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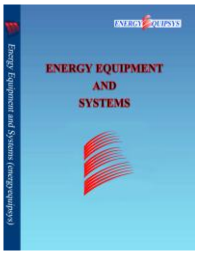

Numerical investigation of clocking in a two-stage gas turbine

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ABSTRACT

Flow in the first two-stage of V 94.2 gas turbine is simulated numerically. In this turbine, the second stator is clocked relative to the first stator to different positions. Steady-state analysis was carried out by varying the circumferential relative position of the consecutive stator vanes to study the effects of the clocking on turbine performance. A density based compressible inviscid flow (Euler equations) solver is used for the flow simulation. The efficiency analysis and entropy generation investigation are performed and the appropriate position of the second stator is obtained. The attained efficiency improvement matches the results obtained by the entropy generation analysis.

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1. Introduction

A way to optimize the turbine efficiency is to clock (position) the blades according to each other appropriately in such a way to minimize the effect of the stator-rotor interaction. Stator blades are stationary blades; therefore they can be clocked easily. This can affect the efficiency of the gas turbine.

In the last decade, there have been many experimental and numerical investigation focused on clocking. An unsteady viscous flow simulation was performed on one and half stage turbine by F. Eulitz et al. in 1996 [1]. The investigation is focused on the shape and the propagation of the wakes from the upstream stator blades and their influence on the turbine losses. It has been found that the overall losses in the exit of the second stator differ for the investigated configuration. F. W. Huber et al. conducted an experimental/analytical and numerical study to determine the performance improvements achievable by circumferentially indexing succeeding rows of turbine stator airfoils [2, 3].

The results from this study indicate that significant increases in stage efficiency can be attained through application of this airfoil clocking concept. In their research an extensive time-accurate CFD simulations completed for the test configurations. Time-accurate Navier–Stokes flow analysis performed for the five different turbine stator positions. A quasi-three dimensional, blade to blade, time-accurate, viscous solver was used for a three-stage low pressure turbine study by A. Arnone et al. [4].

Unsteady analysis of the clocking was conducted and the optimum clocking position was obtained. The effect of the wake interaction mechanism and unsteady blade loading were also studied in their work. The clocking of a high-pressure turbine vane relative to the downstream low-pressure turbine vane was investigated in [5].

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The unsteady three-dimensional (3D) flow through a two-stage turbine was simulated numerically, using a time marching Navier–Stokes computer code with a sliding mesh approach by D. Bohnetal. A relative efficiency variation of 0.52% is concluded among clocking positions. A frozen rotor approach in a steady calculation and a sliding mesh approach in an unsteady simulation were performed in a stator clocking [6, 7].

It is concluded that the frozen rotor approach is valid to search the optimum clocking position in the preliminary design period, although it misses some features of the unsteady flow field in the multistage turbines.

In this work, the V 94.2 industrial gas turbine is simulated. In order to find the optimum stator position, the second stator is clocked in three circumferential positions relative to the first stator using sliding mesh approach.

2. Geometry

The turbine includes two stages: two stators and two rotors, each stator row has 48 cascades and each rotor row has 89 cascades. In order to simulate the flow, one stator and two rotors are selected to model the geometry. In this way, the width of stator domain and the rotor domain are approximately equal. Each cascade is simulated in a periodic domain. Periodic boundaries can be either straight lines or the curved ones. In order to make the appropriate periodic boundaries, they should be the streamlines of the flow field. The periodic boundary curvature is chosen the cascade regression line, which is a parabolic curve. Figure 2 shows this line clearly.

In order to make the clocking series geometry, the second stator is clocked in a step of 2.5 ° in a range of one stator pitch. Therefore, there are three clocking positions: S1, S2, and S3. In the third position the stator is located both on right and left periodic boundaries. The relative position of the second clocked cascade is shown in the Fig.1.

3. Boundary Conditions

To simulate the flow we need five types of boundary conditions: inlet, outlet, periodic, stationary wall, and moving wall. The inlet boundary condition is chosen as the pressure inlet. First stator inlet condition is described as the table 1.

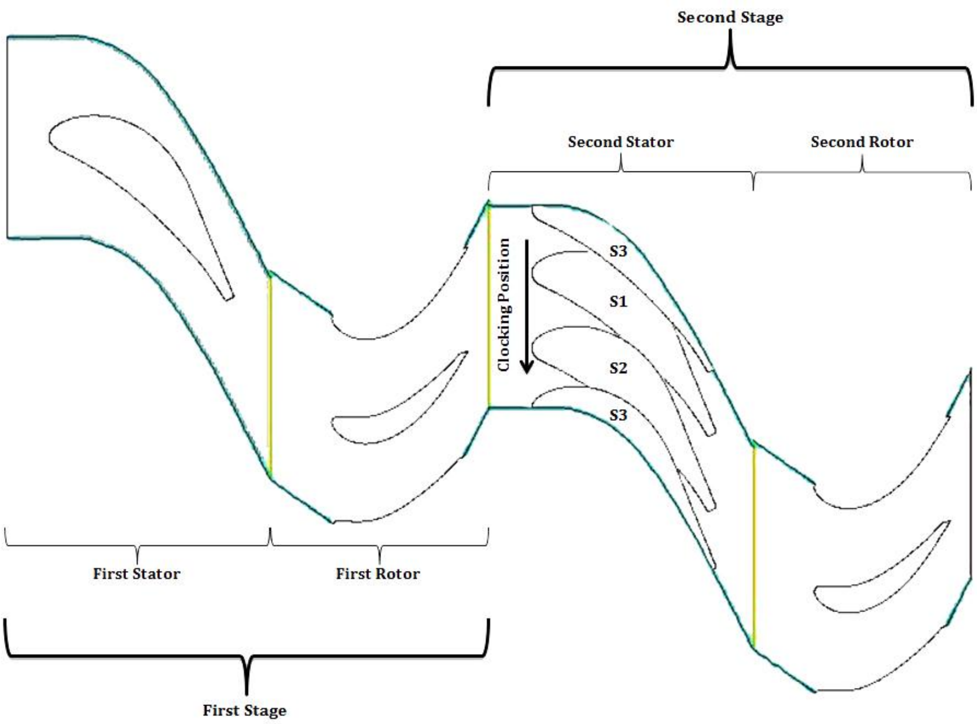

Fig. 1. Two-stage configuration of the investigated turbine

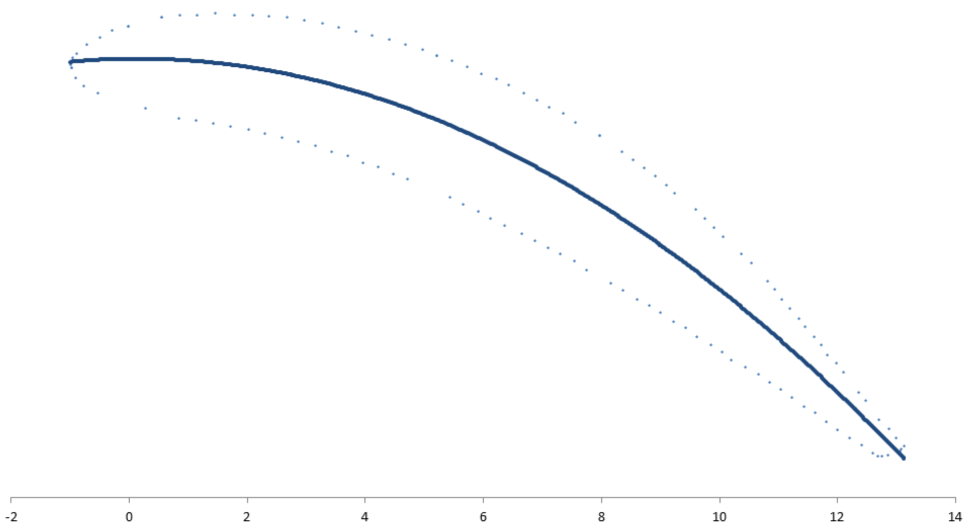

Fig. 2. The regression line of a cascade

Table 1. First stator inlet boundary condition.

Parameter	Magnitude
Static Pressure	1058640 pa
Total Pressure	1102210 pa
Static Temperature	1333K
Total Temperature	1347 K
Mach Number	0.247
Velocity	176 m/s
Mass Flow Rate	509 Kg/s
Density	2.769 Kg/m3

Pressure outlet is used for the exit boundary condition. In order to compute the second rotor outlet boundary condition we use the procedure below:

3.1. Boundary Conditions after the Stator Row

The total temperature is constant in the stator [8, 9]:

$$T \;\; = T_{01} \tag{1}$$

The continuity equation:

$$\rho_2 = \frac{\dot{m}}{A_2 C_{axial}} \tag{2}$$

Using the isentropic relations:

$$T_2 = T_1 \left(\frac{\rho_2}{\rho_1} \right)^{\gamma - 1} \tag{3}$$

$$P_2 = P_1 \left(\frac{\rho_2}{\rho_1} \right)^{\gamma} \tag{4}$$

Mach number can be computed:

$$C_2 = \sqrt{2C_P \left(T_{02} - T_2 \right)} \tag{5}$$

$$M_2 = \frac{C_2}{\sqrt{\gamma R T_2}} \tag{6}$$

The total pressure is computed by:

$$P_{02} = P_2 \left(1 + \frac{\gamma - 1}{2} {M_2}^2 \right)^{\frac{\gamma}{\gamma - 1}} \tag{7}$$

3.2. Boundary Conditions after the Rotor Row

Same as the stator, according to the continuity equation and the isentropic relations [8, 9]:

$$\rho_3 = \frac{\dot{m}}{A_3 C_{axial}} \tag{8}$$

$$T_3 = T_2 \left(\frac{\rho_3}{\rho_2} \right)^{\gamma - 1} \tag{9}$$

$$P_3 = P_2 \left(\frac{\rho_3}{\rho_2} \right)^{\gamma} \tag{10}$$

The stator outlet absolute velocity is known.

The axial velocity of the flow is also nearly constant throughout the turbine. Therefore according to the velocities triangle:

$$W_{axial} = C_{axial} \tag{11}$$

$$W_t = C_t - U \tag{12}$$

$$W = \sqrt{{W_t}^2 + {W_{axial}}^2} \tag{13}$$

The rotor outlet relative velocity is obtained by the relative total temperature relation:

$$T_{02rel} = T_2 + \frac{\overline{|W_2|^2}}{2C_P} \tag{14}$$

The relative total temperature is constant through the rotor, and then velocity after the rotor can be obtained:

$$T_{03rel} = T_{02rel} \tag{15}$$

$$W_3 = \sqrt{2C_P\left(T_{03rel} - T_3\right)} \tag{16}$$

$$\vec{C}_3 = \vec{W}_3 + \vec{U} \tag{17}$$

Finally the Mach number, the total temperature and the total pressure are obtained:

$$M_3 = \frac{C_3}{\sqrt{\gamma R T_3}} \tag{18}$$

$$T_{03} = T_3\left(1 + \frac{\gamma - 1}{2} M_3{}^2\right) \tag{19}$$

$$P_{03} = P_3\left(1 + \frac{\gamma - 1}{2} M_3{}^2\right)^{\frac{\gamma-1}{2}} \tag{20}$$

According to the taken procedure, second rotor outlet boundary is computed. The outlet boundary conditions are:

Table 2. Second rotor outlet boundary condition.

Parameter	Magnitude
Static Pressure	453789 pa
Total Pressure	486415 pa
Static Temperature	1079 K
Total Temperature	1098 K
Mach Number	0.324
Velocity	176m/s
Mass Flow Rate	509 Kg/s
Density	1.466 Kg/m^3

4. Numerical Scheme, Validation and Grid Independency

An Implicit density based compressible inviscid flow (Euler equations) solver is employed for the numerical simulation [7]. The sliding mesh approach is taken to simulate the moving interfaces [7]. Indeed, this approach seems to be necessary for considering the relative motion of the stationary and the rotating components of rotating machinery and it simulates the unsteady interactions.

The method is validated using simulation of the compressible flow over a two-dimensional standard vane of LS95 VKI and by comparison with the experimental test case of Siverding [10].

The grid independency was performed and the resulting mesh resolution was used for our simulations.

5. Results

5.1. Entropy Discussion

Increasing the efficiency of the turbine is the main goal of the clocking. Therefore, analyzing the entropy generation and its minimization is the key in our research. The entropy distribution contour is plotted to understand the physical aspects of the mentioned goal.

The change in entropy can be calculated from [7]:

$$\Delta s = C_P \ln\left(\frac{T}{T_{ref}}\right) - R\ln\left(\frac{P}{P_{ref}}\right) \tag{21}$$

where T is the temperature, P is the static pressure, the T_{ref} is the reference temperature (288.16 K) and the P_{ref} is reference atmospheric pressure (101300 Pa).

The leaving flow from the trailing edge is also displayed in Fig.3. The entropy distribution contours for the clocking positions S1, S2, and S3 are shown in Fig.4, 5 and 6.

The losses due to the stator trailing edge are obvious. According to the Fig.4, 5 and 6 clocking makes a difference in the entropy distribution for different clocking positions. As a result, different entropy distribution is obtained at the outlet boundary.

As Fig.7 shows, it is obvious that the entropy generation is significantly affected by the flow leaving from the stator trailing edge.

5.2. Efficiency Discussion

We define the relative total efficiency as:

$$\eta = \frac{\gamma}{\gamma - 1}\frac{\ln(T_2/T_1)}{\ln(P_2/P_1)} \tag{22}$$

where, T_1 and T_2 are the mass averaged static temperature at the inlet and the outlet boundaries. P_1 and P_2 are the mass averaged static pressure at the inlet and outlet. γ is the specific heat ratio and η is the efficiency. The efficiency and the entropy are compared for the three clocking positions in the Fig.7 and 8.

According to the Fig.8 the efficiency is increased by 1.2 percent due to clocking; i.e. for a 100 MW gas turbine, output power will be increased by 1200 KW.

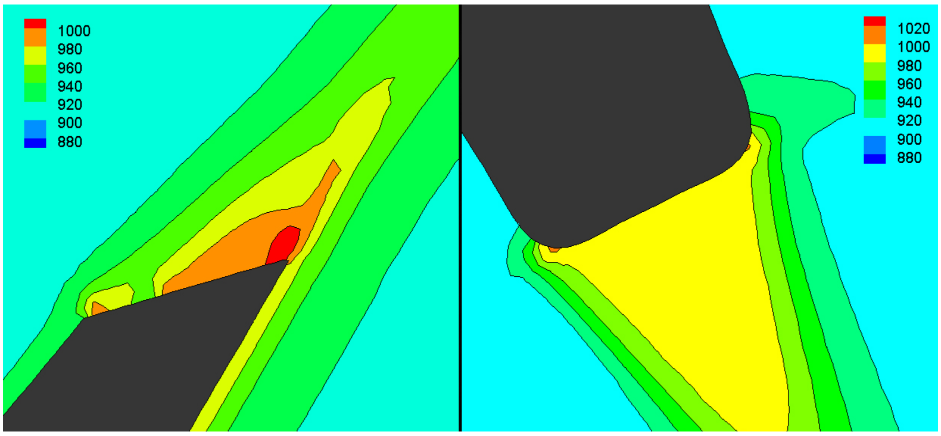

Fig.3. Entropy generation at the trailing edge of the rotor (left) and the stator (right).

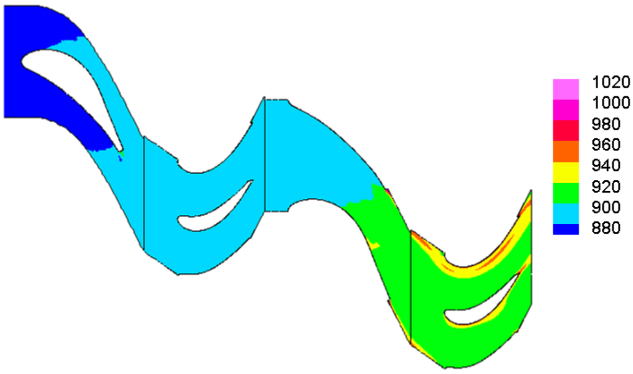

Fig.4. Entropy distribution for the clocking position S1.

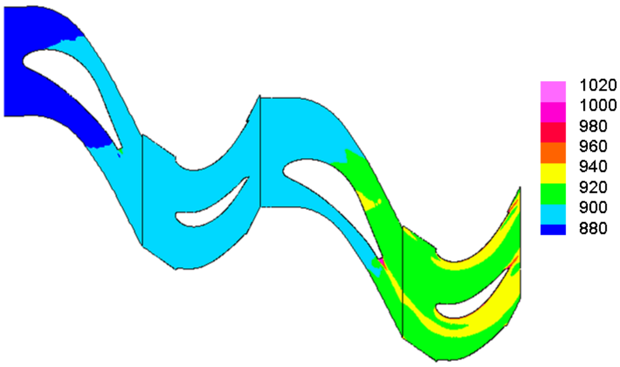

Fig.5. Entropy distribution for the clocking position S2.

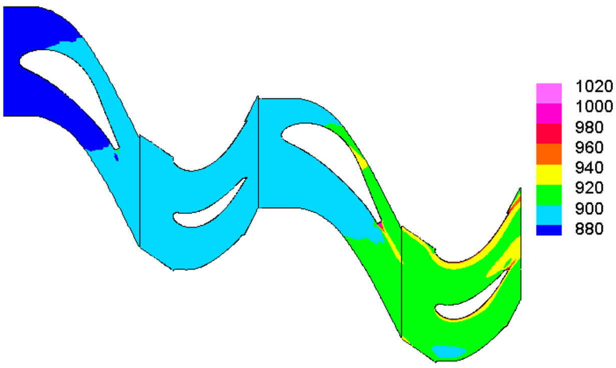

Fig.6. Entropy distribution for the clocking position S3.

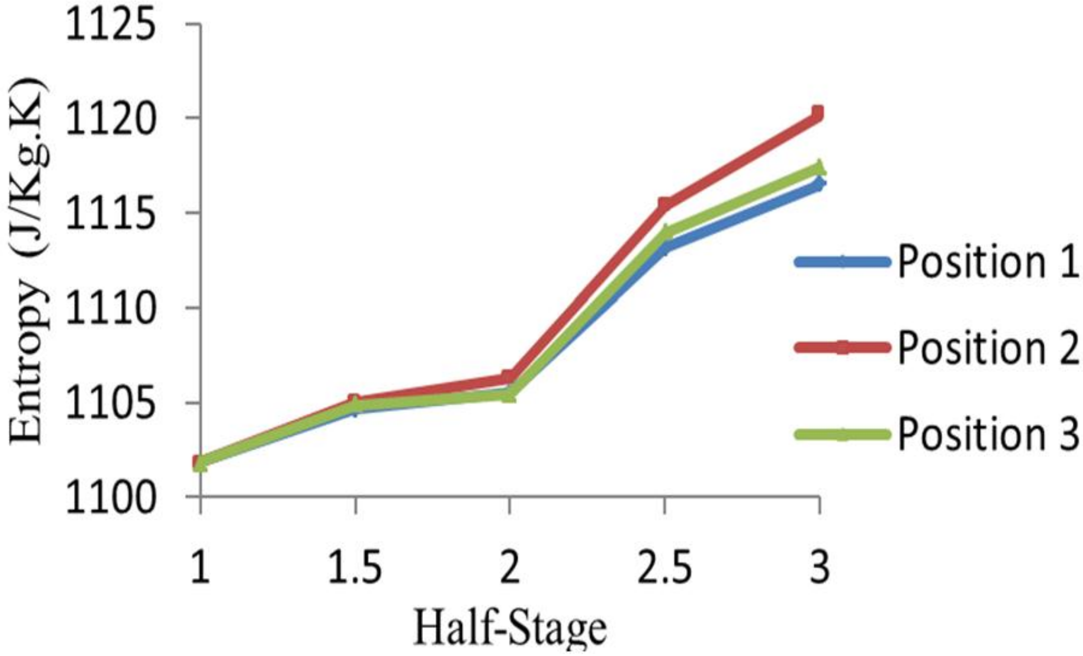

Fig.7. Entropy generation at the interface.

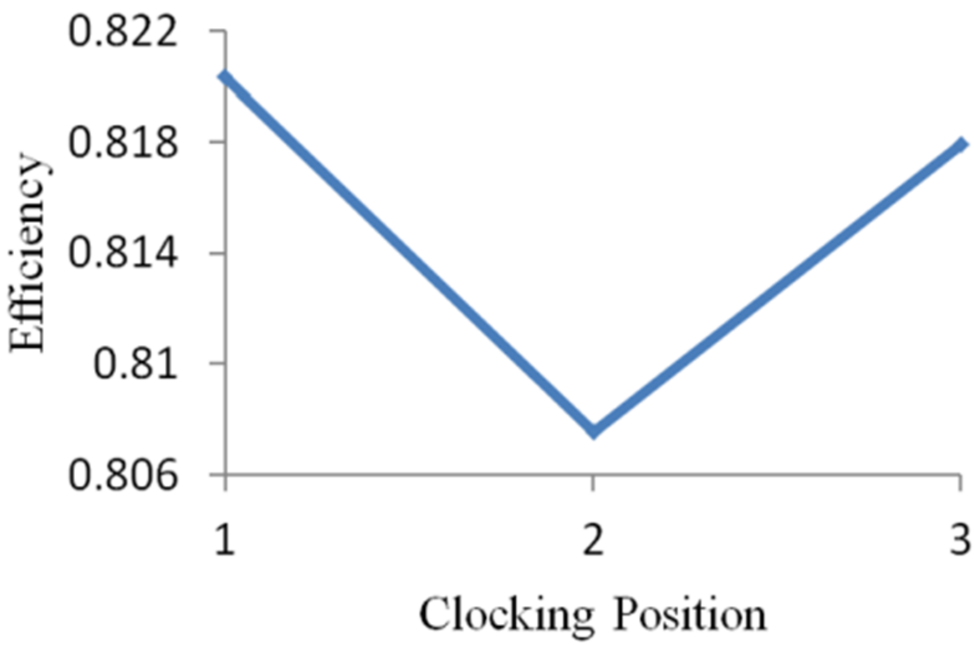

Fig.8. Efficiency of the clocking position.

6. Conclusions

In this paper, the flow field of a two stage gas turbine is simulated numerically. A complete procedure to compute the appropriate boundary conditions is presented. The stator blades of the second stage are clocked in three positions relative to the first stator. The sliding mesh approach is used to simulate the moving interfaces of the gas turbine. The entropy generation and the relative total efficiency are computed for all the clocking positions. According to the results, the efficiency reaches its maximum at position no. 1 where the second stator is shifted to the right side of the first stator. The entropy generation contours confirms the best clocking position where the least losses due to the stator-rotor interactions occur between stages. As a result of our study, the efficiency of the stage was increased approximately by one present, which is a reasonable value considering previous investigations [6, 7].

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