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Influence of blade lean together with blade clocking on the overall aerodynamic performance of a multi-stage turbine

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ABSTRACT

In the present work, an in-house flow solver for multi-stage turbomachines is first applied to compute the unsteady flows in a subsonic 1.5-stage axial turbine with different clocking configurations. The influence of the wake location of the first stator on the overall adiabatic efficiency is presented and analyzed, demonstrating the possibility to improve the aerodynamic performance of the turbine by properly leaning the stator blade. Four different blade lean schemes are then configured based on the given clocking configuration with peak efficiency at 50% span. The overall adiabatic efficiency and the spanwise distributions of aerodynamic performance parameters of different lean configurations are presented and compared to those of the baseline configuration. The adiabatic efficiency of the optimal lean configuration is about 0.35% larger than that of the baseline configuration, while the maximum adiabatic efficiency difference among different clocking configurations is only 0.086%. The improvement on the overall aerodynamic performance of the multi-stage turbine contributed by blade lean is significantly remarkable. The blade lean and blade clocking can be carefully assembled in the unsteady flow regime during the design or optimization phases of multi-stage turbomachines.

1. Introduction

The unsteady nature of the flow in multi-stage turbomachines makes it possible for designers to improve the aerodynamic performance by changing the relative pitchwise positions of consecutive stators/rotors, *i.e.*, the clocking configuration. In the past decades, numerous experiments and numerical computations have been implemented to investigate the influence of blade clocking, including two-dimensional and three-dimensional stator clocking [1–5], rotor clocking and full clocking [6], on the aerodynamic performance of both compressors and turbines with various geometric configurations and operating conditions. It is concluded that at a specific span position the stage efficiency reaches its peak when the wakes of upstream stator/rotor blades pass through the downstream stator/rotor passages extremely near the suction sides. In the authors' previous work [7], the same conclusion is also obtained at the midspan of a subsonic 1.5-stage axial turbine. It is difficult for a three-dimensional multi-stage turbomachine to synchronously achieve the peak efficiency at all span positions within a specified clocking configuration because of the currently available design methods of turbomachinery based on steady flow. However, in the scope of unsteady flow regime, by properly setting the spanwise distribution of blade lean, one can strive to make different span positions to reach their own peak efficiency at the same clocking configuration as far as possible, and thus further improve the overall aerodynamic performance of multi-stage turbomachines.

Although the influence of blade lean has been investigated experimentally [8,9] and numerically [10,11] and considered in the design phase of turbomachine blade [12,13] since many years ago, most of investigations and all blade design methods focus on the lean effects on the aerodynamic performance of single cascade [8–11] or single stage [14–16]. In such situations, time-averaged measurements and steady computations are usually employed. By these, the influence of different kinds of blade lean configurations, such as straight and compound lean, positive and negative lean, on the overall loss and flow angle change is extensively studied. It is concluded that the pressure gradient perpendicular to the endwall would accelerate the flow near the endwall and thus change the stage reaction and dominantly affect the endwall loss. Besides the effects of blade lean alone, the influence of blade lean combined with blade clocking is also studied in the unsteady regime. For example, Chen et al. [17,18] assess experimentally the stator clock-

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ing effects on improving the overall aerodynamic performance of a 2-stage compressor with compound lean stators. However, the lean configuration in their work is a pre-established assignment rather than a purposeful option based on the detailed unsteady flow fields.

In the present study, an in-house three-dimensional flow solver of Unsteady Reynolds Averaged Navier–Stokes (URANS) equations with turbulence model equations is first introduced and implemented. The flow solver is then applied to compute the unsteady flows in a subsonic 1.5-stage axial turbine with five different clocking configurations. Based on the analysis on clocking effects, four alternative blade lean schemes are thus configured. The influence of blade lean on the aerodynamic performance of the multi-stage turbine is presented and analyzed in detail, and the optimal lean configuration with the maximum adiabatic efficiency improvement is obtained.

2. Description of unsteady computation

The three-dimensional URANS equations expressed in the rotating frame of reference [19] are solved by using the cell-centered finite volume method. The turbulent viscosity is resolved by the Spalart–Allmaras turbulence model [20] which is extensively applied in internal flow simulations and is verified by comparisons with experimental data [19,21,22]. The convective and viscous fluxes are discretized by the JST scheme [23] and the second-order central scheme, respectively. The dual time-stepping approach [24] is employed to solve the unsteady problem. The physical time derivative terms are discretized by the second-order backward difference formula. The Lower–Upper Symmetric-Gauss–Seidel (LU-SGS) method [25] is applied to the pseudo time step. The techniques of local time-stepping and multi-grid are adopted to accelerate the convergence of pseudo time marching.

At the inlet, the spanwise distribution of total pressure, constant total temperature and axial flow angle are given. At the outlet, the spanwise variation of static pressure is assigned by the simple radial equilibrium equation with the pressure at the hub fixed. The no-slip boundary condition with zero normal pressure gradient and no heat flux is specified on the hub, casing and blade surface. Following the blade scaling rule proposed by Rai [26], the adjacent blade rows are reduced to have the same pitch angle. In such situations, the periodic boundary conditions can be employed on the two circumferential boundaries of each blade row. The boundary conditions on the interfaces between adjacent blade rows are implemented by the sliding mesh technique. The flow field information across the interfaces is exchanged by a volume-weighted interpolation method, which is described in [7] in detail. The interpolation method can perfectly ensure the flow variable conservation across the interfaces. In fact, in numerous test cases, the relative errors in the conservation of mass, momentum, energy, *etc.* integrated over the interface are always less than 10^{-5}, which is considered to be accurate enough.

A subsonic 1.5-stage axial turbine originally designed by RWTH Aachen University [27,28] is studied in the present work. The turbine consists of three blade rows: the first stator (Stator 1), the rotor (Rotor) and the second stator (Stator 2). The straight low-aspect-ratio blades are used for all blade rows. For the convenience of unsteady computation, the blade counts are scaled from 36 : 41 : 36 to 36 : 48 : 36 following the Rai's rule [26]. Thus, 3, 4 and 3 passages for Stator 1, Rotor and Stator 2, respectively can be configured. The structural multi-block grid is shown in Fig. 1. The cell vertexes on the interfaces among neighbor blocks are completely matched except the interfaces between adjacent blade rows, which are such planes perpendicular to the axial direction. The size of the grid is shown in Table 1. The total cell number is about 11.9 million with the dimensionless wall distance y^+ less than 1 for the first grid point off the wall.

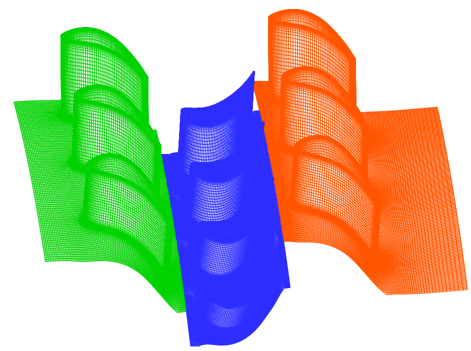

Fig. 1. Grid view over hub and blade surface.

Table 1
Grid configuration for unsteady computation.

	Stator 1	Rotor	Stator 2
Spanwise	97	129	97
Pitchwise	57	57	57
Blade surface	217	241	217
Tip clearance	/	25	/
Cells per passage	789,540	1,727,488	872,448

To check the grid independence, the unsteady flow fields are then computed on the fine and coarse grids generated by changing the cell number in each direction. The total cell numbers of the fine and coarse grids are 23.0 million and 4.0 million, respectively. The spanwise distributions of absolute Mach number at three different axial locations are compared in Fig. 2 for the three grid configurations. The distributions of aerodynamic performance parameters on the medium grid, including absolute Mach number and others not shown here, coincide closely with those on the fine grid, while apparent differences are observed on the coarse grid. Thus the medium grid is used through the present work.

3. Flow solver validation

The flow solver is validated by comparing the computed time-averaged results with the available experimental data [27,28]. To reduce the geometric difference from the original turbine as far as possible, the blade counts are scaled to 36 : 42 : 36 during the flow solver validation. However, the blade counts are still 36 : 48 : 36 in the rest of the paper in consideration of computational costs. The computed spanwise distributions of total pressure and absolute flow angle at three different axial locations are presented and compared with the experimental results in Fig. 3 and Fig. 4, respectively. The computed results agree well with the experiment, especially for the situations after Stator 1. The computation perfectly captures the secondary flow in the Stator 1 passage, as indicated by the total pressure deficits near the hub and casing in Fig. 3(a). All the comparisons prove that the present flow solver is trustable.

4. Results and discussion

4.1. Performance impacts of blade clocking

Fig. 5 displays the sketch of five different clocking configurations named by CC0, CC1, CC2, CC3 and CC4, respectively. Stator 1 is clocked in steps of 2 degrees within one stator pitch. CC0 corresponds to the baseline configuration, in which the leading edge of Stator 1 is aligned with that of Stator 2. CC5 is identical to CC0.