Design and Analysis of a Centrifugal Turbocharger Turbine Postprint

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Abstract

Centrifugal turbocharger turbines possess a structure distinct from conventional turbines. This paper investigates a centrifugal turbine characterized by low flow rate, high rotational speed, and small wheel diameter, encompassing one-dimensional aerodynamic design, three-dimensional flow field simulation and optimization, and off-design performance analysis. Initially, simplifications are applied to the working fluid and flow process to obtain one-dimensional aerodynamic design results. Subsequently, computational fluid dynamics software is utilized to conduct numerical simulation of the centrifugal turbine stage, wherein blade profile optimization is performed by adjusting parameters such as blade number, blade thickness, and camber line shape to achieve a more rational flow field. Finally, the off-design performance of the centrifugal turbine under varying rotational speeds and flow rates (back pressures) is examined.

Full Text

Preamble

The Design and Analysis of Centrifugal Turbocharger Turbine

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Abstract: The centrifugal turbine of a turbocharger possesses a structure distinct from conventional turbines. This paper investigates a small-flow-rate, high-speed, small-wheel-diameter centrifugal turbine through one-dimensional aerodynamic design, three-dimensional flow field simulation and optimization, and off-design performance analysis. First, the fluid medium and flow process are simplified to obtain one-dimensional aerodynamic design results. Next, computational fluid dynamics software is employed to conduct numerical simulation of the centrifugal turbine stage, optimizing the blade profile by varying blade count, blade thickness, and mean camber line shape to achieve a more reasonable flow field. Finally, the off-design performance of the centrifugal turbine under different rotational speeds and flow rates (back pressures) is analyzed.

Keywords: centrifugal turbine; turbocharger; numerical simulation; off-design performance

0 Introduction

Turbochargers deliver comprehensive benefits in improving automotive engine power-to-weight ratio, enhancing torque characteristics, increasing fuel economy, reducing engine noise, and lowering exhaust emissions. With China' s increasingly severe energy situation and increasingly stringent emission regulations, turbocharger development has become an inevitable trend. As the prime mover of a turbocharger, the turbine is key to improving turbocharger performance.

Turbocharger development spans over 100 years of history. In 1905, Swiss engineer Alfred Büchi first proposed the concept of exhaust gas turbocharging and obtained patents in Germany and the United States. The world' s first exhaust-driven supercharger emerged in 1912. During the 1920s, ships began to be equipped with turbocharged diesel engines, and during World War II, the United States first applied turbocharging extensively to military aircraft, enabling mass production of exhaust turbochargers. Turbochargers were initially applied primarily to ships, aircraft, and high-power land-based engines before gradually being extended to smaller engines. However, due to significant difficulties in developing small radial turbochargers with reliable structure, good performance, and low cost, the widespread application of turbochargers in automotive engines came much later. It was not until 1961 that General Motors tentatively installed turbochargers on certain vehicle models. China began developing turbochargers from the late 1950s to early 1960s. Following the development of marine diesel engines after liberation, China' s turbocharger research, design, and manufacturing industries developed accordingly. China' s first radial turbocharger was developed and produced through collaboration between the Marine Product Design Office and Shanghai Qiuxin Shipyard, with design completed in 1954 and certification finished in 1958 [1].

Radial turbines can be classified into centripetal and centrifugal types based on working medium flow direction. In centripetal turbines, Coriolis force contributes to the effective work, resulting in large enthalpy drop. Consequently, radial turbines almost exclusively adopt the centripetal configuration, while research, design, and application of centrifugal turbines remain scarce. Research on centrifugal turbines can be traced back to Ljungström [2], who designed a cantilevered dual-inlet counter-rotating centrifugal steam turbine with power ranges from several hundred kW to MW. Italian researchers G. Persico and M. Pini from Politecnico di Milano published several papers detailing preliminary design and aerodynamics of organic Rankine cycle centrifugal turbines [3-5]. In China, Professor Huang Dian-Gui’ s research group proposed a novel dual-inlet centrifugal turbine [6].

When the working medium expands in a centrifugal turbine, its specific volume increases, which matches the diameter of the centrifugal turbine’ s flow passage. The blades can be designed as constant-height straight blades, and the velocity ratio can be designed near the optimal value, making the flow essentially two-dimensional. This paper proposes applying centrifugal turbines to exhaust turbochargers, using ideal gas as the working fluid to investigate the aerodynamic performance and off-design characteristics of centrifugal turbines at high rotational speeds [7], thereby further exploring their application potential.

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1 One-Dimensional Design of Centrifugal Turbine

Referencing the one-dimensional flow analysis process of axial turbines, one-dimensional thermodynamic calculations are performed for the centrifugal turbine. In the initial design stage and thermodynamic calculations, engine exhaust gas is treated as ideal air, with flow through the blade channels assumed to be axisymmetric, adiabatic, inviscid, and steady [8].

A set of actual operating parameters for a turbocharger is used as design parameters, as shown in Table 1 .

Table 1 Initial Design Parameters of Turbocharger Centrifugal Turbine

Parameter	Value
Nozzle inlet total pressure	p
Nozzle inlet total temperature	t
Rotor exit back pressure	p (Pa)
Exhaust gas flow rate	G (kg · s ^{1})
Design rotational speed	n (r · min ^{1})

Among the various efficiencies of centrifugal turbines, the wheel efficiency can be expressed through design parameters; therefore, appropriate design parameters are generally selected based on wheel efficiency [9].

A C language program was written with rotor inlet flow angle , reaction degree Ω, velocity ratio x , and diameter ratio D as loop conditions for calculation. From the results, a parameter set with relatively high wheel efficiency and appropriate wheel diameter was selected to provide a basis for subsequent turbine numerical simulation, as shown in Table 2 . The nozzle velocity coefficient and rotor velocity coefficient were set as = 0.97 and = 0.93, respectively.

Table 2 One-Dimensional Calculation Parameters of Centrifugal Turbine

Parameter	Value
Reaction degree Ω	
Blade height H (mm)	
Rotor exit diameter D (mm)	
Velocity ratio x	
Nozzle inlet diameter D (mm)	
Diameter ratio D	
Nozzle exit diameter D (mm)	
Wheel power N (W)	
Wheel efficiency (%)	
Rotor inlet absolute flow angle (°)	
Rotor exit absolute flow angle (°)	
Rotor inlet relative flow angle (°)	
Rotor exit relative flow angle (°)	
Nozzle exit velocity c (m · s^{1})	
Rotor exit absolute velocity c (m · s^{1})	
Rotor inlet relative velocity w (m · s^{1})	
Rotor exit relative velocity w (m · s^{1})	
Rotor inlet circumferential velocity u (m · s^{1})	
Rotor exit circumferential velocity u (m · s^{1})	

2 Three-Dimensional Simulation of Centrifugal Turbine

Based on the pressures before and after the stator and rotor obtained from one-dimensional design, preliminary analysis and comparison of the flow fields within the stator and rotor were conducted first to determine a stator maximum thickness of 2.5 mm with 10 stator blades, and a rotor maximum thickness of 4 mm with [incomplete text]

2.1 Blade Profile Parameterization

The blade cross-section consists of four curves: leading edge, trailing edge, pressure surface, and suction surface. The leading and trailing edges are circular arcs, with design parameters including blade height, leading/trailing edge radii, maximum blade thickness, blade count, and inlet/outlet geometric angles [10]. The pressure and suction surfaces are constructed using Bezier curves. Eight variable parameters control the two-dimensional blade profile for both stator and rotor: four parameters control the mean camber line shape, and four parameters control the blade thickness distribution along the radial chord length [11]. The two-dimensional blade profile design parameters are shown in Table 3 .

Table 3 Design Parameters of Two-Dimensional Blade Profile

Parameter	Value
Blade height H (mm)	
Leading edge radius R (mm)	
Trailing edge radius R (mm)	
Maximum blade thickness (mm)	
Inlet geometric angle (°)	
Outlet geometric angle (°)	

The figure below shows the stator blade profile construction and selection of four control points; the rotor blade profile construction method is identical.

Figure 1 Meridional channel [Figure 1: see original paper]
Figure 2 Stator blade profile [Figure 2: see original paper]

2.2 Whole-Stage Optimization Design

Among the blade design parameters, blade height, leading/trailing edge radii, blade count, and inlet/outlet geometric angles are fixed control parameters. During optimization, the positions of variable control points are adjusted, causing corresponding changes in various parts of the Bezier curve to obtain a series of optimized airfoils. The mathematical model for this optimization problem is expressed as [formula garbled]. In the equation, design variables [1-4] represent the thickness at control points of stator and rotor blades, M[1-8] refer to the position coordinates of stator and rotor blades along the radial chord length, and A[1-4] represent the mean camber line tangent angles ([1-4] denotes four variables , , , ; M[1-8] and A[1-4] follow the same notation). A total of 16 variables participate in the optimization, with wheel power as the constraint condition and wheel efficiency as the objective function [12].

Based on preliminary analysis of the rotor and stator, the screening method was employed for optimization. Sample parameters were first integrated, sorted, and grouped, and then calculated one by one to finally obtain several sets of qualified parameter points. This approach improves the quality of initial optimization parameters, thereby enhancing optimization accuracy. The main optimized blade profile parameters are shown in Table 4 .

Table 4 Optimization Results of Control Point Parameters

Blade Type	Control Point

Numerical simulation was used to compare and analyze the internal aerodynamic parameters of the impeller before and after optimization. Results show that after optimization, the rotor exit flow angle becomes more radial, residual velocity loss decreases, wheel power increases by 2.95%, and wheel efficiency increases by 1.8 percentage points. The velocity parameters of the optimized airfoil approach the one-dimensional design values. Due to its small size, its velocity coefficient is relatively small, and the wheel efficiency obtained from numerical simulation is lower than the thermodynamic design value.

Table 5 Comparison of Impeller Parameters Before and After Optimization

Parameter	Before	After
N (W)		
(%)		

Table 6 Speed Triangle of Optimized Blade Profile

Parameter	Value
c (m · s 1)	
c (m · s 1)	
w (m · s 1)	
w (m · s 1)	
u (m · s 1)	
u (m · s 1)	

Figures 5-8 present the eddy viscosity distribution, Mach number distribution, pressure, and velocity contours at 50% blade span.

Figure 5 Eddy viscosity distribution at 50% span [Figure 5: see original paper]
Figure 6 Mach number distribution at 50% span [Figure 6: see original paper]
Figure 7 Pressure distribution at 50% span [Figure 7: see original paper]
Figure 8 Velocity distribution at 50% span [Figure 8: see original paper]

3 Off-Design Performance Study

Turbochargers used in vehicle, agricultural machinery, marine, and construction machinery engines typically operate under variable speed and load conditions, causing the exhaust gas flow rate, pressure, and temperature entering the turbine to change accordingly. Consequently, the turbine operating condition naturally deviates from the design condition. Therefore, off-design performance must be studied in the design of turbocharger centrifugal turbines.

3.1 Off-Design Performance at Design Speed

At design rotational speed, with constant inlet total temperature and pressure, off-design simulation was conducted by varying the exit flow rate to obtain the flow characteristic curve and efficiency characteristic curve of the centrifugal turbine, as shown in Figures 9 and 10. Results indicate that as the pressure ratio p /p decreases, flow rate gradually increases with a slowing trend. When the pressure ratio decreases to a certain value, turbine flow becomes constant and the turbine reaches choking condition. CFX simulation shows that the critical pressure ratio of the centrifugal turbine at design condition is 0.519, with a maximum critical flow rate of 0.0262 kg/s. Wheel efficiency first increases slowly and then decreases rapidly with pressure ratio, reaching maximum at a pressure ratio of 0.921 (the one-dimensional design pressure ratio is 0.932).

Figure 9 Flow characteristic curve at design speed (n = 50000 rpm) [Figure 9: see original paper]
Figure 10 Efficiency characteristic curve at design speed (n = 50000 rpm) [Figure 10: see original paper]

Figure 11 plots the relationship between wheel efficiency and inlet total temperature T at design speed based on the results provided in Table 7 . Results show that inlet total temperature has minimal effect on wheel efficiency. Within material limits, centrifugal turbines can operate at high efficiency across a wide temperature range.

Table 7 Variation of Wheel Efficiency with Inlet Total Temperature

Inlet Total Temperature T	Wheel Efficiency (%)

Figure 11 Variation of wheel efficiency with inlet total temperature T at design speed [Figure 11: see original paper]

3.2 Off-Design Performance at Variable Speeds

The relationship between wheel efficiency and flow rate of the centrifugal turbine stage at different rotational speeds is plotted in Figure 12 [Figure 12: see original paper].

Figure 12 Variation of wheel efficiency with flow rate at different rotational speeds [Figure 12: see original paper]

Results demonstrate that different rotational speeds correspond to different flow rates for maximum efficiency points—higher rotational speeds require larger flow rates. Under variable speed conditions, centrifugal turbines can operate over a wide flow range, demonstrating strong adaptability. Higher rotational speeds increase the minimum operational flow rate, while the maximum critical flow rate changes minimally. Under variable speed conditions, the flow range where wheel efficiency exceeds 70% is relatively large, indicating good off-design performance.

Conclusions

This paper primarily focuses on the optimal design and analysis of a centrifugal turbocharger turbine and its off-design performance. The main conclusions are as follows:

1. Using wheel efficiency as the objective function and blade thickness distribution and mean camber line tangent angles as parameters, airfoil optimization was conducted to obtain a centrifugal turbine suitable for turbocharger applications, achieving a stage efficiency of 80.1%.

2. Off-design performance analysis of this centrifugal turbine stage reveals that at design rotational speed, as the pressure ratio p /p decreases, flow rate gradually increases with a slowing trend. When the pressure ratio decreases to a certain value, turbine flow becomes constant and the turbine reaches choking condition. Wheel efficiency first increases slowly and then decreases rapidly with pressure ratio. Inlet total temperature has minimal effect on wheel efficiency; theoretically, centrifugal turbines can operate at high efficiency across a wide temperature range. Under variable speed operation, maximum wheel efficiency corresponds to different flow rates at different speeds—higher rotational speeds require larger flow rates for maximum efficiency. When rotational speed varies between 40,000-70,000 rpm, the centrifugal turbine can operate over a wide flow range, demonstrating strong adaptability, and the flow range where wheel efficiency exceeds 70% is relatively large. These results indicate that the off-design performance of this type of centrifugal turbine is comparable to that of conventional axial and radial turbines.

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Note: Figure translations are in progress. See original paper for figures.