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Aerodynamic Design of a single stage Axial Flow Compressor using CFD approach

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ABSTRACT

The performance of axial flow compressor has major impact on overall performance of gas turbine engine. The paper deals with numerical analysis of a single stage, subsonic axial flow compressor using commercial CFD code of AxSTREAM. The aerodynamic design and blade profiling has been carried out using CFD software. The research starts with design of the high pressure ratio compressor blade sections which yield a single stage pressure rise up to 1.21, the constant tip diameter of the compressor rotor blade for 15.5 kg/s, 14800 RPM, 276.5 KW power with a tip speed 167.7 m/s. Further the design is optimized for minimum total pressure loss. Analytical results compared with the numerical analysis.

Keywords: Axial flow compressor, aerodynamic design, CFD modeling, Blade design and profiling, optimized performance.

I. INTRODUCTION

In the recent decades, the gas turbines have dominated air transportation by virtue of their high efficiency and reliability. An axial compressor is an important part of any efficient gas turbine. Axial flow compressors are the fluid pumping machinery where the fluid enters and exits axially to the rotor axis. The unique features like high mass flow rate for a small frontal area and high efficiency ratio with higher mass flow rate makes an axial flow compressors a perfect choice for gas turbines used in jet engines. The overall gas turbine engine performance depends on the components' performance like compressor, compressor, combustor and turbine. Among these components the compressor plays a vital role. Hence it is required to know the performance and aerodynamic behaviour of the compressor before it is integrated into the engine. The prime requirement of Gas turbine engine manufacturers is efficiency and power to weight ratio. It is possible in two ways, increase the maximum combustion temperature and increase maximum pressure in compressor. The former is limited to the turbine inlet temperature and turbine blade material. The later can be achieved by running the compressor at higher speed. It results in the either high subsonic or transonic flow. But the sonic flow creates high losses in the cascade because of the formation of shock waves. Hence the other way to achieve an efficient compressor is by improving the compressor blade design. The current trend in compressors is to design an optimized blade with minimal pressure loss and higher pressure ratio. The present work carries out the optimization of the blade profile for the compressor cascade at high subsonic inlet flow conditions. An attempt has been made to design and configure a single stage axial flow compressor to a gas turbine engine producing 276.5 KW power output used for power generation.

II. DESIGN AND PERFORMANCE

In analytical method various relations are used for finding the flow parameters, diffusion factor and power required to run the compressor.

A. Design Specifications:

Overall adiabatic stage efficiency: 90
Number of stages: 1
Working fluid: Ideal gas (Air)
Mass flow rate: 15.5 Kg/s
rotor speed: 14500 RPM
Inlet Temperature: 303 K
Inlet Pressure : 101.325 kPa
Assumptions:

- Axial velocity is constant at mean section throughout the stage.
- Polytropic efficiency is assumed as 90 %.

B. Determination of Annulus dimensions:

For determining annulus dimension it is necessary to select axial velocity. As mass flow rate is known, we can easily find out static temperature, pressure and density at inlet. Using continuity equations for mass flow rate and hub-tip ratio to find out hub and tip diameter. Determine tip speed with help of tip diameter and rotational speed, detail formulas given as follows.

$$\dot{m} = \rho_1 A C_{a1} = \rho_1 \pi r_t^2 \left[1 - \left(\frac{r_h}{r_t}\right)^2\right] C_a \quad \text{………… (1)}$$

$$r_t^2 = \frac{\dot{m}}{\pi \rho_1 C_{a1}\left[1 - \left(\frac{r_h}{r_t}\right)^2\right]} \quad \text{… (2)}$$

$$T_1 = T_{01} - \frac{C_1{}^2}{2*C_P} \quad \text{…… (3)}$$

$$P_1 = P_{01}\left[\frac{T_1}{T_{01}}\right]^{\frac{\gamma}{(\gamma-1)}} \quad \text{……………………… (4)}$$

$$\rho_1 = \frac{P_1}{RT_1} \quad \text{………………… (5)}$$

Substitute ρ_1 in equation number 2. And evaluate tip radius.

$$U = 2\pi r_t N \quad \text{………………………….. (6)}$$

$$V_{1t}^2 = U_{1t}^2 + C_{a1}^2 \quad \text{……………………………. (7)}$$

$$a = \sqrt{(\gamma RT_1)} \quad \text{……………....... (8)}$$

At this stage it is advisable to check relative tip Mach number at inlet of compressor.

$$M_{1t} = \frac{V_{1t}}{a} \quad \text{……………….. (9)}$$

Stage outlet total temperature can be determined with the help of stage pressure ratio as follows.

$$T_{02} = T_{01}\left[\frac{P_{02}}{P_{01}}\right]^{\frac{(\gamma-1)}{n_p\gamma}} \quad \text{…………………………. (10)}$$

$$\eta_p = \frac{\ln\left(\frac{P_{02}}{P_{01}}\right)^{\frac{R}{C_{pc}}}}{\ln\left(\frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{R}{C_{pc}}} - 1}{\eta_{iso}} + 1\right)}$$

Where,

For outlet annuls diameter it is required to find out static temperature, pressure and density at outlet.

$$T_2 = T_{02} - \frac{C_1{}^2}{2*C_P} \quad \text{………. (11)}$$

$$M_2 = \frac{C_a}{\sqrt{(\gamma R T_2)}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (12)$$

$$P_2 = \frac{P_{02}}{\left(1 + 0.2 M_2^2\right)^{3.5}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (13)$$

$$\rho_2 = \frac{P_2}{R T_2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

$$A_2 = \frac{\dot{m}}{\rho_2 C_a} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (15)$$

Get all these parameters known at outlet it is required to assume rotor axial chord, stator axial chord and rotor stator spacing.

Velocity Distribution: In literature survey various method of velocity distribution has been explained and study was carried out for four different methods of distributions. Exponential method of vortex distribution is employed for present design.

For exponential method whirl distribution is given as

$$C_{w1} = a - \frac{b}{R}, \quad C_{w2} = a + \frac{b}{R}.$$

Where value of a, b can be determined by considering radius variation at inlet and outlet Value of a, b is given by following formula.

$$C_{w2m} = \frac{C_p \left(\Delta T_o\right)_{stage}}{\lambda U_m} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (16)$$

$$b = \frac{C_{w2m}}{\left(\dfrac{R_{2m} + R_{1m}}{R_{1m} R_{2m}}\right)},$$

$$a = C_{w2m} - \frac{b}{R_{2m}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (17)$$

$$C_{a1m}^2 = K1 - 2a^2\left(\frac{b}{a r_m} + \ln r_m\right) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

$$C_{a2m}^2 = K2 - 2a^2\left(\ln r_m - \frac{b}{a r_m}\right) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (19)$$

Substituting values of a, b in equation number 18, 19 evaluate value of K1 and K2. Using these values calculate velocity distribution at tip and hub.

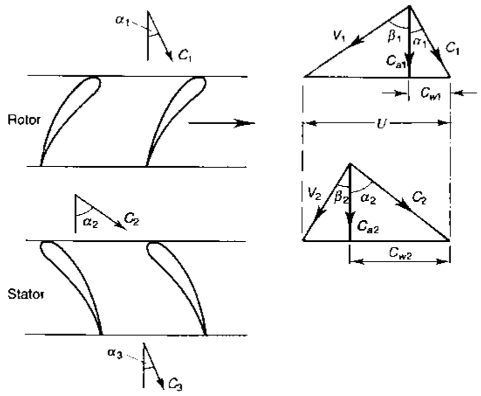


Figure 1. Velocity triangles for one stage

Detail step by step formulas for calculation of inlet and outlet angles given as follows.

$$\tan\beta_{1m} = \frac{U_m}{C_{a1}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (20)$$

$$\tan\beta_{2m} = \frac{U_m - C_{w2m}}{C_{a2m}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (21)$$

$$\tan\alpha_{2m} = \frac{C_{w2m}}{C_{a2m}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22)$$

It is advisable to check de Haller number.

De Haller number = $\dfrac{\cos\beta_1}{\cos\beta_2}$.

Degree of reaction at hub, mean and tip was calculated using following formula.

$$R_m = C_{am}\frac{(\tan\beta_{1m} + \tan\beta_{2m})}{2U_m} \qquad \ldots\ldots\ldots\ldots\ldots (23)$$

Cascade: In the development of the highly efficient modern axial flow compressor the study of the two-dimensional flow through cascades of aerofoil has played an important part. A cascade is a row of geometrically similar blades arranged at equal distances from each other and aligned to the flow direction.

A compressor blade deflects the incoming flow in such a manner that its static pressure increases across it. The flow therefore decelerates from a smaller to a larger cross-section area of a blade passage. Since the flow occurs against a pressure hill, it is more likely to separate, particularly at higher rate of pressure rise. Therefore the flow is handled more carefully in a compressor cascade. Using cascade co-relation find out for each section lift coefficient. Next step find out blade metal angles and this can be achieved by iterating blade deviation given as $\delta = m\theta\sqrt{\frac{s}{c}}$. Once the blade angle is known blade camber and blade stagger angle can be calculated.

Cascade losses and efficiency: The static pressure rise through a compressor cascade depends on the deflection of the fluid through it. Therefore, a maximum value of the fluid deflection is desirable, but on account of stalling and the associated cascade losses, this is carefully chosen.

However Design conditions for a compressor cascade from plots of experimentally obtained values of cascade losses and deflection against incidence. A large number of cascade tests have shown that deviation depends on the degree of guidance or pitch-chord ratio, blade camber, blade exit angle, etc.

first calculation are performed for aspect ratio AR=1, so that the chord length of rotor blade is equal to the rotor blades height, hence, the rotor solidity ratio is inversely proportional to radius. Furthermore, tip solidity is 1.08.

$$\sigma_R = \frac{1.08\, r_{1,t}}{r_1} \quad \ldots\ldots\ldots\ldots\ldots.. \quad (24)$$

With this assumed solidity variation, the rotor diffusion factor is obtained from

$$D_R = 1 - \frac{V_2}{V_1} + \frac{r_2\, C_{W,2} - r_1\, C_{W,1}}{(r_2 + r_1)\sigma_R V_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots.. \quad (25)$$

The rotor total-pressure-loss-parameter is calculated from the empirical curve fit of loss data

$$(\mathrm{TPLP})_R = 0.004 + 0.0639\,(D_R + 0.1)^{2.91} + 0.228 {D_R}^{2.02}\left[1 - \frac{(\mathrm{PBR})_R}{100}\right]^{3.77} \quad \ldots.. \quad (26)$$

Where the percent of blade height from the rotor tip is given by

$$(\mathrm{PBR})_R = \frac{r_{2,t} - r_2}{(r_{2,t} - r_{2,h})}\,(100) \quad \ldots\ldots\ldots\ldots\ldots.. \quad (27)$$

For a given value of rotor diffusion factor, pressure losses in the tip region are substantially higher than at the hub. Finally, the rotor profile loss coefficient is obtained from

$$\omega'_{Pr} = \frac{2\ \sigma_R(\mathrm{TPLP})_R}{\cos\beta_2} \quad \ldots\ldots\ldots.. \quad (28)$$

All the radial positions where the rotor inlet Mach number is less than one, so that the total pressure loss coefficient is equal to the profile loss coefficient

$$\omega' = \omega'_{Pr} \quad \ldots\ldots\ldots\ldots.. \quad (29)$$

Then, the relative total pressure loss parameter ratio becomes

$$\frac{P'_{02}}{P'_{01}} = \left(\frac{P'_{02}}{P'_{01}}\right)_{id} - \omega'\left[1 - \left(1 + \frac{\gamma - 1}{2}(M)^2\right)\right]^{\frac{-\gamma}{\gamma-1}} \quad \ldots\ldots\ldots\ldots.. \quad (30)$$

The aerodynamic design of axial flow compressor yields following results

Static pressure at outlet, kPa	113.4268
Total pressure at outlet, kPa	121.6005
Static temperature at outlet, °C	47.6447
Power, kW	278.25
Total static pressure ratio	1.1194
Diffusion factor (by NASA)	0.2882
Diffusion factor by de Haller (w2/w1)	0.8230
Average flow coefficient (C2s/U2)	0.3446
Total pressure rise, kPa	20.2755
Static pressure rise, kPa	12.1018
Total pressure loss factor	0.0213
Mach number	0.4180

III. NUMERICAL ANALYSIS

Numerical method is used to design and analysis of single stage axial flow compressor through CFD code AxStream. The analysis has been carried out for the constant tip diameter of the compressor blade having an aspect ratio 1 and the pressure loss and flow parameters of the compressor stage obtained. Optimisation of axial flow compressor design was a very tedious task as a small variation in one parameter will have a considerable change in compressor design, also to choose the optimum values is an art, so AXTREAM provides a good option to this problem and design can be optimised in no time by redesigning it as per requirement. The results obtained through it shows very good agreement for both design and off design conditions i.e. getting enough stall and choke margin with required pressure ratio at given mass flow and RPM. Scaled prototypes of blades are cascade tested in low speed wind tunnel. Pressure gradient and flow coefficient values from experiment are in agreement with software data thus validating the optimised design.

A. Design Specification and Assumptions for Stage Design

Based on the problem statement defined from cycle analysis, design specifications for single stage axial flow compressor are

Mass Flow rate at Atmospheric condition 15.5 kg/sec

Rotor speed	14800 RPM
Inlet total Temperature	298 K
Inlet total Pressure	101325 N/m2

B. Preliminary Design

Preliminary design solution generator helps to rapidly select optimal main flow path parameters, such as the number of stages, geometrical dimensions and angles, heat drop distributions etc. Preliminary design procedure performs inverse task calculation i.e. based on boundary conditions and calculates flow path geometry.

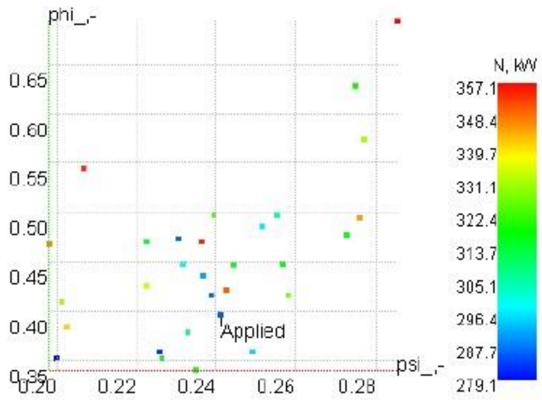


Figure 2. Module design Parameter and Design Space generator with filtered solutions for an axial flow compressor

Preliminary design starts from specification of technical requirement and setting up design task and compressor conceptual layout that includes: Inlet and outlet boundary conditions (inlet pressure, temperature, pressure raise ratio etc); Conceptual design and sizing layout i.e. quantity of modules (group of stages) inside compressor, number of stages in each group, meridional and axial sizes limitations, work coefficient; geometrical parameters should be used as design constraints, i.e. specific diameter and its ranges or exact value, and blade heights or angles based on requirements or assumptions. Next machine parameters selected are as Inlet and outlet condition type and values; Design criterion (power and choice of efficiencies) and Number of modules.

A solution generator generated the possible solution in the design space explorer which is later filtered based on design criterion of power and work efficiency as well as subsonic Mach number. The filtered solutions are validated to provide better coincidence of results to design conditions.

C. Performances Map Generation for Design Point in PD MAP

PDMAP helps PD MAP is the tool to evaluate the created design performances on speed line by calculating compressor curves that are necessary for initial compressor characteristics assessment. At this stage of design process, optimum design solution is tested for the coincidence of the speed line with design point and gas dynamic stability range is evaluated by redefining the pressure ratio.

D. Post-Design of Compressor.

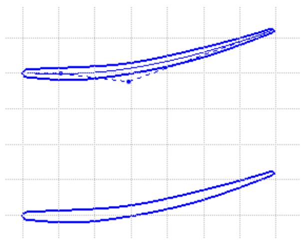
In the post design process, applied design is subjected to compressor flow path editing, adjustment of specific diameters of the whole flow path and adjustment of number of blades; chords and aspect ratio (blade height/chord). It tries to keep solidity (relative pitch) near the value, selected by preliminary design.

E. 1D/2D Streamline Calculation

Two-dimensional profile cascade losses arise primarily from the growth of the boundary layer on the suction and pressure side of the blade [35]. The 1D and 2D calculation will show that the various flow parameters i.e. static pressure distribution in compressor, total pressure and absolute pressure distribution, meridonal velocity and mach number.

F. 3D Blade Design and Profiling

The next step is to perform profiling on plane profiles section to obtained optimum flow characteristics and pressure distributions. On the next step 3D blade design, stacking and shaping are performed and complete geometry which is ready to export is obtained. In this section we adjust the curve of blade make the curvature of blade smooth to get the optimum result. This task is performed by blade parameter editing command i.e. Edit mode. The figure 3 will show the blade editing and figure 4 shows the smooth curve of blade.



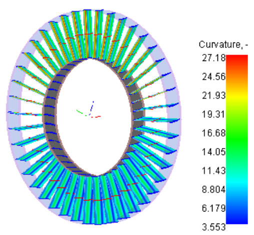


Figure 3. Blade Profiling **Figure 4.** Blade Curvature Smoothness

G. OFF-DESIGN PERFORMANCES ALCULATIONS WITH AxMAP

AxMAP is very effective tool to study the influence of operational parameters on compressor off-design performance. Also it is the ultimate tool to calculate compressor curves that are necessary for turbine-compressor matching. It is also used to predict, at which blade row the stall possibility is highest for current operating mode. AxMAP can be used for this kind of prediction, using indirect, but very accurate criterion of diffusion factor. The applied design is tested for stall formation at two selected speed lines and is checked for critical diffusion factor.

IV. RESULTS AND DISCUSSION

The axial flow compressor redesign and optimization is carried out using the AXSTREAM according to the procedure discussed before and the results are as under. The Finalized Data after redesign in preliminary design and space explorer is as under

Aerodynamic design of an axial flow compressor with

Mass flow 15.5 Kg/s.

Stage pressure ratio = 1.21

Rotational speed = 14800 rpm

The use of inlet guide vane improves the efficiency by almost 10%. The average peak Mach No. Is 0.7690. This indicates it is operating in subsonic range. The performance characteristic curve obtained in AXMAP is shown in Fig.5 indicates the compressor is matching the performance i. e. is delivering the given mass of air at designed outlet pressure; also the compressor has sufficiently wide range for stall and choke. The choking range at designed RPM is 1.2 bar and stalling range is 1.7 bar.

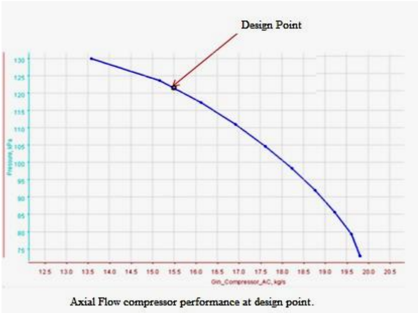


Figure 5. Performance curve of Compressor

Post design shows velocity triangle and HS Diagram for respective section as shown in figure 6. Also it reveals the same for sections of machine which indicates proper flow angles i.e. no excessive turning of blades.

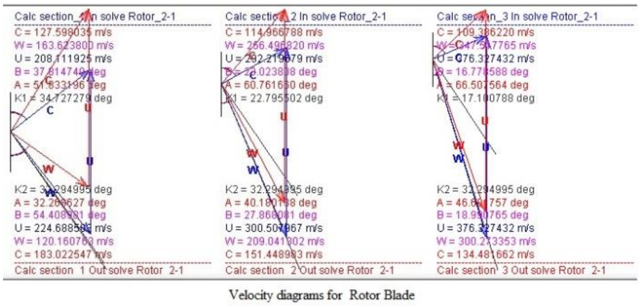


Figure 6. Velocity diagram for Compressor blades

After running the calculations for the post design the machine dimensions are obtained like the radius at each section from hub to tip, the dimensions for machine including shaft.

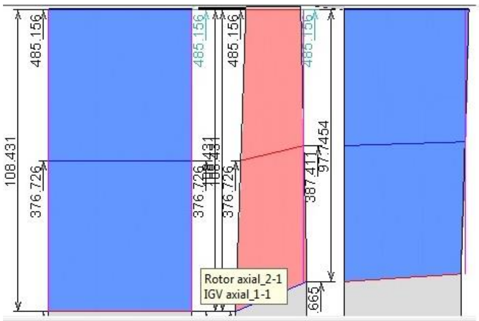


Figure 7. machine dimensions of axial flow Compressor

After running 1D and 2D calculations the parameters obtained are shown in Figure 8. The variation of Parameters such as absolute Mach number, relative Mach Number and Total Pressure are given in a very good agreement between design point and computation results. The results indicate that maximum pressure, absolute maximum pressure, maximum maridonal velocity, absolute Mach number and relative Mach number are at 158.6 kPa, 195.1 kPa 162.7 m/s, 0.41 and 1.081.

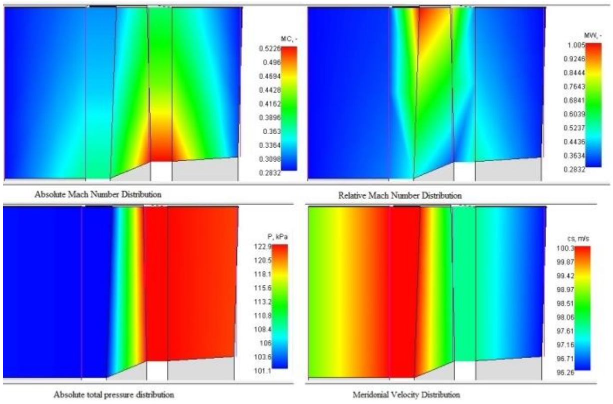


Figure 8. Results of flow analysis of axial flow Compressor

3D structural and modal analysis is carried out using AXSTRESS at different frequency level which shows stress level within range but Campbell diagram indicates one of the natural frequency crosses the mode of operation indicates the compressor needs rotor dynamics to have dynamic balancing and safe operation. Modal analysis will give the frequency at various modes shown in figure 9. The stress is maximum at leading edge and tailing edge.

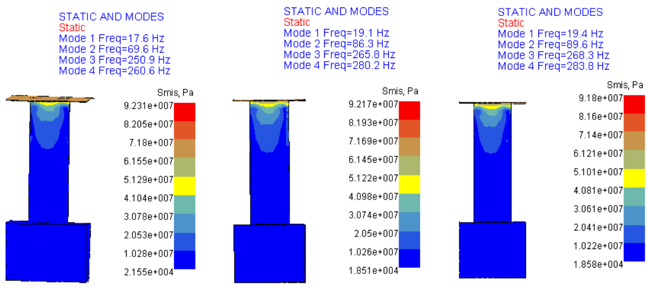


Figure 9. Results of 3D Structural and modal analysis

Using analytical and numerical analysis the results for various properties are compared as follow

Table 1. Analytical and numerical analysis data

Property	**Analytical Results**	**Numerical Results**
Static pressure at outlet, kPa	113.4268	114.4
total pressure at outlet, kPa	121.6005	121.77
static temperature at outlet, °C	47.6447	48.05
Power, kW	278.25	278.91
Total static pressure ratio	1.1194	1.11
diffusion factor (by NASA)	0.2882	0.267
diffusion factor by de Haller (w2/w1)	0.8230	0.8405
Average flow coefficient (C2s/U2)	0.3446	0.33
total pressure rise, kPa	20.2755	20.45
static pressure rise, kPa	12.1018	12.07
total pressure loss factor	0.0213	0.0216
Mach number	0.4180	0.4151

The above comparison made between the numerical and analytical study shows that the pressure rise are nearly closed to each other, it also indicate that the slightly change in diffusion factor by NASA and diffusion factor by De Haller number.

Further wind tunnel cascade experiment results are tabulated along with analytical results and numerical results obtained from AxSTREAM software.

V. CONCLUSION

The single stage axial flow compressor is designed for the constant tip diameter of the compressor rotor blade for mass flow rate 14.4 kg/s, RPM 14800, a single stage pressure ratio 1.21 with a tip speed 167.7 m/s and 276.5 KW power. The aerodynamic design is optimized using blade profiling in CFD software for minimum total pressure loss. Cascade test is carried out in wind tunnel. Results from experiment match with the numerical results, thus confirm the validity the optimized design.

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