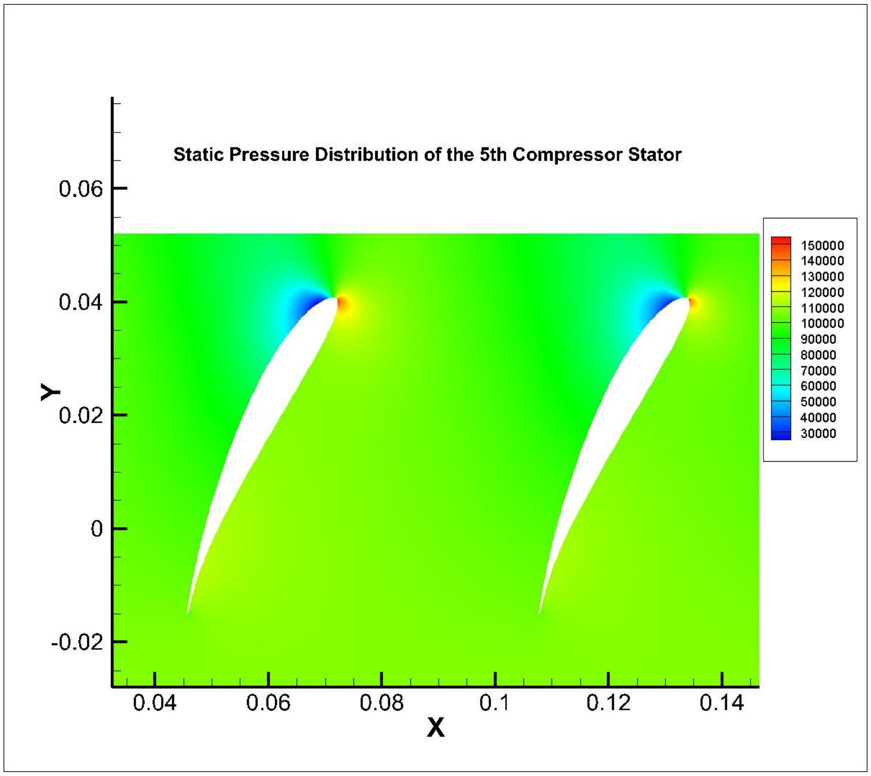

Figure J 2: Pressure contour of 5th compressor stator

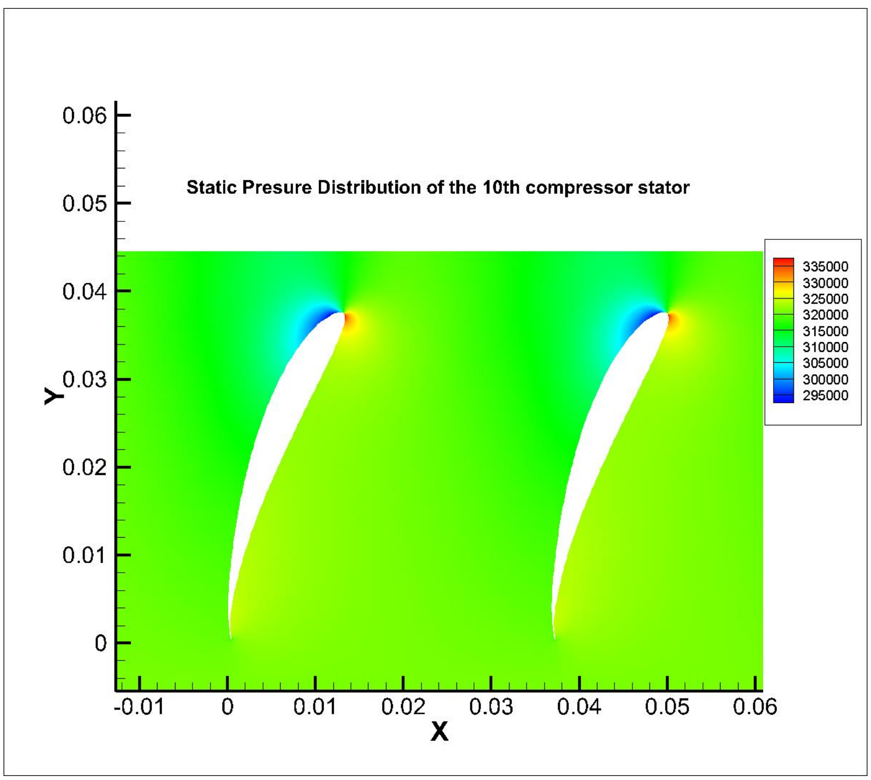

Figure J 3: Pressure contour of 10th compressor stator

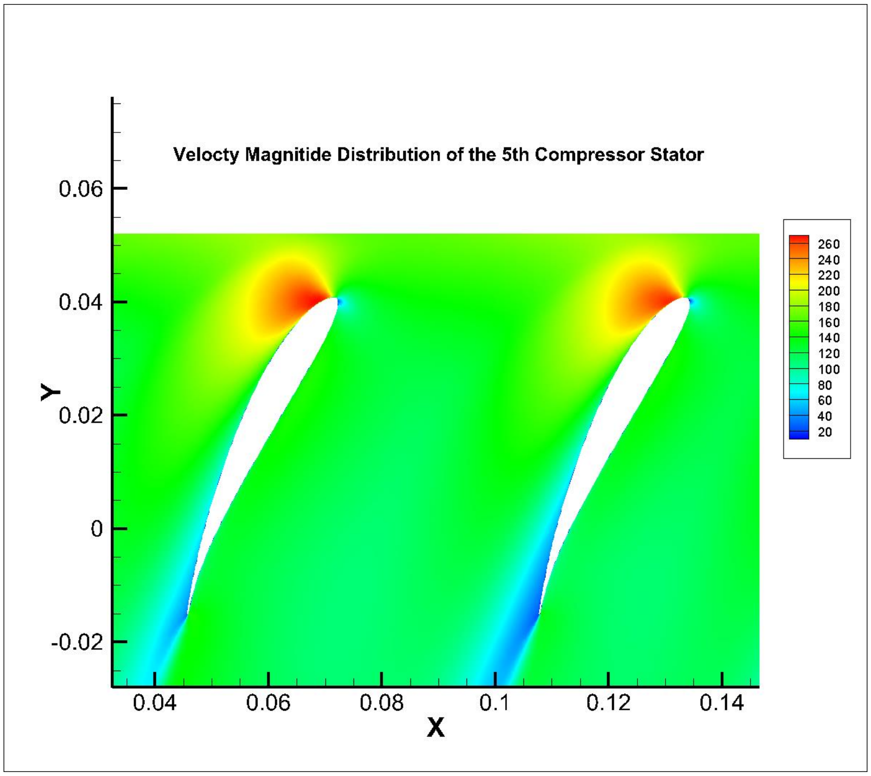

Figure J 4: Velocity Contour of the 5th Compressor Stator

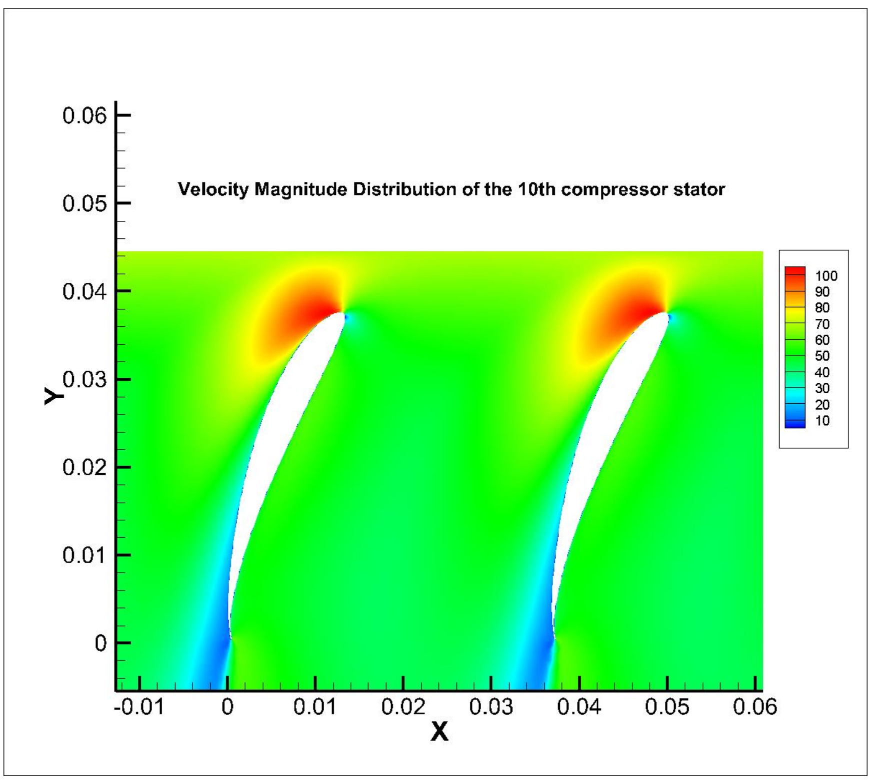

Figure J 5: Velocity Contour of the 10th Compressor Stator

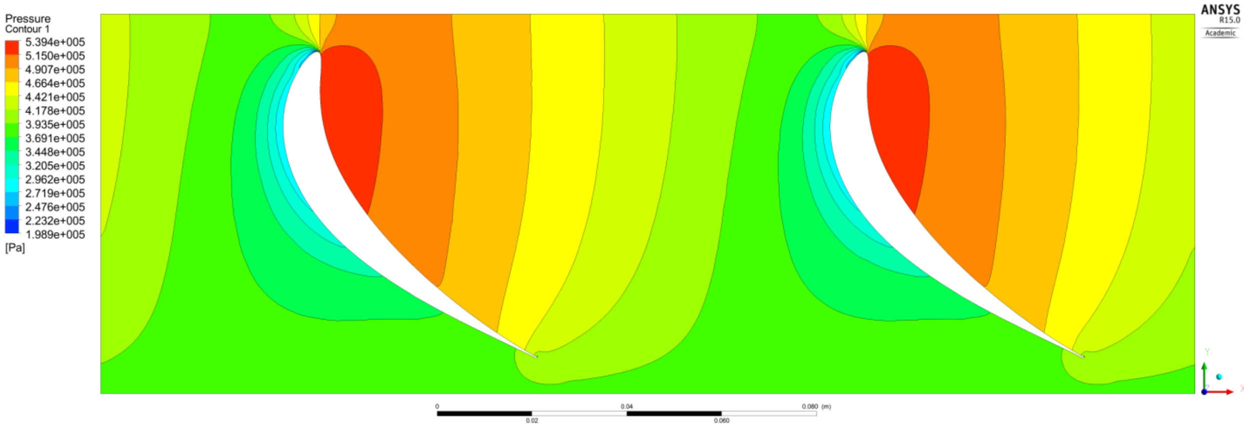

Figure J 6: Pressure Contour of 2nd Stage Turbine Stator

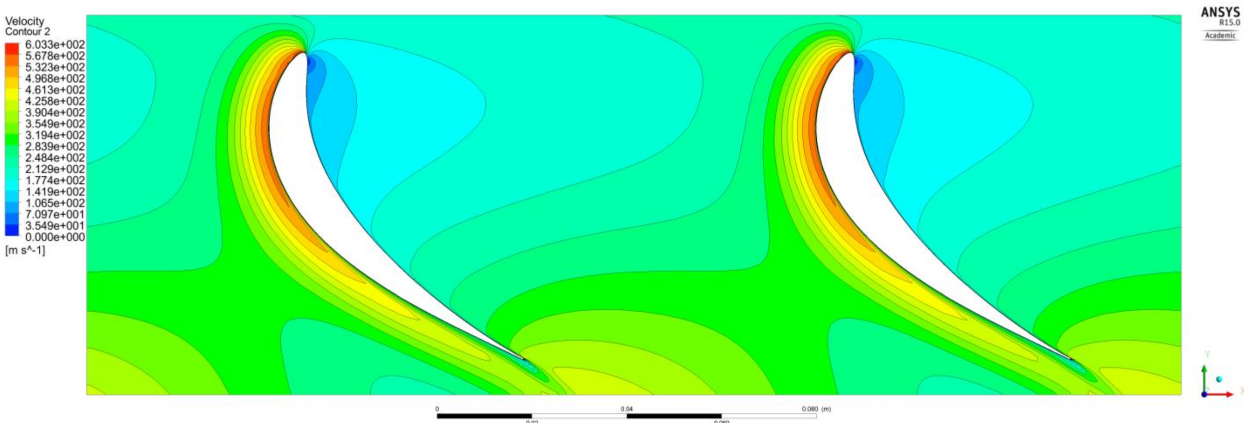

Figure J 7: Pressure contour 2nd turbine stator

From the Figure J 4 and Figure J 5, we get a very good pressure distribution along the compressor blades and also, the velocity magnitude is decreasing from stage 5 to stage 10.

The pressure distribution for both compressor and turbine blade is evaluated based on the following definition:

$$C_p = \frac{p_1 - p_x}{p_1}$$

Where p_1 is the static pressure at leading edge, p_x is the static pressure at the position x, starting from the leading edge along the blade wall. Then we get the Cp distribution in Figure J 8 to Figure J 10.

In Figure J 8 and Figure J 9, the Cp distribution is similar. It shows that the pressure difference at suction side is higher than that at pressure side. The shape of the Cp is abundant and shows good pressure distribution. No flow separation can be predicted from the CFD study.

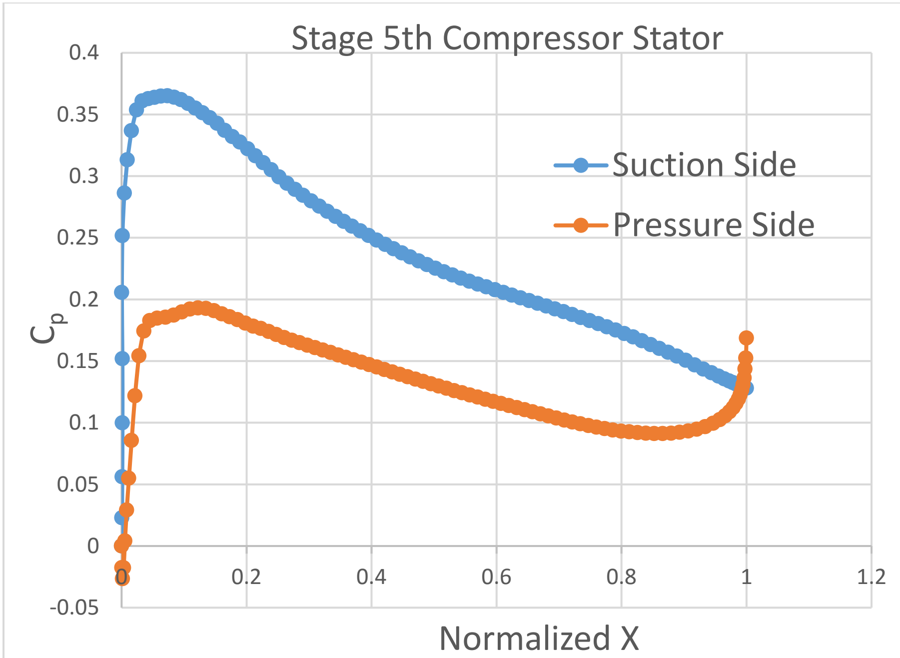

Figure J 8: Cp distribustion of 5th compressor stator

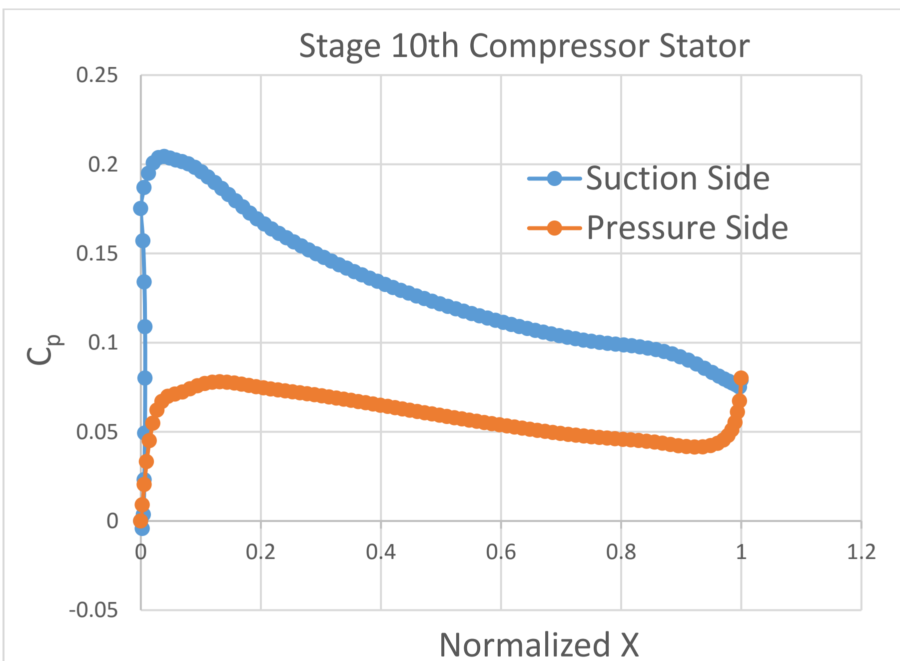

Figure J 9: Cp distribution of the 10th compressor stator

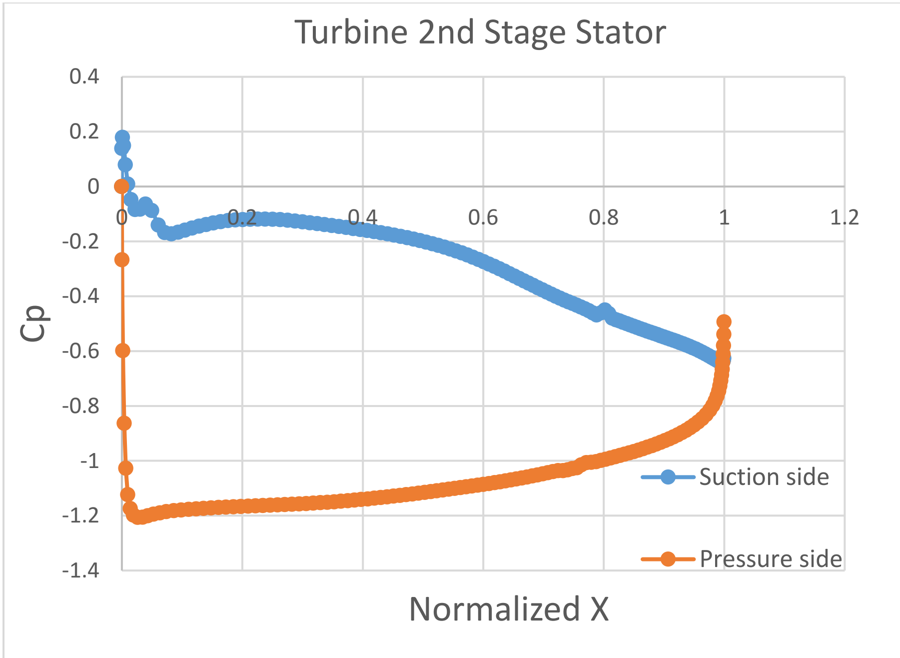

Figure J 10: Cp distribution of the 2nd turbine stator

The above figures show the x/L vs Cp, while X is the position coordinate starting from the leading edge of the blade. The shape of the Cp is very full, which indicates that the turbine blade is well designed to extract enough energy from the hot gas. And also, the pressure gradient doesn't change sharply at the suction side, which represents the flow is not separating.

Conclusions

The axial compressor requires about 40 MW of power to compress the flow from 100 kPa to almost 910 kPa. The total turbine power supplies about 72.5 MW, so the net power of the gas turbine engine is about 32.5 MW. The flow deflection, or difference between α2 and α3, remains under 20° for the low pressure and intermediate pressure compressors, and remains under 40° for the high pressure compressor. The minimized deflection ensures that the flow will not separate, and CFD analysis verifies this as well as the actual pressure distribution.

The 3D mechanical design includes the basic component for a functional gas turbine engine. The manufacturing and maintenance requirement are considered during the design process. The nozzle, compressor, combustion chamber, turbine, diffuser and bearing system are the main components that are designed in this project. Finally, the project provides a detailed steps for designing a gas turbine engine, from the aero-thermodynamics calculation to the 3D mechanical component design.

References

[1] Schobeiri, Meinhard T., *Turbomachinery Flow Physics and Dynamic Performance*, 2nd Ed., Springer, 2012. Print & Online

[2] H.I.H. Saravanamuttoo, G.F.C. Rogers, H. Cohen, and Paul Straznicky, *Gas Turbine Theory*, 6th Ed., Pearson, 2009

Appendix

Appendix_A1-Compressor_Blade_Generation_Code

Appendix_B1-Turbine_Blade_Generation_Code

Appendix_B2-Turbine_Efficiency_Iteration_Code

Appendix_B3-Turbine_Stages_Design_Code

Appendix_C1-Diffuser_Calculation_Code

Appendix_C2-Nozzle_Calculation_Code

```
%                     MEEN-646: Module 2
%           Design of Subsonic Compressor Blade M2-C


% Objective:
%    Develop a design software that enables you to generate subsonic compressor blades
%    using NACA Camberline Equation
% Given Parameters
%    - Base profile
%    - Blade chord C
%    - Lift coefficient
%
% Instruction:
%    Input: prin, K, H, t, s, CL, iZone selection

clc;
clear all;


ln = @(x)(log(x));

%input:
iZone=1; % input 1, 2, 3 (1 for thinnest profile and 3 for thickest profile)
prin=0; % Enter 1 if you want to print the output
% Enter your chord length data and lift coefficient data in K and H
K=[0.021470881].*1000; % chord length
H=-[-0.697243515]; % lift coefficient
% Enter value of t to choose ith value from matrix K and H
t=1;
ch = K(t); %chord length (created by Jita)
CL = H(t); %lift coefficient

%input parameters
C = 1; %blade chord (created by Tran)
s = 1; %spacing
%n_iter = 10000; %number of iteration

n_iter = 10000/10;

n_iter_b1 = 190/10;
n_iter_b2 = 6650/10;
n_iter_b3 = 9955/10; %for zone3 only
x_cam1=n_iter_b1/n_iter*C;
x_cam2=n_iter_b2/n_iter*C;
x_cam3=n_iter_b3/n_iter*C;%for zone3 only

%NACA Camberline Equation
for i=1:1:n_iter
    x_cam(i) = i/n_iter*C;
    y_cam(i) = -C*CL/(4*pi)*((1-i/n_iter)*ln(1-i/n_iter)+i/n_iter*ln(i/n_iter));
    %camber line tangent angle
```

```
    v_cam(i)=atan(CL/(4*pi)*ln((1-i/n_iter)/(i/n_iter)));

    %for the 2nd camberline
    y_cam1(i) = y_cam(i)+s;
end

%blade thickness
if iZone == 1
    for i=1:1:n_iter_b1
        x(i)=i/n_iter;
        t(i)=C*(0.3419*x(i)^0.4929)%zone1
    end

    for i=(n_iter_b1+1):1:n_iter_b2
        x(i)=i/n_iter;
        t(i)=C*(-15.631*x(i)^6 + 38.563*x(i)^5 - 38.22*x(i)^4 + 19.934*x(i)^3 - 6.2802*x
(i)^2 + 1.1333*x(i) + 0.0307)%zone1
    end

    for i=(n_iter_b2+1):1:n_iter
        x(i)=i/n_iter;
        t(i)=C*(75.656*x(i)^6 - 375.15*x(i)^5 + 774.1*x(i)^4 - 850.22*x(i)^3 + 524.07*x
(i)^2 - 172.08*x(i) + 23.628)%zone1
    end

elseif iZone == 2
    for i=1:1:n_iter_b1
        x(i)=i/n_iter;
        t(i)=(C*(0.6128*x(i)^0.4937))%zone2
    end

    for i=(n_iter_b1+1):1:n_iter_b2
        x(i)=i/n_iter;
        t(i)=C*(-35.559*x(i)^6 + 83.97*x(i)^5 - 79.529*x(i)^4 + 39.519*x(i)^3 - 11.876*x
(i)^2 + 2.0934*x(i) + 0.0531)%zone2
    end

    for i=(n_iter_b2+1):1:n_iter
        x(i)=i/n_iter;
        t(i)=C*(93.702*x(i)^6 - 455.68*x(i)^5 + 921.82*x(i)^4 - 991.96*x(i)^3 + 598.52*x
(i)^2 - 192.34*x(i) + 25.931)%zone2
    end

    if t(i)<0
       t(i)=0;
    end

elseif iZone == 3
    for i=1:1:n_iter_b1
        x(i)=i/n_iter;
        t(i)=C*(0.8232*x(i)^0.4941)%zone3
```

```
    end

    for i=(n_iter_b1+1):1:n_iter_b2
        x(i)=i/n_iter;
        t(i)=C*(-56.476*x(i)^6 + 129.12*x(i)^5 - 118.24*x(i)^4 + 56.666*x(i)^3 - 16.456*x
(i)^2 + 2.8703*x(i) + 0.0696);%zone3
    end

    for i=(n_iter_b2+1):1:n_iter_b3
        x(i)=i/n_iter;
        t(i)=C*(65.209*x(i)^6 - 309.36*x(i)^5 + 610.82*x(i)^4 - 641.17*x(i)^3 + 376.88*x
(i)^2 - 118.1*x(i) + 15.713);%zone3
    end

    for i=(n_iter_b2+1):1:n_iter
        x(i)=i/n_iter;
        t(i)=C*(7.1679*x(i)^2 - 14.367*x(i) + 7.1992);%zone3
    end

    if t(i)<0
       t(i)=0;
    end

end


%suction side coordinate
for i=1:1:n_iter
    x_S(i) = x_cam(i) - (t(i)/2)*sin(v_cam(i));
    y_S(i) = y_cam(i) + (t(i)/2)*cos(v_cam(i));
    y_S1(i) = y_S(i) + s;
end

%pressure side coordinate
for i=1:1:n_iter
    x_P(i) = x_cam(i) + (t(i)/2)*sin(v_cam(i));
    y_P(i) = y_cam(i) - (t(i)/2)*cos(v_cam(i));
    y_P1(i) = y_P(i) + s;
end

% Scaling done to match the blade to actual chord length
x_cam=x_cam.*ch;y_cam=y_cam.*ch;
x_S=x_S.*ch; y_S=y_S.*ch;
x_P=x_P.*ch; y_P=y_P.*ch;
y_S1=y_S1.*ch;y_P1=y_P1.*ch;


% plot the blade
if prin==1
figure(1);
plot(x_cam,y_cam,'g')
```

```
hold on
plot(x_S,y_S,'r')
hold on
plot(x_P,y_P,'b')
hold on
plot(x_cam,y_cam1,'g')
hold on
plot(x_S,y_S1,'r')
hold on
plot(x_P,y_P1,'b')
hold on
xlim([-.1*ch 1.1*ch])
% ylim([-0.2 0.8])
ylim([-0.5*ch 0.5*ch])
end

% Z matrix created to organize data for solidworks
Z(:,1)=x_S; Z(:,2)=y_S; Z(:,4)=x_P; Z(:,5)=y_P; Z(:,6)=0;
```

```
%                     MEEN-646: Module 2
%           Design of Subsonic Turbine Blade M2-T

% Objective:
%    Develop a design software that enables you to generate subsonic turbine blades
%
% Given Parameters
%    - Generate a family of profile (alpha1=90, alpha2=160), (alpha1=45, alpha2=160)
%    - Blade chord C
%
% Instruction:
%    Input: alpha1, alpha2 & iZone

clc;
clearvars
iZone=1;
one=[110.69 139.46];
second=[156.63  154.3];
%input:
t=2;
alpha1 = one(t); %in degree
alpha2 = second(t); %in degree
%convert to rad
alpha1 = (alpha1/180)*pi;
alpha2 = (alpha2/180)*pi;

C = 40; %chord
s = 30; %spacing
n_iter = 1000; %number of iteration for camberline

n_iter_b1 = 19;
n_iter_b2 = 665;
n_iter_b3 = 995; %for zone3 only
x_cam1=n_iter_b1/n_iter*C;
x_cam2=n_iter_b2/n_iter*C;
x_cam3=n_iter_b3/n_iter*C;%for zone3 only

%cascade stagger angle
gamma = atan(sin(alpha2)/(-1/2*sin(alpha1-alpha2)/sin(alpha1)+cos(alpha2)));
C_ax = C*sin(gamma);


%define camber line equation:
x_p0 = 0; y_p0 = 0;
x_p2 = C; y_p2 = 0;

%determine P1 coordinates by consider triangle P0P1P3
%a_1 = 1/3*C/sin(alpha1)*sin(gamma); %P1P0 length
%b_1 = 1/3*C/sin(alpha1)*sin(pi-alpha1-gamma); %P1P3 length
%c_1 = 1/3*C; %P0P3 length
%p = (a_1+b_1+c_1)/2;
```

```
%area = sqrt(p*(p-a_1)*(p-b_1)*(p-c_1));

%y_p1 = 2*area/(1/3*C);
%x_p1 = y_p1/tan(pi-alpha1-gamma);

%determine P1 coordinates, formula given in the book (equation 10.40)
phi1=pi/2-alpha1+gamma;
phi2=pi/2+alpha2-gamma;
y_p1 = C*(cot(phi1)/(1+cot(phi1)/cot(phi2)));
x_p1 = C*(1/(1+cot(phi1)/cot(phi2)));



%Bezier Curve
for i=1:1:n_iter
    zeta(i) = i/n_iter;
    x_cam(i) = (1-zeta(i))^2*x_p0 + 2*(1-zeta(i))*zeta(i)*x_p1 + zeta(i)^2*x_p2;
    y_cam(i) = (1-zeta(i))^2*y_p0 + 2*(1-zeta(i))*zeta(i)*y_p1 + zeta(i)^2*y_p2;

    %camber line tangent angle
    v_cam(i)=atan((-2*(1-zeta(i))*y_p0 + 2*(1-2*zeta(i))*y_p1 + 2*zeta(i)*y_p2)/(-2*(1-
zeta(i))*x_p0 + 2*(1-2*zeta(i))*x_p1 + 2*zeta(i)*x_p2));

    %for the 2nd camberline
    y_cam1(i) = y_cam(i)+s;
end

%blade thickness
if iZone == 1
    for i=1:1:n_iter
        x(i)=x_cam(i)/C;
        if (x_cam(i)<x_cam1)
            t(i)=C*(0.3419*x(i)^0.4929);%zone1
        elseif (x_cam(i)>x_cam1) && (x_cam(i)<x_cam2)
            t(i)=C*(-15.631*x(i)^6 + 38.563*x(i)^5 - 38.22*x(i)^4 + 19.934*x(i)^3 -
6.2802*x(i)^2 + 1.1333*x(i) + 0.0307);%zone1
        elseif (x_cam(i)>x_cam2)
            t(i)=C*(75.656*x(i)^6 - 375.15*x(i)^5 + 774.1*x(i)^4 - 850.22*x(i)^3 + 524.07
*x(i)^2 - 172.08*x(i) + 23.628);%zone1
        end
        %t(i)=t(i)/2;
    end
elseif iZone == 2
    for i=1:1:n_iter
        x(i)=x_cam(i)/C;
        if (x_cam(i)<x_cam1)
            t(i)=C*(0.6128*x(i)^0.4937);%zone2
        elseif (x_cam(i)>x_cam1) && (x_cam(i)<x_cam2)
            t(i)=C*(-35.559*x(i)^6 + 83.97*x(i)^5 - 79.529*x(i)^4 + 39.519*x(i)^3 -
11.876*x(i)^2 + 2.0934*x(i) + 0.0531);%zone2
        elseif (x_cam(i)>x_cam2)
```

```
            t(i)=C*(93.702*x(i)^6 - 455.68*x(i)^5 + 921.82*x(i)^4 - 991.96*x(i)^3 +
598.52*x(i)^2 - 192.34*x(i) + 25.931);%zone2
        end

        if t(i)<0
           t(i)=0;
        end
    end
elseif iZone == 3
    for i=1:1:n_iter
        x(i)=x_cam(i)/C;
        if (x_cam(i)<x_cam1)
            t(i)=C*(0.8232*x(i)^0.4941);%zone3
        elseif (x_cam(i)>x_cam1) && (x_cam(i)<x_cam2)
            t(i)=C*(-56.476*x(i)^6 + 129.12*x(i)^5 - 118.24*x(i)^4 + 56.666*x(i)^3 -
16.456*x(i)^2 + 2.8703*x(i) + 0.0696);%zone3
        elseif (x_cam(i)>x_cam2) && (x_cam(i)<x_cam3)
            t(i)=C*(65.209*x(i)^6 - 309.36*x(i)^5 + 610.82*x(i)^4 - 641.17*x(i)^3 +
376.88*x(i)^2 - 118.1*x(i) + 15.713);%zone3
        elseif (x_cam(i)>x_cam3)
            t(i)=C*(7.1679*x(i)^2 - 14.367*x(i) + 7.1992);%zone3
        end

        if t(i)<0
           t(i)=0;
        end
    end
end



%suction side coordinate
for i=1:1:n_iter
    x_S(i) = x_cam(i) - (t(i)/2)*sin(v_cam(i));
    y_S(i) = y_cam(i) + (t(i)/2)*cos(v_cam(i));
    y_S1(i) = y_S(i) + s;
end

%pressure side coordinate
for i=1:1:n_iter
    x_P(i) = x_cam(i) + (t(i)/2)*sin(v_cam(i));
    %x_P_test(i) = x_cam(i) - (t(i)/2)*sin(v_cam(i));
    y_P(i) = y_cam(i) - (t(i)/2)*cos(v_cam(i));
    y_P1(i) = y_P(i) + s;
end


%plot the blade
figure(1);
plot(x_cam,y_cam,'g')
hold on
```

```
plot(x_S,y_S,'r')
hold on
plot(x_P,y_P,'b')
hold on
%plot(x_P_test,y_P,'y')
%hold on
%for i=1:1000:(n_iter)
%    th = 0:pi/50:2*pi;
%    xunit = t(i)/2 * cos(th) + x_cam(i);
%    yunit = t(i)/2 * sin(th) + y_cam(i);
%    h = plot(xunit, yunit,'r');
%    hold on
%end
xf=[x_S';x_P'];
yf=[y_S';y_P'];
zf=zeros(2000,1);
Z=[xf yf zf];
% plot(x_cam,y_cam1,'g')
% hold on
% plot(x_S,y_S1,'r')
% hold on
% plot(x_P,y_P1,'b')
% hold on
%
% x_S=x_S'; x_P=x_P';
% y_P1=y_P1'; y_S1=y_S1';
%
% %plot(x_P_test,y_P1,'y')
% %hold on
% %for i=1000:1000:(n_iter-1000)
% %    th = 0:pi/50:2*pi;
% %    xunit = t(i)/2 * cos(th) + x_cam(i);
% %    yunit = t(i)/2 * sin(th) + y_cam1(i);
% %    h = plot(xunit, yunit,'r');
% %    hold on
% %end
%
% xlim([-4 44])
% %ylim([-0.2 1])
% ylim([-8 12])
%
% gamma*180/pi
```

```
% run the designturbine.m function with initial isentropic efficiency, then
% the iteration will keep going until it doesn't change any more

clc
clearvars
eta_sai=0.88;
eta_sbi=0.88;
eta_sci=0.88;
eta_sdi=0.89;
eta_sei=0.7;
count=0;
while count<=1000
    designturbine(eta_sai,eta_sbi,eta_sci,eta_sdi,eta_sei);
    eta=xlsread('turbine data.xlsx',1,'B71:F71');
    if abs(eta(1)-eta_sai)>eps||abs(eta(2)-eta_sbi)>eps||abs(eta(3)-eta_sci)>eps||abs(eta
(4)-eta_sdi)>eps||abs(eta(5)-eta_sei)>eps
        eta_sai=eta(1);
        eta_sbi=eta(2);
        eta_sci=eta(3);
        eta_sdi=eta(4);
        eta_sei=eta(5);
        count=count+1;
    else break
    end
end
```

```
% turbine design
% known : m, p_in, p_out, T_in, T_out, Dm_in, Dm_out, w angular speed.
% unkonwns: axial velocity, density, stage number, degree of reaction,
% stage load coefficient, flow coefficient.
function []=designturbine(eta_sa,eta_sb,eta_sc,eta_sd,eta_se)

m=151.3; % kg/s combustion gas mass flow rate
p_in=873350; % Pa static pressure at first stage of turbine stator blade.
p_out=102200; % Pa static pressure at exit of turbine, to the atmosphere.
T_in=1222.7; % K temperature at first stage of turbine stator blade.
T_out=806.77; % K temperature at exit of turbine.
D_in=1.062; % m inlet mean diameter
D_out=1.08; % m exit mean diameter
w=469.35; % rad/s angular velocity
k=1.333;
R=287;
cp=1148;
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   preliminary estimation    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
T_out_s=T_in*(p_out/p_in)^((k-1)/k);% isentropic exit temp
%eta=0.89; % total isentropic efficiency (T_in-T_out)/(T_in-T_out_s)
roh1a=p_in/R/T_in; % inlet density
roh_e=p_out/R/T_out; % exit density
% a b c d e corresponding stage 1, 2, 3, 4, 5
Da=D_in;

Db=(D_out-D_in)/4+D_in;

Dc=(D_out-D_in)/2+D_in;

Dd=(D_out-D_in)*3/4+D_in;

De=D_out;

% assume inlet velocity is axial, thus, alpha1 = 90.
U3a=w*Da/2; % rotational speed for first stage rotor m/s assume U1=U2=U3 for the first
stage of turbine
U3b=w*Db/2;
U3c=w*Dc/2;
U3d=w*Dd/2;
U3e=w*De/2; % rotational speed for last stage rotor m/s
% last stage degree of reaction is 0
%r_exit=0;
%r=0.5; % for first,second and third fourth stage degree of reaction
%assume phi = 0.7 as the stage flow coefficient, all the other parameters
%can be solved.
% assume first four stage load coefficient lambdaa
lambda1=1.7;
r=0.5;
phi=0.7;
```

```
alpha1a=90; % first stage stator inlet angle
alpha2=acot((1-r+lambda1/2)/phi)*180/pi;% first four stages rotor inlet angle
alpha3=acot((1-r-lambda1/2)/phi)*180/pi+180;% first four stages rotor outlet angle
alpha1b=alpha3; % stage 2,3,4 stator inlet angle
beta2=180-alpha3; % first 4 stages rotor metal angle
beta3=180-alpha2;% first 4 stages rotor metal angle
Vaxa=U3a*phi;
Vaxb=U3b*phi;
Vaxc=U3c*phi;
Vaxd=U3d*phi;
Vaxe=U3e*phi;
V3a=Vaxa/sin(alpha3*pi/180);
V3b=Vaxb/sin(alpha3*pi/180);
V3c=Vaxc/sin(alpha3*pi/180);
V3d=Vaxd/sin(alpha3*pi/180);

V1a=Vaxa;
V1b=V3a;
V1c=V3b;
V1d=V3c;
V1e=V3d;

W2a=V3a;
W2b=V3b;
W2c=V3c;
W2d=V3d;

U2a=U3a;
U2b=U3b;
U2c=U3c;
U2d=U3d;

V2a=Vaxa/sin(alpha2*pi/180);
V2b=Vaxb/sin(alpha2*pi/180);
V2c=Vaxc/sin(alpha2*pi/180);
V2d=Vaxd/sin(alpha2*pi/180);

W3a=V2a;
W3b=V2b;
W3c=V2c;
W3d=V2d;
% solve inlet area
A1a=m/roh1a/Vaxa;
h_b1a=A1a/pi/Da;
% velocity triangle solved
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% as lambda=lm/U^2=cp*(T01-T03)/U^2
T1a=T_in;
T01a=T1a+V1a^2/2/cp;
H01a=cp*T01a;
T2a=T01a-V2a^2/2/cp;
```

```
T02a=T01a;
M2a=V2a/sqrt(k*R*T2a);

T03a=T01a-lambda1*U3a^2/cp;
T3a=T03a-V3a^2/2/cp;
H03a=cp*T03a;
H02a=H01a;
p1a=p_in;
p01a=p1a*(T01a/T1a)^(k/(k-1));
%T2as=T2a-0.05*V2a^2/2/cp;

eta_n=0.89; % define the nozzle isentropic efficiency
T2as=(1-(1-T2a/T01a)/eta_n)*T01a;
p2a=p01a/((T01a/T2as)^(k/(k-1)));
roh2a=p2a/R/T2a;
p02a=p2a+V2a^2*roh2a/2;
A2a=m/roh2a/Vaxa;
h_b2a=A2a/pi/Da;
% the isentropic efficiency at first stage is eta, which is also the whole thermal
% efficiency
T3as=T1a-(T1a-T3a)/eta_sa;
p3a=p1a*(T3as/T1a)^(k/(k-1));
roh3a=p3a/R/T3a;
p03a=p3a+V3a^2/2*roh3a;
A3a=m/roh3a/Vaxa;
h_b3a=A3a/pi/Da;
dT0a=T01a-T03a;
%%% first stage design finished  %%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% for second stage the velocity triangle should be the same.the only
% difference is the absolute value since U change and the thermodynamic properties.


p1b=p3a;
roh1b=roh3a;
A1b=m/roh1b/Vaxb;
h_b1b=A1b/pi/Db;
T1b=T3a;
T01b=T1b+V1b^2/2/cp;
H01b=cp*T01b;
T2b=T01b-V2b^2/2/cp;
T02b=T01b;
H02b=H01b;
M2b=V2b/sqrt(k*R*T2b);
T2bs=(1-(1-T2b/T01b)/eta_n)*T01b;
T03b=T01b-lambda1*U3b^2/cp;
T3b=T03b-V3b^2/2/cp;
H03b=cp*T03b;
p01b=p1b*(T01b/T1b)^(k/(k-1));
p2b=p01b/((T01b/T2bs)^(k/(k-1)));
```

```
roh2b=p2b/R/T2b;
p02b=p2b+V2b^2*roh2b/2;
A2b=m/roh2b/Vaxb;
h_b2b=A2b/pi/Db;
% the isentropic efficiency at second stage is eta, which is also the whole thermal
% efficiency
T3bs=T1b-(T1b-T3b)/eta_sb;
p3b=p1b*(T3bs/T1b)^(k/(k-1));
roh3b=p3b/R/T3b;
p03b=p3b+V3b^2/2*roh3b;
A3b=m/roh3b/V3b;
h_b3b=A3b/pi/Db;
dT0b=T01b-T03b;

% for third stage
p1c=p3b;
roh1c=roh3b;
A1c=m/roh1c/Vaxc;
h_b1c=A1c/pi/Dc;
T1c=T3b;
T01c=T1c+V1c^2/2/cp;
H01c=cp*T01c;
T2c=T01c-V2c^2/2/cp;
T02c=T01c;
H02c=H01c;
M2c=V2c/sqrt(k*R*T2c);
T2cs=(1-(1-T2c/T01c)/eta_n)*T01c;
T03c=T01c-lambda1*U3c^2/cp;
T3c=T03c-V3c^2/2/cp;
H03c=cp*T03c;
p01c=p1c*(T01c/T1c)^(k/(k-1));
p2c=p01c/((T01c/T2cs)^(k/(k-1)));
roh2c=p2c/R/T2c;
p02c=p2c+V2c^2*roh2c/2;
A2c=m/roh2c/Vaxc;
h_b2c=A2c/pi/Dc;
% the isentropic efficiency at second stage is eta, which is also the whole thermal
% efficiency
T3cs=T1c-(T1c-T3c)/eta_sc;
p3c=p1c*(T3cs/T1c)^(k/(k-1));
roh3c=p3c/R/T3c;
p03c=p3c+V3c^2/2*roh3c;
A3c=m/roh3c/Vaxc;
h_b3c=A3c/pi/Dc;
dT0c=T01c-T03c;

%%%% for fourth stage %%%%%%%%%%
p1d=p3c;
roh1d=roh3c;
A1d=m/roh1d/Vaxd;
h_b1d=A1d/pi/Dd;
```

```
T1d=T3c;
T01d=T1d+V1d^2/2/cp;
H01d=cp*T01d;
T2d=T01d-V2d^2/2/cp;
T02d=T01d;
H02d=H01d;
M2d=V2d/sqrt(k*R*T2d);
T2ds=(1-(1-T2d/T01d)/eta_n)*T01d;
T03d=T01d-lambda1*U3d^2/cp;
T3d=T03d-V3d^2/2/cp;
H03d=cp*T03d;
p01d=p1d*(T01d/T1d)^(k/(k-1));
p2d=p01d/((T01d/T2ds)^(k/(k-1)));
roh2d=p2d/R/T2d;
p02d=p2d+V2d^2*roh2d/2;
A2d=m/roh2d/Vaxd;
h_b2d=A2d/pi/Dd;
% the isentropic efficiency at second stage is eta, which is also assumed as the whole
thermal
% efficiency
T3ds=T1d-(T1d-T3d)/eta_sd;
p3d=p1d*(T3ds/T1d)^(k/(k-1));
roh3d=p3d/R/T3d;
p03d=p3d+V3d^2/2*roh3d;
A3d=m/roh3d/Vaxd;
h_b3d=A3d/pi/Dd;
dT0d=T01d-T03d;


%%%%%% last stage %%%%%%%%
re=0.6;
roh1e=roh3d;
% based on the above lamda_exit=1
lambdae=2*(1-re);
alpha1e=alpha3;
alpha2e=acot((1-re+lambdae/2)/phi)*180/pi;
beta2e=acot((lambdae/2-re)/phi)*180/pi+180;
alpha3e=90;
beta3e=acot(-(lambdae/2+re)/phi)*180/pi+180;
V3e=Vaxe/sin(alpha3e*pi/180);
% Vaxe=Vaxd;
% V3e=Vaxd;
W2e=Vaxe/sin(beta2e*pi/180);
U2e=U3e;
V2e=Vaxe/sin(alpha2e*pi/180);
W3e=sqrt(Vaxe^2+U3e^2);

% solve inlet area
A1e=m/roh1e/Vaxe;
h_b1e=A1e/pi/De;
% velocity triangle solved
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% as lambda=lm/U^2=cp*(T01-T03)/U^2
T1e=T3d;
T01e=T1e+V1e^2/2/cp;
H01e=cp*T01e;
T2e=T01e-V2e^2/2/cp;
T02e=T01e;
H02e=H01e;
M2e=V2e/sqrt(k*R*T2e);
T2es=(1-(1-T2e/T01e)/0.8)*T01e;
T03e=T01e-lambdae*U3e^2/cp;
T3e=T03e-V3e^2/2/cp;
H03e=cp*T03e;
p1e=p3d;
p01e=p1e*(T01e/T1e)^(k/(k-1));
p2e=p01e/((T01e/T2es)^(k/(k-1)));
roh2e=p2e/R/T2e;
p02e=p2e+V2e^2*roh2e/2;
A2e=m/roh2e/Vaxe;
h_b2e=A2e/pi/De;
% the isentropic efficiency at first stage is eta, which is also the whole thermal
% efficiency
T3es=T1e-(T1e-T3e)/eta_se;
p3e=p1e*(T3es/T1e)^(k/(k-1));
roh3e=p3e/R/T3e;
p03e=p3e+V3e^2/2*roh3e;
A3e=m/roh3e/Vaxe;
h_b3e=A3e/pi/De;
dT0e=T01e-T03e;
% output result
p=[p1a p2a p3a p1b p2b p3b p1c p2c p3c p1d p2d p3d p1e p2e p3e];
T=[T1a T2a T3a T1b T2b T3b T1c T2c T3c T1d T2d T3d T1e T2e T3e];
roh=[roh1a roh2a roh3a roh1b roh2b roh3b roh1c roh2c roh3c roh1d roh2d roh3d roh1e roh2e
roh3e];
p0=[p01a p02a p03a p01b p02b p03b p01c p02c p03c p01d p02d p03d p01e p02e p03e];
T0=[T01a T02a T03a T01b T02b T03b T01c T02c T03c T01d T02d T03d T01e T02e T03e];
H0=[H01a H02a H03a H01b H02b H03b H01c H02c H03c H01d H02d H03d H01e H02e H03e];
thermo=[roh;T;p;p0;T0];
D=[Da Db Dc Dd De];
V=[V1a V2a V3a V1b V2b V3b V1c V2c V3c V1d V2d V3d V1e V2e V3e];
W=[0 W2a W3a 0 W2b W3b 0 W2c W3c 0 W2d W3d 0 W2e W3e];
U=[0 U2a U3a 0 U2b U3b 0 U2c U3c 0 U2d U3d 0 U2e U3e];
alpha=[alpha1a alpha2 alpha3 alpha1b alpha2 alpha3 alpha1b alpha2 alpha3 alpha1b alpha2
alpha3 alpha1b alpha2e  alpha3e];
beta=[0 beta2 beta3 0 beta2 beta3 0 beta2 beta3 0 beta2 beta3 0 beta2e  beta3e];
v=[V; W; U;alpha;beta];
h_b=[h_b1a h_b2a h_b3a h_b1b h_b2b h_b3b h_b1c h_b2c h_b3c h_b1d h_b2d h_b3d h_b1e h_b2e
h_b3e];
A=[A1a A2a A3a A1b A2b A3b A1c A2c A3c A1d A2d A3d A1e A2e A3e];
dT0=[dT0a dT0b dT0c dT0d dT0e];
for i=1:5
```

```
    Dt(3*(i-1)+1:i*3)=h_b(3*(i-1)+1:i*3)+D(i);
    Dh(3*(i-1)+1:i*3)=-h_b(3*(i-1)+1:i*3)+D(i);
end
geometry=[Dh; Dt; A; h_b];
designp=[r r r r re;lambda1 lambda1 lambda1 lambda1 lambdae];
% save it to excel
xlswrite('turbine data.xlsx',v,1,'B3:P7');
xlswrite('turbine data.xlsx',geometry,1,'B8:P11');
xlswrite('turbine data.xlsx',thermo,2,'B3:P7');
xlswrite('turbine data.xlsx',D,3,'B4:F4');
xlswrite('turbine data.xlsx',designp,3,'B2:F3');
```

```
clear all;

close all;


%Diffuser inlet ondition,Parameters from Turbine

p2 = 118300;                      %static pressure at inlet of diffuser

Cp = 1099;                        %heat capacity of air at T=804K

k = 1.354                         %heat capacity ratio

T2 = 804.87;

A2 = 1.6655;                      %annulus area at inlet of diffuser

rho2 = 0.51;

mfr = 151.3;                      %mass flow rate at the exit of turbine

R=287;

v2 = mfr/rho2/A2;

P20 = p2+0.5*rho2*v2*v2;

T20 = T2+v2^2/2/Cp;

c2 = sqrt(k*R*T2);

Ma2 = v2/c2;                      %Ma2 = 0.3124

p3 = 102200                       %diffuser exit static pressure

T3 = T20*(p3/P20)^((k-1)/k);


%calculation of nozzle inlet

D2_i = 0.958;

D3_i =0.958;

K = 2.5;                          % total pressure loss coefficient

P30 = P20-K*(P20-p2);             % exit total pressure

v3 = sqrt((P30-p3)*2/rho2);
```

```
rho3 = p3/R/T3;

c3 = sqrt(k*R*T3);

Ma3 = v3/c3;                        % exit Ma = 0.23

A3 = mfr/rho3/v3;

ARNd  = A3/A2;

D3_o = sqrt(4*A3/pi+D3_i^2);        %exit dia
```

```
clear all; close all; clc;

%Nozzle inlet ondition,Parameters from compressor

mfr = 150;                              %mass flow rate of air

k = 1.4;

R = 287;

D1_o = 1.4506;

D1_i = 0.958;

D0_i =D1_i;

A1 = pi/4*(D1_o^2-D1_i^2);              %nozzle exit annulus area

p1 = 98610;

T1 = 288.21;

rho = p1/R/T1;

v1 = mfr/rho/A1;

c1 = sqrt(k*R*T1);

Ma1 = v1/c1;

P1 = p1+0.5*rho*v1*v1;                  %nozzle exit total pressure

%calculation of nozzle inlet

p0 = 101325;

Ploss = 0.05*P1;                        %5percent pressure loss

v0 = sqrt((P1-p0-Ploss)/0.5/rho);

A0 = mfr/rho/v0;                        %nozzle inlet area

T0 = p0/rho/R;

Ma0 = v0/c1;                            %nozzle inlet Ma = 0.197

ARN  = A0/A1;                           %area ratio

D0_o = sqrt(4*A0/pi+D0_i^2);            %nozzle inlet dia
```