EFFECT OF MANUFACTURING TOLERANCE IN FLOW PAST A COMPRESSOR BLADE

S. Venkatesh - Q. Rendu - M. Vahdati - L. Salles

Mechanical Engineering Department, Imperial College London, London, UK
Email: v.suriyanarayanan16@imperial.ac.uk , q.rendu@imperial.ac.uk ,
m.vahdati@imperial.ac.uk , l.salles@imperial.ac.uk

ABSTRACT

This paper presents the effect of manufacturing tolerance on performance and stability boundaries of a transonic fan using a RANS simulation. The effect of tip gap and stagger angle was analysed through a series of single passage and double passage simulation; based on which an optimal arrangement was proposed for random tip gap and random stagger angle in case of a whole annulus rotor. All simulations were carried out using NASA rotor 67 as a test case and AU3D an in-house CFD solver. Results illustrate that the stagger angle mainly affects efficiency and hence its circumferential variation must be as smooth as possible. Furthermore, the tip gap affects the stability boundaries, pressure ratio and efficiency. Hence its optimal configuration mandates that the blades be configured in a zigzag arrangement around the annulus i.e. larger tip gap between two smaller ones.

KEYWORDS

MANUFACTURING TOLERANCE, RANDOM STAGGER ANGLE, RANDOM TIP GAP, OPTIMAL ARRANGEMENT, UNCERTAINTY QUANTIFICATION

NOMENCLATURE

RANS	Reynolds averaged Navier-Stokes	SP	Single Passage
SM	Stall Margin	DP	Double Passage

INTRODUCTION

A real world system is seldom deterministic in nature. There is always an element of uncertainty present in the parameters that define the system. Sometimes these uncertainties are known to trigger the instabilities occurring in the system prior to the deterministic prediction potentially resulting in unwanted oscillations of large amplitude. Though these uncertainties can be broadly classified into aleatory uncertainty (inherent variation) and epistemic uncertainty (insufficient knowledge; more problematic), Kiureghian & Ditlevsen (2009) say that the characterisation of any uncertainty depends on the modeller as the line dividing the two concepts is quite blurred. The uncertainties in a turbo-machinery environment can arise due to different rotation speed of blades, varying atmospheric conditions, eccentrically assembled rotor, casing ovalation, manufacturing tolerances, etc. The quantification of innumerable uncertainties present in a system is overcome by employing reduced order model wherein only the parameters of interest are considered for further evaluation. The present paper deals with only the uncertainties due to manufacturing tolerance which give rise to differing passage geometry and subsequent performance degradation, as pointed out by Lange et al. (2010) using Monte Carlo Simulation. Research carried out in the last 3 decades as listed by Montomoli et al. (2015)

raises a few open ended questions that few researchers have attempted to tackle *e.g.* "Is there an optimal configuration for a given set of random blades? If yes, what is that arrangement and how do we predict the performance?". The authors in the present paper have tried to address the above issues by concentrating on tip gap and stagger angle as the tip gap is known to affect stall margin, efficiency and pressure ratio while stagger angle has a detrimental effect on efficiency.

FLOW SOLVER AND TEST CASE

The simulations were performed using an in-house CFD solver called AU3D. The CFD solver developed at Imperial College London-VUTC with support from Rolls-Royce is a three dimensional, time-accurate, viscous, compressible Reynolds-averaged Navier-Stokes (RANS) solver that uses standard Spalart-Allmaras (SA) model to evaluate the turbulent eddy viscosity. The solver, implicit and second-order accurate in space and time, has been successfully used to predict the design and off-design conditions for last 25 years (Dodds & Vahdati 2014). Computationally, a full annulus rotor with blades of same dimension is solved by considering only a sector of it. This method of solving turbo-machinery problem is ubiquitous to a lot of research papers found in literature (Wilson et al. 2006). The sector is called single passage if it consists of a single blade; double passage in case of two blades, etc. A full annulus simulation has been done when random stagger angle and tip gaps are considered, given the lack of symmetry. The computational domain in this paper consists of (a) Inlet duct (b) Single passage or Double passage or Full annulus rotor (as the case maybe) and (c) nozzle. A variable area nozzle placed downstream of the rotor is adjusted gradually to move the fan operating point from near choke condition to near stall point (Vahdati et al. 2005). The solver operates on a semistructured mesh with hexahedral elements found around the airfoil and prismatic elements in rest of the passage. The boundary layer region in rotor-to-rotor grid has a body-fitted O grid while rest of the region is unstructured. The tip clearance geometry is meshed by triangulating the blade tip and mapping extra layers over the tip. For more details, please refer to Sayma et al. (2000).

NASA rotor 67 was used as a test-case for all the computations. As seen in the literature, there is an uncertainty in the value of nominal tip gap of NASA rotor 67. For the same performance, Strazisar et al. (1989) and Roberts et al. (2013) report different running tip clearance of approximately 1 mm and 0.5 mm respectively. Adamczyk et al. (1991) performed CFD computations at four different clearance to tip chord ratios of zero, 0.25% (≈ 0.225 mm), 0.75% and 1.25%. They considered a clearance of 0.25% of tip chord to be nominal clearance to account for the vena contracta at the tip region. Jennions & Turner (1992) performed CFD computations at three different tip gap values viz. nominal gap of 0.024 inch (≈ 0.61 mm), half nominal and twice nominal. Since then many papers including Arima et al. (1999) and Choi et al. (2009) have adopted the nominal gap (≈ 0.61 mm) in their computation. Moreover, the usage of standard SA model leads to under-prediction of the stall point and requires fine tuning the production and destruction terms (Lee et al. 2018) to make it predict accurately, which is not the main objective of this paper. This paper considers the nominal tip gap to be 0.6 mm and concentrates only on qualitative comparison of CFD results. A similar mesh count for different single passage rotors (≈ 1 million) has been ensured by keeping the number of radial layers (85) and the position of all periodic points exactly the same. One million mesh points per passage were chosen based on previous studies carried out on NASA rotor 67 using AU3D (Vahdati & Imregun 1996, Zhang & Vahdati 2017, Lee et al. 2018). A near wall resolution with $Y^+ \approx 20$ along with wall function was used (Lee et al. 2018).

STEADY RANS SIMULATION FOR COMPRESSOR STAGGER ANGLE

There is a dearth of data in open literature pertaining to the aerodynamic study of mis-staggering as most papers concentrate on the aspects of forced response and the mechanism of flutter. Of the available few, Wilson et al. (2006), Sladojevic et al. (2007) and Zheng et al. (2017) mention that the manufacturing tolerances for stagger angle are allowed up to $0.5°$ to $1°$.

The present paper considers the stagger variation to be within $\pm1.5°$, also used by Zheng et al. (2017). The fan blade with tip gap of 0.6 mm was considered and the stagger was gradually changed in steps of $\pm0.5°$ to study the effect of change in stagger of single passage (**SP**) vs. the double passage (**DP**) rotor performance. The first step was to generate three **SP** cases to see variation in results: (**I**) **SP** rotor 67 blade (also called as $0°$ stagger) (**II**) **SP** $+1.5°$ staggered blade (**III**) **SP** $-1.5°$ staggered blade. The second step entails the evaluation of results of **DP** mesh vis-a-vis the **SP** results. The following are the two **DP** mesh that were created: (**A**) **DP** with one $0°$ stagger blade and other $+1.5°$ staggered blade (**B**) **DP** with one $0°$ stagger blade and other $-1.5°$ staggered blade.

For **SP** cases, a change $\pm1.5°$ causes the mass flow rate to change by $\pm2\%$ on either side of the baseline **SP** $0°$ stagger case. Hence the mass flow rate was non-dimensionalised for each case with respective choking mass flow rate which makes a one-to-one comparison between different cases simpler. For brevity, the results of only **SP** $0°$ stagger, **SP** $-1.5°$ stagger and **DP** $0°$ and $-1.5°$ stagger together are presented in figure 1(a) and a comparison is made at the non-dimensional mass flow rate of 0.986 (marked by vertical blue dashed line). These points depict the behaviour of the respective cases as they near maximum efficiency. It was seen that the peak efficiency of 92.5% is observed for all the single passage cases; whereas for a double passage case, the peak efficiency drops to about 91.5%. Figure 1(b) shows that as the mis-stagger decreases, the peak efficiency increases.

In figure 2(a), (b) and (c), a blade-to-blade view at 75% span from hub is shown for **SP** $0°$ stagger, **SP** $-1.5°$ stagger and **DP** $0°$ and $-1.5°$ stagger respectively. It can be seen that both the single passage cases have symmetric shock waves which are at the starting of the blade passage while the double passage case (with two different stagger blades) has unsymmetrical shock waves (Wilson et al. 2006). This implies that when one of the blade is having peak efficiency, the other blade is choked and hence the assembly will always be less efficient than the single passage cases. Hence the pressure ratio of both **SP** $0°$ stagger and **SP** $-1.5°$ stagger exceed the **DP** $0°$ and $-1.5°$ stagger case as seen in figure 1(a). A schematic explaining the mechanism of shock wave movement for a double passage case with different stagger blades is shown in figure 2(d). It can be seen that in case of extreme mis-staggering one of the passage behaves like a nozzle while the other behaves like a diffuser thus resulting in unsymmetrical shock waves. Moreover, due to asymmetrical aerofoil used in the compressor blades, the area of nozzle and diffuser formed in a **DP** with $0°$ stagger blade and $+1.5°$ stagger blade is different from the area of nozzle and diffuser formed in a **DP** with $0°$ stagger blade and $-1.5°$ stagger blade resulting in different losses in the two **DP** cases. This can be clearly seen in figure 1(b) which is unsymmetric about the $0°$ mis-stagger.

Thus, for a given set of uniformly distributed random staggered blades between $\pm1.5°$ as shown in figure 3(a), if the arrangement is smooth as shown in figure 3(b) and figure 3(c), the mis-stagger between successive blades is minimum which results in corresponding increase in efficiency and pressure ratio. However, if the mis-stagger between successive blades is maximum as is the case for zigzag arrangement (larger stagger blade between two smaller ones) shown in figure 3(d), there will be a drastic reduction in efficiency and corresponding decrease
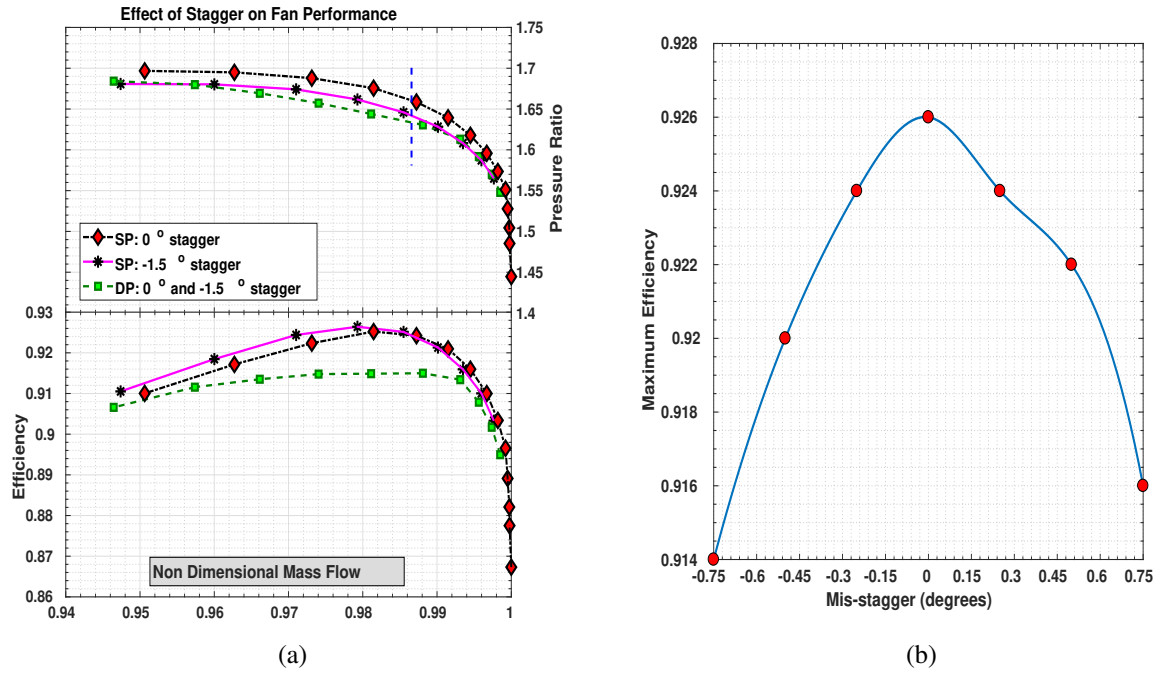
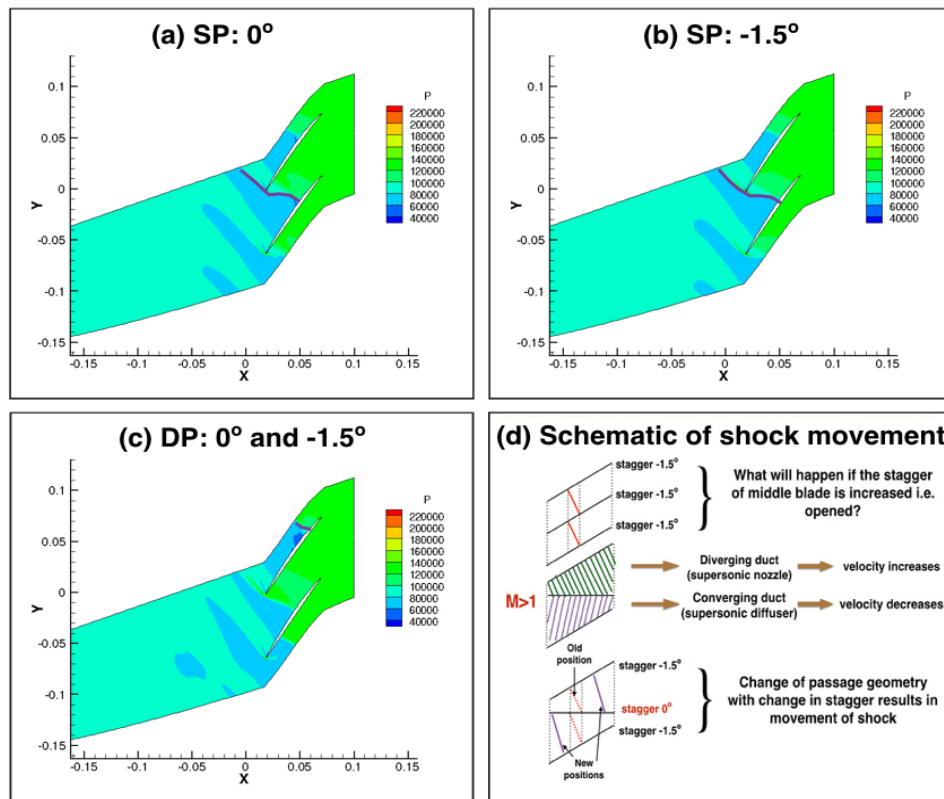Figure 1: **(a) Fan performance due to mis-staggering (b) Max. efficiency vs. mis-staggering**

Figure 2: **Shock wave position at 75% span and non-dimensional flow rate of 0.986 for (a) Single passage 0° stagger (b) Single passage -1.5° stagger (c) Double passage 0° and -1.5° stagger and (d) Schematic explaining the mechanism of shock wave movement due to mis-staggering**
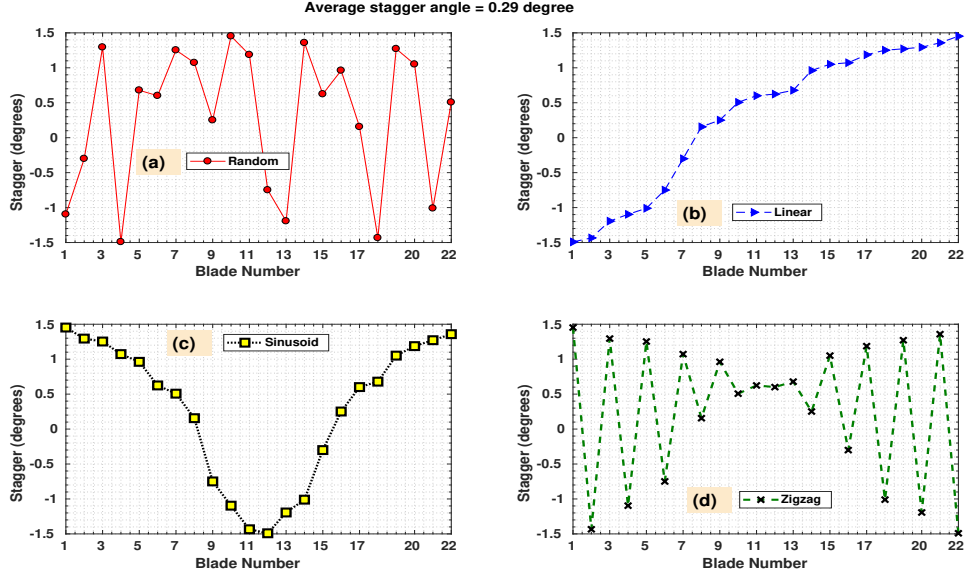
Figure 3: **Different arrangements for same set of random staggered blades with average stagger 0.29° (a) Random arrangement (b) Linear arrangement (c) Sinusoid arrangement (d) Zigzag arrangement**

in pressure ratio. This can be clearly seen in figure 4 wherein sinusoidally and linearly arranged blades around the whole annulus have the performance highest and closest compared to single passage performance for average stagger of $0.29°$ stagger; while for the zigzag arrangement it is vice-versa. The performance of all other random arrangements lie between these two extremities. No difference in the stall mass flow is observed for different configurations. Zheng et al. (2017) also observed that the sinusoidal arrangement works best though they have not evaluated the minima of possible performance. While there maybe other combinations which give similar high performance, the aim of this paper was to establish a rule of thumb for the best possible arrangement. The sinusoidal arrangement can also have an added advantage of a reduced "Alternate Passage Divergence" as shown in Lu et al. (2018).

STEADY RANS SIMULATION FOR COMPRESSOR TIP GAP

A **SP** rotor domain has been generated for five different tip clearances: $\mathbf{0.3}$ mm ($\approx 0.188\%$ span), $\mathbf{0.6}$ mm, $\mathbf{0.9}$ mm, $\mathbf{1.2}$ mm and $\mathbf{1.5}$ mm. In addition to this, **DP** rotor configurations with two different tip gaps for all the possible combinations from above tip gaps have been generated. As can be seen in figure 5(a), the stalling mass flow increases with a corresponding decrease in the pressure ratio and efficiency for an increase in **SP** tip gap. These trends are similar to those obtained by Adamczyk et al. (1991) and Beheshti et al. (2004). As seen in figure 5(b), the increased tip gap in a **SP** rotor causes loss of stall margin (**SM**) which matches with observation of Moore (1982) and Beheshti et al. (2004). On the other hand, the **SM** improves for **DP** with tip gaps 0.3 mm and 1.5 mm compared to **SP** tip gap of 0.9 mm. The **SM** is defined below:

$$\mathbf{SM} \;=\; \left[\frac{(\mathbf{PR})_{\mathbf{stall}} \times (\dot{\mathbf{m}})_{\mathbf{ref.}}}{(\mathbf{PR})_{\mathbf{ref.}} \times (\dot{\mathbf{m}})_{\mathbf{stall}}} - 1 \right] \times \mathbf{100} \tag{1}$$

where $\mathbf{PR}$ is Pressure ratio and $\dot{\mathbf{m}}$ is mass flow rate (Moore 1982, Beheshti et al. 2004).
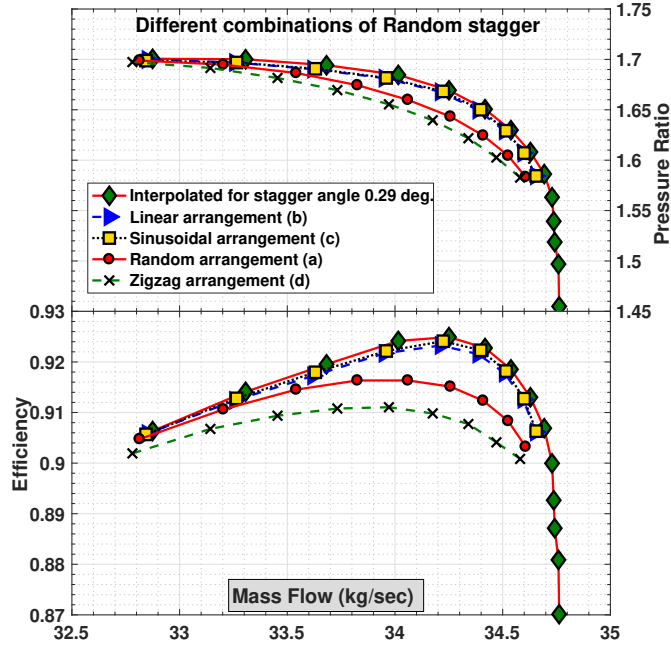
Figure 4: **Performance for different combinations of random stagger**

For brevity, the Mach number contours of only **SP** tip gaps of 0.3 mm, 1.5 mm and **DP** with tip gaps of 0.3 mm and 1.5 mm together has been attached in figure 6(a), (b) and (c). An increase in the tip gap causes the size and strength of tip leakage vortex to increase that in turn creates a large zone of low energy fluid due to the shock wave and vortex interaction as seen in figure 6(a) and (b). It can also be seen from figure 6(c) that the low energy fluid in **DP** (0.3 mm + 1.5 mm) is much less compared to **SP** 1.5 mm. These pockets of low energy fluid produce a region of blockage which causes the main flow to be diverted around it. The altered main flow due to the blockage creates an adverse pressure gradient which leads to flow separation and hence the blade stall (Thompson et al. 1997, Beheshti et al. 2004). A summary of the maximum pressure ratio and maximum efficiency with respect to the tip gaps is shown in figure 7. As discussed earlier, it can be seen that as the tip gap increases, the maximum pressure ratio as well as efficiency in the **SP** cases starts decreasing though the performance **DP** 0.3 mm and 1.5 mm is better than **SP** 0.9 mm.

A schematic explaining the mechanism of movement of shock waves for a **DP** with tip gaps of 0.3 mm and 1.5 mm together is shown in figure 6(d). It can be seen that the higher tip leakage flow due to the larger tip gap blade causes one of the passages to stall while the other passage undergoes choke. Hence the shock waves in the two passages moves in opposite direction to each other. Due to the movement of shock wave in the double passage case, the larger tip gap blade becomes "unloaded" while the smaller tip gap blade becomes "more loaded" as seen in figure 8 showing the blade loading of **SP** 0.3 mm and **SP** 1.5 mm together with **DP** tip gaps of 0.3 mm and 1.5 mm. The increase in static pressure on the blade surface due to the shock has been marked and compared in figure 6(a), (b) and (c). The tip leakage flow is strongly dependent on both the pressure difference between Suction side (SS) - Pressure side (PS) as well as on the tip gap size. Lambda-2 method has been used (Jeong & Hussain 1995) to visualise the tip leakage vortex. The areas marked as "A" and "B" in figure 9(b), are clearly absent or reduced
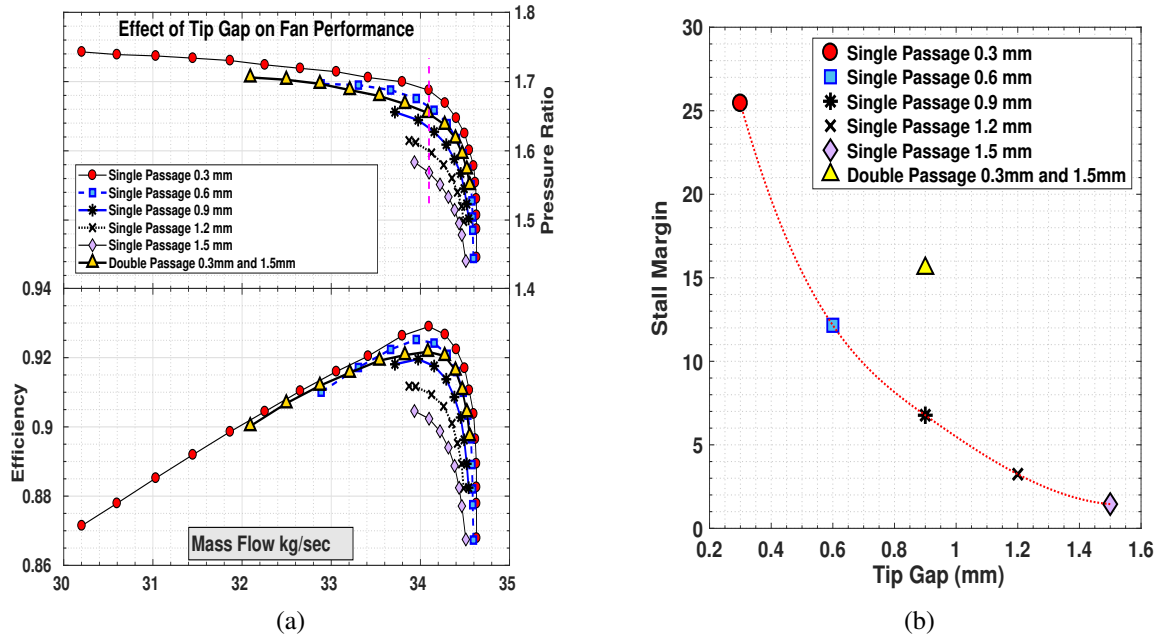
Figure 5: **(a) Fan performance for different tip gaps (b) Stall margin for different tip gaps**
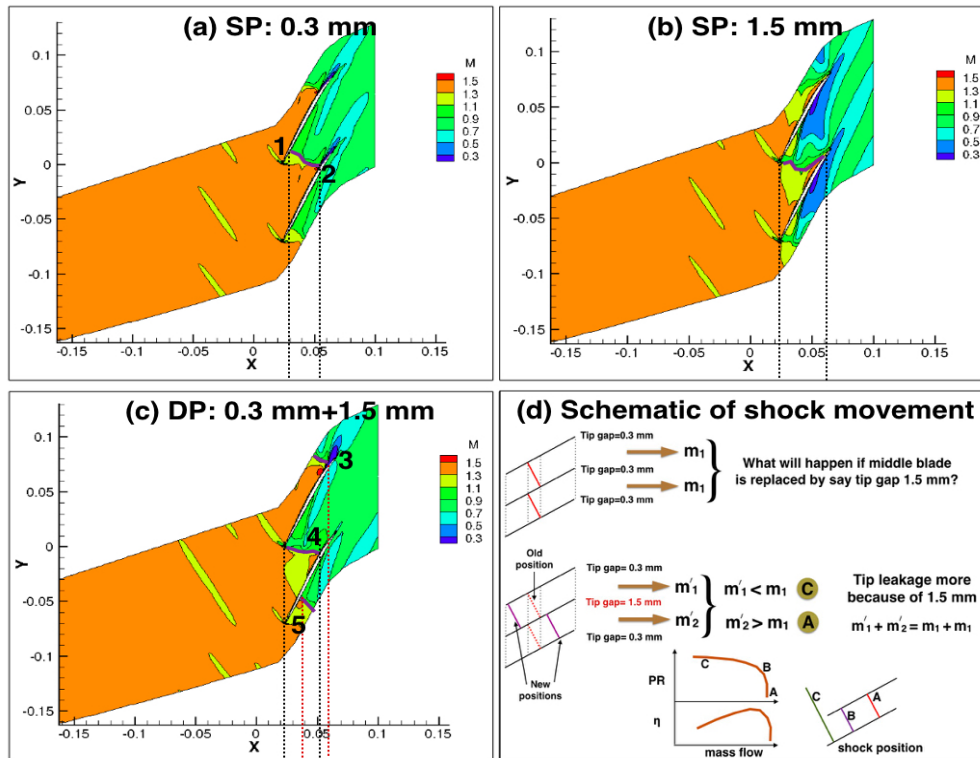

Figure 6: **Shock wave position at 99% span and mass flow rate of 34.1 kg/s for (a) Single passage 0.3 mm (b) Single passage 1.5 mm (c) Double passage 0.3 mm and 1.5 mm and (d) Schematic explaining the mechanism of shock wave movement due to different tip gaps**
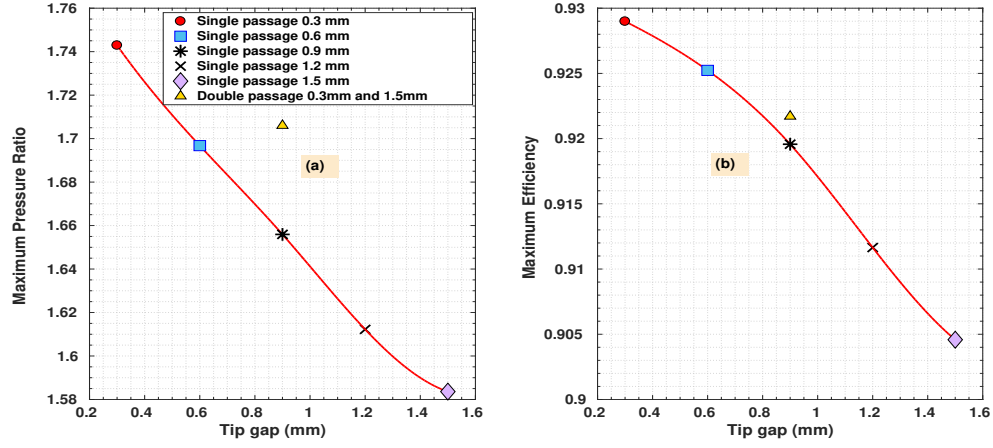
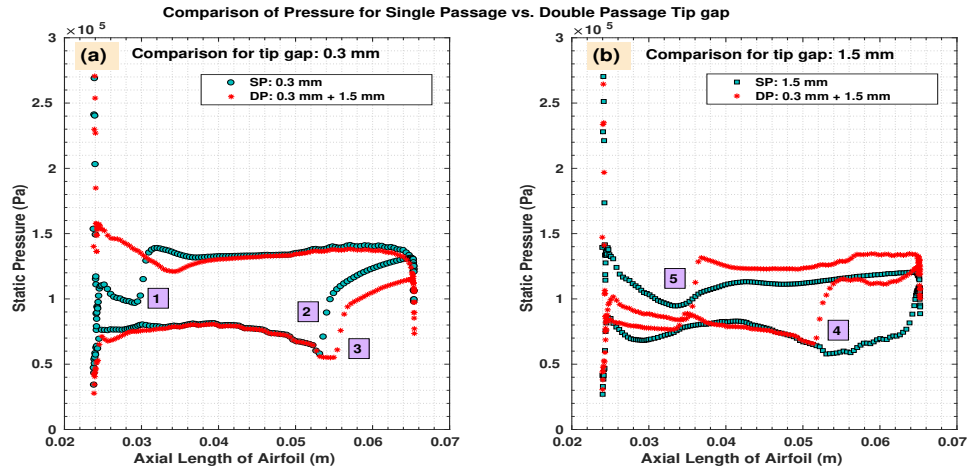Figure 7: **(a) Tip gap vs maximum pressure ratio (b) Tip gap vs maximum efficiency**

Figure 8: **Blade loading: SP 0.3 mm and 1.5 mm vs. DP 0.3 mm and 1.5 mm together**

(a) Leakage vortex for 0.3 mm tip gap

(b) Leakage vortex for 1.5 mm tip gap

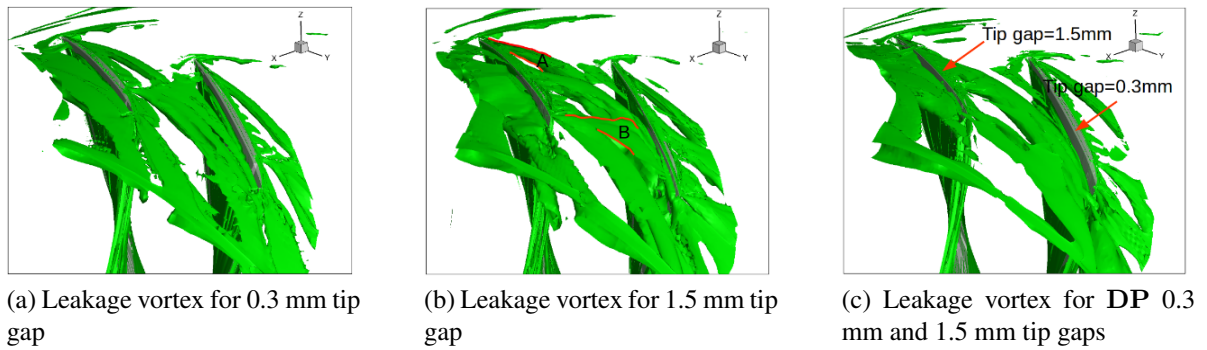(c) Leakage vortex for **DP** 0.3 mm and 1.5 mm tip gaps

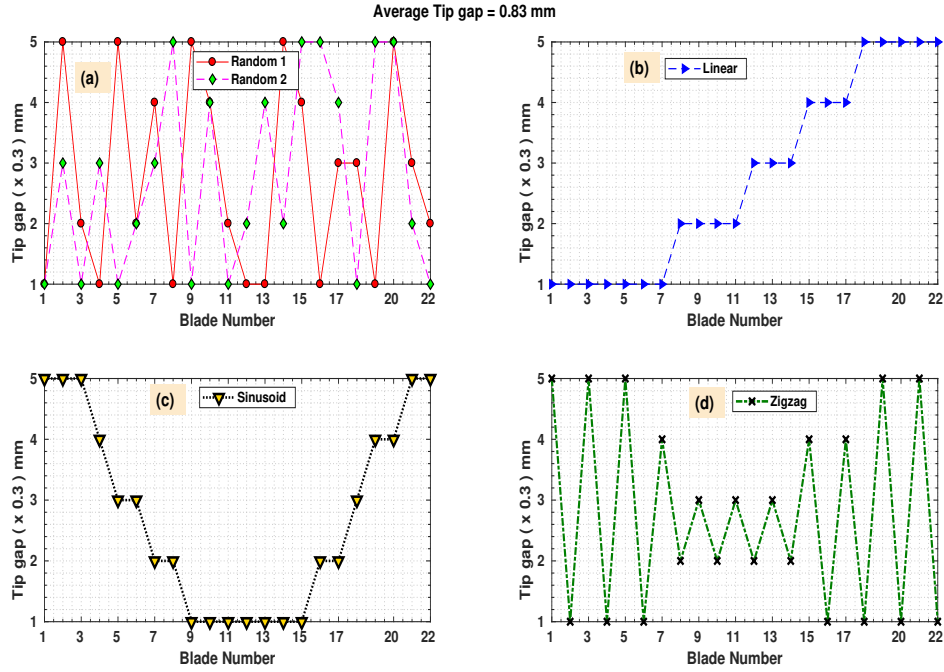Figure 9: **Lambda-2 method for flow visualisation over the tip**

Figure 10: **Different combinations for same set of random tip gap blades with average tip gap of 0.83 mm (a) Random arrangement (b) Linear arrangement (c) Sinusoid arrangement (d) Zigzag arrangement**

in intensity in figure 9(c). Thus it can be said that in the presence of smaller tip gap blade (0.3 mm), the larger tip gap blade (1.5 mm) has less tip leakage vortex than it would have were all the blades of the same tip gap size (1.5 mm).

Another study was conducted to see the effect of arranging the 11 blades of 1.5 mm in one half of the annulus and the rest 11 blades of 0.3 mm in the other half of annulus (also called "half-half" arrangement) and was compared with double passage 0.3 mm and 1.5 mm. Similar tests were done for blade with tip gaps 0.6 mm and 1.2 mm with the results summarised in figure 11(a). It can be seen that the performance and the aerodynamic stability of the double passage 0.3 mm and 1.5 mm was the best while the half-half arrangement of 0.3 mm and 1.5 mm tip gap blades was the worst. The performance of double passage 0.6 mm and 1.2 mm matches closely with the single passage 0.9 mm case as does the half-half arrangement of 0.6 mm and 1.2 mm. This implies that the type of arrangement plays no further role in eking out any extra performance when the manufacturing tolerance is very close to the mean value. It can thus be hypothesised that when faced with random tip gaps in a whole annulus which are not close to mean value, the best arrangement would be to have a larger tip gap between two smaller ones also called as "zigzag arrangement" around the full annulus to obtain maximum efficiency, pressure ratio and stall margin. The word "random" in the context of tip gaps is a slight misnomer as there are only 5 discrete tip gaps in this study. Instead, a uniformly distributed random integer between 1 to 5 was generated with "1" implying 0.3 mm tip gap and "5" implying 1.5 mm. Figure 10(a) shows the random arrangement of the different tip gaps while figures 10(b), (c) and (d) show the linear, sinusoid and zigzag arrangement respectively for the same set of random blades. Figure 11(b) shows the corresponding performance of all
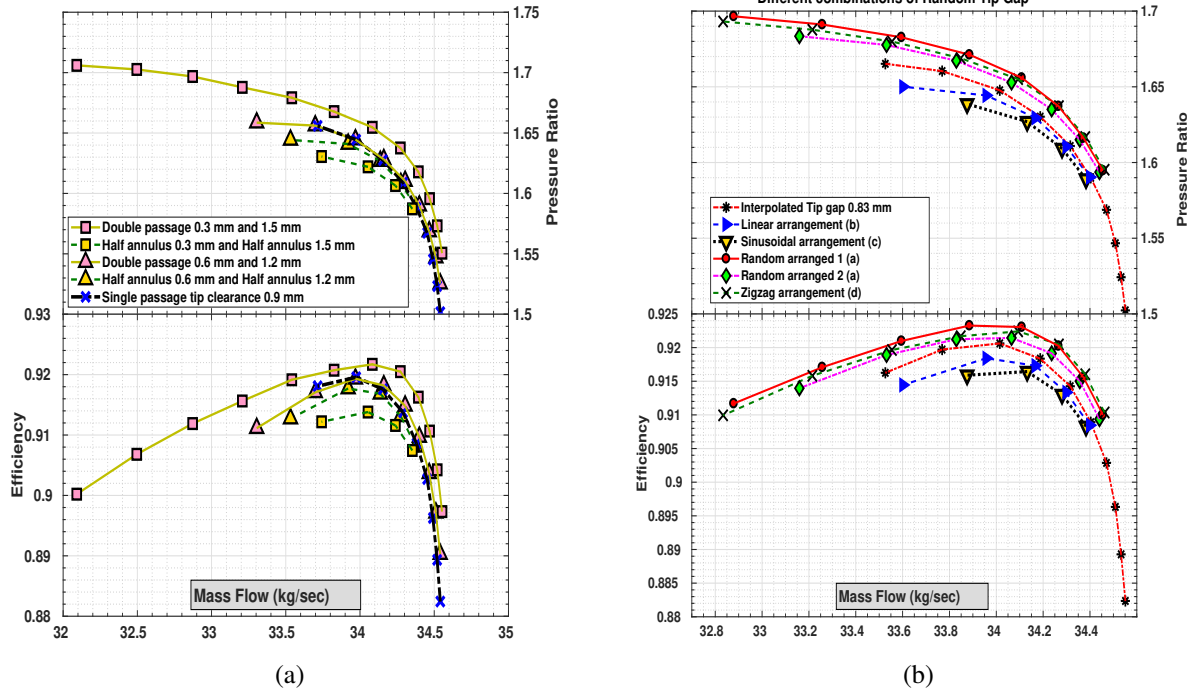
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Figure 11: **(a) Comparison of DP vs. Half-Half combination (b) Performance for different combinations of random tip gaps**

the different arrangements described in figure 10. It can be seen that sinusoidal arrangement stalls earliest whereas the zigzag arrangement stalls the last when compared to interpolated performance of single passage average tip gap of 0.83 mm which implies that the performance of all the random combination lie in between these two extremities as seen from the performance of "Random arranged 1" and "Random arranged 2". It should be noted that for any arrangement, there are two combinations that are possible: (a) Clockwise (b) Anticlockwise. There were no major difference in the performance curve exhibited by the two combinations. Future work would consider the effect of coupling between of tip gap and stagger angle on the similar lines of the study by Montomoli et al. (2011) and also develop a low fidelity model to predict of the performance of compressors without resorting to full annulus simulation.

CONCLUSION

The study on the effect of manufacturing tolerance on the performance and stability boundaries of a compressor was carried out using stagger angle and tip gaps as the parameters of interest. It can be seen that in case of stagger angle, the change in passage area leads to the movement of the shock which affects efficiency and hence pressure ratio. This change of passage area is minimum when the mis-stagger between two successive blade is minimum. Hence the most optimal arrangement when confronted with random stagger angle would be a sinusoidal arrangement wherein the circumferential variation is smooth throughout the annulus. A linear arrangement also gives a smooth circumferential variation bar the variation between the first and the last blade. Performance wise, linear arrangement is equivalent to sinusoidal arrangement. In real turbo-machinery application, the random stagger results would apply to both fan and compressors equally.

In case of tip gaps, it was seen that in the presence of smaller tip gap blade, the tip leakage flow from larger tip gap blade is reduced and thus the stalling can be delayed. Hence the most optimal arrangement when confronted with a random tip gap would be to arrange it in a zigzag arrangement wherein the larger tip gap blade is placed between two smaller tip gap blades. It was also seen that close manufacturing tolerance to the designed one would nullify the effect of different type of arrangements. In real turbo-machinery application, the random tip gap results would be more significant in case of compressors. This study, to the best knowledge of the authors, covers all possible combinations and gives an upper limit and lower limit to the compressor performance if persisted with random arrangement.

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