Analysis of Flow through Vaneless Contra-Rotating Turbine of Jet Propulsion Engine

Harshil Patel,
Parul Institute of Engineering &Technology
Email: harshilpatel2109@gmail.com

Dr. A S Mohite.
Assistant Professor in the Mechanical Engineering Department.
Faculty of Technology & Engineering.
The M.S University of Baroda.

Abstract: As per the increasing demand of Jet propulsion Engine, it is required to improve the efficiency, reduction of weight and consideration of fuel consumption, which lead the researchers to arrive at an unconventional turbine known as vaneless contra rotating turbine (VCRT). The major advantage of this is in its compactness or in its ability to give high power/total weight. Research in this area is being pursued since many years. Scientists are putting their best to get the aim to its zenith. Such placement of flow path elements gives benefits, but needs special approach to organize flow inside the turbine. Modern aerodynamic designs, computational and optimization methodologies allow us to fulfill this task in the shortest period of time with the highest gain in turbine performance. The Aim of this topic is to understand the significance of blade design, geometries & domains and its effect in turbine efficiency & performance at various operating conditions. In this paper, three-dimensional multiblade row Navier-Stokes (3D RANS) simulations have been performed to investigate the flow characteristics of a VCRT. Bladegen modular are used to generate the Blades. TurboGrid modular is used for meshing. The turbine components are modeled for all the three spacing. In present work, 3D viscous flow simulation with SST k-Ԑ turbulence model is carried out in ANSYS CFX14.5. The flow in VCRT is very complex including several flow phenomena, such as turbulence, separation, swirling flow and unsteadiness flow. The variation of flow parameters from hub to tip of blades are presented in graphical form and average circumferential area (ACA) value of cascade parameters from inlet to outlet of the blades are computed at different operating regimes. The results of this analysis shows a good prediction of the flow behavior inside the blades and this lead to acceptable blade design, which can be used in VCRT.

Keywords: *VCRT, 3D Rans, ACA, Bladegen*

*****

1.0 Introduction

A gas turbine plant is a compact power-producing unit comprising of compressor, combustor, turbine and power or thrust generator. It is having wide range of applications like, a power production plant, in aircrafts as a propulsive plant, etc. Currently research is going on to its micro size; the major advantage of this is in its compactness or in its ability to give high power/total weight ratio. Higher the power/weight ratio it can achieve higher is its work delivering capacity and its value. Research in this area is being pursued since many years. Scientists are putting their best to get the aim to its zenith. There are two ways to achieve the aim, keep the size of the plant same and increase its power production or lessen weight as much as you can from the plant to get the same power production. The major components like compressor, combustor and turbine, are undergoing continuous research to reduce its weight and increase its effectiveness.

In compressor, research is the aspirated compressor as well as on casing treatment. In combustor currently research is going on the orbiting combustor. In an orbiting combustor the swirl coming out of the compressor is not diffused and the combustion is done with the swirl and then directly hot gas is utilized in the turbine. One more emerging area of combustion is to burn the fuel in the turbine stator blade passage so that whole combustion chamber is eliminated. Apart from the core parts of the plant, the research in petroleum engineering to get more refined aviation fuel with the help of chemical research is also considerable. Another major element to achieve this target was with the metallurgy department. In turbine part of the gas turbine power plant there are many developments have been done so far. There are three main types of turbine design currently being utilize in civil and military engines, i.e.,

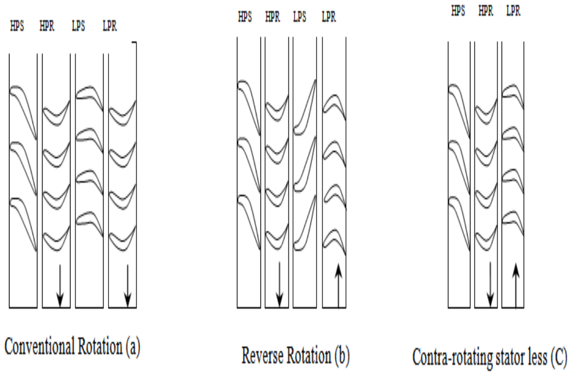

Figure:1 Types of turbine design

Conventional rotation – HP and LP turbines rotate in same direction as shown in Fig: 1 (a). Reverse rotation –

HP and LP turbines rotate in opposite directions resulting in reduced turning in LP vane in Fig: 1 (b) and Contra-rotating stator less- HP and LP turbines rotate in opposite direction with the deletion of the LP vane Fig: 1 (c) Turbine with Conventional Rotation. Industrial standard design, Fig: 1 (a). Companies: RR, P&W, GE etc. Within Rolls-Royce used in the civil RB211 and Trent series of engines and a range of military designs, EJ200, Adour, etc. In conventional rotation designs the nozzle guide vanes turn and accelerate the flow into the next blade row.Turbine with Reverse RotationInnovative technique to reduce the turning in the second vane (90 – 40 degrees). Resulting in a significant reduction in the secondary loss, and hence improvement in stage efficiency. Fig: 1 (b)Rolls Royce demonstrated technique on high-speed cold flow rig, and has now designed a reverse-rotation IP turbine for the T900 (Airbus A380 – first engine run March 2003). Although reverse-rotation means that the exit flow from the first rotor is in the correct sense for the second rotor, the second vane is still required to achieve the required whirl velocity for the second rotor, i.e., comparable work on the 1st and 2nd stages.

Removal of 2nd vane, resulting in a potential improvement in efficiency(aerodynamic & reduced cooling), and reduced cost & weight. Fig: 1 (c)Technique used in military turbines, Rolls-Royce XJ99 lift fan, 120/136 JSF engines (GE & PW). Possible future Rolls-Royce turbine design for civil aircraft.Limitation of this type of turbine is that the work produced by first rotor is around 2 to 3 times greater than that of second (at conventional shaft speeds).First rotor exit whirl velocity is limited by exit Mach number supersonic design and high reaction. It is clear from the above discussion that reverse-rotation and contra-rotating vaneless design has the potential to improve the overall turbine efficiency when compared with a conventional turbine design. However, these designs appear to offer aerodynamic benefits, the designer must make careful consideration on its applicability.

Reverse rotation may improve the turbine efficiency but can compression system be designed to operate with one of the shafts reversed? Does it make sense from a weight and cost point of view to have a second stage turbine in a contra-rotation statorless concept, which only provides approximately one-third of the overall turbine power? [16].The answer to all these questions is experimentation. A full proof design can come through experimentation only. Once we have the experimental database of VCRT, we can utilize it for many other applications too. In order to target the experimentation, we need to have a good foundation of theoretical and analytical database of the system. Therefore, objective during this project is to get a preliminary design methodology of VCRT with rich literature backup.

Another research area in turbine part is statorless or vaneless turbine stage. This stage consists of only rotor after combustor and the stator is eliminated. Literature indicates, the stator contributes to typically 30-40% of total energy losses through the turbine and thus by eliminating it the turbine efficiency will increase[14]. Pratt & Whitney’s F119 turbofan engine is the world’s first fifth-generation fighter engine having contra rotating turbine. The F119 combines stealth technologies and vectored thrust performance to provide unprecedented maneuverability and survivability with a high thrust-to-weight ratio. The ability to operate supersonically without afterburner supercruise gives the F-22 exceptional combat performance without compromising mission range. It is twinspool augmented turbofan engine having annular combustor, Three-stage fan, Six-stage high-pressure compressor, one stage LP turbine and one stage HP Turbine Rotating in Opposite Direction.

1.1 Vaneless Contra-Rotating Turbine Stage (VCRT):

Vaneless Contra-Rotating Turbine (VCRT) comprise of first rotor followed by a stator and then second rotor with vaneless space. First rotor and second rotor rotate in opposite directions. The main apparent attributes of the contra-rotating turbine are said to be [2]:

1. The increase of work per unit mass flow resulting from the large change in angular momentum made possible by the large relative rotational velocity.
2. The elimination of stator vanes between wheels reduces the weight of the stage, and increases the stage efficiency since it eliminates the pressure drop, leakage and the cooling losses normally associated with vanes.
3. The stage can be torque less and eliminate gyroscopic effects.
4. Negligible erosion rate through practical low speed operation.
5. Reduction of particulates deposition through control of surface shear flow.

Design parameters like coefficients of loading and flow, total-to-total efficiency, off design performance of VCRT, are evaluated and compared with conventional two-stage turbine.

1.2 Loading coefficient and flow coefficient of VCRT stage:

VCRT stage consists of two rotors that may or may not rotate at equal speed. Therefore, careful consideration must be taken while defining its loading coefficient. Louis [3] considered the VCRT stage with both the rotors rotating at equal speeds. Based on that the loading coefficient defined as the total stage enthalpy drop (specific work output of a stage) divided by the square of the peripheral velocity: $\psi = \Delta h0T/U^2$. In case of unequal peripheral velocities of the rotors, Cai [7] used the higher peripheral speed of both the rotors. That is loading coefficient $\psi = \Delta h0T/(Umax)^2$. Where Umax is the maximum of Ua and Ub. In order to fairly compare load capacity of different configurations, he defines an additional criterion, the average load factor of unit blade row given by $\psi^* = \psi/m$, where m is the blade row number of the stage. For example, for an ordinary stage m = 2, for a curtis stage m = 4 and for a common counter-rotating stage m = 2 or 3. This factor can be used to evaluate the work output per unit engine length or per unit engine weight more reasonably.

The most important part of this theory is to compare the performance of it with the conventional turbine stage. It is necessary to make certain assumption of the design parameter for comparison.

1.3 Off design performance evaluation of VCRT:

The evaluation of the off design performance was carried out by Louis [2]. Assumptions made during the evaluations were, relative leaving angles, α and β remains constant. The enthalpy extraction by

All parameters other than U and α are fixed in above equation. Therefore, in order to get equal work output from both the rotors, the guide vanes would have to be variable if the wheel speeds were not controlled. Which has hub loading coefficient ψh = 4, and hub flow coefficient φh = Cx/U = 0.73.The other most general parameter selected are, axial flow velocity, Cx = 100 m/s and the exit blade angle of rotor cascade β = 70 [4, page 6]. Cx / U = 0.73, and U = 136.98 m/s. Considering the variation of vane exit angle required to maintain same work output with the change in blade speed is shown in table:1. With the change in blade speed. Therefore, in order to get same work output from both the rotors with change in blade speed, the vane angle should be changed with respective blade velocity as mentioned in the table: 1.Similarly, if we want constant work output from the stage with different speed ratios with constant rotor blade angle, there is a need of variable stator angle arrangement.If we want the required work output at all speed of any rotor than there is a need of a speed control mechanism that controls the speed of the other rotor in order to get the designed speed ratio.Loading coefficient and flow coefficient of conventional two-stage turbine. This consists of two conventional stages having two stators and two rotors. In order to study the effect of removal of a stator, 100% reaction turbine stage is selected in which the whole expansion is taken place in rotor only and the stator is used just to guide the flow in proper direction. In order to compare the design parameters of conventional turbine stage with VCRT, it is necessary to take parameters similar to that of VCRT. They are,As the second stage is also 100% Reaction and second rotor rotates in same direction as of the first one, the stator of it just turns the flow to 2α degree to maintain continuity. Speed of both the rotor and the inlet absolute flow angle of them are same therefore relative flow angle at the inlet of rotors will be same, i.e. β2 =β4.This shows the similarity of design parameter assumed for VCRT and conventional two-stage unit. Therefore, if the work output of conventional two-stage unit is equal to that of VCRT, the two-stage conventional turbine unit is ready for one-to-one comparison with that of VCRT.Figure: 2 shows the comparison of temperature (T) – entropy (s) diagram of VCRT with conventional two-stage unit.

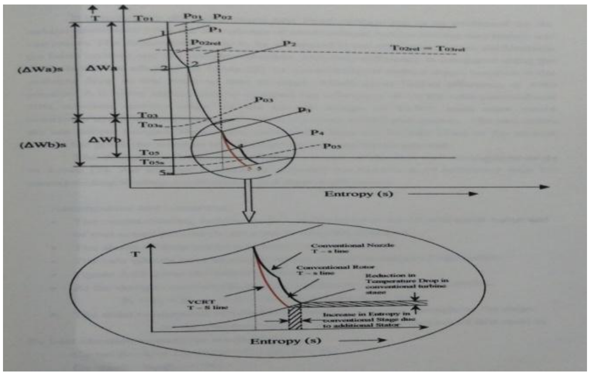

Figure 2

U	Cx / U	α
136.98	0.73	54.03
130	0.77	55.36
110	0.91	58.74
100	1.00	60.22
90	1.11	61.57
80	1.25	62.82
70	1.43	63.97
66.7	1.50	64.33

Table:1

From the analysis it is observed here that as the absolute flow angles of both VCRT and conventional two-stage unit are same, the intermediate stator used in conventional two-stage unit just turns the flow to 2α degrees in order to provide proper inlet to the second stage rotor (b) for the same work output. It is well known that if there is a turning of flow, there will be significant pressure loss associated with it. Therefore, in conventional unit the intermediate stator in turn adds loss within the system and decreases the overall efficiency of the unit. If the intermediate stator is eliminated and the second rotor is made to rotate in the opposite direction to that of the first one, same work output can be achieved with higher turbine efficiency. Hence, the significant pressure loss associated with large flow deflection of ε = 2α in intermediate stator and the mandatory exit swirl make single rotation of 100% reaction stage turbines unattractive.

Stage-loading coefficient (ψ) as a function of flow coefficient (φ) for VCRT having rotors rotating at same speed. For the loading coefficient ψ = 4 and flow coefficient φ = 0.6 as evaluated by Louis [2], the efficiency falls close to 94%. Louis also made a general comparison and indicated that the efficiency advantage of the contra-rotating turbine increases with stage loading coefficient and to lesser extent with flowcoefficient. For a flow coefficient φ = 0.6 and a one wheel loading coefficient ψ = 1.5 i.e. 3 for a contra-rotating stage made of two wheels, the stage efficiency is η = 93.0 % for the 50 % reaction stage and η = 95.3 % for the contra-rotating stage. He concluded a 1.5% or better efficiency advantage of the contra-rotating stage over the other two single rotation designs. For the same flow coefficient φ = 0.6 and ψ = 2 i.e. 4 for the complete contra-rotating stage, the stage efficiency is, η = 91.3 % for the 50 % reaction stage and 94.2 % for the contra-rotating stage, an efficiency advantage of at least 1.9 % for the contra-rotating stage over the other. This efficiency advantage increases with both ψ and φ.

In the present design of VCRT having different rotational speeds of both the rotors with loading coefficient, ψ = 1.6714 and flow coefficient, φ = 0.3896, the efficiency obtained is 94.7%. It is observedthat for the ψ = 1.6714 and φ = 0.3896, the efficiency is above the range of 95%. The difference between these efficiencies could be due to difference in rotational speed of both the rotors of VCRT at design conditions. In order to get more insight of the VCRT performance, specific speed (Ns) and specific diameter (Ds) of VCRT are calculated based on maximum rotational speed and total isentropic enthalpy drop.[23} The values of Ns and Ds of present VCRT design condition are 0.525 and 2.867.

2 3D Detailed design

Modified details of Geometric configuration obtained from Dring et al. [5][28] are shown in Table:2.BladeGen Modular of Ansys software is used to generate the blade profile of stator and rotors Blade.

In initial Meridional configuration, Hub -Tip Radius and axial chord has to be insert to generate Meridional plane .Which are shown in Figure:3& Figure: 4. Various parameter of Blade profile are inserted in pressure/suction Dialog to generate Blade profile.

Table :2 Blade profile Parameters

Parameters	Nozzle	Rotor 1	Rotor 2
Hub diameter (mm)	610	610	610
Tip diameter (mm)	762-776	776-790	790-805
Number of vanes	22	28	28
Stagger angle(Deg)	51	33	33
Leading edge diameter(mm)	11.3	8.8574	8.8574
Trailing edge diameter(mm)	2.79	4.826	4.826
Inlet blade angle(Deg)	0	-42.1864	47.81
Inlet wedge angle (Deg)	31.79	25.97	25.97
Exit blade angle(Deg)	67.55	68.76	31.24
Exit wedge angle(Deg)	6.85	5.31	5.31

Table 2

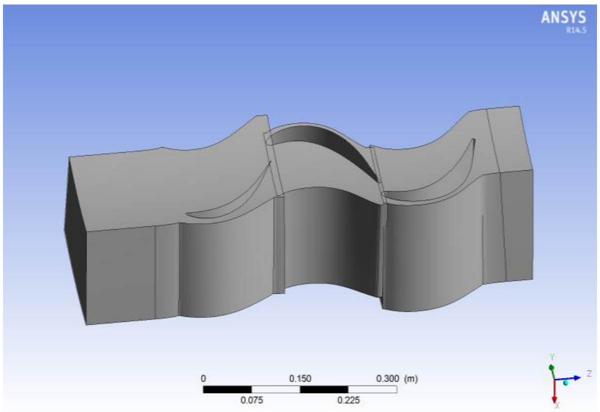

Figure 3Fluid Control Volume

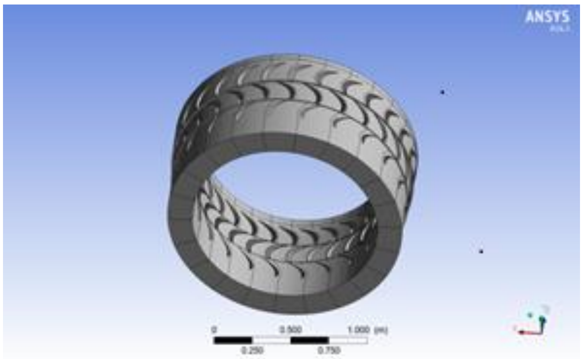

Figure 4 3D View of VCRT

3 CFD Simulation

Mean-line analysis is a preliminary design method that assumes one-dimensional flow and depends on empirical correlations to estimate many aspects of the flow. In order to examine the flow in more detail, a full viscous three-dimensional Computational Fluid Dynamics (CFD) code, ANSYS-CFX, was used to solve the complex flow in the rotor blade passages of the Vaneless Contra-Rotating turbine.

The results of a CFD simulation are highly dependent on the models and the methods that are specified when setting up the simulation. Solving the Navier-Stokes equations, Reynolds Averaged Navier Stokes (RANS) is used for steady simulation and Unsteady Reynolds Averaged Navier Stokes (URANS) for transient. The SST k-ω turbulence model is used due to its ability to switch between the k-ε and k- ω model and in order to combine the most accurate range of each model. At this stage of the design, RANS is suitable as it is much less computationally expensive than other models such as LES.

One of the objectives of this paper is to compare results from the mean-line analysis at design and off design operating points with CFD results. By performing a three-dimensional CFD analysis of the turbine, the degree of accuracy of the codes in mean-line analysis can be investigated.

CFX 14.5 is used for simulations. Flow is treated as periodic and domain of nozzle and rotors having one blade is taken to reduce computational cost substantially. Total pressure at nozzle inlet and static pressure at rotor2 outlet are specified as boundary condition. Flow parameter are shown in Table:3. Inlet flow is assumed to be uniform with no swirl. Air Ideal Gas is considered as working fluid. All surface viz.hub and tip end walls and blade are given smooth wall with no slip boundary condition. Rotational periodicity enforced about the axis of rotation. Standard k-ω based shear stress transport (SST) Model is used as it accounts for the transport of turbulent shear stress and amount of flow separation under adverse pressure gradients. Boundary conditions were applied as shown in Figure:5.

Parameters	Details
Analysis	Steady State
Mesh	Tetra with smoothening algorithm adapted
Refinement	At LE,TE, Blade
Turbulence model	Shear Stress Transport (SST)
Interface	Frozen rotor
Convergence criteria	0.0001
Inlet total temperature (K)	550
Inlet total pressure (atm)	2.218
Outlet Static Pressure	1.0456
Rotors speed(rpm)	600
Fluid	Air ideal gas
Inlet turbulence	1%

Table : 3

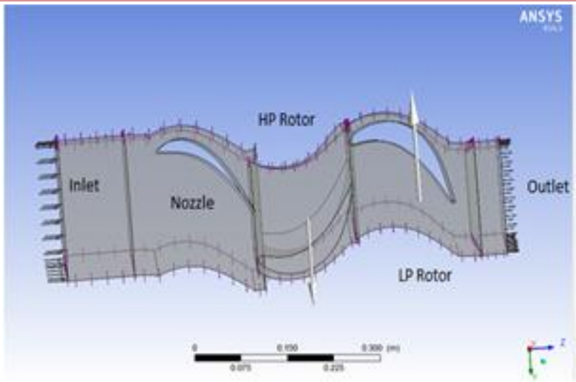

Figure 5

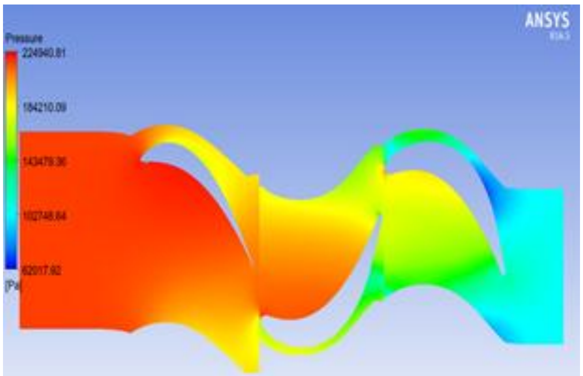

Figure 6

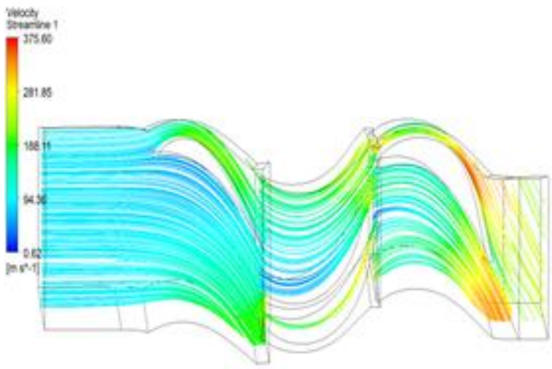

Figure 7

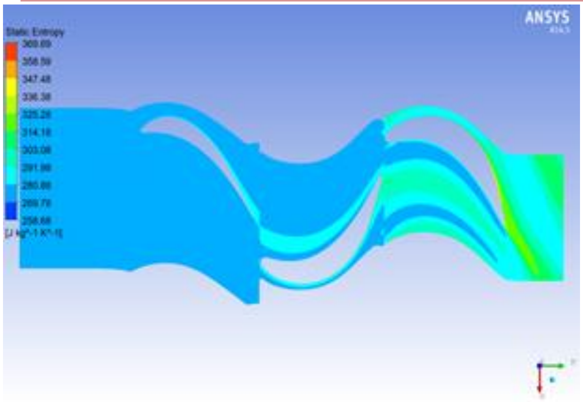

Figure 8

Parameter	HP Rotor	LP Rotor
Torque(one Blade row)	832.969 J	1234.94J
Torque(All Blades)	23.3 kJ	34.57 kJ
Power	1.46 M W	2.17 M W
ΔH	18kJ/Kg	28kJ/Kg
Dimensionless massflow(theta)	15.86	18.85
Pressure Ratio	1.25	1.52
Total-to-total isen. efficiency	89.46%	81.58%
Total-to-Static isen. efficiency	36.89%	31.22%
Total-to-total Poly. efficiency	89.35%	81.08%
Total-to-total Poly. efficiency	36.19%	30.15%

Table :4

4 Results

The criterion of solution convergence was determined by the residuals of the mass, momentum in U-V-W directions, energy and turbulence equations. The MAX normalized residuals method is more conservative than the root mean square (RMS) normalized residuals method. However, the values of MAX residuals may be large in localized areas of the grid (eg. where the mesh quality values are poor). Therefore, the RMS residuals are generally used to judge convergence. The results of RMS and MAX residuals at the design operating point simulation are plotted. The solution technique is a time marching algorithm that starts from an initial (default) guess of the solution field, and marches the solution in time until the convergence criteria are met, thus arriving at the steady value. The time increments are effectively iterations. As seen, all of the solution convergence criteria in RMS residuals method are near to 10^{-4} after 90 time steps.Simulation results are validated with the experiments conducted in a one and one-half stage large scale rotating turbine rig at the United Technologies Research Center (UTRC), USA by Dringet al..Pressure coefficient (Cp) distribution of rotor and stator are used for comparison with the experimental values of Dring et al. [12]. .Overall, the match between the simulation and experimentation is good in both rotor and stator cases.Figure:6 shows pressure distribution across pressure and suction side of blade for various span 20%,40%,50%,60%and 80%.which shows uniform pressure distribution from hub to tip. Figure:7 shows the variation of streamlines and Figure:8 shows the Entropy distribution. Velocity, Pressure, Entropy distributions across turbine stage is analyzed. The flow in VCRT is very complex including several flow phenomena, such as turbulence, separation, swirling flow and unsteadiness flow. The variation of flow parameters from hub to tip of blades are presented in graphical form and average circumferential (ACA) value of cascade parameters from inlet to outlet of the blades are computed at different operating regimes. Table: 4 shows the comparison between HP rotor and LP Rotor.

5 Conclusion

Pressure distribution across pressure and suction side of blade for various span 20%,40%,50%,60% and 80% are presented in graphical form which shows uniform pressure distribution from hub to tip. Results confirm that the pressure drops gradually form the inlet to outlet due to the extraction. The results indicate that the combined effects of vane wake, tip leakage flow, complicated wave systems and rotor wake induce the remarkable blade-to-blade variations.Entropy values enable us to identify losses as well as wake generation regions. The entropy generation inside the rotor is due to the rotational effects and due to non-uniform inlet flow from the nozzle. HP Rotor has Total to Total isentropic efficiency is 89% while LP Rotor has only 82% Total to Total isentropic Efficiency, which indicates higher losses in LP rotor compared to HP Rotor.Numerical result gives Total Power Output of VCRT is 3.63MW and work output is 46 KJ/Kg of working fluid. The results of this analysis shows a good prediction of the flow behavior inside the blades and this lead to acceptable blade design, which can be used in VCRT.

NOMENCLATURE:

α Absolute flow angle, degrees
β Relative flow angle, degrees
ε Flow turning through cascade, degrees
φ Flow coefficient, (Cx/U)
ψ Loading Coefficient $\psi = \frac{Cp\Delta Tos}{U^2}$, (NGTE, $\psi = \frac{Cp\Delta Tos}{\frac{1}{2}U^2}$)

γ Specific heat ratio of working fluid i.e. $\gamma = Cp/Cv = 1.4$
ξ loss coefficient
η_{TT} Total to total efficiency
Cx Axial component of gas velocity, m/s
U Mean section blade speed, m/s

References:

[1]. 1958, Wintucky, William T. and Stewart, Warner L., "Analysis of two-stage counter rotating turbine efficiencies in terms of work and speed requirements."NACA-RM-E57 L05
[2]. 1980, Kacker, S.C. and Okapuu, U., "A Mean Line Prediction Method for Axial Flow Turbine Efficiency."ASME Journal of Engineering for Power, 81-GT-58
[3]. 1985, Louis, J. F., "Axial flow contra-rotating turbines." ASME 85 – GT – 218
[4]. 1985, Pritchard, L.J., "An Eleven Parameter Axial Turbine Airfoil Geometry Model."ASME – 85 – GT – 219
[5]. 1987, Dring RP, Joslyn HD, Blair MF. The effect of inlet turbulence and rotor/stator interactions on the aerodynamics and heat transfer of large scale rotating turbine model. NASA Report 1987;CR 179469.
[6]. 1990, Sotsenko, Y.V., "Thermogasdynamic Effects of the turbines with the contra-rotating rotors." ASME 90-GT-63
[7]. 1990, Cai, R.X., al, "Basic analysis of Counter-Rotating Turbine." ASME 90 – GT – 108
[8]. 1991,Sharma PB, Pundhir DS, Chaudhary KK. A study of speed ratio affecting the performance of a contra rotating axial compressor stage. Defence Science Journal 1991;41(2):165-180.
[9]. 1992, United states Patent: Inventor Bruce R Branstrom Patent number 5077968
[10]. 1993, Davidson, D.P., Finke, A.K., "The design and fabrication of a small instrumented counter-rotating turbine rig." ASME 93-GT-396
[11]. 1993, Huber, F.W., al, "Design and test of a small two stage counter-rotating turbine for rocket engine application." AIAA 93-2136
[12]. 2000, Weaver, Matthew M., "Forcing Function Measurements and Predictions of a Transonic Vaneless Counter Rotating Turbine." ASME 2000-GT-0375
[13]. 2000, Haldeman, C.W., Dunn, M.G., "Experimental and computational Investigation of the Time – Averaged and Time Resolved Pressure Loading of a Vaneless Counter – Rotating Turbine." ASME 2000 -GT- 0445
[14]. 2001, Lucheng Ji, "A Vaneless Counter – Rotating Turbine Design towards Limit of Specific work Ratio." ISABE 2001 – 1062
[15]. 2000,Keith, B.D., Basu, D.K., and Stevens, C., "Aerodynamic Test Results of Controlled Pressure Ratio Engine (COPE) Dual Spool Air Turbine Rotating Rig," ASME Paper 2000-GT-0632, 2000.
[16]. 2003, M. Taylor, "HP Axial flow turbine aerodynamic design – Aero Engine Design: A State Of The Art." Von Karman Institute for Fluid Dynamics Lecture Series 2003-06
[17]. 2003, Lucheng Ji, "The Revelations from the Research about the Vaneless Counter – Rotating Turbine." ISABE 2003 – 1040
[18]. 2003, Pempie Pascal, "Counter – Rotating Turbine designed for Turbopump Rocket Engine." AIAA 2003 – 4768
[19]. 2004, Lucheng Ji, "Numerical Investigations about the Vaneless Counter – Rotating Turbine for the Cold – Air Test." ASME 2004 – GT – 53333
[20]. 2005, Wang, Huishe, "Numerical Investigations on performance of Vaneless Counter – Rotating Turbine." ISABE 2005 – 1159
[21]. 2005, Lior, D., Priampolsky, R., "Stator-Less Turbine Design." AIAA 2005-4220
[22]. 2006, Wang, Huishe, "Numerical simulation of Shock systems of Low Pressure Turbine in Vaneless Counter-Rotating turbine." ASME 2006-GT-90397
[23]. 2006,YagneshHathiwala Aero thermodynamic design of an experimental vaneless contra-rotating turbine.NAL thesis.
[24]. 2007, ZhaoQJ, Wang HS, Zhao XL, Xu JZ. Numerical analysis of 3-D unsteady flow in a vaneless counter rotating turbine. Journal of Engineering Thermophysics (1)3:352-358
[25]. 2009 Qingjun Zhao Experimental and Numerical Investigation on Flow Characteristics of a Vaneless Counter-Rotating Turbine at Off-Design Conditions AIAA 2009 4835
[26]. 2009 Qingjun Zhao Influence of Tip Clearance Size on Flow Characteristics of a Vaneless Counter-Rotating Turbine AIAA 2009 4836
[27]. 2013, Rayapati Subbarao The effect of speed ratio on the performance and flow field of contra rotating turbine ICAER 2013
[28]. 2013, Rayapati Subbarao Effect of axial spacing between the components of a counter Rotating Turbine. IJFMS 2013
[29]. 2013, Rongchao Zhao Design CRT to improve off design performance of Turbo-compounding systems GT 2013-94412
[30]. 2015,Wei Zhao Aerodynamically design and analysis a multistage vaneless counter-rotating turbine ASME 2015
[31]. 1982, Balje O.E., " Turbomachines – A guide to design, selection, and theory", A Wiley-Interscience Publication
[32]. 2002, S.M. Yahya, "Turbines compressors and fans." Second edition, Tata McGraw-Hill Publishing Company Limited, New Delhi
[33]. 1996, Cohen, H., Rogers, G.F.C., Saravanamuttoo, H.I.H., "Gas Turbine Theory", Longman Group Limited, England
[34]. ANSYS. Ansys cfx-solver modeling guide. PA 15317, 15th Edition, 2013.
[35]. ANSYS. Ansys cfx-solver theory guide. PA 15317, 15th Edition, 2013.
[36]. ANSYS. Ansys turbogrid user's guide. PA 15317, 15th Edition, 2013.
[37]. F Menter Improved two equation [k-w] turbulence model for aerodynamics flows.
[38]. www.pw.utc.com
[39]. www.google.com
[40]. www.rolls-royse.com