A numerical study of automotive turbocharger mixed flow turbine inlet geometry for off design performance

T Leonard\(^1\), S Spence\(^1\), J Early\(^1\) and D Filsinger\(^2\)

\(^1\)Queens University, Belfast BT9 5AH, UK
\(^2\)IHI Charging Systems International GmbH, Heidelberg 69126, Germany

E-mail: tleonard06@qub.ac.uk

**Abstract.** Mixed flow turbines represent a potential solution to the increasing requirement for high pressure, low velocity ratio operation in turbocharger applications. While literature exists for the use of these turbines at such operating conditions, there is a lack of detailed design guidance for defining the basic geometry of the turbine, in particular, the cone angle – the angle at which the inlet of the mixed flow turbine is inclined to the axis. This investigates the effect and interaction of such mixed flow turbine design parameters. Computational Fluids Dynamics was initially used to investigate the performance of a modern radial turbine to create a baseline for subsequent mixed flow designs. Existing experimental data was used to validate this model. Using the CFD model, a number of mixed flow turbine designs were investigated. These included studies varying the cone angle and the associated inlet blade angle. The results of this analysis provide insight into the performance of a mixed flow turbine with respect to cone and inlet blade angle.

1. **Introduction**

Reducing engine mass is an effective means of improving the fuel economy of a vehicle; any reduction in engine mass can, in turn, lead to a reduction in chassis, drivetrain and suspension masses. The downside of this engine downsizing is an increased reliance on forced induction to produce adequate engine power. This means that inlet boost is required not just at peak torque engine speed but also at lower engine speeds. This can be difficult, as little mass flow is available at low engine speeds to extract turbine work and high efficiencies are therefore required. Typically, however, this is where turbine efficiencies are lowest and the flow at the inlet of the turbine can have a high degree of swirl, that is that the flow is highly tangential.

To operate efficiently in this highly tangential flow, an inlet blade angle, $\beta$, is required to achieve optimum incidence angles that are typically quoted as values between $-10^\circ$ and $-40^\circ$ [1]. For a radial inlet turbine, however, achieving this non-zero inlet blade angle requires interrupting the radial fibres of the blade[2], as illustrated in Figure 1. Interrupting this radial fibre induces additional bending stresses within the blade that can pose significant structural issues in comparison to the purely tensile stresses encountered in a radially fibred blade. With the given trends for increasing rotation speed and turbine inlet temperature, this increase in stress is not desirable and may require a redesign of the rotor to the detriment of aerodynamic performance. Therefore, interrupting this radial fibre is not desirable and it is difficult for radial turbines to operate efficiently with strong positive incidence that results from highly tangential flows.
Figure 1. Comparison of uninterrupted (left) and interrupted (right) radial fibres.

Given the wide range of operation required, from low to high engine speeds, it is also necessary that the turbocharger demonstrate good transient performance. This transient performance is closely related to the acceleration of the rotating mass, which is heavily governed by the inertia of turbine rotor, which can account for up to 80% of the total rotating mass. If the turbine inertia is relatively high then this, together with the efficiency and mass flow issues at low engine speeds, can result in a perceived ‘turbo-lag’ in a heavily boosted engine. Therefore, any reduction in turbine rotor inertia will be beneficial to the turbocharger’s transient performance and, hence, the drivability of the engine [3]. For transient performance there is effectively a trade-off between turbine inertia and efficiency since some reduction in efficiency could be tolerated if it compensated by a sufficient reduction in rotor inertia that produces and overall better transient response.

One suggested solution to these issues is a mixed flow turbine [4-5]. A mixed flow turbine differs from a radial turbine as its inlet is not at a constant radius but rather is inclined at an angle, commonly referred to as the cone angle, as shown in Figure 2. The most immediate benefit of this is a reduction in the mass of the rotors back disk. As inertia is a function of radius squared, Equation 1, this reduction of mass at the outer radii of the rotor leads to a significant reduction in rotor inertia.

\[ I_m = m \cdot r^2 \]  

(1)

Figure 2. Comparison of radial (left) and mixed flow (right) turbine rotors.
The second benefit of a mixed flow turbine is the introduction of an axial flow component at the inlet to the turbine rotor. The advantage of this axial flow component is that it sees the camber angle of the blade, the angle of blade to the rotational axis. For a radially fibred blade, the incoming flow therefore sees an inlet blade angle that is a function of both the camber angle and the cone angle as illustrated in Figures 3 and 4 and Equation 2. In this way, a mixed flow turbine can achieve a non-zero inlet blade angle without interrupting the blade’s radial fibre and inducing structural issues.

\[
\tan \beta = \frac{A}{B}, \quad \cos \lambda = \frac{C}{B}, \quad \tan \Phi = \frac{A}{C} \\
\Rightarrow \tan \beta = \tan \Phi \cdot \cos \lambda
\]  

(2)

The definition of cone angle, given the symbol \( \lambda \), describes the angle of a cone tangential to a point on a stream line as shown in Figure 5 [6]. From this description, cone angle is something that can be defined for all points on a rotor blade and is a core parameter of calculating the shape of 3-dimensional blades. This is the correct value of \( \lambda \) when calculating blade angle from Equation 2. The problem exists, however, that this value of cone angle may vary along the leading edge from hub to shroud, depending on the direction of the flow streamlines. It is also possible, and intuitive, to define a value specifying the angle of the leading edge of the blade in the meridional plane, as shown in Figure 5 by the symbol \( \Lambda \). To distinguish these two values by name, the angle represented by \( \lambda \) is referred to as the flow cone angle and the angle represented by \( \Lambda \) is referred to as the blade cone angle in this paper. Often, the two angles can be similar but there are many cases where they may differ considerably, and confusion of the two angles may also lead to misconceptions of the operation of a mixed flow turbine. For example, any value of blade cone angle, \( \Lambda \), in radial duct, with purely radial streamlines, will not produce any inlet blade angle as the flow cone angle, \( \lambda \), will be 90°. In this study, the angle \( \Lambda \) has been used to intuitively name the turbines rotors studied, with a small angle representing a design close to a radial turbine and a large angle representing a more axial design.
2. Methods

The computation model used has been progressively developed and used extensively in previous projects and has shown good correlation in many different cases, with many validations against experimental results completed to-date. The model was comprised of multiple domains representing the major components of the test rig. A set of preswirl vanes serve to produce swirl, similar to the function of a scroll on a turbocharger turbine, without producing the non-axisymmetric flow features associated with a normal scroll. This allowed a single passage computational model to be used that was directly comparable to the experimental setup, greatly reducing the computational effort required.

The analysis was performed using steady-state Reynolds-Averaged Navier-Stokes (RANS) with the Shear Stress Transport (SST) turbulence model within the ANSYS commercial CFD system. Y+ values have been successfully controlled within the limits of the model (<11) throughout all domains, with the vast majority of boundary layers resolved to more than 3 cells. Boundary conditions of total inlet temperature and pressure have been specified at the preswirl vane and an average static pressure boundary condition was applied at the exhaust diffuser outlet. The computation was completed using High Performance Computing (HPC) facilities available at the Queen’s University of Belfast.

3. Design Space

The designs were initially based on a state-of-the-art radial turbine which was subsequently scaled to a diameter of 90mm to permit instrumentation in future experimental work. Each of the designs in the space share common exducer geometries, shroud profiles and blade numbers, and, therefore, the exducer throat area, which typically controls the maximum mass flow rate achievable by a rotor, is identical for all rotors. As the cone angle and camber angle of the blade are intrinsically linked, as given in equation 2, simply creating an inlet cone angle by removing blade material, analogous to machining a portion of the inlet of a rotor, would produce blade angles that would vary from design to design and that would be uncontrollable. Therefore, the inlet blade angle was set for each design, with a constant value used for the complete span of the blade inlet. These design modifications were completed using the ANSYS Blade Modeler package.

The design space was set to encompass both varying inlet cone angle and varying inlet blade angle as outlined by the 9 points in Figure 6. Figure 6 uses the simplification that the blade and flow cone angles are normal to one another, which was approximately observed for the designed rotors. The first blade cone angle selected was 30° followed by steps of 15° to 45° and 60° blade cone angles and each of these designs were produced with inlet blade angles of 10°, 20° and 30°. A blade cone angle of less than 30° was not selected as it would require impractical camber angles to achieve the required inlet blade angles up 30° as evident in Figure 6. These 9 rotors have been given the naming convention cXXiYY where XX denotes the blade cone angle and YY denotes the inlet blade angle, β.

Figure 6. Contours of inlet blade angle achieved by combining cone and camber angles.
The 3-D models of the rotors in the design space are shown in Figure 7. The greatest camber angle required was for the 30° inlet blade angle combined with the 30° cone angle, which was 25% more than for a 45° cone angle rotor and 44% more than the 60° cone angle rotor. The blade cone angle has been applied by maintaining the blade leading edge position at the shroud in the meridional plane, and moving the hub leading edge location along the existing hub contour, in the meridional plane, to achieve the required angle. This, therefore, led to a shortening of the blade’s length at the hub. This shortened hub profile, which drives the shroud profile as the blade is radially fibred, gives the impression that 60° cone angle rotors have the greatest degree of camber, however, as shown in Figure 6, they are significantly less so than the lower cone angle rotors. Despite this strong curvature, the radial fibres of the blade have still been maintained, as shown in Figure 8 for c60i30, and no additional bending stresses should be induced in these designs. While it may be possible to counter this strong blade curvature by modifying the exducer geometry and extending the blade length at the hub, this is outside the scope of this current work.

![Figure 7](image_url)  
**Figure 7.** Design space of mixed flow rotors varying cone angle and inlet blade angle.

![Figure 8](image_url)  
**Figure 8.** Radial fibres of c60i30 turbine rotor.

### 4. Results

Figure 9 shows the velocity ratio at which peak efficiency occurred for all the rotors within the design space. For a given cone angle, the first of these trends is the velocity ratio at which peak efficiency occurs which shifts to lower value for increasing inlet blade angle. This is as expected since optimum
incidence will now occur at a more tangential flow condition, owing to the greater non-zero inlet blade angle achieved. Crucially, in the context of this work, this shift in peak efficiency to a lower velocity ratio has been achieved without interrupting the blade’s radial fibre, demonstrating a mixed flow turbine’s suitability to such operating conditions.

Comparing the rotors of different cone angles it can also be seen that, with increasing cone angle, the location of the peak efficiency also shifts to a lower velocity ratio. However, to interpret this as a clear gain may not be correct as the definition of velocity ratio, that is the ratio of the inlet blade velocity to the isentropic spouting velocity, is highly dependent on the blade radius defined for the inlet of the device. For a radial turbine, the choice is simple as inlet radius is constant, but for a mixed flow turbine a number of options exist such as the shroud tip radius, the mean radius or, as used in this case, the Route Mean Square (RMS) radius. Therefore, depending on which of these values is selected, a number of different peak velocity ratio efficiency values could be deduced.

![Figure 9](image-url)  
**Figure 9.** Change in peak efficiency for mixed flow rotors.

To remove any uncertainty arising from velocity ratio calculations, the efficiency trends of 3 of these rotors have been plotted in Figure 10 with respect to rotational speed for a constant pressure ratio of 2.7. It can be seen that increasing blade cone angle reduces efficiency at lower rotational speeds and benefits it at higher rotational speeds. As a lower rotational speed corresponds to a lower velocity ratio, and a higher rotational speed to high velocity ratio, the observed trend in efficiency is actually counter to what would be expect from Figure 9. These trends in efficiency could be explained by looking at the inlet incidence, Figure 11, at both high and low speeds at these operating conditions, namely at 40k rpm and 80k rpm at a pressure ratio of 2.7.

![Figure 10](image-url)  
**Figure 10.** Percentage change in efficiency rate relative to c30i10 for a constant pressure ratio of 2.7.
At the 40k rpm speed, representing low velocity ratio conditions, all 3 rotors show very positive values of incidence, in the region of +60° and +80°. This can be expected through simple analysis of the vectors at the inlet and the relatively low blade velocity at this speed. The effects of the reducing blade inlet radius are also evident as the incidence values tend to become more positive when moving from shroud to hub. With decreasing radius, the conservation of angular momentum of the fluid is also applicable and the tangential component of its velocity tends to increase before reaching the leading edge of the blade, as demonstrated in Figure 12 for the 40k rpm operating speed. It is also seen that the incidence of c30i10 and c45i10 are quite similar, as would be expected from the smaller difference in the efficiency at this speed shown in Figure 10. However, in comparing both these rotors to c60i10, a difference of up to approximately 15° is evident, with the maximum difference near the hub surface. This diverging trend could be explained by the compounded effect of both the change in blade velocity and increasing tangential component simultaneously to give a non-linear response to changing radius.

Comparing the incidence at conditions typical of a higher velocity ratio conditions at a rotational speed of 80k rpm, the majority of the values now lie in the negative incidence region. Again, in analysing the inlet velocity triangle, this is as expected as the blade velocity component has now doubled in magnitude. The effects of the conservation of angular momentum and changing blade velocity are again evident as the incidence values tend to become more positive moving towards the hub, evident in the divergence of both c60i10 and c45i10 from the c30i10 rotor. In this case, however, these more positive values of incidence for increasing cone angle are bringing these values closer to typically optimal incidence values between -10° and -30°, yielding the improvements in efficiency that are predicted. Comparing the c45i10 and c60i10 rotors, a divergence in incidence angles is apparent in

![Figure 11. Spanwise variation of incidence angle at the rotor leading edge near peak efficiency conditions.](image1)

![Figure 12. Meridional plot of increasing swirl angle (to meridional streamline) with decreasing radius for 60° cone angle.](image2)
the hub region, with the c60i10 showing positive incidence angles for a portion of the span. This large shift in incidence appears to be a result of the appearance of an amount of ‘over-the-blade’ flow occurring, as the incidence becomes more positive towards $0^\circ$ incidence and then shifts from negative to positive incidence.

The incidence for changing inlet blade angle, with constant cone angle, has been plotted in Figure 13 for the $45^\circ$ blade cone angle rotors for a pressure ratio of 2.7 at 60k rpm rotating speed.

![Figure 13. Inlet incidence for changing inlet blade angle.](image)

The effects of changing blade speed and conservation of angular momentum are again evident as incidence becomes more positive towards the hub for all rotors, as well an evident strong shift from negative to positive incidence as the flow structures near the leading of the blade change. For increasing inlet blade angle, the incidence can be said to improve. Near the shroud, the three rotors, c45i10, c45i20 and c45i30, have incidence angles in region of $-30^\circ$, $-20^\circ$ and $-10^\circ$ respectively, all falling within the typically values for maximum efficiency. Towards the hub, however, significant improvements are made where the flow conditions are more tangential. The rotors c45i10 and c60i10 both show more negative values of incidence in this region as a result of their greater inlet blade angle and these large improvements could outweigh any smaller changes near optimal incidence in the shroud region.

5. Conclusions

While this work has highlighted a number of fundamental benefits of mixed flow turbines, it has also highlighted a number of issues that require further consideration and that should form future work.

The shift in peak efficiency to lower velocity ratios for increasing inlet blade angle was evident and demonstrated that a mixed flow turbine can achieve such operation without interrupting the blade radial fibre. However, when comparing velocity ratio between turbine rotors, it highlighted shortcomings in the use of velocity ratio in comparing mixed flow turbines of varying cone angle geometry. Further work will therefore be carried on assessing this issue and possible solutions or alternatives.

When analysing the change in efficiency with changing cone angle, a reduction in performance for low velocity ratio operating conditions was actually observed, contrary to what would be suggested by the velocity ratio equation when a smaller inlet radius is applied. This was seen to be a result of both changing blade speed and the conservation of angular momentum increasing the tangential component of the flow. This would suggest that the blade cone angle of a rotor is in fact not a driver of turbine efficiency but rather a means to enable efficiency at lower velocity ratios by other means, such as by the addition of a non-zero inlet blade angle without interrupting the blade’s radial fibre. This was evident in the shift in velocity ratio at which peak efficiency occurred when such an inlet blade angle
was applied and as was illustrated by the improvements in incidence angle as a result. Therefore, it is evident that the simple addition of a blade cone angle to a design alone is not adequate to yield efficiency improvements at low velocity ratio conditions.

References
[1] Japikse D and Baines N C 1994 *Introduction to turbomachinery* (Vermont & Oxford: Concepts ETI)
[2] Walkingshaw J, Spence S, Ehrhard J and Thornhill D 2011 An investigation into improving off-design performance in a turbocharger turbine utilizing non-radial blading *Proceedings of the ASME Turbo Expo Conference* (Vancouver, Canada, 6-10 June 2011)
[3] Fredriksson C and Baines N 2010 The Mixed Flow Forward Swept Turbine for Next Generation Turbocharged Downsized Automotive Engines *ASME Conference Proceedings* (Vancouver, Canada, 12-18 Nov 2010)
[4] Rajoo S and Martinez-Botas R 2008 *Journal of turbomachinery* 130(4) 044001
[5] Lüddecke B, Filsinger D and Ehrhard J 2012 *International Journal of Rotating Machinery* 2012 589720
[6] Watson N and Janota M S 1982 *Turbocharging the internal combustion engine* (Houndmills: MacMillan Press)