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Optimal Vaneless Diffuser Design For A High-End Centrifugal Compressor

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ABSTRACT: Turbochargers are widely used in the automotive industry to reduce engine emissions and to increase the power. Centrifugal compressors are an integral part of turbochargers. Centrifugal compressors comprises primarily of inducer, impeller, diffuser and volute. The diffuser has an important role in the isentropic efficiency of the compressor. Over the past few decades, researchers have been trying to increase the total-to-total compressor stage efficiency by altering the diffuser's geometries. Many different methods have been adopted for this purpose, like pinching the diffuser walls, tilting the diffuser walls etc. Pinching increases the outer width of the diffuser while tilting provides an increased radial length. In the present study, both these methods have been used simultaneously. The primary benefit of doing so is to make the turbocharger compressor stage more compact in design, which is the current requirement of the automotive market. In order to investigate the effect of pinching and tilting of diffuser walls, a Computational Fluid Dynamics based solver has been used to predict the flow phenomena within the compressor, especially in the vaneless diffuser. Design of Experiments, using Taguchi's method, has been incorporated in this study to statistically define the scope of the numerical work, and to obtain the optimal configuration of pinching and tilting that leads to maximum total-to-total compressor stage efficiency. The results depict that the compressor stage efficiency increases up to a tilt angle of 6.25º, after which it starts to decrease. Furthermore, the stage efficiency increases with increase of diffuser outlet width i.e., pinching the diffuser passage, however, this increasing trend has been observed up to an outlet width ratio of 1.23, after which the stage efficiency starts to decrease. Hence, the optimal diffuser model, based on the combined tilting and pinching results obtained, which leads to the maximum total-to-total compressor stage efficiency, has been identified and analysed.

Keywords: Computational Fluid Dynamics (CFD), Centrifugal Compressor, Turbocharger, Diffuser, Design of Experiments (DoE).

1. Introduction

Automotive industries require high performance and power from small size engines, and similarly, legislation demands low emissions from combustion engines. (1) To achieve this, most particularly in diesel engines, turbochargers are the best solution that can provide increased air density in the engine cylinder, which means more fuel is burnt in the combustion engine. Turbochargers conventionally have a turbine side and a compressor side. Centrifugal compressors are used in turbochargers, which comprises of the impeller, diffuser, map width enhancement cavity and the volute. The diffuser is one of the most important components where the static pressure is partially recovered by converting the kinetic energy of the flow coming out of the impeller into static pressure. Henceforth, this leads to enhanced total-to-total efficiency of the system. High-end pressure ratio is required from the turbocharger compressor in order to provide increased air density to the combustion chamber. However, high pressure ratio leads to reduction in the stable operating range of turbocharger compressors. Various techniques have been employed to maintain the operating range of the compressor as the pressure ratio increases. (2) Various studies regarding diffusers have been carried out by Japikse, where the performance of a centrifugal compressor stage has been optimised by employing divergence to the diffuser sidewalls. (3) Adachi et al. studied the optimisation methods for annular vaneless diffusers in order to improve the overall stage performance. Improvement in both the surge margin and the efficiency at high flow

rates has been achieved by adding a tapered passage at the vaneless diffuser outlet section. [4] Jaatinen carried out investigations analysing different vaneless diffuser designs where pinching has been introduced at both the shroud and the hub walls, where pinching has been used to decrease the diffuser outlet width. Significant improvement in stage efficiency over a wide range of operating conditions has been observed at all rotational speeds. Furthermore, it has been stated that the pinch on the diffuser walls assists in reducing the secondary flow losses caused by the tip clearance flow from the impeller. Pinch accelerates and pushes the flow towards the centre of the diffuser, thereby decreasing the size of the boundary layer, which leads to reduction in pressure losses in the diffuser. Moreover, it has been noticed that by reducing the diffuser width, the flow becomes more radial at the impeller outlet, thus the margin to the critical flow angle (stall) is greater.
[5] Jaatinen and Gronmen carried out investigations on the effects of tip clearance on the pressure distribution within the diffuser. It has been noticed that the circumferential pressure distribution is minimum near the volute tongue area at low mass flow rates, whereas it is higher in other regions. Furthermore, as far as radial pressure distribution is concerned, the rate of pressure rise is at its highest at the start of the diffuser, and decreases downstream the diffuser. It has been concluded that the reduction in diffuser width has a minor effect on the radial pressure distribution. Furthermore, higher tip clearance has a significant effect on the pressure rise. The rate of pressure rise is lower, when the tip clearance is higher, and the rate of pressure rise is higher throughout the diffuser when the tip clearance is lower.
[6] Mykola and Oleg investigated the flow phenomena in a vaneless diffuser of a centrifugal compressor stage experimentally, numerically and analytically. In the analytical investigations, the time-averaged boundary layer parameters have been considered. Furthermore, two boundary regions have been used; one with laminar flow and the other with turbulent flow. It has been found from the numerical analysis that there is an average difference of 17.3% and 14.5% in total pressure loss coefficient in predicted and measured results. Moreover, the average difference between the measured and predicted static pressure recovery coefficients is 2.3% and 4.7%. The investigations show that at low mass flow rates, the pressure losses are caused by the flow separation close to the diffuser hub wall. It is due to the higher frictional losses and jet wake mixing. Similarly, at high flow rates, the pressure losses are only caused by the jet wake mixing.

In the present study, the effect of increased diffuser outlet width and the tilt angle on the total-to-total efficiency of a compressor stage has been critically analysed. Pinch has been implemented at different locations on the hub wall of the diffuser. Furthermore, the diffuser passage has been tilted towards the shroud wall of the diffuser passage. Pinch has been used to increase the width of the diffuser outlet. The increased diffuser width helps to increase the surge margin, whereas tilting makes the compressor stage more compact. Therefore, this study offers new insight about centrifugal compressors with varying diffuser outlet widths and tilt angles. Design of Experiments (DoE), using Taguchi's methodology, has been employed in the present study in order to reduce the number of numerical simulations performed by selecting specific designs with the help of orthogonal arrays. Signal-to-Noise ratio (S/N) has been used, which enables to obtain the best diffuser design with optimum pinch and tilt angle. Total-to-total stage efficiency has been used as the prime factor to carry out S/N ratio analysis, where total-to-total stage efficeincy is a function of both the pressure ratio and the temperature ratio. Total-to-total stage efficiency can be computed as:

$$\eta_C = \frac{\left[PR^{\frac{k-1}{k}} - 1\right]}{\left[\frac{T_{o.out}}{T_{o.in}} - 1\right]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

where PR is the pressure ratio between inlet and outlet of the compressor stage, k is ratio of specific heats, $T_{o,out}$ is total temperature at the outlet and $T_{o,in}$ is total temperature at the inlet of the compressor.

2. Numerical Setup

This section provides the details of the numerical modeling that has been used in the present study. The numerical setup is categorised into two sub-sections i.e. (i) creation of geometry, and (ii) mesh generation with solver settings.

2.1. Geometry of the Compressor Model

The baseline compressor stage geometry is shown in figure 1. It can be seen that the model comprises of various sections, such as inlet duct, map width enhancement cavity (MWE), impeller wheel, diffuser and outlet duct. The MWE duct consists of two bleed slots, which assist in extreme conditions such as surge and choke. The impeller consists of seven full and seven splitter backswept blades. The backswept blades are used to derive the high speed flow radially out to the diffuser. The tips of the impeller blades are enclosed in a shroud with a very small gap in order to prevent tip leakage losses. The schematic of the baseline diffuser is shown in figure 2(a). As the focus of this study is on the diffuser of the compressor stage, it can be seen in the figure that the diffuser consists of two parallel walls, where these walls are termed as the hub and the shroud walls.

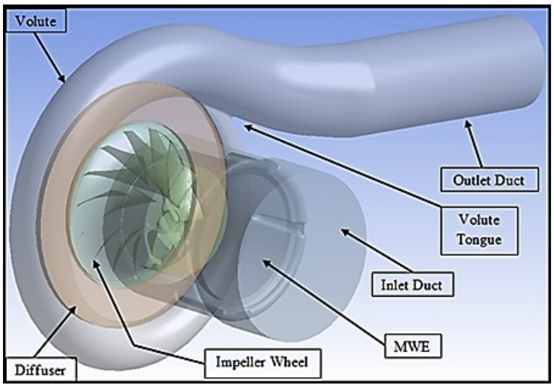


Figure 1. Geometry of the Centrifugal Compressor

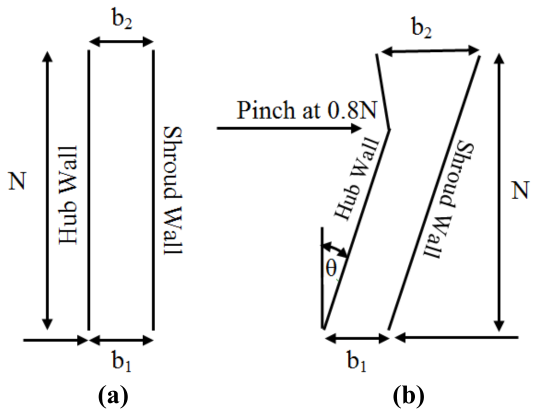


Figure 2. Schematic of the (a) Baseline diffuser, and (b) Pinched diffuser

Further to the baseline compressor model, various other models have also been created by altering the diffuser geometry. In order to carry out DoE studies, some of the geometric parameters of the diffuser have been considered variable, such as the outlet-to-inlet width ratio (b_2/b_1), the pinch location (N) and the tilt angle of the diffuser passage (θ). The schematic shown in figure 2(b) depicts these variables. The diffuser passage has been tilted to make the compressor stage more compact.

Table 1. Diffuser passage design configurations

Diffuser Designs	b_2/b_1	θ (°)	Pinching Point
Baseline	1.00	0.00	N/A
D1	1.08	0.00	Diffuser Inlet
		6.25	0.2N
		12.50	0.4N
		18.75	0.6N
		25.00	0.8N
D2	1.15	0.00	Diffuser Inlet
		6.25	0.2N
		12.50	0.4N
		18.75	0.6N
		25.00	0.8N
D3	1.23	0.00	Diffuser Inlet
		6.25	0.2N
		12.50	0.4N
		18.75	0.6N
		25.00	0.8N
D4	1.31	0.00	Diffuser Inlet
		6.25	0.2N
		12.50	0.4N
		18.75	0.6N
		25.00	0.8N
D5	1.38	0.00	Diffuser Inlet
		6.25	0.2N
		12.50	0.4N
		18.75	0.6N
		25.00	0.8N

Taguchi's approach, based on a statistical Design of Experiments, has been incorporated in this study in order to identify appropriate combinations of the aforementioned variables. These combinations have been summarised in table 1. The geometry for the different diffuser models have been altered by impelemeting pinch on the diffuser hub wall at diffuser inlet (0N), 0.2N, 0.4N, 0.6N and 0.8N from the inlet of the diffuser passage. Similarly, b_2/b_1 ratio has been increased to 1.08, 1.15, 1.23, 1.31 and 1.38 in the present study. Furthermore, the diffuser passage has been tilted by 0°, 6.25°, 12.5°, 18.75° and 25° towards shroud wall as depicted in figure 2(b).

2.2. Mesh Generation and Solver Settings

Prism layers have been employed in the present study as they are capable of accurately predicting the complex flow phenomena occuring wihtin the compressor stage. These prism layers consist of hexahedral elements, and have been concentrated in near wall regions in order to accuratley model the boundary layer formation. The compressor stage comprises of 7.7 million elements.

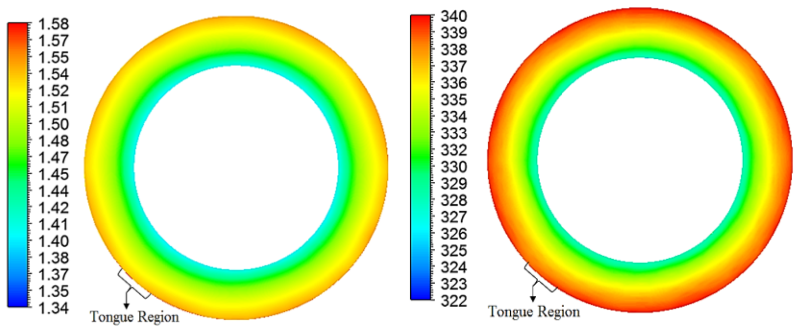
Three-dimensional Navier Stokes equations, alongwith the continuity and energy conservation equations, have been numerically solved, in an iterative manner, for the turbulent flow of air through the compressor stage. The rotation of the impeller wheels has been modelled using a well-established methodology, known as Multiple Reference Frame (MRF) modelling. It is a steady-state approximation in which individual cell zones move at different rotational speeds. At the interfaces between cell zones, a local reference frame transformation is performed to enable flow variables in one zone to be used to calculate fluxes at the boundary of the adjacent zone. Turbulence in the air has been modelled using Shear Stress Transport (SST) k-ω turbuelnce model, which is a two equation turbulence model which solves for the turbulent kinetic energy and the turbulent dissipation rates. As it is a combination of standard k-ω and k-ε algorithms, it behaves more accurately in predicting flow parameters than any of these in flow domains comprising of extreme adverse pressure gradients and boundary layer separation. (7) The development of the numerical modelling approach enables to explore the flow phenomena in various types of turbomachines. A commercial CFD package, CFX, has been used in the present study to carry out the simulations of the compressor stage. This numerical investigation is focused on a high-end compressor, since the impeller has a dimensionless rotational speed of 58.9. Additionally, this investigation has been carried out on the best efficiency point. This study investigates the best pinch distance, best tilt angle and best outlet diffuser width, whereby the maximum total-to-total efficiency of the compressor stage is achieved.

3. Results and Discussions

This section provides qualitative and quantitative details of the complex flow phenomenon within the turbocharger compressor stage, primarily focusing on the vaneless diffuser for the various geometrical configurations considered in the present study.

3.1. Baseline Model

Figure 3 depicts variations in static gauge pressure and static temperature within the diffuser passage of the baseline compressor model. For both these flow parameters, it can be seen that the values of these parameters are lower at the inlet of the baseline diffuser, whereas both these parameters increase significantly as the flow propagates towards the outlet of the diffuser passage. The static gauge pressure increases wihtin the diffuser passage as the flow velocity reduces, reducing the dynamic pressure of the flow. It can be seen that the average static gauge pressure increases from 1.39atm to 1.58atm from the inlet to the outlet section of the diffuser. Similarly, the average static temperature of the flow increases from 323K to 340K from the inlet to the outlet section of the baseline diffuser passage.



(a) (b)

Figure 3. Flow parameters variations within the baseline diffuser passage (a) Static gauge pressure (atm) (b) Static temperature (K)

3.2. Diffuser Models of the Compressor Stage

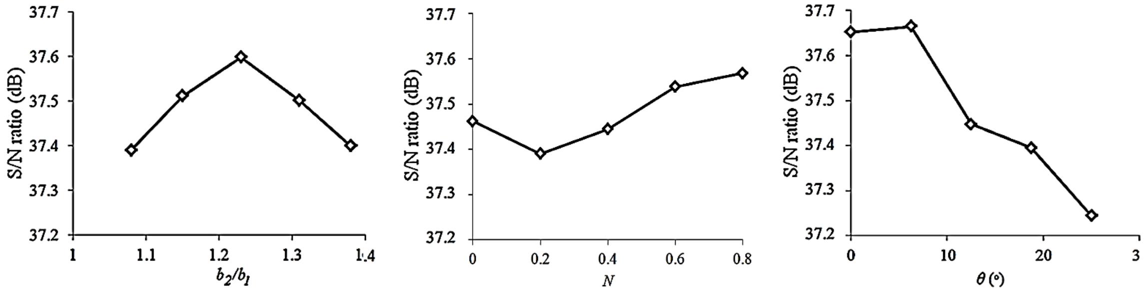
As mentioned earlier, the main focus of this research is to make the turbocharger compressor stage compact by tilting the diffuser passage, and at the same time increasing the total-to-total compressor stage efficiency by increasing the outlet width of the diffuser passage. Therefore, tilted diffusers with pinched hub walls have been investigated using Taguchi's method, based on a statistical Design of Experiment method. Total-to-total efficiency is considerably affected by the design of the diffuser passage. Improper arrangement of the diffuser channel causes reduction in the total-to-total efficiency and pressure ratio of the turbocharger compressor stage. In this study, optimal combination of diffuser's geometric design parameters, determined by Taguchi's method, have been identified. This has been acheieved by following three significant steps. Initially, Signal-to-Noise (S/N) ratio is determined in terms of quality characteristics. Subsequently, S/N ratio is used to evaluate the quality characteristics of different combinations of parameters in order to predict the optimal combination of parameters. Finally, the verification of numerical simulations is conducted to show that the predicted combinations achieve the quality characteristics. Bigger-the-better approach has been used to obtain the highest total-to-total compressor stage efficiency. The S/N ratio for bigger-the-better approach is defined as:

$$\mathrm{S/N} = -10\log\left(\frac{1}{N}\sum_{i=1}^{N}\left(\frac{1}{y_i^2}\right)\right) \quad \text{.............................. (2)}$$

where N is the frequency of the experiment in each group and y_i is the value of the *i*th experiment in each group. Investigations have been carried out using the above numerical setup on all the compressor models, where the results of Taguchi's method are based on the S/N ratio. The S/N ratio has been obtained from the achieved total-to-total compressor stage efficiency, as shown in figure 4, where increases in S/N ratio means increase in total-to-total compressor stage efficiency. It can be seen from the figure that the S/N ratio increases with increase in b_2/b_1 ratio until b_2/b_1=1.23, after which the S/N ratio starts to decrease. This is an indication that b_2/b_1=1.23 is the optimal diffuser outlet-to-inlet width ratio, corresponding to maximum total-to-total efficiency.

The S/N ratio decreases with an increase in pinching location from the inlet of the diffuser, upto N=0.2, after which the S/N ratio starts to increase. Hence, more the distance of the pinching location from the inlet of the diffuser, higher the total-to-total efficiency of the compressor stage. Furthermore, increase in tilt angle of the diffuser passage increases the S/N ratio upto θ=6.25°, after which it starts to decrease. Hence, the optimum tilt angle of the diffuser passage is 6.25° that corresponds to the maximum total-to-total efficiency of the compressor stage. It can be concluded from this analysis that the optimal diffuser design, based on the total-to-total compressor stage efficiency, corresponds to b_2/b_1=1.23, N=0.8 and θ=6.25°.

Based on the statistical analysis carried out, new compressor stage model, with optimised geometric configuration, has been modeled and investigated, and the results presented in the next section.

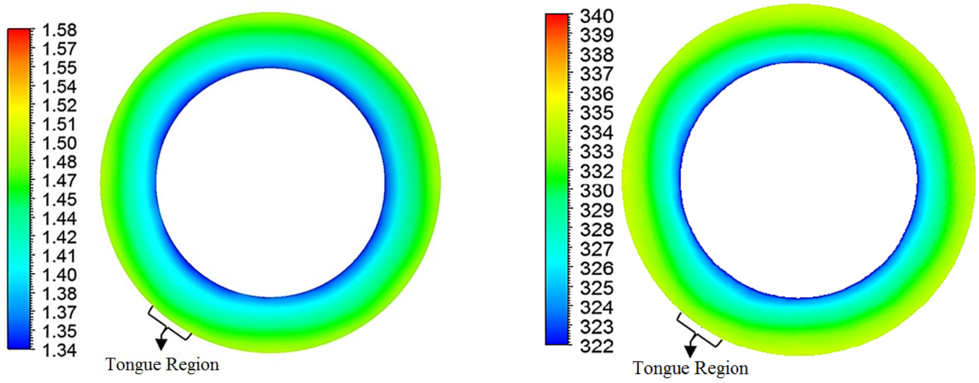


(a) (b) (c)

Figure 4. Signal-to-noise ratios w.r.t. (a) Outlet-to-inlet width ratio (b) Pinching location (c) Tilt angle

3.3. Optimised Diffuser Model

Figure 5(a) depicts the static gauge pressure variations across the diffuser passage for the optimised diffuser design. The scale of the contour has been kept the same as in case of baseline diffuser, for effective comparison purposes. It can be seen from the figure that the static gauge pressure throughout the diffuser region is significantly lower as compared to the baseline diffuser in figure 3(a). However, the trends observed remains the same i.e. the static gauge pressure is lower at the inlet of the optimised diffuser, whereas it increases significantly as the flow propagates towards the outlet of the diffuser passage. The average static gauge pressure increases from 1.34atm to 1.50atm from the inlet to the outlet section of the optimised diffuser model.



(a) (b)

Figure 5. Flow parameters variations within the optimised diffuser design (a) Static gauge pressure (atm) (b) Static temperature (K)

The work done by the impeller is responsible for the energy transfer, which affects the thermodynamic properties of the operating fluid. Figure 5(b) depicts the variations in the static temperature across the diffuser passage for the optimised model. The scale is again the same as in case of baseline model, and the trends observed are also the same i.e. the temperature across the diffuser increases significantly from the inlet to the outlet sections of the diffuser passage. The average static temperature increases from 322K to 335K from the inlet to the outlet section of the optimised diffuser model. Furthermore, it has been noticed that although the pressure ratio of the optimised model is 1.24% lower than the baseline model, the total-to-total stage efficiency is 6.62% higher. This is because the temperature difference between the inlet and the outlet of the optimized diffuser passage is 23.5% lower than the baseline diffuser model.

4. Conclusions

Detailed numerical investigations have been carried out on the effects of pinching and tilting the diffuser passage on the total-to-total efficiency of a turbocharger compressor stage. Various geometric configurations of a diffuser have been created and critically analysed by comparing them with the baseline diffuser configuration. Taguchi's design of experiments method has been employed, using bigger-the-better approach, to identify the optimised diffuser configuration that corresponds to the maximum total-to-total compressor stage efficiency. Local flow field analysis has been carried out on the optimised diffuser model in order to understand the underlying flow phenomena occuring within the diffuser passage of the compressor.

The results obtained indicate that as the outlet-to-inlet width ratio of the diffuser passages increases, the total-to-total efficiency of the compressor stage increases upto a certain value of width ratio, after which it starts to decrease. The overall trends observed for increasing tilt angle and the pinching location from the inlet of the diffuser passage indicate increased and decreased total-to-total stage efficiency respectively. The qualitative comparison of the static gauge pressure and static temperature distribution within the optimised and the baseline diffuser passages reveals that the pressure ratio of the optimised model is 1.24% lower than the baseline model, however, the total-to-total stage efficiency is 6.62% higher, because the temperature difference between the inlet and the outlet of the optimized diffuser passage is 23.5% lower than the baseline diffuser model.

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