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ORIGINAL ARTICLE

Design of a high-performance centrifugal Q1 compressor with new surge margin improvement Q2 technique for high speed turbomachinery

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Received 17 February 2017; accepted 14 December 2017

KEYWORDS

Centrifugal compressor;
Aerodynamic performance;
Surge margin;
Blade angles;
Stress analysis;
Computational fluid dynamics

Abstract This paper presents the design of a centrifugal compressor for high-speed turbomachinery. The main focus of the research is to develop a centrifugal compressor with improved aerodynamic performance. As a meridional frame has a significant effect on overall performance of the compressor, special attention has been paid to the end wall contours. The shroud profile is design with bezier curve and hub profile with circular arc contour. The blade angle distribution has been arranged in a manner that it merges with single value at impeller exit. The rake angle is positive at leading edge and negative at trailing edge with identical magnitude. Furthermore, three-dimensional straight line element approach has been used for this design for better manufacturability. The verification of the aerodynamic performance has been carried out using CFD software with consideration of a single blade passage and vaneless diffuser. The result has been compared with matching size aftermarket compressor stage gas stand data. The compressor stage with Trim 55 provides 34% increase in choke flow at 210000 RPM as compared to gas stand data with 87% peak stage efficiency at 110000 RPM. In addition, new surge margin improvement technique has been proposed by means of diffuser enhancement. This technique provides an average of 16% improvement in surge margin compared to standard diffuser stage with 55 trim compressor impeller. The mechanical integrity has been validated at maximum RPM with the aluminum alloy 2014-T6 as a fabrication material.

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Peer review under responsibility of National Laboratory for Aeronautics and Astronautics, China.

https://doi.org/10.1016/j.jppr.2018.02.004

Nomenclature

I	total work factor/work input coefficient
H	enthalpy (unit: J)
U	blade tip speed (unit: m/s)
B	fractional aerodynamics area blockage, hub to shroud passage width (unit: mm)
$\overline{\varpi}$	total pressure loss coefficient
Pv	velocity pressure, P_0-P (unit: Pa)
Z	number of blades
P	pressure (unit: Pa)
d	diameter (unit: mm)
L	length of mean camber line (unit: mm)
η	efficiency
C_p	static pressure recovery coefficient

Greek letters

α	angle between streamline slop with axial direction
λ	impeller tip distortion factor
σ	slip factor
δ	clearance gap parameter

Subscripts

B	blade
c	compressor
D	diffuser
SF	skin friction
v	velocity
H	hydraulic
CL	clearance
1	impeller blade inlet condition
2	impeller blade outlet condition
3	diffuser outlet condition

1. Introduction

The development of high-performance centrifugal compressor for the application of high-speed turbomachinery such as turbochargers and micro gas turbines needs to be addressed urgently due to the fact that it offers enormous opportunities to control the exhaust emission in automobile engines and makes it more environmentally friendly. Specifically, the development of high-performance turbochargers for automobile engines are attracting more attention since it has a direct impact on the engine power, specific fuel consumption and exhaust emission level. As global emission authorities are issuing strict guidelines for exhaust emission levels, it is necessary for the automotive industry to alleviate an emission rate to comply with the global emission standard. This could be achieved by incorporating turbochargers into automobile engines, which essentially increases the mean effective pressure (the ratio of torque to the volume of the engine) and enhance engine performance. In addition to that, this target can be more efficiently achieved by designing a centrifugal compressor for turbochargers with enhanced performance and additional distinct qualities as lower wheel inertia to promote a rapid acceleration and better transient response of compressor during the course of engine operation [1]. Moreover, as an application of centrifugal compressor is found in various domains such as aerospace, automotive industry, oil and gas industry, and power generation, the demand for improving performance is enormous, particularly in terms of compressor efficiency and mass flow range [2].

Interestingly, development in a centrifugal compressor is more appealing to turbocharger industry due to the fact that existing compressor wheels are unable to meet the required performance target in terms of efficiency and flow range. Currently, passenger vehicle turbochargers provide 75% to 78% peak stage efficiency with wheel peak efficiency around 82% to 84%, considering the fact that most of the total pressure loss takes place in diffuser and volute and significantly affect overall stage efficiency. Therefore, in this scenario, increase in stage efficiency and compressor map flow range is highly demanded. Nevertheless, development of a diesel engine is rapidly being done to meet the emission standard; therefore, compatible turbocharger with better flow range and efficiency is essentially anticipated. This fact promotes the development of improved performance compressor wheel with lower inertia which subsequently results in better transient response.

The proposed centrifugal compressor design is targeted for turbochargers applicable to 1.6 - 2 liter diesel engines with an intent of providing a required boost pressure, wide flow range, and improved peak efficiency. The design has been accomplished by firstly designing the meridional frame and subsequently converting it into a three-dimensional design with three-dimensional straight line elements. An approach proposed by Aungier [3] has been implemented for preliminary design. Through this approach distribution of hub and shroud blade angles has been devised. However, special attention has been given for hub blade angle since it has significant effect on wheel efficiency [4]. Likewise, impeller key dimensions such as an exducer b-width, an inducer diameter, and flow angles have been established through correlation proposed by Aungier [3]. Meanwhile, some of the design constraints have been imposed in terms of fixed values in order to take account of compactness of impeller.

Apart from that, greater challenge lies in concluding on compressor exducer maximum tip speed and back sweep angle due to structural limitations. In order to achieve the required pressure ratio, back sweep angle needs to be minimized which penalize the flow range. Work input

Please cite this article as: Sagar Pakle, Kyle Jiang, Design of a high-performance centrifugal compressor with new surge margin improvement technique for high speed turbomachinery, Propulsion and Power Research (2018), https://doi.org/10.1016/j.jppr.2018.02.004