70(10A): 437-441; 2022. DOI: 10.18311/jmmf/2022/32947

Print ISSN : 0022-2755

Journal of Mines, Metals and Fuels

Contents available at: www.informaticsjournals.com/index.php/jmmf

CFD Analysis of a Transient Rotor-Stator Axial Steam Turbine Stage

C Rajesh Babu[1] and Sukanta Roy[2]

[1]*School of Mechanical Engineering, REVA University Bengaluru, India. *E-mail: rajeshbabu.c@reva.edu.in*
[2]*School of Mechanical Engineering, Curtin University, Miri, Sarawak, East Malaysia. E-mail: sukanta.roy@curtin.edu.in*

Abstract

An axial steam turbine stage has both stationary and rotating components, which are coupled together in a CFD model by multiple frames of reference. There are three types of interface techniques available to exchange the information between the different frames of reference namely frozen rotor, stage, transient stator rotor. Both frozen rotor and stage interfaces are used in steady state analysis. The operation of axial turbine is inherently an unsteady process. The aerodynamic interaction between the rotating part and the stationary parts are the important contributor to the unsteadiness of the flow present in the turbine. Neither of the two interfaces implemented in the steady state CFD analysis is capable of predicting the unsteady effects resulting from the rotor-stator interaction due to their relative position change. The third type of interface, the transient stator-rotor interface, is available to simulate the fluid motion caused by the relative movement between a rotor and stationary components in axial turbine. Although enormous computer resources are needed for this modelling, it simulates real flow physics best of all, while steady or quasi-steady numerical approaches only approximate the real flow, because they neglect important effects of blade row interactions. The paper presents the CFD analysis carried out for an axial steam turbine stage with three different interface techniques and compares the results.

Keywords: *Multiple frames of reference, Transient analysis, efficiency, Axial Turbine.*

1.0 Introduction

CFD model for axial steam turbine stage analysis uses multiple frames of reference model in which the guide blade is modelled in a stationary frame of reference and whereas moving blade is assigned to a rotating frame of reference. An interface applied at the junction where a change in frame of reference takes place. There are three types of interface techniques available to exchange the information between the different frames of references. The first type of interface is called stage averaging, where the upstream flow velocity profile is first averaged circumferentially before transferring the information to the downstream region or frame of reference. This method assumes the flow going to the downstream region is steady and axi-symmetric [1]. Since it circumferentially averages the values at the interface before imposing them on the neighboring reference frame, any upstream flow non-uniformity or distortion in the circumferential direction will not be preserved in the next inter region. The second type of interface implemented in the MFR analysis of the steady state CFD simulation is called frozen rotor. The flow profile variation in the circumferential direction is now preserved across the interface. However, the relative position between the two components modelled in the inter frames of reference is fixed in time and space, so this interface transfers the non axi-symmetric flow distribution developed only at the given relative position between the rotor and the stationary components to the neighbouring region. Any circumferential flow distribution change due to the variation

*Corresponding Author

of the relative position between the two involved components is not considered in this interface. In axial turbine the aerodynamic interaction between the rotating part and the stationary parts are the important contributor to the unsteadiness of the flow present in the turbine [3]. The steady state CFD analysis is not capable of predicting the unsteady effects resulting from the rotor stator interaction due to their relative position change. A third type of interface, the transient stator-rotor interface also known as sliding mesh interface, is available to simulate the fluid motion caused by the relative movement between a rotor and stationary components in turbo machinery. In this approach, a sliding interface is used between the moving mesh of the rotor and the non-moving mesh of the stationary parts. During such transient solution process, the moving mesh is made to slide past the stationary one by a certain degree during each time step according to the defined rotational speed and the time step size, and the information exchange continuously across the sliding interface [4]. The flow field variation in both time and space, specifically in the circumferential direction, due to the unsteady aerodynamic interaction and the coupling effects between the rotating component and the stationary ones, is fully taken into account in the transient stator-rotor methodology. Although enormous computer resources are needed for this modelling, it simulates real flow physics best of all, while steady or quasi-steady numerical approaches only approximate the real flow, because they neglect important effects of unsteadiness introduced due blade row interactions. However the steady state models are in use, mainly due to relative short time compared to transient calculations needed to obtain numerical results. The paper presents the results of frozen rotor, stage and transient stator rotor models for a stage with everything else of the physics being kept the same [2].

2.0 Methodology

The turbo grid mesh of blades and ICEM mesh for seal flow domain is imported to CFX Pre for defining the simulation. The total domain consists of global number of hexahedral elements of 316369. Domain definitions, interfaces between different regions and boundary conditions are defined in the CFX Pre. Seal regions are attached to blades with GGI interface as shown in Fig.1. Seals for the stationary blade are declared as stationary domain and for the moving blades as rotating domain[4]. The shaft wall at the stationary blade seals is declared as rotating wall and the casing wall at the moving blade seal is declared as counter rotating wall as shown in Fig.1. The blade walls are treated as smooth walls. steam 5 from RGP table of CFX is used as material. Stage interface is defined between the stationary blade and moving

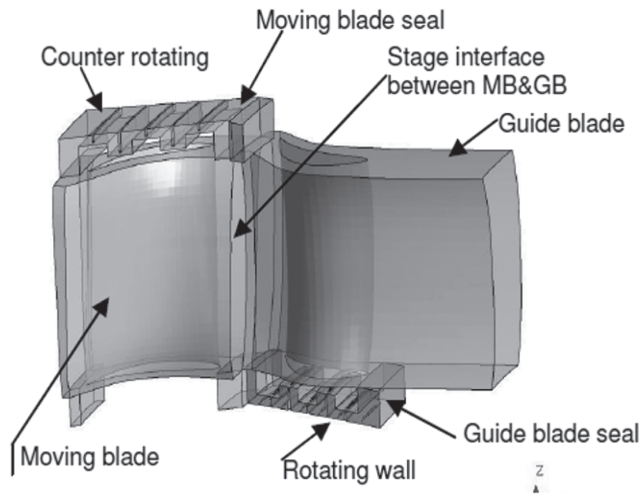

Fig.1 Turbine stage geometry model

blade using multiple frames of reference. For the stage simulation mass flow at inlet and stati pressure at exit at design conditions are used as boundary conditions [7]. The simulation is solved with K-turbulence model and converged to 1e-5 resolution of the second order. Similar analysis is carried out with frozen rotor interface between the stationary blade and guide blade.

After obtaining the solution for the stage and frozen rotor analysis, the interface is modified to define the transient stator-rotor simulation. The transient simulation is specified through time duration and time step. Time duration is set to total time option such that rotor blades pass through 1 pitch using 10 time steps. The time step size is calculated as follows:

3.0 Simulation

Rotational speed = 314.159 rad/sec, Rotor pitch modelled = 1 Time to pass through 1 pitch = 1*2*π/No. of moving blades (85) sec = 0.0739 sec Total time to pass through 1 pitch = 0.0739/314.159 = 0.00023529 sec. Since 10 time steps are used over this interval each time step should be = 0.00023529 /10 = 0.000023529 sec. The solver will continue to compute solutions at each time step iteration until specified number of time steps are reached. At each time step the solver performs several coefficient iterations to a specified maximum number [6]. The maximum number of iterations per time step may not be reached if the residual target level is achieved first. The no. of time steps can be based on whether flow has reached steady state or still developing. second order backward euler scheme, an implicit time-stepping scheme, is used for obtaining the solution. The solution is initialized using results file of stage analysis and converged to 1e-5.

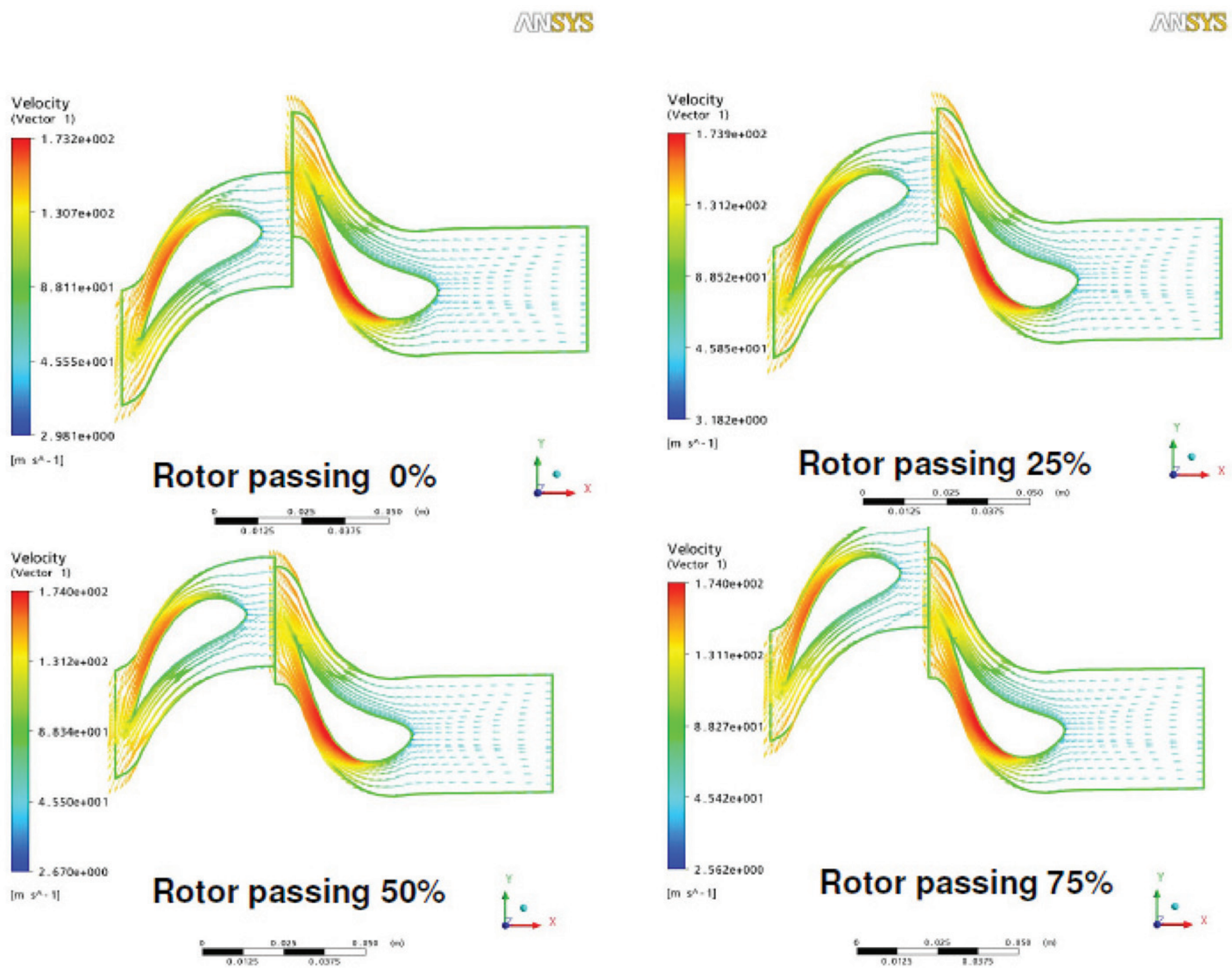

Fig.2 Velocity vector plot at different Rotor positions

4.0 Results and Discussion

To verify the convergence of a transient rotor stator case transient statistics are examined. This is done by obtaining averaged variable data over the time taken for a blade to move through one pitch. By comparing consecutive data sets, one can examine if a pseudo steady-state situation has been reached. Variable data averaged from integer pitch changes should be the same if convergence has been achieved. The results from the transient rotor stator analysis presented in the Table 1 shows variable data for torque, mass flow at outlet, etc obtained for each time step. The data confirms good convergence of the solution.

Figure 2 shows the velocity vector plot of the stage at 50% span of the blades. The Figure shows different rotor passing positions and minor changes in the velocity vector with respect to change in rotor position. Figure 3 shows the pressure contour plot at 15% span for the 100% rotor passing time step. The pressure fluctuations are studied on the rotor blade at 50% span for the transient rotor stator analysis as a function of rotor passing event. The pressure values are collected at the different points shown in Fig.4.

Figure 5 shows the pressure fluctuations on the rotor blade at 50% span of rotor. The torque developed by the moving blade is calculated from the results and found to be 260.52 Nm. Similarly the Total to total isentropic efficiency of the stage is found to be 94.18%.

The simulation is modified by changing the interface first to stage and later to frozen rotor. Everything else of physics of simulation is kept same. The solution is converged to 5e-

Table 1: Variable data for different time steps

	Time step no	Torque (N-m)	Mass flow at outlet per passage(kg/sec)	Temperature at outlet	Seal leakage per passage (kg/sec)
1	0	258.13	4.85699	796.86	0.0437
2	1	257.18	4.85611	796.86	0.0435
3	2	257.01	4.8553	796.86	0.0436
4	3	257.97	4.85515	796.86	0.0438
5	4	259.04	4.8544	796.86	0.044
6	5	259.36	4.85183	796.86	0.0441
7	6	259.35	4.84814	796.86	0.0443
8	7	259.21	4.8452	796.86	0.0445
9	8	258.63	4.84463	796.86	0.0447
10	9	257.72	4.8462	796.86	0.045
11	10	256.96	4.8485	796.86	0.0452

5 high resolution scheme. The results of the different interface models are studied and compared. Figure 6 shows the rotor blade loading for the three interface models. The major parameters of the analysis results show minor changes and presented in Table 2.

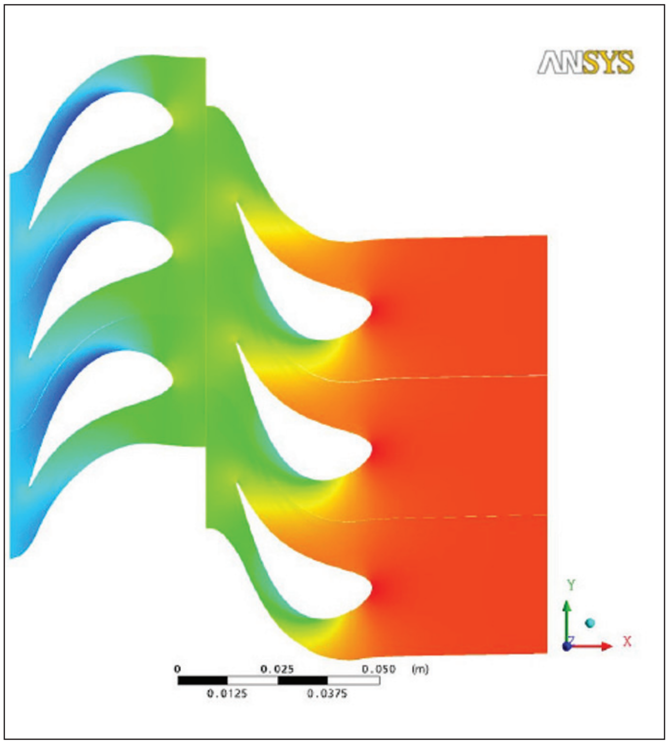

Fig.3 Pressure contour plot at 15% span

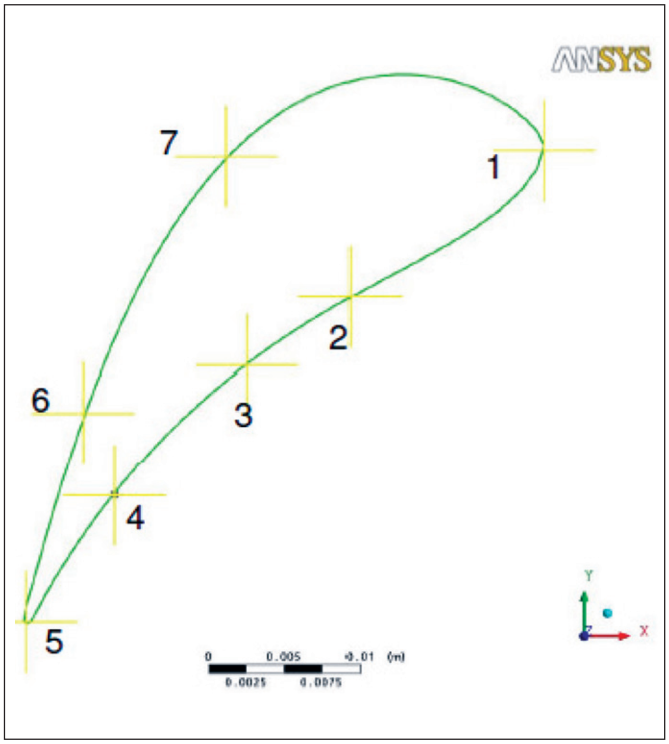

Fig.4 pressure values are collected at the different points

Table 2: Parameter comparison

	Parameters	Stage interface	Frozen rotor interface	Transient rotorinterface
1	Torque(Nm)	258.598	258.645	26.52
2	Mass flow at outlet per passage(kg/sec)	4.85699	4.85624	4.85847
3	Temperature at outlet	796.862	795.889	796.84
4	Pressure at inlet(bar)	154.829	154.806	154.938
5	Seal leakage per passage (kg/sec)	0.043742	0.04365	0.04566
6	Total to total efficiency	93.48	93.48	94.18

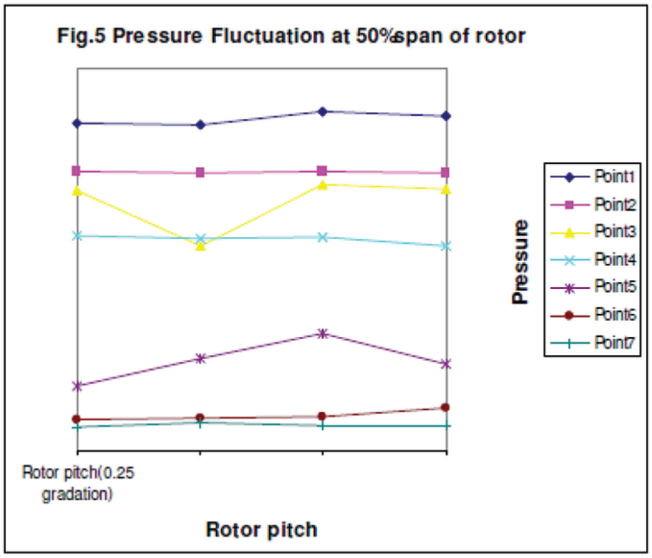

Fig.5 Pressure Fluctuation at 50% rotor span

5.0 Conclusions

Three types of interface models to couple the stationary frame and rotating frame of an axial turbine stage are studied and discussed. The study is carried out for a proven design at designed boundary conditions. All the three interface models for these conditions gave similar results with minor changes. For general purpose analysis the stage or frozen rotor interface model may give fairly accurate results and sufficient to evaluate the performance of the stage. In case the numerical analysis results are to be compared and validated by experimental results, the transient rotor stator interface model may be used. Although transient rotor stator model requires enormous computational resources it simulates the real flow physics best of all, while steady state analysis like stage or frozen rotor interface models approximates the real flow.

References

[1] Lakshminarayana, Budugur. Fluid dynamics and heat transfer of Turbomachinery. John Wiley & Sons, 1995.

[2] Ansys, C. F. X. Ansys cfx user Manual.(2005).

[3] Liu, Zheji, and D. Lee Hill. Issues surrounding multiple frames of reference models for turbo compressor applications. (2000).

[4] Zhang, L. Y., L. He, and H. Stüer. A numerical investigation of rotating instability in steam turbine last stage. *Journal of Turbomachinery* (2013) 135.1.

[5] Toebben, Dennis, et al. Numerical investigation of the heat transfer and flow phenomena in an ip steam turbine in warm-keeping operation with hot air. Turbo Expo: Power for Land, Sea, and Air. *American Society of Mechanical Engineers,* (2017) 50954..

[6] Cao, Lihua, et al. Strain-life estimation of the last stage blade in steam turbine during low volume flow conditions. *Engineering Failure Analysis* 125 (2021) 105399.

[7] Xu, Meichao, et al. Effect of water spray in exhaust passage of steam turbine on flow field of the last stage during windage. *International Journal of Heat and Mass Transfer* 161 (2020) 120296.