Department of Aerospace Engineering

AE - 651: AERODYNAMICS OF COMPRESSORS AND TURBINES

Design of Low speed, Single-stage Axial Compressor

Instructor

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1 Introduction

This project involves designing of an axial, low speed, single-stage compressor according to certain given parameters and meeting constraint of a particular inlet and outlet mass flow rate.

1.1 Axial Compressors

Gas turbine engine compressors provide the compression part of the gas turbine engine thermodynamic cycle. There are three basic categories of gas turbine engine compressor: i)axial compressor, ii)centrifugal compressor and iii)mixed flow compressor. Of these axial compressors are commonly used in aircraft engines for its excellent reliability and high efficiency.
In an axial flow compressor the gas or working fluid principally flows parallel to the axis of rotation, or axially. This differs from other rotating compressors such as centrifugal compressor and mixed-flow compressors where the fluid flow will include a "radial component" through the compressor. A typical axial compressor has a series of rotating blades known as rotors followed by a stationary set of blades called stators. Each rotor stator pair forms a stage. Since the flow in axial compressor has to face an adverse pressure gradient, pressure rise per stage cannot be too high to prevent flow separation from the blades. Thus, in order to achieve higher pressure ratio, multiple stages are used in axial compressors which results in longer size of axial compressors than centrifugal compressors.

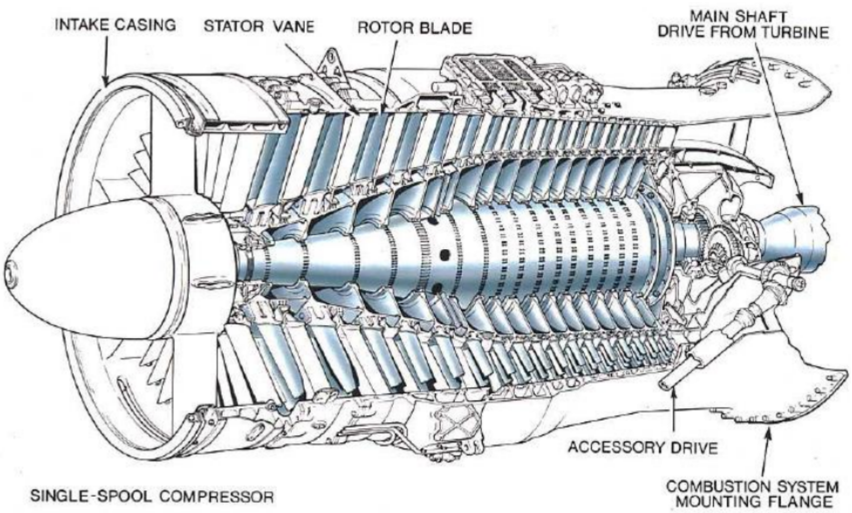

Figure 1: An Axial compressor

1.1.1 Rotor blade

Rotor blades are a set of rotating blades mounted on a shaft which imparts kinetic energy to the flow. The rotor reduces the relative kinetic head of the fluid and adds it to the absolute kinetic head of the fluid i.e., the impact of the rotor on the fluid particles increases its absolute velocity and thereby reduces the relative velocity between the fluid and the rotor. Total and static thermodynamic quantities increase across a rotor blade.

1.1.2 Stator blade

The stator blades are fixed and no energy addition occurs between the flow and the stator. The stationary airfoils, also known as vanes convert the increased kinetic energy into static pressure through diffusion and redirect the flow direction of the fluid to prepare it for the rotor blades of the next stage. The total temperature remains constant across a stator and the absolute velocity reduces. One common design condition is that the absolute velocity exiting the stator is same as the absolute velocity entering the rotor.

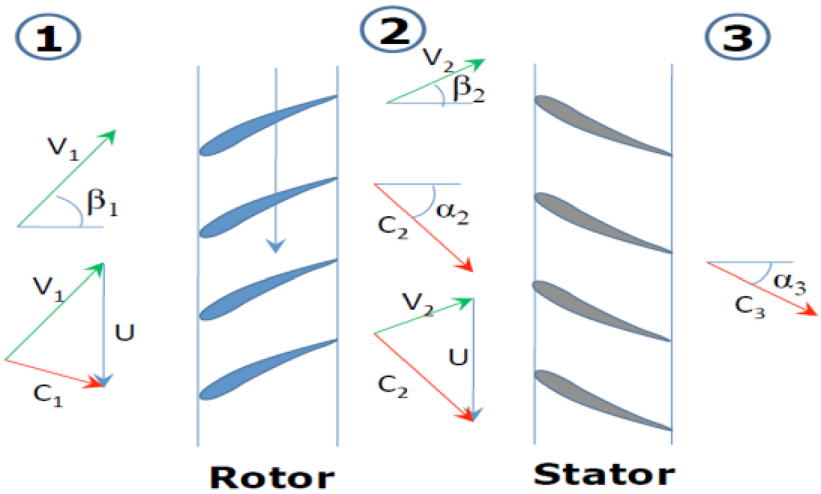

Figure 3: Velocity Triangles [1]

Figure 2: Velocity Triangles

2 Design of Single-stage Axial Compressor

2.1 Given data

Pressure ratio, π_c	1.2
Outer diameter (m)	0.56
Hub-tip ratio	0.56
Rotor speed, N (rpm)	3800
Mass flow rate (kg/s)	7.7

2.2 Design Methodology

2.2.1 Setting up design criteria for low speed stage

- $(\Delta T_o)_{subsonic}$ **= 10 -30 K**
  The total temperature rise of the subsonic stage is to be kept within this limit as per literature as mentioned in references. If the rise is being either far beyond this range or much lower than this range, then the stage is more likely to become a transonic or supersonic stage respectively.

- $(M_{1rel})_{tip} \leq 1.2$
  The tip relative Mach number is to be kept within the above range. An increase above the upper threshold leads to unnecessary shock effects, leading to drastic efficiency drop.

- $\epsilon \leq$ **0.4 - 0.6**
  Since fans tend to handle larger mass flowrates, the hub to tip ratio is usually kept low and it is seen that the given specification itself is meeting this criteria by default.

- **(β1 - β2)** is to be kept within reasonable limit (say $\leq 50^o$)
  Since compressors operate under adverse pressure gradient, it is more likely to be prone to flow separation if flow turning is too large.

- **M**$_a\leq$ **0.8**
  The axial Mach number is to be kept in the above range. Alleviation from this design limit might lead to incidence losses.

2.2.2 Design step by step procedure

- The design system is based on the three-dimensional (3D) computational fluid dynamics (CFD) solver Multall. It consists of 3 parts: a mean line solver or Meangen, stagen for generating the stage parameters, CFD solver Multall

- Meangen predicts the blading parameters on a mean stream surface and writes an input file for Stagen.

- For our design, initially, design parameters like flow coefficient, stage loading degree of reaction are calculated at mean radius.

- Stagen is a blade geometry generation and manipulation program which generates and stacks the blading, combines it into stages, and writes an input file for Multall.

- Multall is the 3D CFD Solver which solves for the Navier Stokes' Equation with appropriate turbulence models.

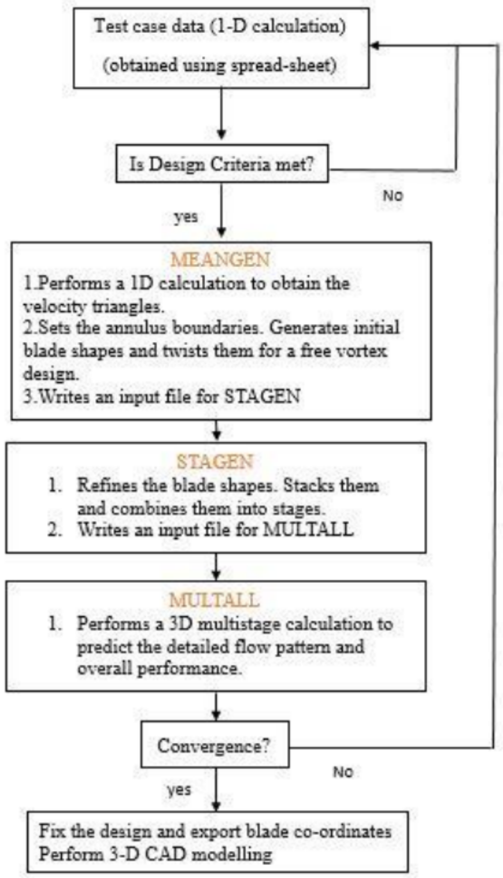

Figure 3: Design Process

- Once the Multall file is executed it is necessary to look for convergence to desired mass flow rate. If not, the assumed flow parameters need to be changed and then reiterate until the convergence is attained.
- If convergence is achieved at lower mass flow rate, reduction in number of blades can be useful as it reduces the blockage factor.

2.3 Mean Line design

Given Parametrs

Pressure ratio, π_c	1.2
Outer diameter (m)	0.56
Hub-tip ratio	0.56
Rotor speed, N (rpm)	3800
Mass flow rate (kg/s)	7.7

Assume parameters

Inlet static Temperature T_1 (K)	300
Inlet static Pressure P_1 (bar)	1
Inlet absolute Mach no.	0.11
Compressor efficiency	90%
α_1(No IGV)	0^o

Preliminary, the design process involved calculation of various flow and blade parameters at the mean-section using empirical relations as follows.

The design is initiated by calculating the parameters at mean section as follows:

1. Inlet absolute velocity (C1) is calculated from the assumed Mach number:

$$C_1 = M_1 * \sqrt{\gamma RT} \tag{1}$$

2. Axial absolute velocity is calculated from the inlet absolute velocity (C1):

$$C_a = C_1 * \cos\alpha_1 \tag{2}$$

3. Blade speed at mean diameter:

$$U_{mean} = \frac{\pi D_{mean} N}{60} \tag{3}$$

4. Work done on the fan/Compressor is calculated based on the pressure ratio assumed efficiency:

$$\triangle h_o = \frac{C_p T_{o1}}{\eta_f}(\pi_c^{\frac{\gamma-1}{\gamma}} - 1) \tag{4}$$

5. Now, by equating Thermodynamic work and Aerodynamic work:

$$\delta h_o = U_{mean} C_a * (tan(\alpha_2) - tan(\alpha_1)) \tag{5}$$

$$\alpha_2 = tan^{-1}\left(\frac{\triangle h_o}{U_{mean} C_a} + tan(\alpha_1)\right) \tag{6}$$

6. Using axial flow velocity and peripheral velocity, Flow coefficient is calculated

$$\phi = \frac{C_a}{U_{mean}} \tag{7}$$

7. Blade angles at inlet and exit can be calculated by:

$$\frac{1}{\phi} = \frac{U_{mean}}{C_a} = tan(\alpha_1) + tan(\beta_1) = tan(\alpha_2) + tan(\beta_2) \tag{8}$$

$$\beta_1 = tan^{-1}(\frac{1}{\phi} - tan(\alpha_1)\Big) \tag{9}$$

$$\beta_2 = tan^{-1}(\frac{1}{\phi} - tan(\alpha_2)\Big) \tag{10}$$

8. Stage loading is calculated from stage enthalpy change and peripheral velocity:

$$\psi = \frac{h_o}{U_{mean}} \tag{11}$$

9. Degree of Reaction is calculated from:

$$R_x = 1 - \frac{\phi}{2}(tan(\alpha_1) + tan(\alpha_2)) \tag{12}$$

10. Diffusion Factor for Rotor Stator:

$$D*_{rotor} = 1 - \frac{V_2}{V_1} + \frac{|V_{w1} - V_{w2}|}{2(\frac{C}{S}V_1)} \tag{13}$$

$$D*_{stator} = 1 - \frac{C_2}{C_1} + \frac{|C_{w1} - C_{w2}|}{2(\frac{C}{S}C_1)}$$

(14)

2.3.1 Calculations

2.3.2 Mean Line Design: Test cases (1-5)

The above preliminary design calculations were made with 5 test cases which are shown in appendix -A. It was found that cases (1,2,3,4) cannot be carried out for further 3-D design approach, because of them being either not satisfying the low speed stage design criteria as discussed earlier or convergence criterion was not more likely to be met. But test-case (5), proved to be satisfying both the criteria and their respective aerodynamic parameters at the mean are carried out to the 3-D design. The design calculations the finalized case (5), is shown below.

The given RPM (3800) had to be changed to 6000 because as seen in test case 6, the values for degree of reaction is coming out to be negative which implies that certain design conditions need to be changed.

Case No.	Inlet static Temperature T_1 (K)	Inlet static pressure P_1 (Pa)	Ca (from M)	Inlet staganatic pressure P_01 (Pa)	Inlet stagnatioi Temperature T_01 (K)	rho_01	Pressur Ratio Pr_ratio	Isentropic Efficiency Isen_Effi	RPM	Mach No. M	Mass flow rate M_dot (kg/s)
1	300	100000	104.184	106302.195	305.400	1.212	1.200	0.900	6000.00	0.300	7.700
2	300	100000	86.820	104376.524	303.750	1.197	1.200	0.900	6000.00	0.250	7.700
3	300	100000	69.456	102800.975	302.400	1.184	1.200	0.900	6000.00	0.200	7.700
4	300	100000	52.092	101575.549	301.350	1.174	1.200	0.900	6000.00	0.150	7.700
5	300	100000	38.201	100847.295	300.726	1.168	1.200	0.900	6000.00	0.110	7.700
6	300	100000	38.201	100847.295	300.726	1.168	1.200	0.900	3800.00	0.110	7.700

Figure 4: Test cases

Case No.	Tip_dia	Hub/tip ratio	Hub_dia	Area	Mean_dia	Mean_radiu	rho_1	U_mean	Flow_coeff	delta_h0	delta_T0	stage_loadi	alpha_2	Deg of reac Rx
1	0.560	0.560	0.314	0.169	0.437	0.218	1.161	137.225	0.759	18235.743	18.145	0.968	0.906	0.516
2	0.560	0.560	0.314	0.169	0.437	0.218	1.161	137.225	0.633	18137.217	18.047	0.963	0.990	0.518
3	0.560	0.560	0.314	0.169	0.437	0.218	1.161	137.225	0.506	18056.606	17.967	0.959	1.085	0.521
4	0.560	0.560	0.314	0.169	0.437	0.218	1.161	137.225	0.380	17993.907	17.904	0.956	1.193	0.522
5	0.560	0.560	0.314	0.169	0.437	0.218	1.161	137.225	0.278	17956.647	17.867	0.954	1.287	0.523
6	0.560	0.560	0.314	0.169	0.437	0.218	1.161	86.909	0.440	17956.647	17.867	2.377	1.388	-0.189

Figure 5: Test cases

Case No.	beta_1	beta_2	Vrel_1	Mrel_1	Vrel_2	Vtang_1	Vtang_2	Spacing (S)	Chord (C)	Diff_fact_ro	C_2	Ctang_2	Spacing (S)	Chord (m)	Diff_fact_stator
1	0.921	0.042	172.293	0.496	104.274	137.225	4.335	0.025	0.050	0.588	168.861	132.890	0.025	0.050	0.580
2	1.007	0.058	162.383	0.468	86.967	137.225	5.053	0.025	0.050	0.668	158.136	132.172	0.025	0.050	0.660
3	1.102	0.081	153.801	0.443	69.685	137.225	5.641	0.025	0.050	0.761	148.790	131.584	0.025	0.050	0.754
4	1.208	0.117	146.779	0.423	52.448	137.225	6.098	0.025	0.050	0.866	141.095	131.127	0.025	0.050	0.863
5	1.299	0.165	142.443	0.410	38.728	137.225	6.369	0.025	0.050	0.958	136.318	130.856	0.025	0.050	0.960
6	1.157	-1.262	94.934	0.273	125.653	86.909	-119.705	0.025	0.050	0.221	210.116	206.614	0.025	0.050	1.064

Figure 6: Test cases

2.4 Mean Line Design for test case 5

Parameters given:

- Single stage, low speed compressor
- Pressure Ration $\pi_f = 1.2$
- Outer Diameter $D_o = 0.56$ m
- Hub-tip ratio = 0.56 m
- Rotor speed = 6000 RPM
- Mass flow rate = 7.7 kg/s

Parameters assumed:

- Inlet Static Temperature $T_1 = 300$K
- Inlet Static Pressure $P_1 = 1$bar
- Inlet Absolute Mach Number M_1=0.11
- Isentropic Efficiency $\eta_{st} = 0.9$
- (No IGV) α_1=0^o

From inlet absolute Mach number, $M_1 = 0.11$

$$C_1 = M_1 * \sqrt{\gamma RT} \tag{15}$$

$$C_a = C_1 * \cos\alpha_1 = 38.2m/s \tag{16}$$

Mean peripheral velocity (U_{mean}):

$$U_{mean} = \frac{\pi D_{mean} N}{60} = 137.224 m/s \tag{17}$$

Work done /Stage Enthalphy change from pressure ratio :

$$\delta h_o = U_{mean} C_a * (tan(\alpha_2) - tan(\alpha_1)) = 17.956 kJ/kg \tag{18}$$

Flow angle from stage enthalpy change:

$$\alpha_2 = tan^{-1}\left(\frac{\triangle h_o}{U_{mean} C_a} + tan(\alpha_1)\right) = tan^{-1}\left(\frac{179656.64}{86.9 * 38.2} + tan(0)\right) = 73.72^o \tag{19}$$

Flow coefficient:

$$\phi = \frac{C_a}{U_{mean}} = 0.27838 \tag{20}$$

Blade angles at inlet and exit of rotor:

$$\beta_1 = tan^{-1}(\frac{1}{\phi} - tan(\alpha_1)\Big) = 74.44^o \tag{21}$$

$$\beta_2 = tan^{-1}(\frac{1}{\phi} - tan(\alpha_2)\Big) = 9.465^o \tag{22}$$

Stage loading:

$$\psi = \frac{h_o}{U_{mean}} = 0.9535 \tag{23}$$

Degree of Reaction:

$$R_x = 1 - \frac{\phi}{2}(tan(\alpha_1) + tan(\alpha_2)) = 0.5232 \tag{24}$$

Diffusion Factor for Rotor Stator:

$$D*_{rotor} = 1 - \frac{V_2}{V_1} + \frac{|V_{w1} - V_{w2}|}{2(\frac{C}{S} V_1)} = 0.9577 \tag{25}$$

$$D*_{stator} = 1 - \frac{C_2}{C_1} + \frac{|C_{w1} - C_{w2}|}{2(\frac{C}{S} C_1)} = 0.9597 \tag{26}$$

2.5 3D design of low-speed ,axial compressor

2.5.1 Free Vortex law assumptions

- Constant *specific work* at all radii,

$$\frac{dh_o}{dr} = 0 = \rightarrow C_a \frac{dC_a}{dr} + C_w \frac{dC_u}{dr} + \frac{C_u^2}{r} = 0 \tag{27}$$

- Constant axial velocity across the radii,

$$\frac{dC_a}{dr} = C_w \frac{dC_u}{dr} + \frac{C_u^2}{r} = 0 - \rightarrow \frac{dC_u}{C_u} + \frac{dr}{r} \tag{28}$$

```
STAGE No, ROW No, No. BLADES          1          1          26
STAGE No, ROW No, No. BLADES          1          2          36

******************************************************
 CONDITIONS FOR THE FIRST BLADE ROW OF THE STAGE.
 THIS IS A COMPRESSOR ROTOR
******************************************************
FIRST BLADE INLET AND EXIT ANGLES   -74.4444809      -9.43560410
 FIRST BLADE AXIAL VELOCITY         38.1196785
 FIRST BLADE INLET MACH NUMBER     0.409060597
 FIRST BLADE EXIT MACH NUMBER      0.109519340
 FIRST BLADE EXIT DENSITY           1.23862112
 FIRST BLADE EXIT PRESSURE          1.10143530
 FIRST BLADE INLET STAGN PRESS      1.00800025
 FIRST BLADE EXIT STAGN PRESS       1.22045767
 FIRST BLADE REL INLET STAG PRES    1.12161386
 FIRST BLADE EXIT TEMPERATURE       309.301941
 FIRST BLADE EXIT STAGN TEMP        318.504181
 FIRST BLADE TIP RADIUS =          0.281625867
 FIRST BLADE INLET SPAN =          0.123231091
 FIRST BLADE AXIAL CHORD=           5.00000007E-02
 FIRST BLADE ASPECT RATIO =         2.46462178

******************************************************
 CONDITIONS FOR THE SECOND BLADE ROW OF THE STAGE.
 THIS IS A COMPRESSOR STATOR
******************************************************
 SECOND BLADE INLET AND EXIT ANGLES    73.7330551       5.14313541E-02
 SECOND BLADE AXIAL VELOCITY        38.1196785
 SECOND BLADE INLET MACH NUMBER    0.385691613
 SECOND BLADE EXIT MACH NUMBER     0.106586382
 SECOND BLADE DENSITY               1.31239140
 SECOND BLADE EXIT PRESSURE         1.19903183
 SECOND BLADE INLET STAGN PRES.     1.22045767
 SECOND BLADE REL INLET STAG PR.    1.22045767
 SECOND BLADE ROW EXIT STAG PRES    1.20859420
 SECOND BLADE TEMPERATURE           317.782135
 SECOND BLADE STAGN EXIT TEMP.      318.504181
 SECOND BLADE TIP RADIUS =         0.274183929
 SECOND BLADE EXIT SPAN =          0.115784615
 SECOND BLADE AXIAL CHORD =         4.00000066E-02
 SECOND BLADE ASPECT RATIO=         2.89461493
******************************************************
******************************************************


 THE RADII THROUGH THE STAGE ARE -
ROW 1 HUB RADIUS   0.1544  ROW 1 TIP RADIUS   0.2816  ROW 2 HUB RADIUS   0.1585  ROW 2 TIP RADIUS   0.2775

******************************************************
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Figure 7: Final design results

- Constant α_1 across the blade length

Free vortex law : Intergration eq(28) we get,

$$r * C_w = constant \tag{29}$$

(*Here the subscript w denotes the tangential/whirl component.*)
Thus the whirl velocity component of the flow varies inversely with radius, which is known as free vortex. Keeping with these implications, free vortex law is used to calculate flow properties along the annulus at the rotor inlet and outlet as shown below.

- The tangential absolute velocity along the annulus is calculated using,

$$r.c_u = rm.c_m \tag{30}$$

- The absolute flow angles at rotor inlet and outlet along annulus are calculated by,

$$r.tan(_1)_r = r.tan(_1)_m \tag{31}$$

- The relative flow angles at rotor inlet and outlet along annulus are calculated from,

$$tan(\alpha_1)_r + tan(\beta_1)_r = \frac{U_r}{(C_a)_r}, whereU_r = U_m \frac{D}{D_m} \tag{32}$$

Radius	Ctang_1	Ctang_2	Radius ratio (	U	Deg of reactic Rx	Ca (from M)	delta_T0	U * delCw	alpha 1	alpha_2
0.1568	0	287.784	0.718	98.520	0.075	38.201	17.867	28352.400	0	78.163
0.17	0	265.438	0.778	106.814	0.213	38.201	17.867	28352.400	0	77.198
0.18	0	250.692	0.824	113.097	0.298	38.201	17.867	28352.400	0	76.472
0.19	0	237.497	0.870	119.380	0.370	38.201	17.867	28352.400	0	75.750
0.2	0	225.622	0.916	125.663	0.431	38.201	17.867	28352.400	0	75.033
0.21	0	214.879	0.962	131.946	0.484	38.201	17.867	28352.400	0	74.320
0.2184	0	206.614	1.000	137.224	0.523	38.201	17.867	28352.442	0	73.726
0.23	0	196.193	1.053	144.512	0.570	38.201	17.867	28352.400	0	72.911
0.24	0	188.019	1.099	150.796	0.605	38.201	17.867	28352.400	0	72.214
0.25	0	180.498	1.145	157.079	0.636	38.201	17.867	28352.400	0	71.522
0.26	0	173.556	1.190	163.362	0.664	38.201	17.867	28352.400	0	70.836
0.27	0	167.128	1.236	169.645	0.688	38.201	17.867	28352.400	0	70.155
0.28	0	161.159	1.282	175.928	0.710	38.201	17.867	28352.400	0	69.481

Figure 8: Applying free vortex law

Radius	beta_1	beta_2	beta_1 - be	Vrel_1	Vrel_2	Vtang_1	Vtang_2	C_2	Diff_fact_ro	stage_loadi	Flow_coeff	Mrel_1
0.1568	68.806	-65.479	3.327	105.667	92.045	98.520	-83.744	186.224	0.560	1.850	0.388	0.306
0.17	70.321	-58.069	12.252	113.439	72.227	106.814	-61.298	172.397	0.734	1.574	0.358	0.328
0.18	71.337	-50.092	21.244	119.374	59.544	113.097	-45.675	163.303	0.834	1.404	0.338	0.346
0.19	72.256	-39.091	33.164	125.343	49.219	119.380	-31.035	155.190	0.907	1.260	0.320	0.363
0.2	73.091	-24.279	48.812	131.341	41.907	125.663	-17.231	147.913	0.953	1.137	0.304	0.380
0.21	73.853	-6.191	67.662	137.365	38.425	131.946	-4.144	141.350	0.968	1.031	0.290	0.398
0.2184	74.444	9.464	64.979	142.442	38.728	137.224	6.368	136.318	0.958	0.954	0.278	0.412
0.23	75.193	27.935	47.258	149.476	43.239	144.512	20.256	129.996	0.919	0.860	0.264	0.433
0.24	75.784	39.702	36.083	155.559	49.651	150.796	31.717	125.056	0.872	0.790	0.253	0.450
0.25	76.331	48.225	28.106	161.657	57.341	157.079	42.763	120.529	0.822	0.728	0.243	0.468
0.26	76.838	54.443	22.395	167.769	65.692	163.362	53.443	116.368	0.772	0.673	0.234	0.486
0.27	77.310	59.088	18.222	173.893	74.360	169.645	63.797	112.530	0.725	0.624	0.225	0.503
0.28	77.749	62.652	15.097	180.028	83.155	175.928	73.861	108.982	0.680	0.580	0.217	0.521

Figure 9: Applying free vortex law

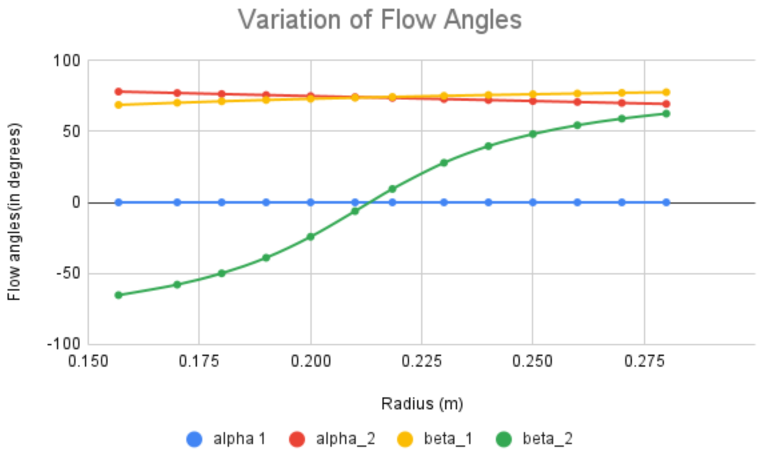

Figure 10: Variation of Flow angles

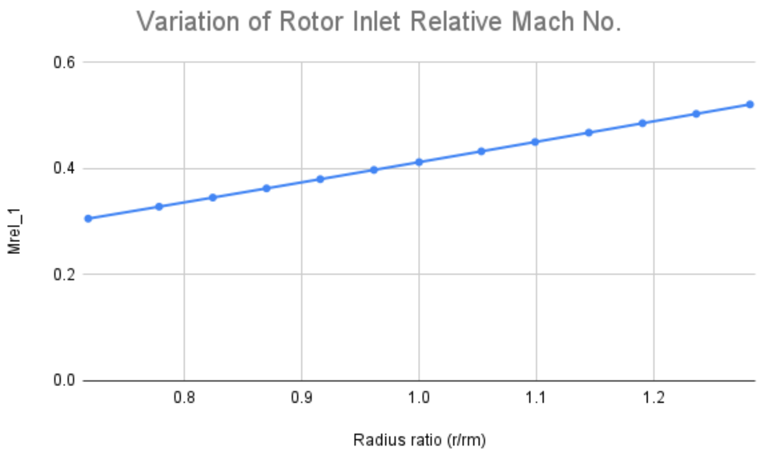

Figure 11: Mach no. relative increases from hub to tip

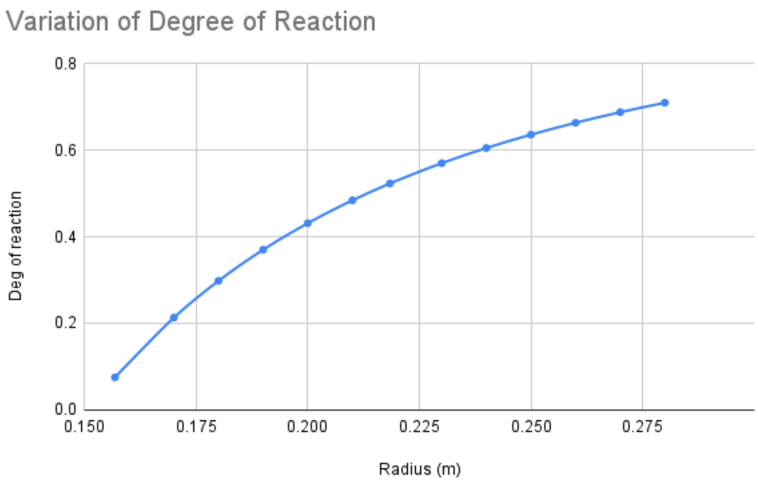

Figure 12: Degree of Reaction increase form hub to tip

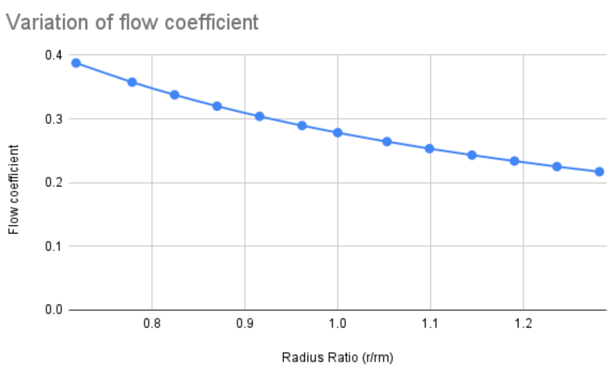

Figure 13: Flow coefficient decreases from hub to tip

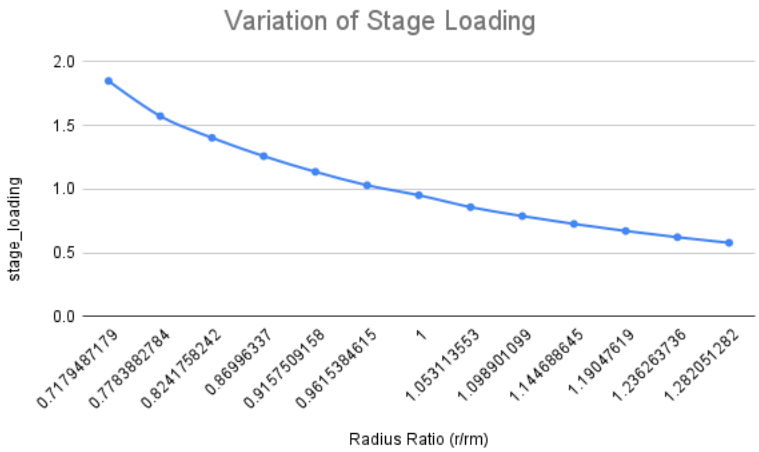

Figure 14: Stage loading decreases form hub to tip

2.6 Velocity triangles

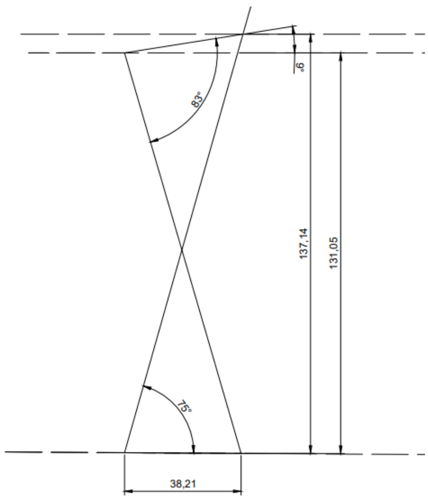

Figure 15: Velocity triangle for mean

3 Conclusion

- Manual and computational analysis have been carried out for design of low speed single stage axial compressor having hub-to-tip ratio (0.56) with a rotor rotational speed of 6000 rpm.
- Certain reasonable assumptions were made during the design process and a check has been made whether such assumptions could satisfy the preliminary design criteria. Parameters varied are inlet pressure, temperature and RPM and Mach no.
- Manually, calculated 1-D design values were compared with the obtained 'meangen' values for given specifications and the design process is carried further.
- 3-D design analysis were carried using 'Free Vortex Law' and similarity comparison has been made with the 'Stagen' values.
- The output from 'Stagen' file has been given as input to 'Multall' file to carry out the simulation and convergence is checked (Inflow=Outflow)
- Once the convergence is met, blade geometry is finalized and velocity triangles are generated.

4 References

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