

Research on Transonic Centrifugal Compressor Blades Tip Clearance Distribution of Vehicle Turbocharger

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ABSTRACT

Flow induced by blades tip clearance is important for centrifugal compressor, especially for the high charging ratio transonic centrifugal compressor of vehicle. Based on three-dimensional CFD method, the flow fields of two opposite impeller blades tip clearance distribution compressor stages with vaneless diffuser are analyzed. The law and action mechanism of tip clearance distribution effect on the high charging ratio transonic centrifugal compressor impeller and vaneless diffuser performance are researched. It is verified that the pressure ratio and efficiency of compressor stage whose aft axial clearance height is smaller than fore radial clearance height is higher than that of compressor stage whose aft axial clearance height is larger than fore radial clearance height. The smaller aft axial clearance height weakens the flow in aft half of impeller passage, improves impeller aft passage power input and obtains higher impeller pressure ratio and compressor stage pressure ratio. The smaller aft axial clearance height gets higher efficiency because it reduces impeller aft blades tip clearance leakage vortices intensity and imposes the vortices breakdown and dissipation loss less in impeller passage exit and vaneless diffuser passage. The research of the impeller blades tip clearance distribution effect on vehicle transonic centrifugal compressor performance and flow structure lays a foundation for transonic centrifugal compressor flow control.

INTRODUCTION

The tendency that the vehicle diesel engine pursues higher power density drives turbocharger charging ratio to be higher. Followed by the increase of charging ratio, because of the limitations on the outer diameter, the inlet blade height of the impeller is substantial, causing the relative Mach number to surpass unity on a localized basis, thus creating transonic inflow condition. The leakage flows originating from the gap between the casing and rotor blade tip is affected by shock wave

position and intensity. At the same time, the evolutionary and breakdown of the clearance leakage vortices are affected by clearance height and clearance distribution. The height of impeller blades tip radial and axial height imposes impeller and diffuser power input and flow loss changed, and that the clearance height distribution controls the efficiency and pressure ratio of the whole compressor stage including impeller and vaneless diffuser. Furthermore, the research on the clearance height distribution effect under compressor stage performance and flow field structure is the premise to apply casing treatment flow control method.

The effect of tip clearance height on the centrifugal compressor flow field has been investigated for a long time. It is known that a larger tip clearance results in a larger efficiency drop. Senoo and Ishida [1] have measured deterioration of compressor pressure rise due to inlet tip clearance at the impeller in centrifugal impellers. Schleer [2] reports degradation in pressure rise and stability due to the increase in the tip clearance; Pampreen [3] investigated six different impellers with different tip clearances. He estimated that there was a 3% efficiency drop when the relative tip clearance (t/b_2) increased by 10%. Klassen et al. [4, 5] tested centrifugal compressors of pressure ratio 6, a backswept bladed impeller and a vaned and vaneless diffuser. The axial tip clearance was varied and compressor characteristics were measured. Similar results on the effect of tip clearance on compressor characteristics have also been studied by Mashimo et al. [6] and Schumann et al. [7]. Most previous investigations were focused on the overall characteristic, not on the components (inlet, impeller, and diffuser). Only a few results show the components performance affected by clearance height. Backman et al. [8] considered that impeller fore radial clearance had little effect on the pressure rise of vaneless diffuser. Sato [9] and Weiß [10] testified that decreasing the impeller aft clearance made wake close to blades suction surface in the aft part of passage based on test and simulation. Therefore, the sensitivity of component characteristic and flow structure to clearance distribution variations remains to be investigated.

In this paper, based on three-dimensional CFD method, the flow fields of two opposite impeller blades tip clearance distribution compressor stages with vaneless diffuser are analyzed. The mechanism of clearance leakage vortices generation and breakdown are researched and the sensitivity of impeller and vaneless diffuser pressure ratio drop and efficiency loss on radial clearance height and axial clearance height are discussed. The research guides the application of high charging ratio transonic compressor casing treatment flow control method.

NUMERICAL MERHODS

The code used in this simulation is based on a 3-D steady compressible finite difference scheme to solve the Reynolds-averaged Navier-Stokes equations in conservative formulation. Modified Jameson central difference scheme [11] is applied to spatial discreteness to receive satisfaction from shock wave resolution and viscosity resolution. An explicit Jameson forth-order Runge-Kutta scheme [12] was used. Convergence of the solver is first accelerated by local time stepping, residual average methods and multigrid procedure. For turbulence model, Spalart-Allmaras [13] is adopted, which is theoretically robust in numerical calculation and has numeration accuracy for the calculation of adhesive boundary layer turbulent flow, separated flows of small or medium scale and free shear turbulent flow except for jet. However, in Reynolds-averaged computations of separated turbulent flow, to find a proper yet simple turbulence model is an extremely delicate and difficult issue.

Total pressure, temperature and velocity components were imposed at the inlet boundary. At the outlet boundary the static pressure was imposed at choke and self adaptive mass flow rate was imposed at other operation conditions. The impeller surface is defined as rotational non-slip solid boundary and that the casing is defined as static non-slip solid boundary.

Figure 1 shows the external appearance of the test compressor stage used in the work reported here. The test compressor includes transonic impeller, vaneless diffuser and volute. The impeller is a backward swept open impeller with an outer diameter of 144mm, having 9 full blades and 9 splitter blades. The inlet tip diameter is 103 mm, the inlet blade height is 36.5 mm and the tip clearance is 0.5 mm from the leading edge to the trailing edge. The leading edge tips of the impeller have a Mach number of approx. 1.0. The design speed of test compressor is 60,000 (impeller outlet peripheral speed of 452m/s), the pressure ratio is 3.8, and the design mass flow rate is 1.0kg/s.

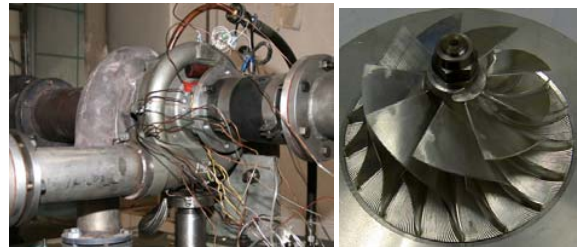


Fig.1 Vehicle turbocharger and transonic impeller

For validation the CFD code, the grid of the whole compressor stage is built. As indicated in Figure 2, the compressor stage grid is made up of 9 full blades passages grid and volute grid. Inlet of compressor, impeller passages and vaneless diffuser are calculated in relative rotation coordinate, and volute is solved in absolute coordinate. For It is found that impeller simulation performance varies little if single full blades passage grid is more than 150,000 in grid independence test, the single full blades passages grid cell number is set as 202,046 to minimize the calculation time. The whole stage grid is about 1.99 million: 9 full blades passages grid is 1,818,414 and volute grid is 167,960.

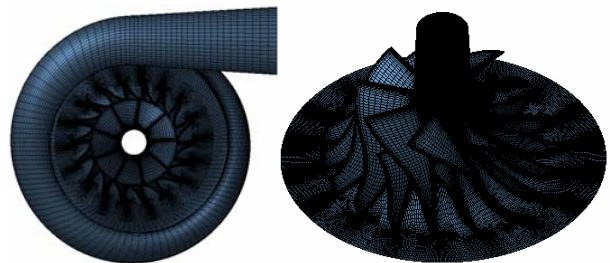


Fig 2 Grid of vehicle turbocharger and transonic impeller

For the design speed the characteristics of the impeller have been evaluated in more detail and are plotted in Figures 3 versus mass flow rate. First, the test compressor characteristics and the characteristics calculated for the compressor stage including volute should be compared. As the pressure ratios have some difference and the resulting efficiencies shows the corresponding difference, the work input to the fluid is higher for the calculations than for the measurements. Characteristics curve has the same tendency as Krain impeller test and simulation [14]. Krain considers that perhaps the turbulence models cannot solve this problem satisfactorily. We think that there are three reasons: first, the leakage loss of clearance between compressor back disc and diffuser is not considered in simulation. Second, the roughness of test compressor volute internal wall is high, but that the simulation has the hypothesis of smooth volute internal wall. Third, the turbulence models cannot satisfy the calculation completely, especially for jet and turbulent separation under intensive reverse pressure gradient. These reasons lead to the pressure and corresponding efficiency of calculation than the test data. But calculation coincides with test at surge and choke boundary, besides the pressure and efficiency curves distribution tendency of calculation match with test result. So the calculation is credible.

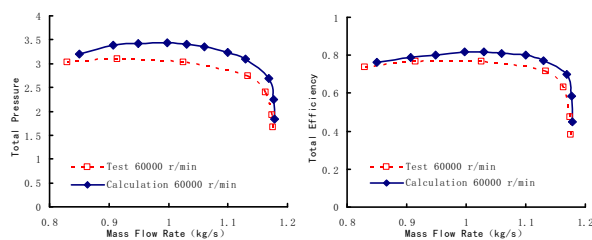


Fig 3 Designed speed line performance of stage

RESULTS AND DISCUSSION

DETAILED FLOW FIELD ANALYSIS GRID

In order to improve the flow field resolution at finite computer condition, the detailed flow field analysis calculated only 1 full blade pitch, including 1 splitter blade. A multi-block computational grid was used, creating a 12 block inter-blade pitch for inlet passage, full blades and splitter blades, thus inserting 4 blocks of the grid for the tip clearance of each blade. Each block was composed of an H-type structural grid. There were 240 cells in the flow direction, 56 cells in the spanwise direction, and 96 cells in the pitchwise direction, with a total of 1.74 million cells in 12 blocks. The blade tip clearance grid contained approx. 107,520 cells, with a total of approx. 1.85 million cells for 1 blade pitch.

CLEARANCE DISTRIBUTION EFFECT ON PERFORMANCE

This paper takes the uniform 0.5mm blades tip clearance height transonic impeller as baseline impeller to research the different clearance height distribution effect on compressor stage performance. Fig 4 is the impeller blades tip clearance height distribution sketch map. As indicated in Fig 4(a), the radial clearance height is the clearance height at full blade leading edge, the axial clearance height is the clearance height at full blade trailing edge, and the blades tip clearance height changes equably from the leading edge to the trailing edge along the full blade meridian chord direction. The tip clearance of splitter blade leading edge has the same height as that of corresponding the full blade 30% meridian chord length, and the tip clearance of splitter blade trailing edge has the same height as that of corresponding the full blade trailing edge. In this paper, nine impellers which have different clearance distribution with the 0.2mm, 0.5mm and 1.0mm radial and axial clearance height respectively are analyzed and the clearance height linear distribution along the full blade meridian chord is showed in Fig 4(b).

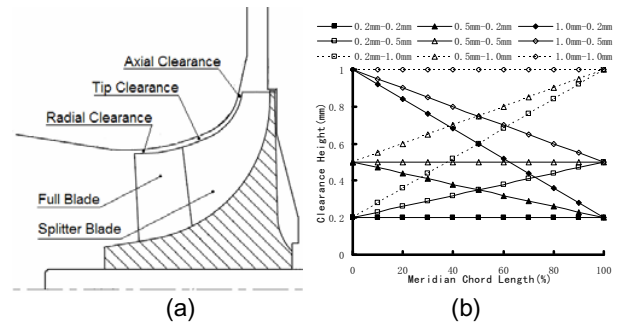


Fig 4 Blades tip clearance distribution

The performance of the nine compressor stages as indicated in Fig 4 with the different clearance distribution impeller is calculated. Each compressor stage includes compressor inlet, impeller with different radial-axial clearance distribution and the same vaneless diffuser. Fig 5 shows the compressor stage performance curve with the impellers under the different blade tip clearance distribution. As indicated in Fig 5, the compressor stage total pressure ratio and efficiency with the minimum uniform clearance distribution (0.2mm-0.2mm) is the best, also the pressure ratio and efficiency will fall down following the height increase of uniform clearance. In order to show different clearance distribution effect on pressure ratio and efficiency, Fig 6 is used to indicate the relative performance curve that nine compressor stages having different clearance distribution compared with the compressor stage having minimum uniform clearance (0.2mm-0.2mm). As showed in Fig 6, total pressure ratio and total efficiency decrease when radial and axial clearance height increasing at the meantime, which is proved in references[1-7]: when clearance distribution increase from [0.2mm-0.2mm] to [0.5mm-0.5mm], the compressor stage pressure ratio drops 0.2, the efficiency drops 1%, and when clearance distribution increase from [0.5mm-0.5mm] to [1.0mm-1.0mm], the compressor stage pressure ratio drops 0.25, the efficiency drops 5% steeply. When radial or axial clearance height is fixed, the performance change law is as following: When the impeller aft axial clearance is fixed and the fore radial clearance height increase from 0.2mm to 0.5mm and from 0.5mm to 1.0mm, pressure ratio drops 0.05~0.15, efficiency drops 1%~3%. When the impeller fore radial clearance is fixed and the aft axial clearance height increase from 0.2mm to 0.5mm and from 0.5mm to 1.0mm, pressure ratio drops 0.05~0.25, efficiency drops 1%~3%. The compressor stage with uniform clearance distribution [0.5mm-0.5mm] is special, the pressure ratio and efficiency is higher than that of non-uniform small clearance, such as [0.2mm-0.5mm] and [0.5mm-0.2mm] in some mass flow condition. It is indicated that the clearance distribution difference leads to different flow field structure, and performance. As indicated in Fig 6, the pressure ratio and efficiency of the [0.2mm-1.0mm] compressor stage is higher than that of [1.0mm-0.2mm] compressor stage obviously. The clearance height at various impeller meridian chord positions leads to different power input ability and flow loss for compressor stage including vaneless diffuser.