Cairo University Faculty of engineering
Aerospace department

Compressor design
And off-design project

Team (14)

SEC	BN	الاسم
1	4	احمد صلاح محمد
1	33	عبدالله هاني عبدالله
2	4	ماركو جمال فكري
2	17	محمد شيخ الدين اسماعيل
2	35	نور الدين مصطفى احمد

Contents

List of figures

List of Tables

1. Design problem

For the design problem we had to design a compressor achieving a pressure ratio equal to 3. The rotor relative Mach number is limited to 0.75 and the diffusion factor DF is limited to 0.5. The mass flow rate is equal to 50 kg/s and the design rotational speed is 10000 rpm. The design should maximize the compressor efficiency and minimize the number of blades which has a significant effect on the product cost.

1.1 Givens

```
pi_c = 3;
mdot = 50;
N = 10000;
DF = 0.5;
Tt1 = 288;
Pt1 = 101325;
Cp = 1004.5;
```

1.2 assumption

```
Cx = 150;
eta_c = 0.9;
```

1.3 Choosing values from given table.

Compressor	Lower limit-Upper limit
Flow coefficient ϕ	0.4 - 0.6
Loading coefficient Ψ	0.1 - 0.4
Degree of reaction R	0.5 - 1
Diffusion factor DF	0.4 - 0.6
Hub to tip ratio ζ	0.6 - 0.75
Height to chord ratio h/c	2 - 3.5

Table 1: given value ranges.

```
phi = 0.6;
epsi = 0.379285;
R = 0.5;
```

```
h_c = 3.5;
```

For maximizing the efficiency and decreasing the number of stages the following was chosen:

Coefficient	Value chosen	Reason
Flow Coefficient Φ	Maximum (0.6)	Decreases the number of stages.
Loading Coefficient Ψ	0.379285	Changing the number until the number of stages is a whole number.
Degree of reaction R	Minimum (0.5)	Decreases the number of stages.

Table 2: chosen coefficients values.

1.4 Calculations

1.4.1 Compressor total temperature ratio

```
Taw_c = 1+(pi_c^(0.4/1.4)-1)/eta_c
```

Taw_c = 1.4097

```
dela_t_c = (Taw_c-1)*Tt1
```

dela_t_c = 117.9962

1.4.2 Velocity triangles

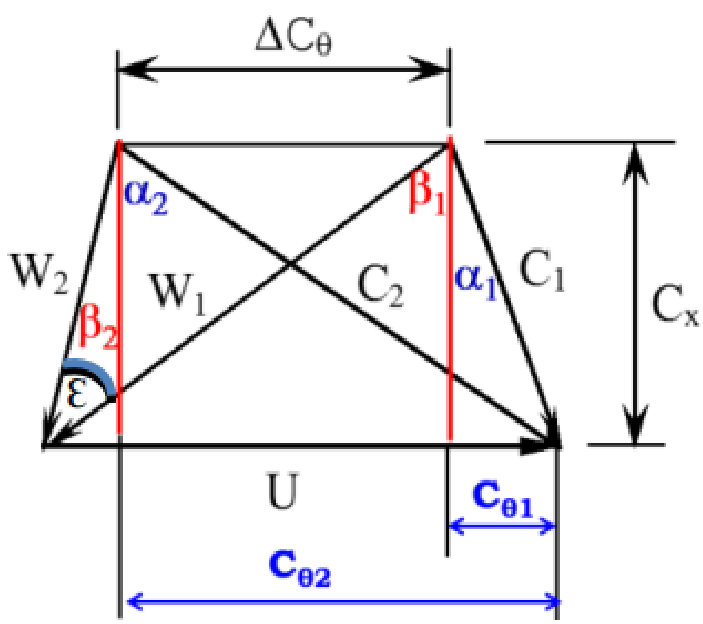


Figure 1: velocity triangle.

1.4.3 Velocity triangle 1

```
beta1 = atan((R+0.5*epsi)/phi)*180/pi
```

beta1 = 48.9762

```
alpha1 = atan((1-R-0.5*epsi)/phi)*180/pi
```

alpha1 = 27.3508

```
Ctheta1 =Cx*tan(alpha1*pi/180)
```

Ctheta1 = 77.5894

```
Wtheta1 = Cx*tan(beta1*pi/180)
```

Wtheta1 = 172.4106

```
W1 = Cx/cos(beta1*pi/180)
```

W1 = 228.5288

```
C1 = Cx/cos(alpha1*pi/180)
```

C1 = 168.8790

1.4.4 Velocity triangle 2

```
beta2 = atan((R-0.5*epsi)/phi)*180/pi
```

beta2 = 27.3508

```
alpha2 = atan((1-R+0.5*epsi)/phi)*180/pi
```

alpha2 = 48.9762

```
Ctheta2 =Cx*tan(alpha2*pi/180)
```

Ctheta2 = 172.4106

```
Wtheta2 = Cx*tan(beta2*pi/180)
```

Wtheta2 = 77.5894

```
W2 = Cx/cos(beta2*pi/180)
```

W2 = 168.8790

```
C2 = Cx/cos(alpha2*pi/180)
```

C2 = 228.5288

```
U = W1*sin(beta1*pi/180)+C1*sin(alpha1*pi/180)
```

U = 250

```
rm = 60*U/2/pi/N                    %blade mean radius
```

rm = 0.2387

1.4.5 Stage Work

```
W_s = U*(Ctheta2-Ctheta1)*mdot
```

W_s = 1.1853e+06

1.4.6 Stage temperature increase

```
Tt2 = Tt1 + W_s/mdot/1004.5
```

Tt2 = 311.5991

```
delta_t_s = Tt2-Tt1
```

delta_t_s = 23.5991

1.4.7 number of stages

$$Number\ of\ stages\ Z_{st} \cong \Delta T_c / \Delta T_{st}$$

```
Z = dela_t_c/delta_t_s
```

Z = 5.0000

1.4.8 Total compressor work

```
W = W_s*Z
```

W = 5.9264e+06

1.4.9 calculation of solidity

$$DF_R = 1 - \frac{W_2}{W_1} + \frac{|\Delta W_\theta|}{2\sigma_R W_1}$$

$$DF_S = 1 - \frac{C_3}{C_2} + \frac{|\Delta C_\theta|}{2\sigma_s C_2}$$

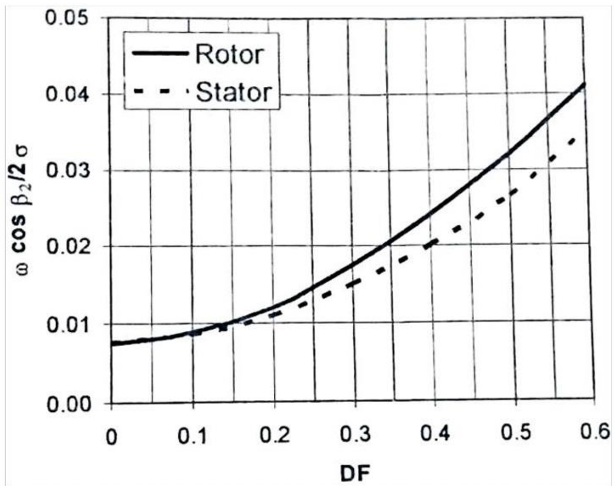


Figure 2: DF vs efficiency graph.

```
Solidity_r = abs(Wtheta1-Wtheta2)/2/W1/(DF+W2/W1-1)
```

Solidity_r = 0.8681

```
Solidity_s = abs(Ctheta1-Ctheta2)/2/C2/(DF+C1/C2-1)
```

Solidity_s = 0.8681

```
%from graph
omega_r = 0.0315;
omega_s = 0.0265;
```

1.4.10 calculation of temperatures and pressures

For rotor

```
T1 = Tt1-C1^2/2/1004.5
```

T1 = 273.8038

```
Ttrel1 = T1+W1^2/2/1004.5
```

Ttrel1 = 299.7996

```
T2 = Tt2-C2^2/2/1004.5
```

T2 = 285.6034

```
Ttrel2 = T2+W2^2/2/1004.5
```

Ttrel2 = 299.7996

```
P1 = Pt1*(T1/Tt1)^(1.4/0.4)
```

P1 = 8.4895e+04

```
Ptrel1 = P1*(Ttrel1/T1)^(1.4/0.4)
```

Ptrel1 = 1.1661e+05

$$\omega_R = \frac{P_{tr2}^s - P_{tr2}}{P_{tr1} - P_1} \rightarrow get\ P_{tr2}$$

```
Ptrel2 = Ptrel1-omega_r*(Ptrel1-P1)
```

Ptrel2 = 1.1562e+05

```
P2 = Ptrel2*(T2/Ttrel2)^(1.4/0.4)
```

P2 = 9.7561e+04

```
Pt2  = P2*(Tt2/T2)^(1.4/0.4)
```

Pt2 = 1.3234e+05

For stator

$$\omega_S = \frac{P_{t2} - P_{t3}}{P_{t2} - P_2} \rightarrow get\ P_{t3}$$

```
Pt3 = Pt2-omega_s*(Pt2-P2)
```

Pt3 = 1.3142e+05

1.4.11 Stage adiabatic efficiency

$$\pi_{st} = P_{t3}/P_{t1}, \qquad \tau_{st} = T_{t3}/T_{t1}, \qquad \eta_{st} = \left(\pi_{st}^{\frac{\gamma-1}{\gamma}} - 1\right)/(\tau_{st} - 1)$$

```
pi_s = Pt3/Pt1
```

pi_s = 1.2970

```
Taw_s = Tt2/Tt1
```

Taw_s = 1.0819

```
eta_s = (pi_s^(0.4/1.4)-1)/(Taw_s-1)
```

eta_s = 0.9413

1.4.12 Geometry

$$\dot{m} = \rho C_x A$$

$$A_n = \pi(r_t^2 - r_h^2) = 2\pi\left(\frac{r_t + r_h}{2}\right)(r_t - r_h) = 2\pi r_m h$$

$$\zeta = \frac{r_h}{r_t} \quad \text{(hub to tip ratio)}$$

```
rho1 = P1/287/T1
```

rho1 = 1.0803

```
rho2 = P2/287/T2
```

rho2 = 1.1902

```
A1 = mdot/rho1/Cx
```

A1 = 0.3085

```
A2 = mdot/rho2/Cx
```

A2 = 0.2801

```
h = A1/2/pi/rm
```

h = 0.2057

```
r_hub = (2*rm-h)/2
```

r_hub = 0.1359

```
r_tip = h+r_hub
```

r_tip = 0.3416

```
zeta = r_hub/r_tip
```

zeta = 0.3978

1.4.13 Calculation of Mach number at different stages

$$M = \frac{C}{\sqrt{\gamma RT}} \rightarrow absolute\ Mach\ no., \qquad M_r = \frac{W}{\sqrt{\gamma RT}} \rightarrow relative\ Mach\ no.$$

$$M_t = \frac{C}{\sqrt{\gamma RT_t}} \rightarrow absolute\ total\ M, \qquad M_{rt} = \frac{W}{\sqrt{\gamma RT_t}} \rightarrow relative\ total\ M$$

$$M_{ut} = \frac{U}{\sqrt{\gamma RT_t}} \rightarrow tangential\ total\ Mach\ number$$

For state 1

```
M1 = C1/sqrt(1.4*287*T1)
```

M1 = 0.5092

```
Mt1 = C1/sqrt(1.4*287*Tt1)
```

Mt1 = 0.4964

```
Mr1 = W1/sqrt(1.4*287*T1)
```

Mr1 = 0.6890

```
Mrt1 = W1/sqrt(1.4*287*Tt1)
```

Mrt1 = 0.6718

```
Mut1 = U/sqrt(1.4*287*Tt1)
```

Mut1 = 0.7349

For state 2

```
M2 = C2/sqrt(1.4*287*T2)
```

M2 = 0.6746

```
Mt2 = C2/sqrt(1.4*287*Tt2)
```

Mt2 = 0.6459

```
Mr2 = W2/sqrt(1.4*287*T2)
```

Mr2 = 0.4985

```
Mrt2 = W2/sqrt(1.4*287*Tt2)
```

Mrt2 = 0.4773

```
Mut2 = U/sqrt(1.4*287*Tt2)
```

Mut2 = 0.7065

1.4.14 Blade angles

using NACA 65-(15)10 airfoil

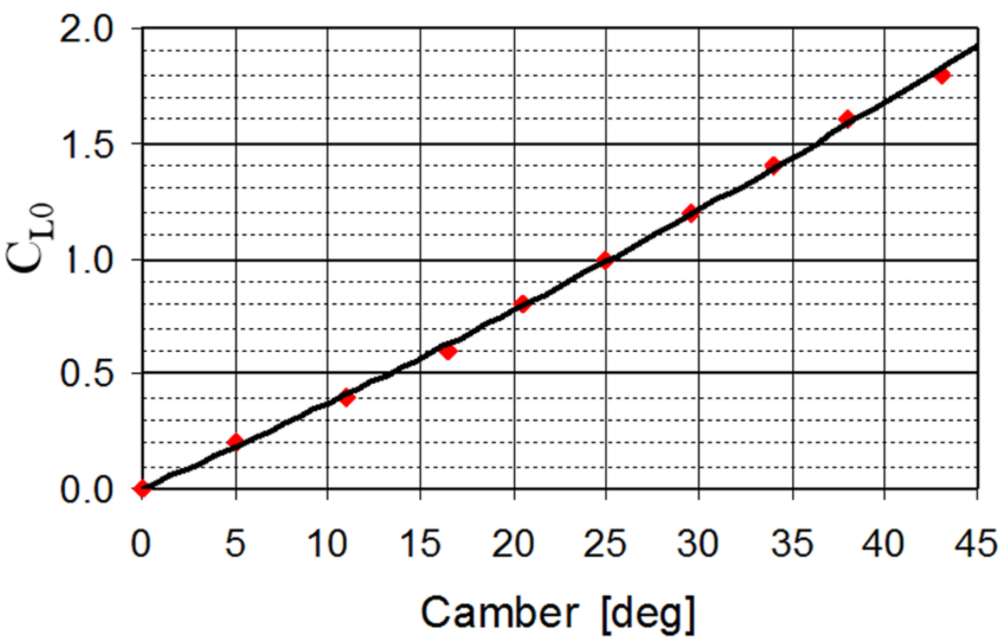


Figure 3: CLo vs Camber graph.

```
% from graphs
camber_angle = 36
```

camber_angle = 36

$$\delta = m\theta / \sigma^n$$

$$m = 0.23 \left(\frac{2a}{c}\right)^2 + \frac{\alpha_2}{500}$$

β_1	Inlet flow angle.	β_2	Outlet flow angle.
β_1'	Inlet blade angle.	β_2'	Outlet blade angle.
i	Incidence angle, $(\beta_1 - \beta_1')$.	δ	Deviation angle, $(\beta_2 - \beta_2')$.
θ	Camber angle, $(\beta_1' - \beta_2')$.	ε	Deflection angle, $(\beta_1 - \beta_2)$.
AoA	Angle of attack, $(\beta_1 - \beta_s)$	β_s	Stagger angle, $(\beta_1' + \beta_2')/2$

Table 3: angles calculations

```
m = 0.23*(2*0.5)^2+alpha2/500
```

m = 0.3280

```
deviation_angle = m*camber_angle/Solidity_r^0.5
```

deviation_angle = 12.6716

```
blade_angle_2 = beta2-deviation_angle
```

blade_angle_2 = 14.6793

```
blade_angle_1 = camber_angle+blade_angle_2
```

blade_angle_1 = 50.6793

```
incidence_angle = beta1-blade_angle_1
```

incidence_angle = -1.7031

```
deflection_angle = beta1-beta2
```

deflection_angle = 21.6254

```
stagger_angle = (blade_angle_1+blade_angle_2)/2
```

stagger_angle = 32.6793

```
AoA = beta1-stagger_angle
```

AoA = 16.2969

1.4.15 estimating axial compressor length and number of blades

```
chord = h/h_c
```

chord = 0.0588

```
length = 2*chord*Z+chord
```

length = 0.6465

```
Z_blades = ceil(2*pi*rm*Solidity_r/chord)
```

Z_blades = 23

2. Off design problem

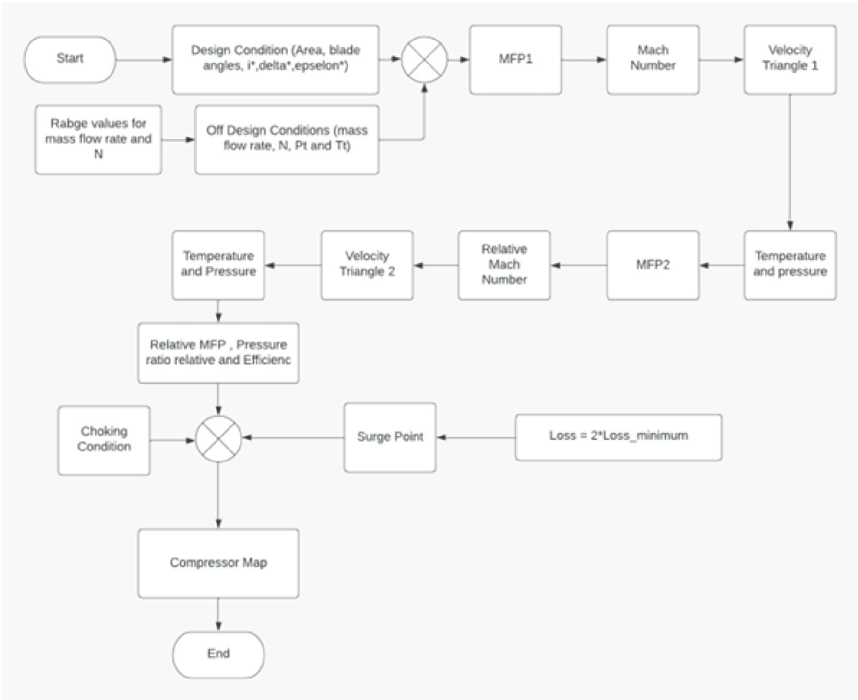


Figure 4: problem approach.

2.1 Inlet Gide vanes' function

In this section we design a MATLAB function that calculate outlet condition of inlet Gide veins

Step1: Evaluate the MFP_1 from the inlet total conditions and area A_1, then get M_1

$$MFP_1 = \frac{\dot{m}_1\sqrt{RT_{t1}}}{A_1 P_{t1}\cos\alpha_1} = \sqrt{\gamma}\, M_1\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\frac{1+\gamma}{2-2\gamma}} \rightarrow get\ \ M_1$$

$$\frac{T_t}{T} = 1 + \frac{C^2}{2\,c_p\,T} = 1 + \frac{\gamma-1}{2}M^2$$

$$\frac{P_t}{P} = \left(\frac{T_t}{T}\right)^{\frac{\gamma}{\gamma-1}} = \left(1+\frac{\gamma-1}{2}M^2\right)^{\frac{\gamma}{\gamma-1}}$$

$$Y_{IGV} = \frac{P_{t0}-P_{t1}}{P_{t1}-P_1}$$

```
function [Pt1,Tt1,alpha1,P1,M1,P0,T0]=igvsoff(Pt0,Tt0,md,A,alpha1_des)
y_loss=0.03;                        % offdesign loss coefficient for IGV
options = optimset('Display','off');
alpha1=alpha1_des;                  %For inlet guide vane the losses remains constant for M1<=0.5
[Mo]=fsolve(@(Mo) (md*sqrt(287*Tt0))/(Pt0*A*cosd(alpha1))-
(sqrt(1.4)*Mo/((1+.2*Mo^2)^3)),.35,options);
P0=Pt0/((1+.2*Mo^2)^3.5);
T0=Tt0/(1+.2*Mo^2);
Tt1=Tt0;
% To calc the pressure substitute in loss equation for IGV
[x]=fsolve(@(x) [x(1)-x(2)*(1+.2*x(3)^2)^3.5; y_loss-(Pt0-x(1))/(x(1)-x(2));...
md*sqrt(287*Tt1)/(x(1)*A*cos(alpha1*pi/180))-sqrt(1.4)*x(3)*(1+.2*x(3)^2)^(-3)],[10^5 9*10^4
.5],options);
Pt1=x(1);
P1=x(2);
M1=x(3);
```

2.2 Off design Calculations function

```
function [Pt3,Tt3,alpha3,max_eff,inced1,inced2]=offdesign(Pt1,Tt1,mdot,u1,A1,A2,alpha1)

options = optimset('Display','off');
```

2.3 Fitting of The Losses Curve

```
x_inc=[-.8 -.6 -.4 -.2 0 .2 .4 .6 .8];
y_loss=[.47 .3 .19 .1 .07 .1 .19 .3 .47];
w_loss=polyfit(x_inc,y_loss,2);
```

2.4 Fitting of The Deviation Curve

```
x_inc=[0 0.2  0.4 0.6 0.8];
y_dev=[0 0.05 0.2 0.4 0.7];
d_dev=polyfit(x_inc,y_dev,2);
```

2.5 At design

```
delta_des = 12.76; % [deg] eps=beta2-beta2d
eps_des = 21.64;   % [deg] eps=beta1-beta2
i_star=-1.703;      % [deg] same incidence angle for stator and rotor due to symmetry
```

2.6 Across Rotor

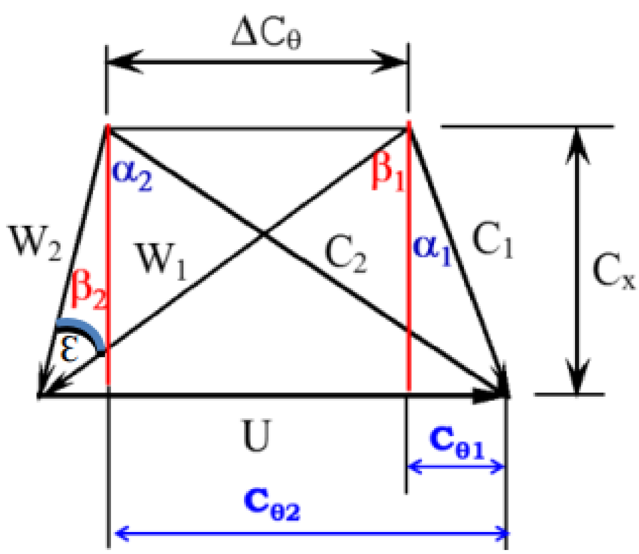
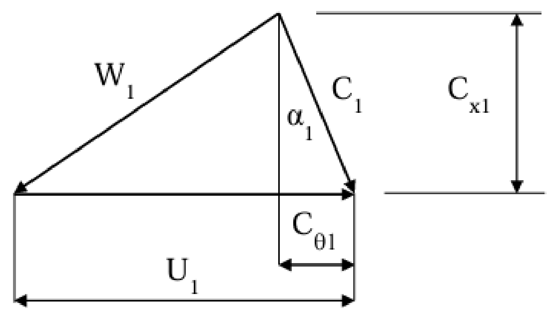


Figure 5: velocity triangle (off design)

Step2: From $T_{t1}, M_1, \alpha_1, U_1 \rightarrow$ get T_1, C_1, W_1, β_1 (complete vel. triangle **1**) and i_R

$$T_1 = \frac{T_{t1}}{\left(1 + \frac{\gamma - 1}{2} M_1^2\right)}, \quad C_1 = M_1 \sqrt{\gamma R T_1}, \qquad C_{x1} = C_1 \cos \alpha_1, \qquad W_1 = \frac{C_{x1}}{\cos \beta_1}$$

$$(\tan \beta_1 + \tan \alpha_1)\, C_{x1} = U_1 \rightarrow \text{get } \beta_1, \qquad \text{then} \rightarrow \text{get } i_R = \beta_1 - \beta_1'$$



Step3: From T_1, W_1 get T_{tr1}, then from isentropic relations with P_{t1} get P_1, P_{tr1}

$$T_{tr1} = T_1 + \frac{W_1^2}{2\, c_p}, \qquad P_1 = P_{t1} \left(\frac{T_1}{T_{t1}}\right)^{\frac{\gamma}{\gamma-1}}, \qquad P_{tr1} = P_{t1} \left(\frac{T_{tr1}}{T_{t1}}\right)^{\frac{\gamma}{\gamma-1}}$$

Recall that, the Rothalpy is constant across rotor which indicates that,

$$T_{tr2} = T_{tr1} + \frac{U_2^2 - U_1^2}{2 c_P}, \qquad \frac{P_{tr2}^s}{P_{tr1}} = \left(\frac{T_{tr2}}{T_{tr1}}\right)^{\frac{\gamma}{\gamma-1}}, \quad \text{so if } U_1 = U_2 \rightarrow T_{tr2} = T_{tr1}$$

Step 4: Knowing the rotor incidence (i_R), from Fig. 3. 31 or Fig. 3. 32 we can get

- Rotor deviation δ_R (or deflection ε_R) then calculate β_2.
- Rotor non-dimensional loss coefficient (ω_R), then calculate P_{tr2}.

$$\delta_R = \beta_2 - \beta_2' \rightarrow get\ \beta_2, \qquad \omega_R = \frac{P_{tr2}^s - P_{tr2}}{P_{tr1} - P_1} \rightarrow get\ P_{tr2}$$

Step 5: Evaluate MFP_{r2} from the total relative outlet conditions (T_{tr2}, P_{tr2}), β_2 and the rotor outlet area A_2, hence get M_{r2}

$$MFP_{r2} = \frac{\dot{m}_2 \sqrt{R T_{tr2}}}{A_2 P_{tr2} \cos \beta_2} = \sqrt{\gamma}\, M_{r2} \left(1 + \frac{\gamma - 1}{2} M_{r2}^2\right)^{\frac{1+\gamma}{2-2\gamma}} \rightarrow get\ \ M_{r2}$$

Step 6: From $T_{tr2}, M_{r2}, \beta_2, U_2 \rightarrow$ get T_2, W_2, C_2, α_2 (complete vel. triangle **2**)

$$T_2 = \frac{T_{tr2}}{\left(1 + \frac{\gamma - 1}{2} M_{r2}^2\right)}, \qquad W_2 = M_{r2}\sqrt{\gamma R T_2}, \qquad C_{x2} = W_2 \cos\beta_2$$

$$(\tan\alpha_2 + \tan\beta_2) C_{x2} = U_2 \rightarrow \alpha_2, \qquad C_2 = C_{x2}/\cos\alpha_2$$

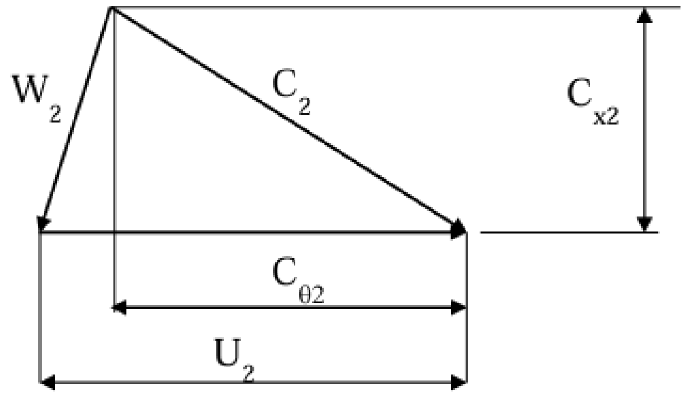


Figure 6: outlet velocity triangle:

Step 7: Evaluate rotor outlet total temperature T_{t2} and outlet pressures P_{t2}, P_2

$$T_{t2} = T_2 + \frac{C_2^2}{2\,c_p}, \qquad P_{t2} = P_{tr2}\left(\frac{T_{t2}}{T_{tr2}}\right)^{\frac{\gamma}{\gamma-1}}, \qquad P_2 = P_{tr2}\left(\frac{T_2}{T_{tr2}}\right)^{\frac{\gamma}{\gamma-1}}$$

```
beta1d=41.7; % [deg]
beta2d=16.7; % [deg]
[M1]=fsolve(@(M1) (mdot*sqrt(287*Tt1))/(Pt1*A1*cos(alpha1*pi/180))-(sqrt(1.4)*M1*(1+0.2*M1^2)^-
3),.5,options);
T1=Tt1/(1+.2*M1^2);
P1=Pt1/((1+.2*M1^2)^3.5);
c1=M1*sqrt(1.4*287*T1);
cx1=c1*cos(alpha1*pi/180);
cth1=c1*sind(alpha1);
wth1=u1-cth1;
beta1=atand(wth1/cx1);       %[deg]
i_r=beta1-beta1d;
inc_r=(i_r-i_star)/(eps_des);
max_eff=0;
% In Order To Give Indication That The Incidence Exceed The Staling limits
% (I Assume The Limits Are That of The Range Of The Given Curves)
if (inc_r > .8) || (inc_r < -.8)
    max_eff=1;
end
if inc_r <=0
```

```
    dev_r=0;
else
dev_r=polyval(d_dev,inc_r);
end
deltar=dev_r*eps_des+delta_des; % [deg]
beta2=(beta2d+deltar);          %[deg]
wr=polyval(w_loss,inc_r);
w1=cx1/cos(beta1*pi/180);
Ttr1=T1+(w1^2/(2*1004.5));
Ptr1=P1*((Ttr1/T1)^3.5);
Ptr2=Ptr1*(1-wr)+P1*wr;
Ttr2=Ttr1;                     %Since Rothalpy is constant across the Rotor
[Mrel2]=fsolve(@(Mrel2) (mdot*sqrt(287*Ttr2))/(Ptr2*A2*cos(beta2*pi/180))-
(sqrt(1.4)*Mrel2*((1+.2*Mrel2^2)^-3)),0.4,options);
T2=Ttr2/(1+.2*Mrel2^2);
P2=Ptr2/((1+.2*Mrel2^2)^3.5);
w2=Mrel2*sqrt(1.4*287*T2);
cx2= w2*cos(beta2*pi/180);
wth2=w2*sin(beta2*pi/180);
cth2=u1-wth2;
c2=sqrt(cx2^2+cth2^2);
M2=c2/sqrt(1.4*287*T2);
Tt2=T2*(1+.2*M2^2);
Pt2=P2*((1+.2*M2^2)^3.5);
```

2.7 Across Stator:

```
alpha2d=60; %[deg]
alpha3d=40; %[deg]
Tt3=Tt2;                      %Since No Heat added and no work across the rotor
alpha2=atan(cth2/cx2)*(180/pi);       % [deg]
i_s=alpha2-alpha2d;
inc_s=(i_s-i_star)/(eps_des);
if (inc_s > .8) || (inc_s < -.8)
    max_eff=1;
end
if inc_s <=0
    dev_s=0;
else
dev_s=polyval(d_dev,inc_s);
end
deltas=dev_s*eps_des+delta_des;%[deg]
alpha3=alpha3d+deltas; % [deg]

ws=polyval(w_loss,inc_s);
Pt3=Pt2*(1-ws)+ws*P2;
inced1=(beta1-beta1d);   %[deg] Incidence to rotor
inced2=(alpha2-alpha2d); %[deg] Incidence to stator
```

from design problem

```
U=250;
m_dot_des=50;
Pt0=10^5;
Tt0=288;
A=[ 0.2801 0.3085 0.2801 0.3085];
alpha1_des=27.3508;
n_stage=1;
pi_c_des=3;
i_star=-1.703;
delta_star=12.62;
epsi_star=21.64;
```

Calculations

Group	1	2	3	4	5	6	7	8	9	10	11	12
% N_{rel}	50	55	60	65	70	75	80	85	90	95	105	110

Table 4: Nrel for teams

But we are group 14 so we use N_{rel}=120

```
N_rel=[0.5:0.1:1.4];
mdot_rel=[0.1:0.005:1.4];
for count1=1:length(N_rel)
    U_off=U*N_rel(count1);
    Limt1=0;
    for count2=1:length(mdot_rel)
        mdot=mdot_rel(count2)*m_dot_des;
        [Pt1,Tt1,alpha1,P1,M1,P0,T0]=igvsoff(Pt0,Tt0,mdot,A(1),alpha1_des);
        for n=1:n_stage

[Pt3,Tt3,alpha3,max_eff,ince_off(2*n),ince_off(2*n+1)]=offdesign(Pt1,Tt1,mdot,U_off,A(2*n),A(2*n+
1),alpha1);
            Pt1=Pt3;
            Tt1=Tt3;
            alpha1=alpha3;
        end
        mfpout_off=mdot*sqrt(Tt3)/Pt3;
        pic_off=Pt3/Pt0;
        toic_off=Tt3/Tt0;
        etac_off=(pic_off^(.4/1.4)-1)/(toic_off-1);
        % Applying The Condition That Efficiency is Between 0 & 1 %
        if (etac_off > 1) | (etac_off < 0) | (max_eff == 1)
        else
            pic_rel(count1,count2)=pic_off/pi_c_des;
            etac_rel(count1,count2)=etac_off;
            % To Get The surge points %
            if pic_rel(count1,count2) > Limt1
                picmax(count1)=pic_rel(count1,count2);
```

```
                mdmin(count1)=mdot_rel(count2);
                beg(count1)=count2;
                Limt1=pic_rel(count1,count2);
            end
            mdrr(count1,count2)=mdot_rel(count2);
            en(count1)=count2;
        end

    end
end
```

plots

```
for count1=1:length(N_rel)

plot(mdrr(count1,beg(count1):en(count1)),pic_rel(count1,beg(count1):en(count1)),'color',rand(1,3)
,'LineWidth',3)
    hold on
end
surge=polyfit(mdmin,picmax,5);
plot(mdot_rel,polyval(surge,mdot_rel),'b-.','LineWidth',2)
axis([0.2 0.6 0.2 0.6])
legend('At Part Speed = 50%','At Part Speed = 60%','At Part Speed = 70%','At Part Speed =
80%','At Part Speed = 90%','At Part Speed = 100%','At Part Speed = 110%','At Part Speed =
120%','At Part Speed = 130%','The Surge Line','The Operating Line')
title('Compressor Map','FontWeight','bold','color','b')
xlabel('MFP)_{rel}','FontWeight','bold','color','r')
ylabel('\pi_{c})_{rel}','FontWeight','bold','color','r')
grid on

figure(2)
for count1=1:length(N_rel)
    xi = mdrr(count1,beg(count1):en(count1));
    zi = etac_rel(count1,beg(count1):en(count1));
    plot(xi,zi,'color',rand(1,3),'LineWidth',3)
    hold on
end
legend('At Part Speed = 50%','At Part Speed = 60%','At Part Speed = 70%','At Part Speed =
80%','At Part Speed = 90%','At Part Speed = 100%','At Part Speed = 110%','At Part Speed =
120%','At Part Speed = 130%','The Surge Line')
title('Efficiency Variation','FontWeight','bold','color','r')
xlabel('MFP)_{rel}','FontWeight','bold','color','b')
ylabel('\eta_{c})_{rel}','FontWeight','bold','color','b')
grid on
```

3.0 compressor Surge

3.1 Introduction

Compressors are workhorses in various industries, playing a critical role in processes ranging from refrigeration and air conditioning to natural gas transportation and power generation. Their function is simple: to increase the pressure of a gas. However, operating a compressor comes with a crucial limitation – surge. This part delves into the intricacies of compressor surge, exploring the surge line, surge point, and surge cycle. It further examines the detrimental effects of the surge, the limited benefits it offers, and various strategies for preventing and detecting this unwanted phenomenon.

3.2 Understanding the Limits: Surge Line and Surge Point

3.2.1 Surge Line:

Compressor performance is often visualized on a performance map with mass flow rate on one axis and pressure ratio on the other. The surge line on this map represents the boundary between stable and unstable operating conditions for the compressor. It's not a single point but a curve that encompasses various operating conditions where instabilities like surges can occur. The shape and position of the surge line depend on various factors, including compressor design, speed, and inlet conditions.

3.2.2 Surge Point:

While the entire surge line signifies potential instability, a specific point on this line marks the critical limit. This point represents the operating condition where the compressor can no longer maintain stable flow against the system's resistance (pressure drop across the compressor). Beyond this point, a surge sets in. Operating a compressor even close to the surge line is risky, as any minor disturbance can push the operating point into the unstable zone.

3.3 The Unwanted Guest: Surge Cycle and its Effects

3.3.1 Surge Cycle:

When a surge occurs, the flow through the compressor momentarily reverses. This creates a characteristic cycle on the performance map. The operating point oscillates rapidly between the high-pressure discharge side and the low-pressure suction side, causing significant pressure fluctuations and vibrations. This cycle can repeat several times per second, leading to a highly unstable operating condition.

3.3.2 Effects of Surge:

Surge is highly detrimental to compressor performance and can lead to several negative consequences:

- **Reduced Efficiency:** The flow reversal and instability during surge disrupt the normal compression process, leading to a significant decrease in compressor efficiency.
- **Increased Wear and Tear:** The rapid pressure fluctuations and vibrations caused by the surge put immense stress on the compressor's internal components, accelerating wear and tear on blades, impellers, bearings, and seals.

- **Potential Damage:** In severe surge events, the intense forces can lead to structural damage to blades, impellers, and other compressor components. This can necessitate costly repairs or even complete compressor replacement.
- **System Instability and Process Upsets:** Surge can cause significant pressure fluctuations throughout the entire system connected to the compressor. This can disrupt downstream processes and potentially lead to safety hazards.

3.4 Surge: A Double-Edged Sword? (Pros and Cons)

There's no upside to the surge itself. However, understanding the surge line provides some benefits:

- **Safe Operational Boundaries:** Knowing the surge line allows engineers to set safe operating limits for the compressor. By keeping the operating point well within the stable region of the performance map, surge can be prevented.
- **Compressor Design:** During the design phase of a compressor, surge line data is crucial. It helps engineers optimize the compressor's performance for expected operating conditions and ensure it can handle the required pressure ratios without entering the surge zone.

3.5 Keeping the Flow Going: Surge Control Strategies

Since surge is highly undesirable, various strategies are employed to prevent it:

3.5.1 Inlet Guide Vanes (IGVs):

These adjustable vanes are located at the compressor inlet. By adjusting the angle of the IGVs, the flow entering the impeller can be regulated. This allows for fine-tuning the compressor's performance and keeping the operating point away from the surge line.

3.5.2 Anti-Surge Valves:

These valves act as a safety measure. During potential surge situations, an anti-surge valve opens a bypass route, diverting some gas flow from the compressor discharge back to the inlet. This reduces the pressure rise across the compressor, preventing it from entering the surge zone. However, using an anti-surge valve comes at the cost of reduced compressor efficiency.

3.5.3 Variable Speed Drives (VSDs):

Modern compressors can be equipped with variable speed drives. By adjusting the compressor's speed, the operating point on the performance map can be shifted. This allows for adapting the compressor's performance to varying system demands while keeping it away from the surge line.

3.5.4 Control Systems:

Sophisticated control systems are employed in modern compressor applications. These systems continuously monitor various compressor parameters, such as pressure, flow rate, and speed. By analyzing these parameters in real time, the control system can take corrective actions.

3.6 How do we plot the surge line in MATLAB?

1- In each operating line detect the surge point -the point at which the maximum pressure ratio and minimum mass flow rate-
2- Save all of these points in a vector
3- Use poly-fit order in MATLAB to get the polynomial equation of the surge line. Give it surge points and the polynomial degree and it will return the equation
4- Use poly-Val order in MATLAB to evaluate the value of surge at different mass flow rates. Give it the surge equation and mass flow rate vector.
5- Plot surge line equation after evaluation in the y-axis with a mass flow rate in the x-axis

```
surge=polyfit(mdmin,picmax,5);
plot(mdot_rel,polyval(surge,mdot_rel),'b-.','LineWidth',2)
```

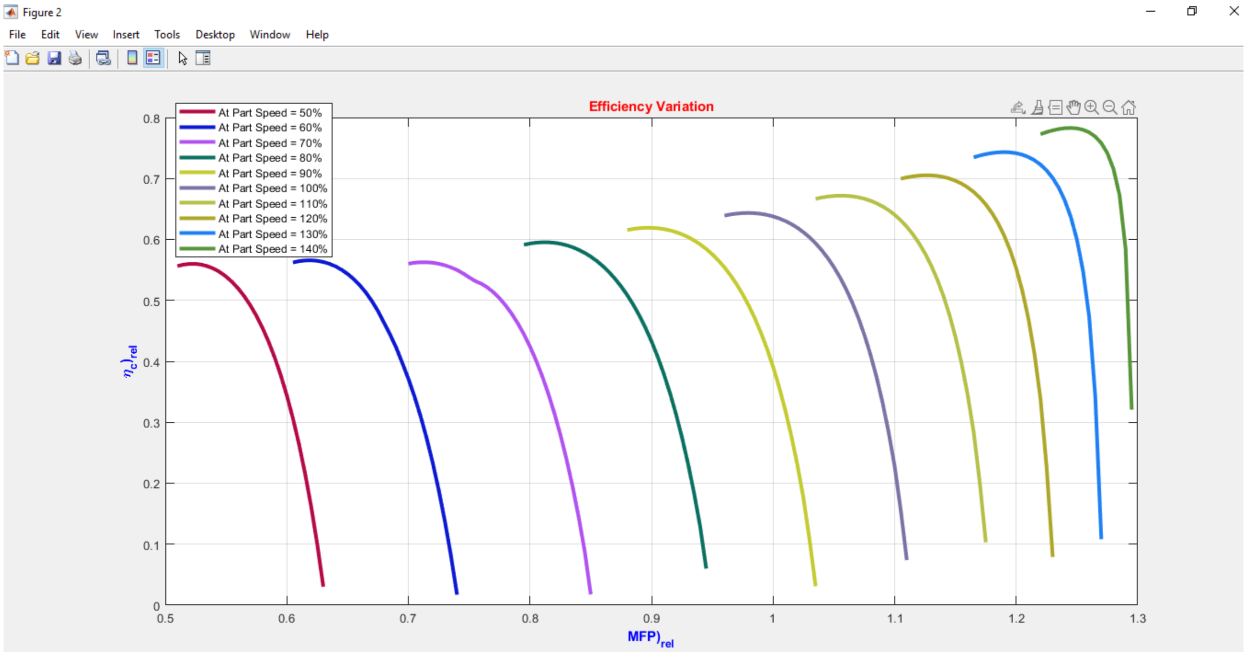


Figure 7: Efficiency variation.

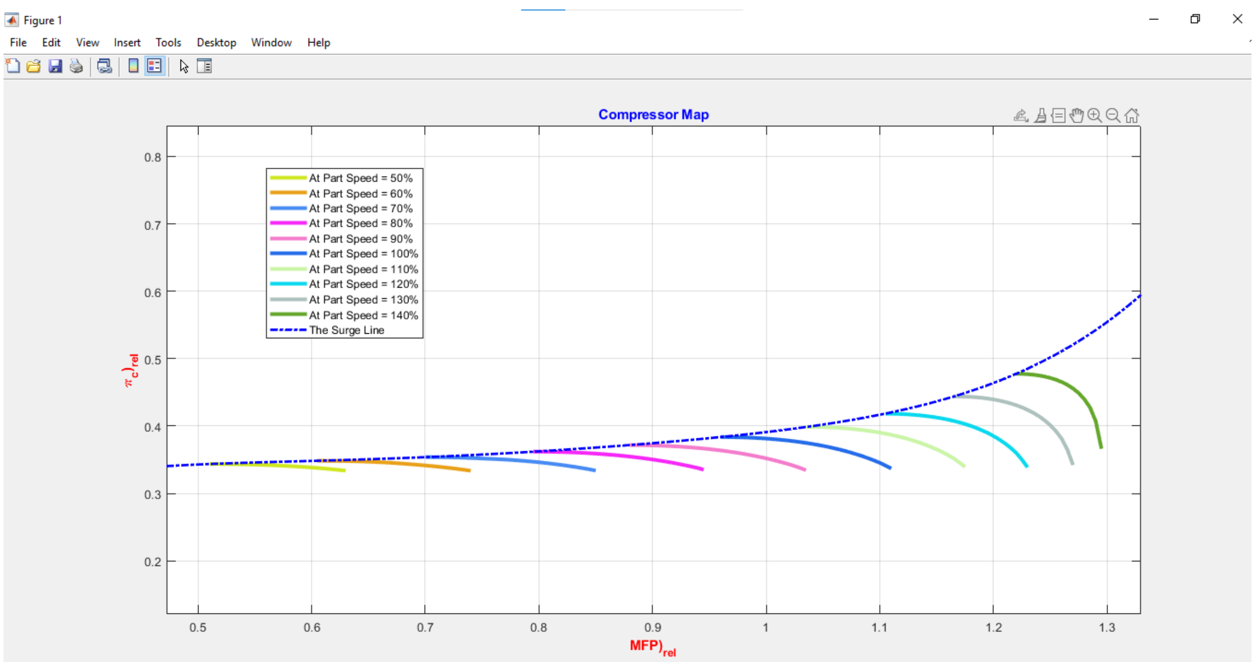


Figure 8Compressor map.

4.0 CFturbo

4.1 Introduction

CF Turbo is a user-friendly computational fluid dynamics (CFD) software tailored for analyzing and optimizing turbomachinery components like pumps and fans. It offers intuitive geometry creation, automated mesh generation, comprehensive simulation setup, and efficient post-processing tools. With CF Turbo, engineers can swiftly design, simulate, and optimize fluid flow systems across various industries, saving time and enhancing product performance. Our main focus while using it was the geometry which will be later on used on Ansys software for further CFD analysis.

Figure 9: CFturbo Logo

4.2 Methodology

The design methodology involves iterative steps, starting from initial geometry creation to final optimization. CFturbo software is utilized for geometric modeling, blade design, and flow path optimization. The methodology emphasizes performance-driven design iterations to achieve the desired compressor characteristics.

4.3 Geometry Creation

The geometry creation phase involves defining the compressor's basic geometry, including blade profiles, hub, and shroud contours. CFturbo's intuitive interface is utilized to create and modify the compressor geometry according to design requirements and constraints. A design case was solved using a MATLAB code giving us the parameters needed to start working on the CFturbo software. The code provided necessary inputs for the geometry including rm which was calculated to be 238.7 mm with a hub and shroud radius of 135.9 mm and 341.6 mm and blade angles for IGV (0 ,14.679), Rotor (50.6793, 14.6793) & Stator (50.6793, 14.6793) these were the main values needed to commence the work.

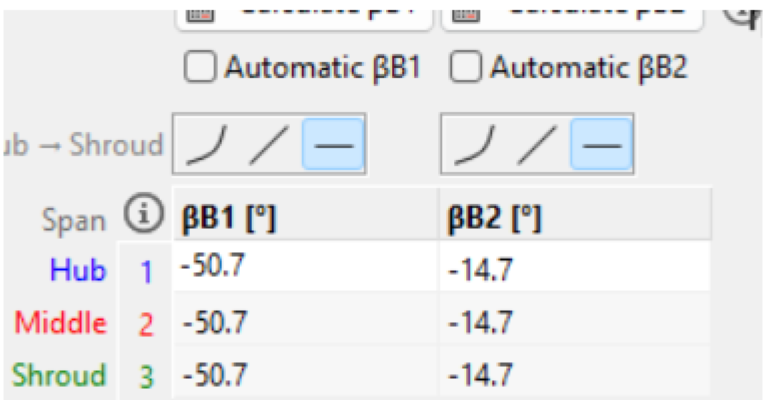


Figure 10: inserting rotor blade angles in CF turbo.

4.4 Blade Design

Blade design is a critical aspect of axial compressor performance. CFturbo's blade design module allows for the creation of custom blade profiles, including chord length, twist angle, and stacking parameters. The blade design process focuses on optimizing aerodynamic performance and minimizing losses.

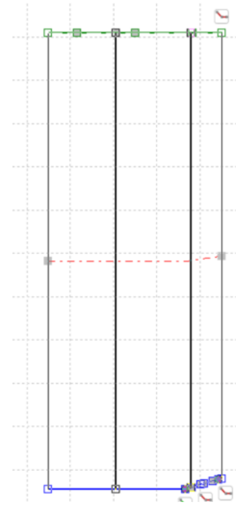


Figure 11: IGV.

4.5 Meridional Flow Path

The meridional flow path design determines the overall shape and dimensions of the compressor's flow channel. CFturbo's meridional contour editor is employed to define the flow path geometry, including inlet and outlet shapes, throat area, and diffusion characteristics. The meridional flow path is optimized to achieve uniform flow distribution and minimize losses.

Figure 12: 3D Model.

4.6 Optimization

Optimization techniques, such as parameter sweeps and genetic algorithms, are employed to refine the compressor design iteratively. CFturbo's optimization tools facilitate rapid exploration of design space and identification of optimal configurations. The optimization process aims to maximize compressor performance while satisfying design constraints.

4.7 Conclusion

In conclusion, the utilization of CFturbo software in the design and optimization of the axial compressor geometry has been instrumental in achieving our project objectives efficiently. By leveraging its user-friendly interface and robust features, we were able to create, simulate, and optimize the compressor geometry with precision and accuracy. The initial geometry parameters obtained from the MATLAB code provided a solid foundation for our design process, ensuring that we started with relevant and accurate inputs. Through iterative steps, focusing on blade design, meridional flow path optimization, and overall performance-driven design iterations, we were able to refine the compressor geometry to meet the desired performance criteria. The optimization techniques available in CFturbo enabled us to explore the design space effectively and identify optimal configurations that maximize compressor performance while adhering to design constraints. Overall, CFturbo has proven to be a valuable tool in our workflow, streamlining the design process and paving the way for further analysis using ANSYS software.

With the optimized compressor geometry in hand, we are well-positioned to conduct comprehensive CFD analysis and further refine our design for enhanced performance and efficiency.

5.0 Turbo grid and CFX

5.1 CFturbo Software Analysis

- The following figure shows the CF turbo meridian plane with accurate dimensions and blade angles.

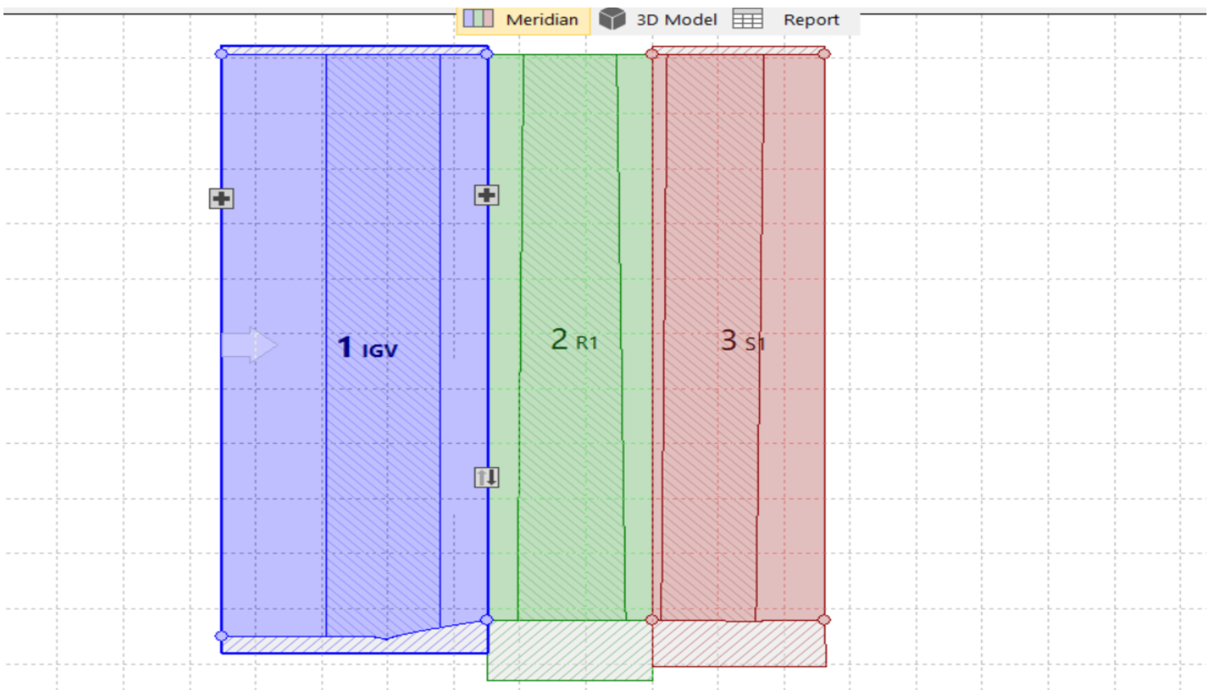


Figure 13: blades on CFturbo.

- The following is the Ansys workbench we constructed

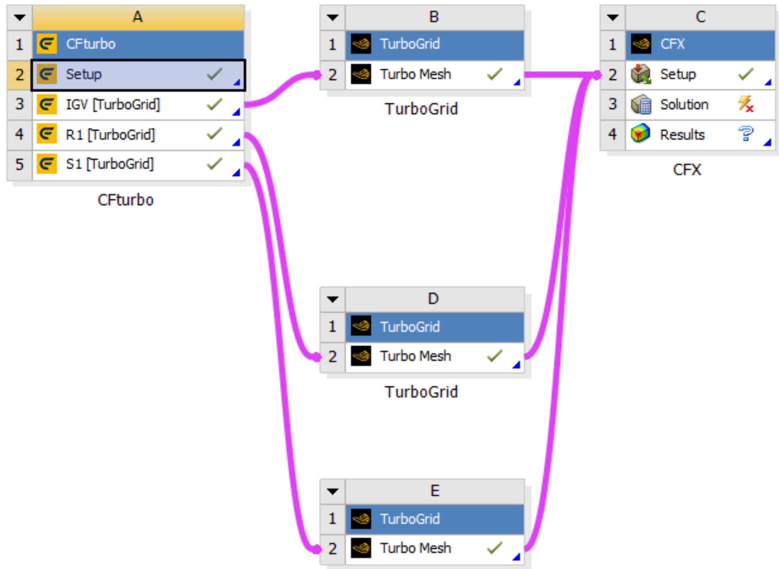


Figure 14: Ansys workbench.

- Meshed full Compressor with rotor stator and IGVs.

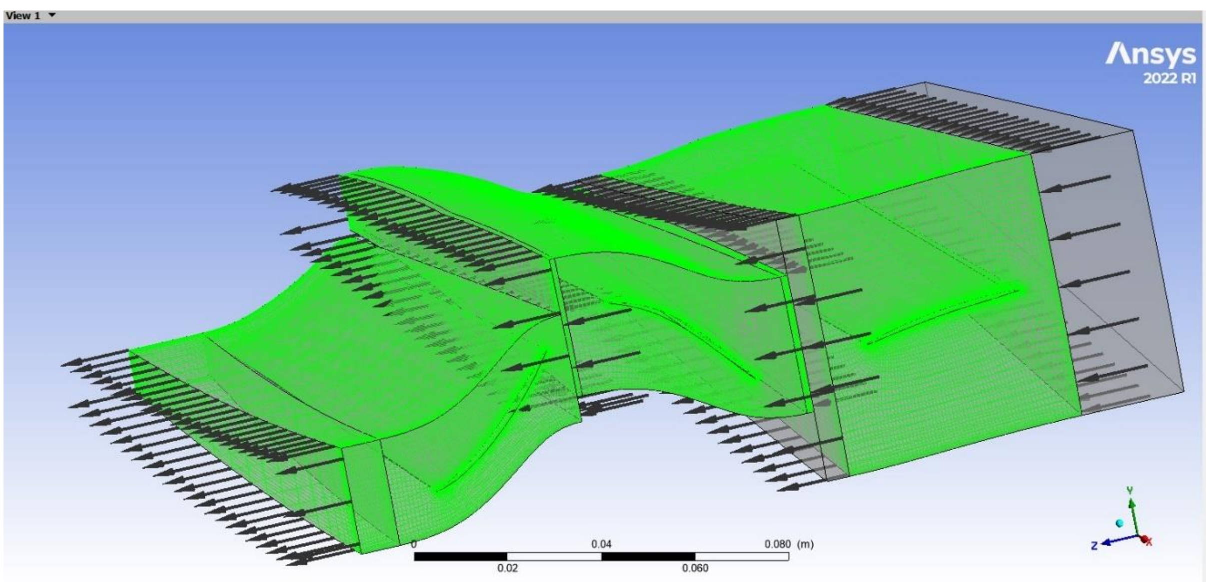


Figure 15: segment ready for analysis.

- Established boundary conditions including inlet conditions, rotational speed, and geometric parameters.

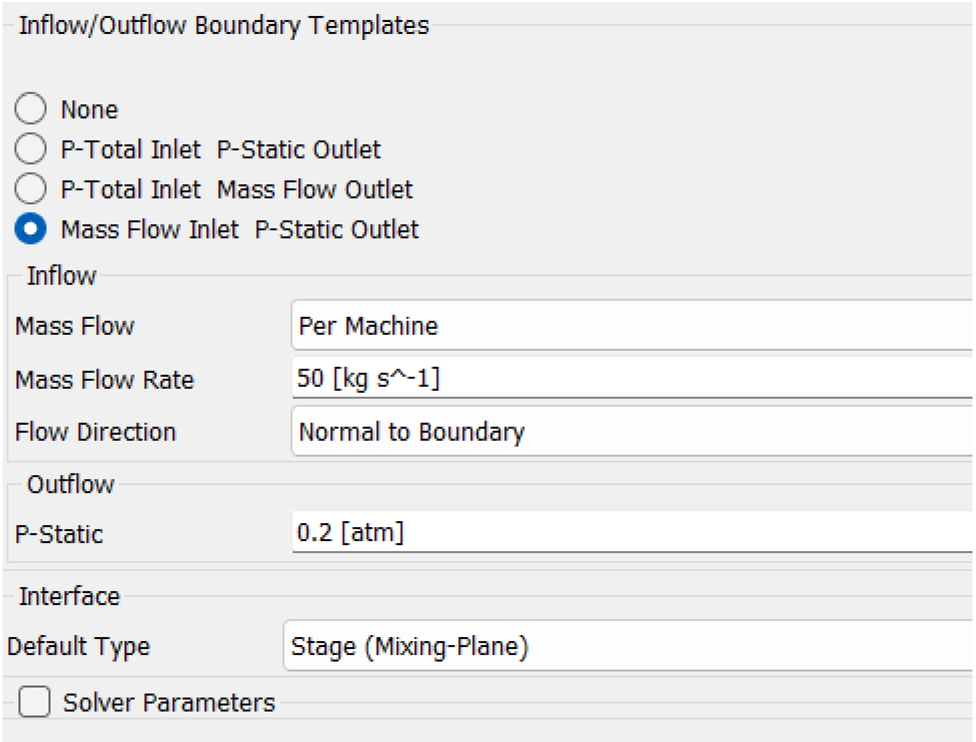


Figure 16: establishing boundary conditions.

- Evaluated compressor performance metrics such as pressure ratio, efficiency, and flow characteristics.
- Imported the geometric model from CF Turbo into Ansys.
- Meshed the geometry to ensure computational accuracy and efficiency.
- Applied boundary conditions including inlet pressure and temperature, rotational speed, and material properties.
- Conducted steady-state simulations to analyze fluid flow, pressure distribution, and temperature distribution within the compressor stage.
- Evaluated the performance of the compressor stage by comparing simulation results with CF Turbo predictions.

5.2 Ansys Simulation Run

5.2.1 Velocity calculations on ANSYS

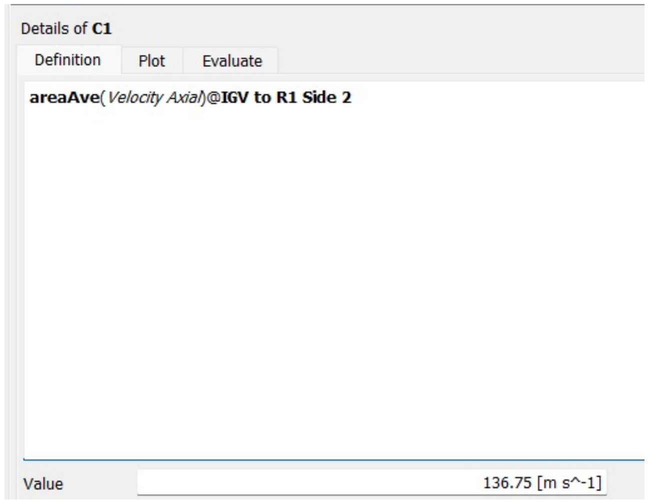


Figure 17: C1 on Ansys.

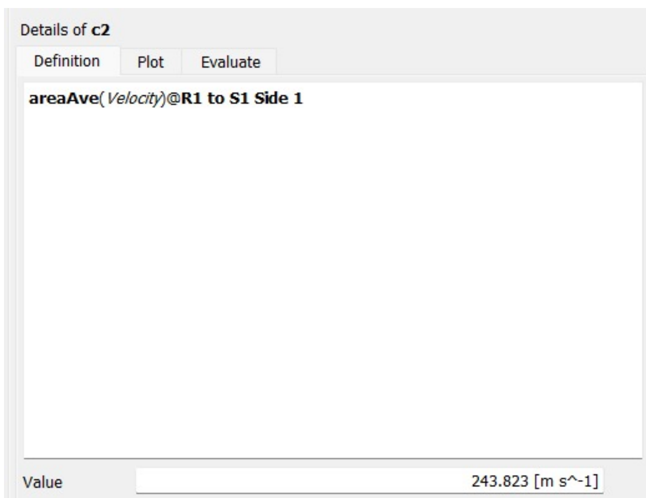


Figure 18: C2 on Ansys.

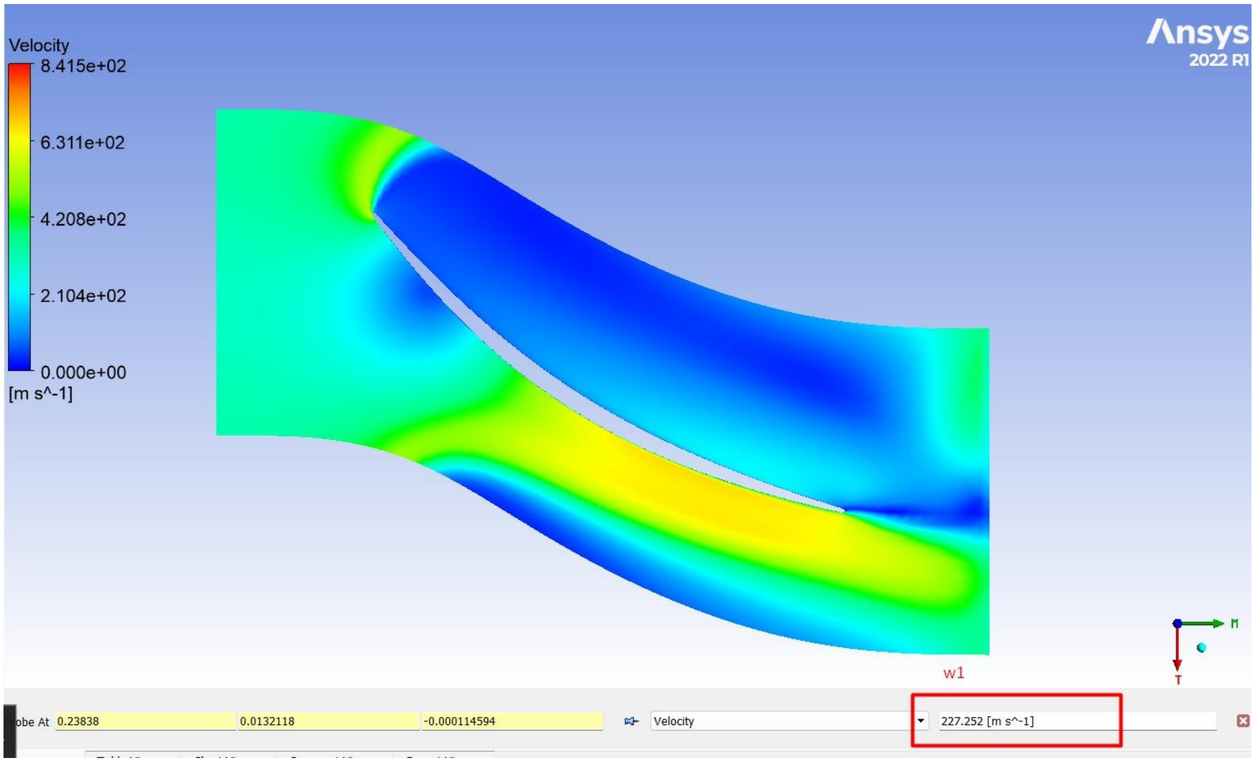


Figure 19: W1 on Ansys.

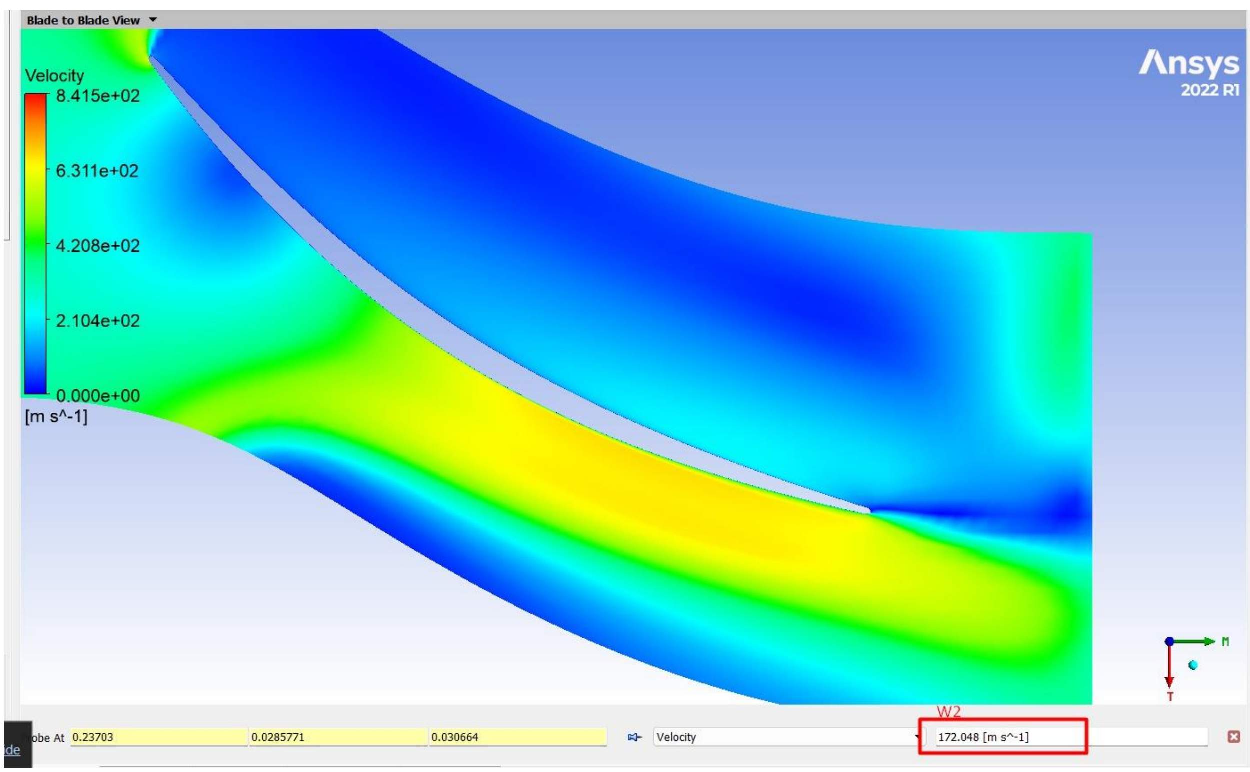


Figure 20: W2 on Ansys.

5.2.2 From MATLAB calculations

```
W1 = 228.5288
C1 = 168.8790
beta2 = 27.3508
alpha2 = 48.9762
Ctheta2 = 172.4106
Wtheta2 = 77.5894
W2 = 168.8790
C2 = 228.5288
```

Figure 21: velocities calculations from MATLAB.

5.2.3 For the pressure calculations

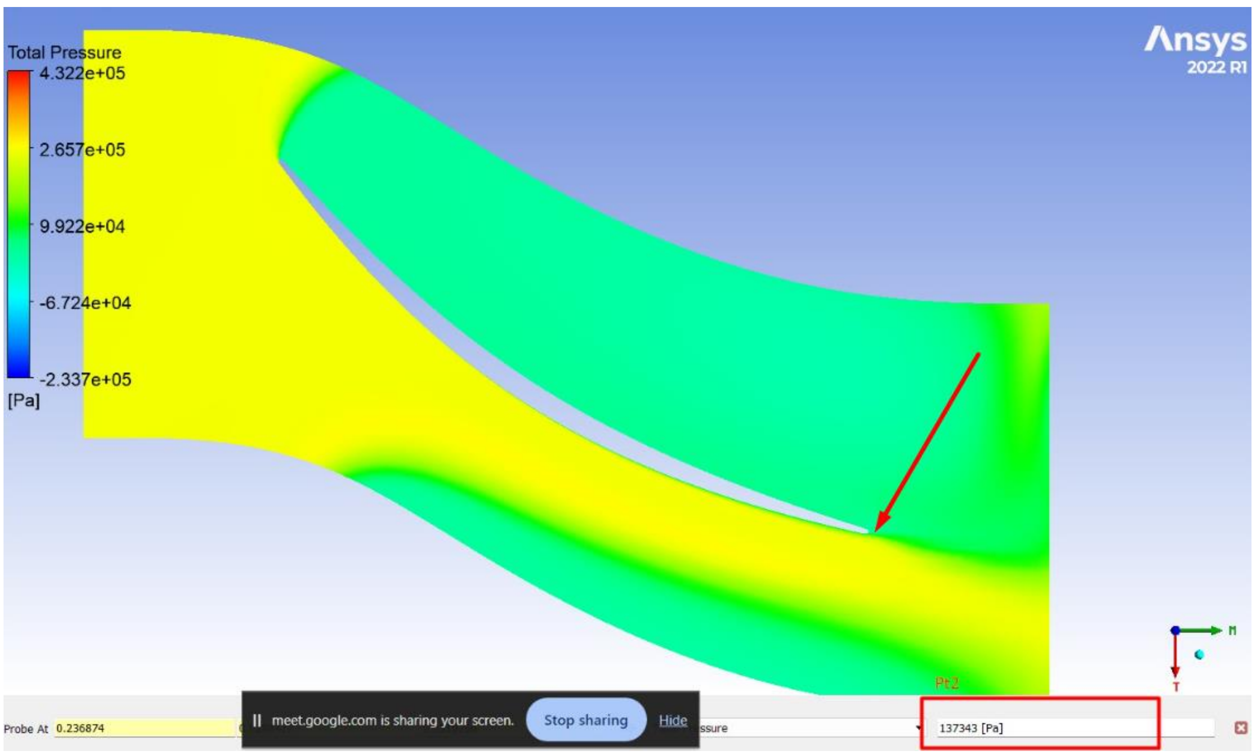


Figure 22: PT2 on Ansys.

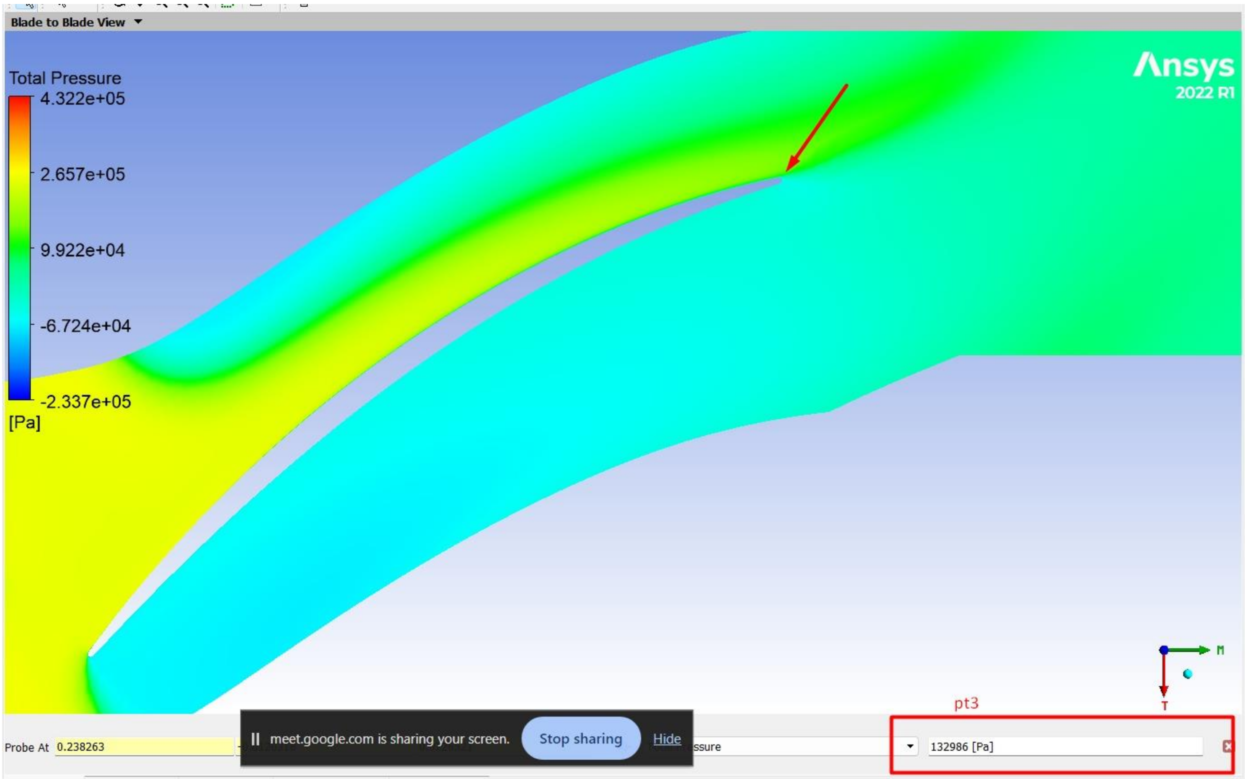


Figure 23: PT3 on Ansys.

5.2.4 From the MATLAB calculations

```
T1 = 273.8038
Ttrel1 = 299.7996
T2 = 285.6034
Ttrel2 = 299.7996
P1 = 8.4895e+04
Ptrel1 = 1.1661e+05
Ptrel2 = 1.1562e+05
P2 = 9.7561e+04
Pt2 = 1.3234e+05



Pt3 = 1.3142e+05
```

Figure 24: pressures from MATLAB.

5.2.5 Comparison Table

Parameter	MATLAB	CFD	Error
C1	168.87	136.75	19%
C2	228.5288	243.823	6.7%
W1	228.5288	227.252	0.55%
W2	168.878	172.048	1.877%
PT2	1.3234e+05	1.3734e+05	3.77%
PT3	1.3142e+05	1.32985ee+05	1.2%

Table 5: comparison between MATLAB and Ansys values for velocities and temperatures.

5.2.6 Conclusion

The analysis of an axial compressor stage with Inlet Guide Vanes using CF Turbo software and Ansys provided valuable insights into the performance characteristics and design optimization of such systems.

CF Turbo Analysis:

- Optimal geometric parameters were identified to maximize compressor efficiency while maintaining a desirable pressure ratio.

Ansys Simulation:

- Simulation results demonstrated good agreement with CF Turbo predictions, validating the accuracy of the preliminary analysis.
- Pressure and temperature distributions across the compressor stage were visualized, highlighting areas of high stress and potential flow separation.
- Efficiency and pressure ratio obtained from Ansys simulations corroborated the findings from CF Turbo analysis, providing further confidence in the design.

6 References

https://youtu.be/p5ky0cqn1TM?si=xbYNbLw5D5tk0oqm