EFFECT OF GEOMETRIC UNCERTAINTY ON A ONE STAGE TRANSONIC COMPRESSOR OF AN INDUSTRIAL GAS TURBINE

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ABSTRACT

The geometrical uncertainties can result in flow asymmetry around the annulus of compressor which in turn can detrimentally affect on the compressor stability and performance. Typically these uncertainties arise as a consequence of in-service degradation and/or manufacturing tolerance, both of which have been dealt with in this paper. The paper deals with effects of leading edge damage and tip gap on rotor blades. It was found that the chord-wise damage is more critical than radial damage.

It was found that a zigzag pattern of arranging the damaged rotor blades (i.e. most damaged blades between two least damaged blades) would give the best possible performance and stability when performing maintenance and overhauling while a sinusoidal pattern of arrangement had the worst performance and stability. This behaviour of zigzag arrangement of random damaged blades is consonant with the behaviour of zigzag arrangement in random tip gaps. It is also shown in this work that the level of damage has a bigger impact on the compressor performance and stability than the number of damaged blades.

NOMENCLATURE

RANS	Reynolds Averaged Navier-Stokes.
SM	Stall Margin.
CMF	Corrected Mass Flow Rate.
NCMF	Normalised Corrected Mass Flow Rate.
η	Isentropic Efficiency Of Compressor.
RE	Relative efficiency.

INTRODUCTION

The in-service degradation is an important variable that the designers need to take into consideration because of its potential to adversely affect the aerodynamics of the compressor, its performance and stability. A better understanding of the aerodynamic behaviour under such conditions can lead to better post overhauling performance of the compressor. However, this area of research has not been given its due and of the available few open access literature, many of them pertain to bird strike problem with an emphasis on flutter stability and forced response characteristics [1].

The in-service degradation can occur due to various reasons like foreign body impact (leading edge damage), erosion due to ingestion of sand or ash particles due to faulty air filtration system in an industrial gas turbine (leading edge damage), sudden surge in compressor causing excessive blade rubbing against the casing (trailing edge damage). Both, leading edge damage as well as trailing edge damage can deteriorate the performance and stability margins though the later can be more detrimental as it often leads to other problems like stage mis-matching. The damage produced can typically vary both radially and in chord-wise direction. In fact, the compressor blades of TV3-117 turboshaft suffer from chord length and airfoil thickness reduction above 66% span when operating in extreme desert terrains of Algeria [2].

During the engine overhaul, such damaged blades are typically “boroblended” as well as “re-contoured” to restore blade performance [3, 4]. In fact, new airfoils are sometimes mixed with reused parts resulting in wide variety of airfoil geometries

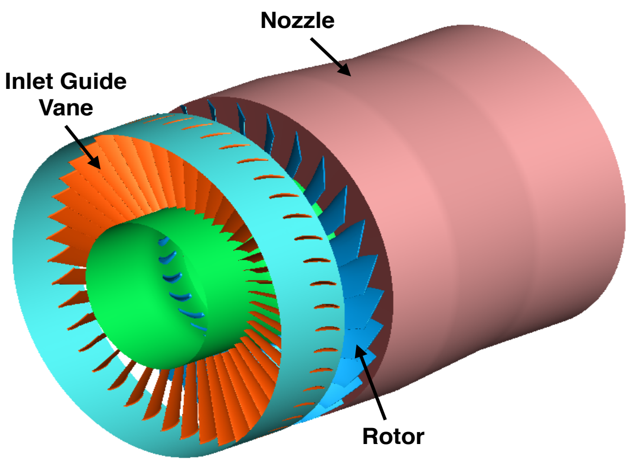

FIGURE 1. COMPUTATIONAL DOMAIN SHOWING IGV, ROTOR AND NOZZLE

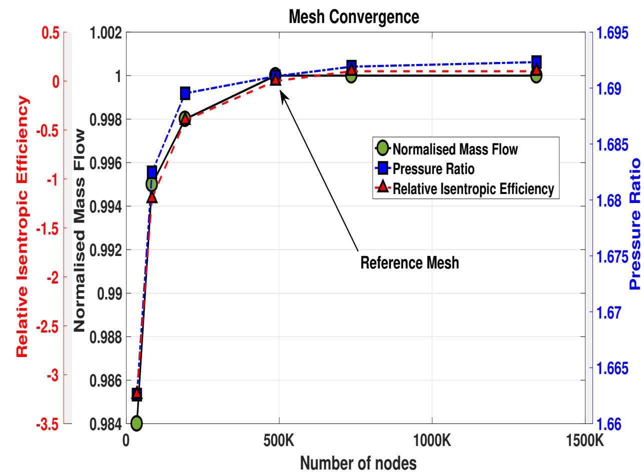

FIGURE 2. MESH CONVERGENCE STUDY FOR ROTOR NEAR PEAK EFFICIENCY

with a distinct statistical distribution found within an engine after operation [5]. Roberts *et al.* [3] report that the current maintenance practice is to distribute these blades of varying chord length around the disk with a check for balance. With the given background, an apposite question as asked in [6] would be: "Is there an optimal configuration for a given set of random damaged blades that can eke out maximum performance without compromising stability boundaries? If yes, what is that arrangement?". This is a very pertinent question as it is known that even a 0.5% increase in efficiency across fan and compressor can result in fuel savings worth tens of thousands of pounds per engine per year [7]. It is important to note that Roberts *et al.* [3] also reasoned on the similar lines in their research paper. In previous study [6], it was shown that the tip gap affects pressure ratio, efficiency and stability margins. The behaviour of tip gap vis-a-vis damaged blades for a 1-stage transonic compressor of an industrial gas turbine has been presented in this paper.

FLOW SOLVER AND TEST CASE

The simulations were performed using the in-house CFD solver AU3D, developed at Imperial College London-VUTC with support from Rolls-Royce. AU3D is a three dimensional, time-accurate, viscous, compressible Reynolds Averaged Navier-Stokes (RANS) solver that uses Spalart-Allmaras (SA) model to evaluate the turbulent eddy viscosity. The flow solver is based on cell-vertex finite volume methodology. The central differencing scheme has been applied along with a mixture of second and fourth-order artificial dissipation for stabilisation. Additionally, a pressure switch which reverts to first order Roe scheme in the vicinity of discontinuities, has been used for numerical robustness. The resulting system of equations is advanced in time using an implicit scheme with Jacobi iteration and dual time stepping. The solver has been successful in predicting the design and off-design conditions [8–11].

Computationally, only a sector of a whole annulus rotor is considered if all the blades are of same geometry [12]. The sector is called single passage if it consists of a single blade; double passage in case of two blades and so on. A whole annulus simulation has been performed when the given geometry lacks symmetry. The computational domain shown in fig. 1 consists of (a) IGV (b) Rotor and (c) Nozzle. Depending on the case in hand, the IGV can be single passage (*SP*) or whole annulus (*WA*) and the rotor can be single passage (*SP*) or double passage (*DP*) or whole annulus (*WA*). The nozzle is either single passage (*SP*) or whole annulus (*WA*).

An inherent unsteadiness to the flow is introduced due to the interaction of rotating and stationary blades. There are two ways of treating the adjacent blade rows in relative rotation frame viz. mixing planes and sliding planes. A comparative study between these two methods for a turbo-machinery environment [13] has shown that both mixing plane and sliding plane predict identical loading up to the mid-span region but differ towards the tip section. This is because, a mixing plane carries out circumferential averaging which cuts off the shock propagation upstream and tip vortex propagation downstream. However, the mixing plane approach has been used widely by many researchers due to its simplicity to calculate the performance in case of multistage machines [13].

All the performance characteristic for 1-stage compressor shown in this paper are computed using a steady RANS solver with mixing planes between IGV, rotor and nozzle. At inlet, the total pressure, total temperature and flow angles are specified. The present computation uses a chocked nozzle which drives the outlet boundary condition. As the rotor operating point moves from choke to stall, the nozzle is closed gradually in discrete steps which ensures a gradual reduction of mass flow rate accompanied by a corresponding increase in pressure ratio. A chocked nozzle ensures that disturbance from the exit does not propagate upstream [14]. At stall point, the slope of the performance characteristic becomes positive with a massive drop in mass flow rate and pressure ratio.

The solver operates on a semi-structured 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. More details about this technique is given in [15]. A transonic 1-stage compressor designed by the second author during his stay at Imperial College London has been used as a test-case for all the computations. This geometry represents the front stage of a modern industrial gas turbine. The IGV has 44 blades and the rotor has 28 blades with a nominal tip gap of 0.33% of span.

A number of rotor meshes were generated by systematically varying the total number of radial levels (45 to 80) and also the blade-to-blade mesh for rotor domain (700 to 18,000 nodes per radial layer). The radial spacing was finalised after testing both the largest as well as smallest tip gap size that would be encountered. The reference mesh with tip gap of 0.33% of span, consists of a total of 50 radial layers such that there are 40 radial layers on the blade and 10 radial layers in the tip gap region. The position and the total number of radial layers in the passage of largest tip gap blade remains the same as that in the passage of the smallest tip gap blade. However compared to the smallest tip gap blade, the number of radial layers on the blade with largest tip gap decreases with a corresponding increase in the radial layers in the tip gap region. Fig. 2, shows the variation of normalised mass flow rate, pressure ratio and relative isentropic efficiency plotted against the number of grid points for rotor near peak efficiency. The reference mesh with approximately 0.5 million nodes per passage for rotor can be treated as fine enough with converged values. A reference mesh of 0.44 million nodes per passage for IGV is generated using the above mentioned principles.

Seven levels of damage have been considered in the rotor with each level indicating progressively higher damage both chord-wise as well as radially. Mesh-morphing tool has been used to generate damaged blades from the datum mesh of blade having tip gap of 0.33% of span. The damage could be classified as either leading edge curl damage or leading edge erosion damage as shown in Fig. 3(A) [1, 3]. The present paper does not consider the leading edge tip curl damage at the rotor tip. Fig. 3(B) and (C) show the mesh around the undamaged blade and leading edge damaged blade respectively. In addition to this, as discussed earlier, three tip gap cases have been considered viz. 0.33% of span, 0.66% of span and 1% of span.

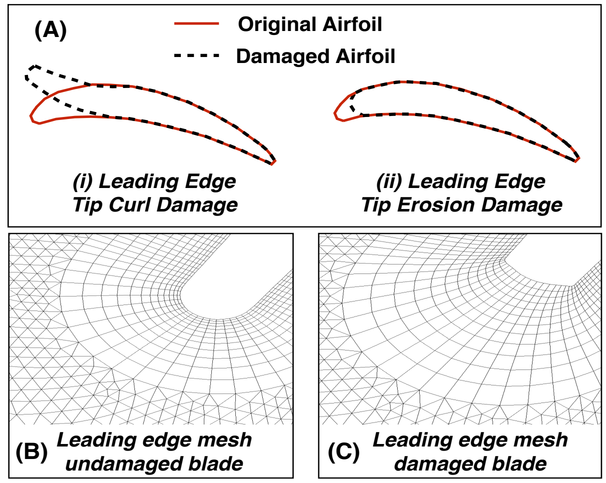

FIGURE 3. UNDAMAGED BLADE VS. DAMAGED BLADE: (**A**) SCHEMATIC OF DIFFERENT LEADING EDGE DAMAGE (**B**) MESH OF LEADING EDGE UNDAMAGED BLADE (**C**) MESH OF LEADING EDGE DAMAGED BLADE

BEHAVIOUR OF DAMAGED BLADES AND BLADES WITH DIFFERENT TIP GAPS

The maximum chord damage that can be acceptable in a civil aviation turbofan engine is between 4-6% of the nominal chord [3]. In fact, NASA stage 35 was tested for erosion of leading edge of the blades [3] with a gradual increase of damage from onwards of 50% span up to the tip of the blade where 5% chord damage was observed. In author's experience of working in an industrial gas turbine industry, blades in the field are inspected regularly according to a preset damage criteria (which is typically less than 10% chord). The current practice is to replace the compressor blades having damage more than the allowable limit. This research focuses on a way to reuse them by rearranging them favourably and therefore the highest damage considered in terms of percentage of chord at the tip is 18% of nominal chord with the damage starting from approximately 65% span. The damage has been represented in terms of a hyperbolic tangent function. This testing of damaged blades of an industrial gas turbine helps in re-certification criteria with a potential for reducing the number of scrapped blades. This section covers the parallels and anomalies between the performance behaviour of damaged blades as well as blades with different tip gaps.

The study of stall margin (*SM*) of damaged blades has been split into two parts : (A) radially increasing damage with same chord-length (B) chord-wise increasing damage with same radial level. The *SM* is defined as follows:

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

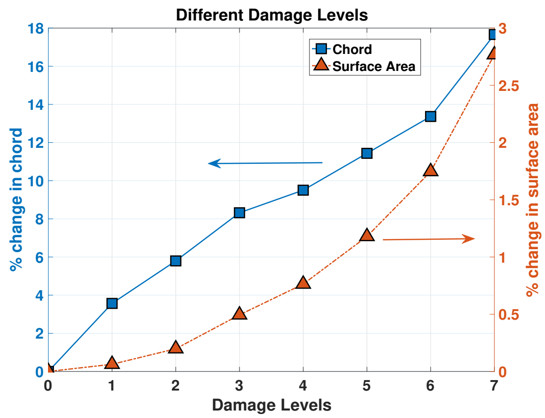

FIGURE 4. DIFFERENT DAMAGE LEVELS IN TERMS OF PERCENTAGE CHANGE IN CHORD AND PERCENTAGE CHANGE IN AREA

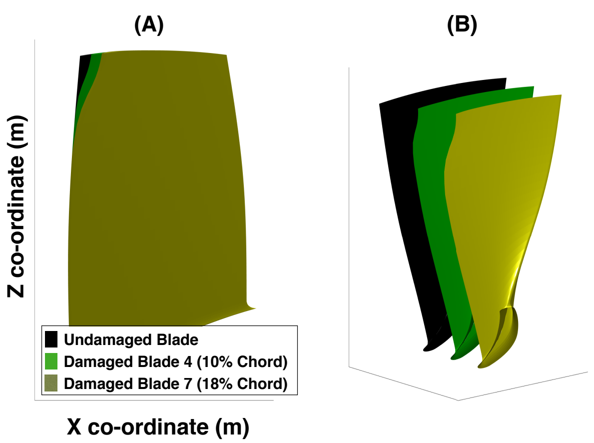

FIGURE 5. DIFFERENT VIEWS OF DAMAGED BLADE: (**A**) TANGENTIAL VIEW (**B**) EXPLODED VIEW

where **PR** is Pressure ratio and $\dot{\mathbf{m}}$ is mass flow rate [6].

All the simulations for testing the stability limits of individual damaged blades is done using single passage simulation unless mentioned otherwise. This study would further the knowledge in identifying the more crucial type of damage. A total of 7 types of damaged compressor blades were generated, namely damage blade 1, 2, 3, 4, 5, 6 and 7. Each of these blades represent increased intensity of damage both radially as well as chord-wise as shown in Fig. 4. The three dimensional visualisation of these damaged blades is shown in Fig. 5 (the scale of X, Y and Z axes are not equal in Fig. 5). Moreover at damage level 4 and damage level 6, additional blades are generated such that same chord-wise damage but different radial level damage as shown in Fig. 6(A). They are divided into 4 categories : "Most damaged, More damaged, Less damaged and Least damaged".

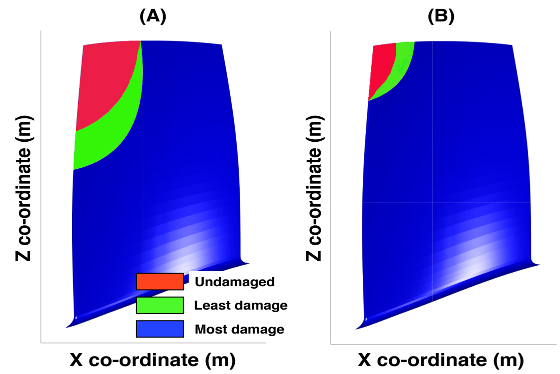

FIGURE 6. SCHEMATIC OF DAMAGE DECOMPOSITION INTO: (**A**) DIFFERENT RADIAL LEVEL DAMAGE. (**B**) DIFFERENT CHORD-WISE DAMAGE.

Two additional damaged blades with varying chord-wise damage are generated at a radial level corresponding to damage 5 (see Fig. 6(B)). These three blades are divided into 3 categories : "Most damaged, Intermediate damaged and Least damaged".

The sensitivity of each of the above two parameters has been studied in terms of stall margin. Also, three blades with different tip gaps viz. 0.33% of span, 0.66% of span and 1% of span have been generated. All the results have been reported in terms of Normalised corrected mass flow (NCMF) and Relative efficiency (RE) where:

$$NCMF = \frac{CMF}{C_1}$$
$$RE = \eta - C_2 \tag{2}$$

In the above equations, C_1 and C_2 are mass flow and efficiency at design point and are used to normalise the parameters.

In the first stage, the computations were performed by assuming all the blades have the same damage (single passage simulation). From Fig. 7, it can be seen that an increase in the damage intensity causes a drop in performance along with the stability margin. As can be seen in the Mach number contour plot shown in Fig. 8, an increasing damage causes a larger zone of low energy fluid to be created due to the interaction of shock-waves and increased tip leakage flow. This low energy fluid produces a region of blockage which creates an adverse pressure gradient causing the blade to stall. Also, the location of shock wave in 18% chord damage case as compared to an undamaged blade makes it more prone to stall [16]. A similar kind of behaviour with respect to the increasing size of tip gaps has been observed in the literature [6, 17]. As seen in Fig. 9, increasing the size of tip gap (single passage simulation) causes the perfor-

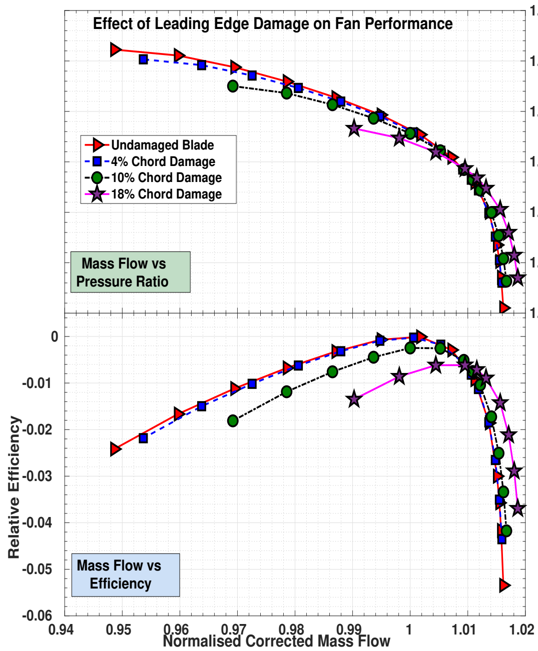

FIGURE 7. EFFECT OF LEADING EDGE DAMAGE ON PERFORMANCE OF DIFFERENT BLADES

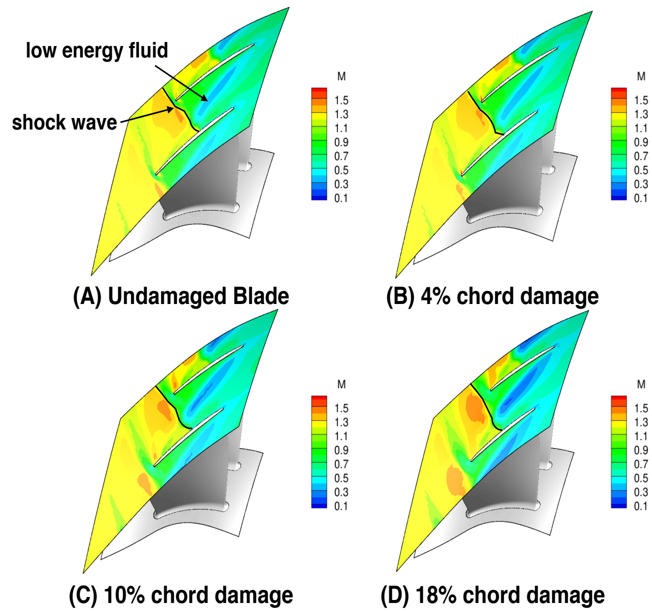

FIGURE 8. MACH NUMBER CONTOURS AT **SAME** NON-DIMENSIONAL CORRECTED MASS FLOW RATE $\approx \mathbf{1}$: (**A**) UNDAMAGED BLADE. (**B**) CHORD DAMAGE OF 4%. (**C**) CHORD DAMAGE OF 10% AND (**D**) CHORD DAMAGE OF 18%

mance and the stability margin to deteriorate similar to the previous study [6]. Also seen in Fig. 9 and Fig. 10, wherein the double passage (which models alternate blades with tip gaps 0.33% of span and 1% of span) has better pressure ratio, efficiency, stalling mass flow rate and *SM* compared to the single passage blade with tip gap 0.66% of span similar to [6].

One major difference between the performance of different single passage cases for damaged blades and blades with different tip gaps as observed in Fig. 7 and Fig. 9 is that of the choked mass flow rate in the two cases. While the choked mass flow rate for different tip gap remains mostly unchanged (also observed in [6]), in case of damaged blades, a rightward shift of choked mass flow occurs for an increase in the damage intensity. Reason for this kind of behaviour is the prevalence of a bigger throat area from approximately 65% span onwards for the higher damage intensity blades. This can be clearly seen in Fig. 11 wherein the choked mass flow rate for 18% chord has higher mass flow rate from 60% span onwards compared to undamaged blade. The increase in annulus area with an increase in the span causes a discernible rightward shift of choked mass flow for higher damage intensity blade.

Having established that higher damage causes higher loss in performance and stability boundaries, the study of the chord-wise damage and radial damage was undertaken. All the calculations in this case is performed using single passage computation of IGV, Rotor and Nozzle. It can be seen from Fig. 12, that the performance curves of the all the radially damaged blade profiles at chord length corresponding to damage level 4 (viz. most damaged blade, more damaged blade, less damaged blade and least damaged blade) look very similar with negligible difference in the efficiency. Consequently, Fig. 13 shows that all these damaged blades have exactly the same *SM*. A similar set of results are obtained when the computation is performed for radially damaged blades at chord length corresponding to damage level 6 (not shown for the sake of brevity). Thus the radial extent of the damage does not have significant influence on performance. On the flip side, it can be seen from Fig. 14 that the chord-wise damage at a constant radial level has impact on the stability of compressor. The pressure ratio and efficiency curves of the most damaged blade follows that of the least damaged blade before stalling prematurely. Consequently, the *SM* plot shows a decreasing trend as the damage intensity increases as seen in Fig. 15. It can thus be surmised that the *SM* is affected majorly by the chord-wise damage compared to radial damage. More importantly, the loss in surface area is a derived quantity resulting from either chord-wise damage and/or radial damage and hence should not be used as a parameter for determining the loss in *SM*.

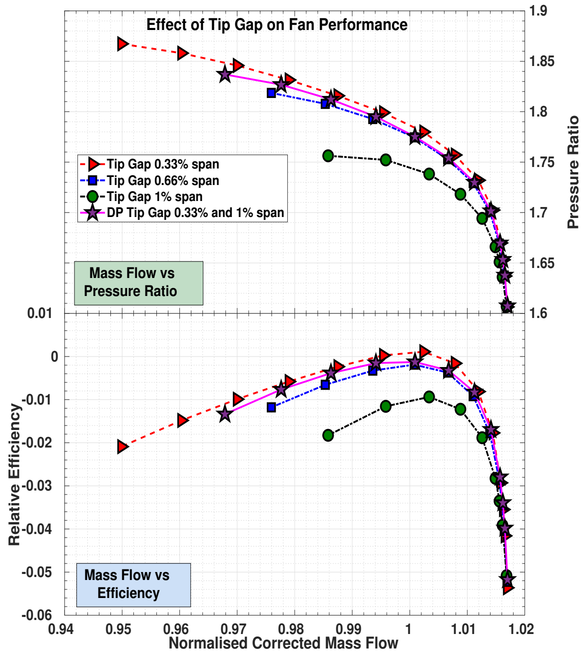

FIGURE 9. PERFORMANCE OF THE BLADES WITH DIFFERENT TIP GAPS

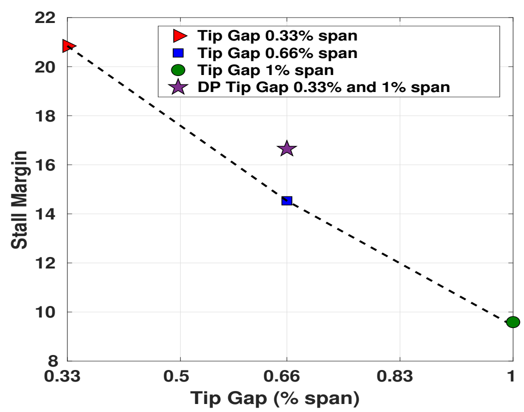

FIGURE 10. STALL MARGIN OF THE BLADES WITH DIFFERENT TIP GAPS

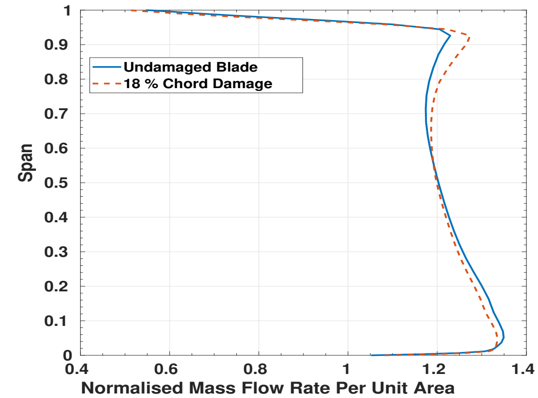

FIGURE 11. SPAN-WISE COMPARISON OF CHOKED MASS FLOW RATE FOR 18% CHORD DAMAGED BLADE AND UNDAMAGED BLADE

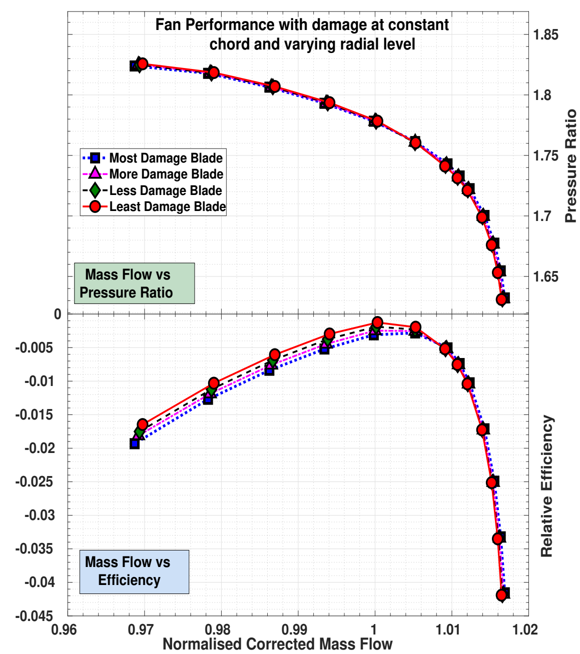

FIGURE 12. EFFECT OF VARYING RADIAL LEVEL DAMAGE WITH CONSTANT CHORD ON STAGE PERFORMANCE

OPTIMISED ARRANGEMENT OF RANDOM DAMAGED BLADES

The problem of optimised arrangement is a subset of combinatorial optimisation which has its genesis in the field of applied mathematics, computer science and management studies; with applications ranging from logistics, vehicle routing and gene synthesis to name a few. Its application to the field of turbo-machinery bedecks it with another layer of complexity : understanding the flow physics. In the earlier study regarding optimised arrangement of random tip gaps [6], it was proved that a zigzag arrangement (largest tip gap sandwiched between two smallest tip gap) works better than any other arrangement. Though the random tip gap was initially introduced as an effect of manufacturing tolerance, after further deliberation it was felt that it could also be considered as a manifestation of in-service degradation. Hence it was argued that the best possible performance from a compressor can be eked out by arranging the blade

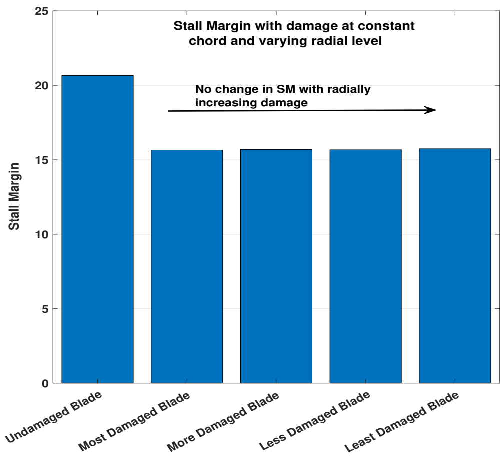

FIGURE 13. EFFECT OF VARYING RADIAL LEVEL DAMAGE WITH CONSTANT CHORD ON STALL MARGIN

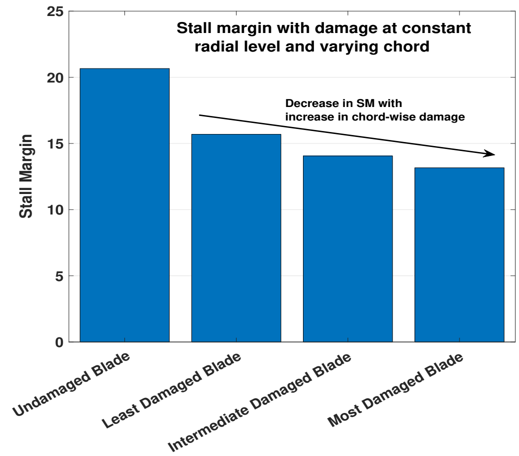

FIGURE 15. EFFECT OF VARYING CHORD-WISE DAMAGE AT A CONSTANT RADIAL LEVEL ON STALL MARGIN

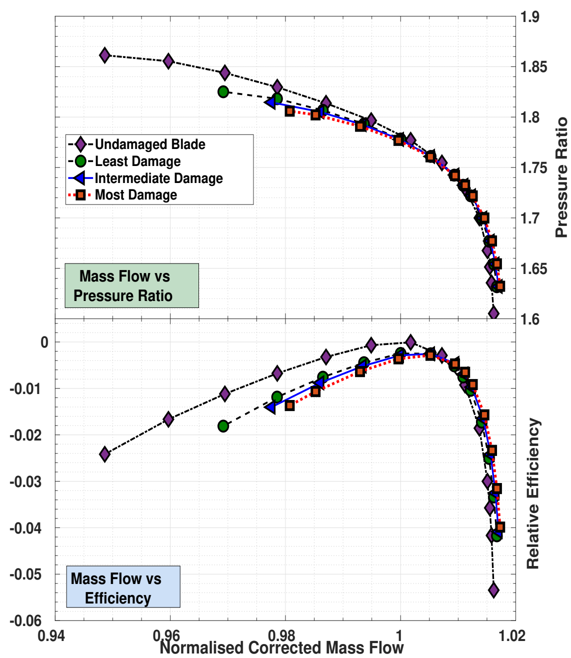

FIGURE 14. EFFECT OF VARYING CHORD-WISE DAMAGE AT A CONSTANT RADIAL LEVEL ON STAGE PERFORMANCE

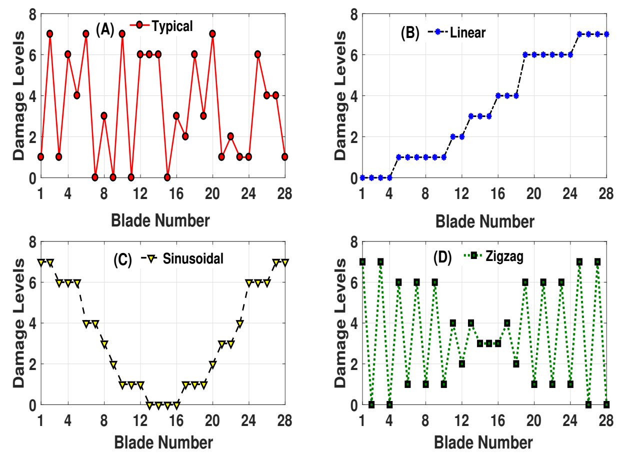

FIGURE 16. DIFFERENT TYPES OF DAMAGED BLADE ARRANGEMENT : (**A**) TYPICAL ARRANGEMENT. (**B**) LINEAR ARRANGEMENT. (**C**) SINUSOIDAL ARRANGEMENT AND (**D**) ZIGZAG ARRANGEMENT

with the worst SM in between the blades with best SM.

In regard to answering the question asked in the first section about the optimal arrangement for a given set of random damaged blades, a set of uniformly distributed random integers between **0** to **7** were generated, where " **0** " represents undamaged blade and " **7** " represents damaged blade 7. These set of damaged blades are then arranged in four different ways viz. random (typical), linear, sinusoidal and zigzag as shown in Fig. 16. This way of studying the problem of combinatorial optimisation was also carried out in previous study [6] wherein the effect of manufacturing tolerance was dealt with. All the calculation in this section was carried out using whole annulus simulation as shown in fig. 1.

As seen in Fig. 17, the zigzag arrangement has better pressure ratio and stability margin compared to rest of the arrangements and represents a maxima for the given set of random damaged blade. on the other hand, sinusoidal arrangement performs the worst and it stalls prematurely. This behaviour of zigzag ar-

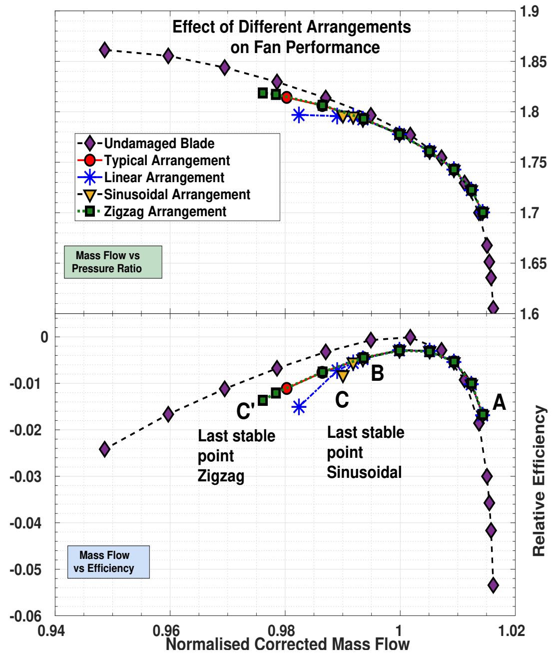

FIGURE 17. EFFECT OF DAMAGED BLADE ARRANGEMENT ON THE STAGE PERFORMANCE

rangement (for the given set of random damaged blades) wherein it gives a maxima performance is consonant with the zigzag arrangement for random tip gaps seen in previous paper [6]. However, it is to be noted that this result is also in complete contradiction to the results obtained by Roberts *et al.* [3] who said that the type of arrangement has no significant difference between the various performances. Roberts *et al.* [3] have considered 3 different arrangements : (1) Alternate nominal chord blade and damaged short chord blade around the annulus (2) Half annulus of nominal chord blades and half annulus of damaged short chord blade (3) Alternating quadrants of nominal chord blades and damaged short chord blades. However, the maximum chord damage that they have considered is 5% which is too small a value to affect the performance significantly. In the previous research [6], a similar study conducted for tip gaps shows that the type of arrangement has negligible effect on the performance if the manufactured blades with tip gaps are very close to nominal value. However, if the manufactured blades with tip gaps are not close to the blade with nominal tip gap, the type of arrangement has a significant impact on the performance. Extending the same argument, herein maximum chord damage of 18% is almost 4 times the value used by Roberts *et al.* [3] and hence a significant deviation in the results. It can be seen from Fig. 7 that the difference in performance between undamaged blade and damaged blade 1 having 4% chord damage is minimal. Usage of these two blades for testing performances of various combinations would

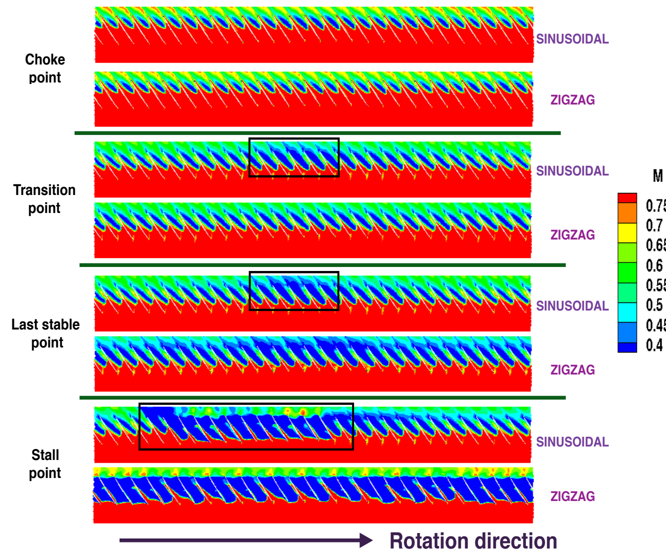

FIGURE 18. FLOW-FIELD AT THE TIP LAYER SINUSOID ARRANGEMENT VS ZIGZAG ARRANGEMENT

draw similar conclusions as that obtained by Roberts *et al.* [3].

It is also important to dwell upon the reason that triggers premature stall in sinusoidal arrangement. It can be seen in Fig. 17 that the performance curve of sinusoidal arrangement follows that of zigzag arrangement from point A (choke side) up to point B (intermediate point) before the sinusoidal arrangement stalls prematurely. Point C and C′ are the last stable points sinusoidal arrangement and zigzag arrangement respectively after which both of them stall. The flow-field at these points is shown in Fig. 18. It can be seen from this plot that at choke point, both the flow fields seem similar. However at the intermediate point, some localised blockage (marked by a rectangle) is introduced in sinusoidal arrangement which is not seen in a zigzag arrangement. This blockage can be seen to grow at the last stable point for sinusoidal arrangement, though the difference in the mass flow rate between intermediate point and last stable point is not huge. At stall point, it can be seen that half of the annulus is blocked with low momentum fluid in a sinusoidal arrangement thereby causing it to stall. On the contrary, the zigzag arrangement can be seen to be undergoing an almost uniform stall inception and hence is able to have better stability margin. The *SM* of each of these four cases is plotted against the different *SP* damaged blade cases in Fig. 19. The average chord damage for the given set of random damaged blades was calculated and the *SM* for all these arrangements were plotted against average chord damage. It can be seen that the *SM* for all the above arrangements is less than the single passage case of blade having the same chord damage. This result was contrary to the expectations. In an earlier study [6], it was shown

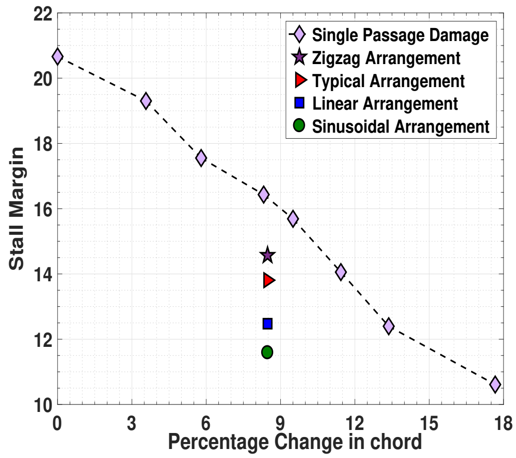

FIGURE 19. STALL MARGIN COMPARISON OF DIFFERENT WHOLE ANNULUS CASES WITH SINGLE PASSAGE DAMAGED BLADES

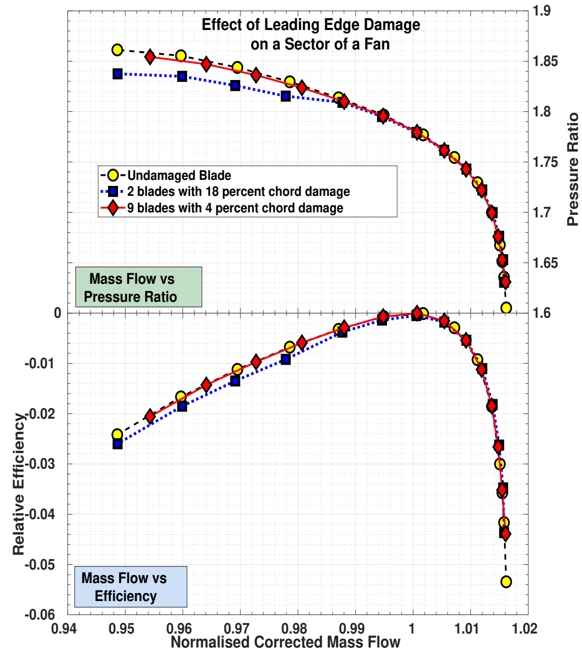

FIGURE 20. EFFECT OF TOTAL DAMAGE IN A SECTOR ON THE FAN PERFORMANCE

that when encountered with blades having random tip gaps, the zigzag arrangement works the best amongst all the arrangements and always performs better than the single passage blade having average tip gap (in the limiting case when manufacturing tolerances are close to nominal, all the arrangements give same performance as the nominal blade). The authors at this point of time have left it as an observation without any conjecture.

Finally, the authors recognise that only a sector of rotor annulus may be damaged due to foreign body impact and all the damaged blades may or may not suffer from the same damage. Different rotors can also have same total damage but with different distribution of damage intensities on damaged blades. On the similar lines of [4], two cases were considered : (**A**) 2 blades with 18% damage (**B**) 9 blades with 4% damage. The intent was to see if there was any difference in the behaviour between the present 1-stage transonic compressor of an industrial gas turbine and the low speed compressor test rig used by Taylor *et al.* [4]. It can be seen from Fig. 20 that case (**A**) and case (**B**) have negligible difference in the stalling mass flow rate though the performance curve of case (**B**) is closer to undamaged rotor blade in the entire range of operation than compared to case (A). This behaviour is different from the low speed compressor of Taylor *et al.* [4] wherein their equivalent of case (**B**) initially follows the performance curve of undamaged blade but stalls earlier compared to case (**A**). This study has been performed using steady RANS computation and therefore the unsteady effects which maybe important has been ignored. Fig. 21 shows the comparison of the performance of Case(A) and Case(B) above plotted against exit mass flow function, which indicates a slight loss in stall margin for case (A).

The authors have also tested a case wherein the damage around the annulus increases progressively. In this study, the number of adjacent blades that undergo damage increases progressively until it covers half of the annulus. It can be seen from Fig. 22 that the stalling mass flow rate for the case with half of the annulus consisting of damaged blades is almost the same as the case where all the blades have undergone same damage. The threshold limit for performance in this case is 6 damaged blades next to each other and the performance is same as the case where all the blades are damaged. Overhauling of the case with half the annulus filled with damaged blades with an intention to eke out the maximum performance and stability results in an alternately arranged undamaged blade and damaged blade. It can be seen from Fig. 22 that the pressure ratio, efficiency and stalling mass flow rate for an alternate arrangement of damaged blades and undamaged blades (zigzag arrangement) is better than the case with half of annulus consisting of damaged blades.

CONCLUSION

The geometric variability arises due to manufacturing tolerance and/or in-service degradation. The effect of damaged blades and tip gap on the performance of a 1-stage transonic compressor used in an industrial gas turbine were studied as parameters of interest. The blade is both chord-wise damaged and radially damaged. It was seen that chord-wise damage was a more crucial parameter compared to radial damage and affects stall margin

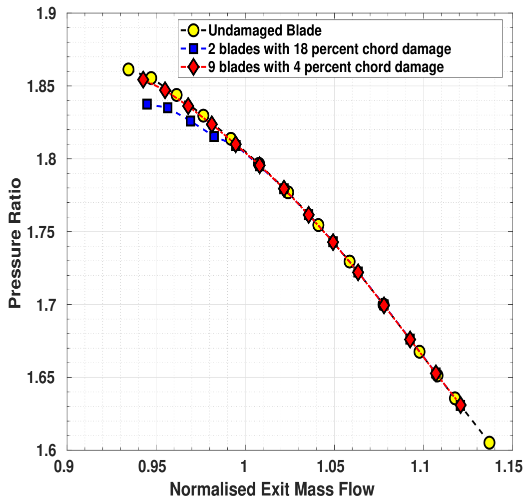

FIGURE 21. EFFECT OF TOTAL DAMAGE IN A SECTOR ON THE FAN PERFORMANCE IN TERMS OF EXIT MASS FLOW FUNCTION

more adversely. It was also noted that loss in surface area should not be used as a parameter for measuring loss of stall margin.

The behaviour of damaged blades and random tip gaps has certain commonalities as well as divergences. Higher chord-wise damage intensity resulted in loss of stall margin as did the presence of blade with larger tip gaps. For a given set of random damaged blades, zigzag arrangement was the best while sinusoidal arrangement was the worst. This study shows that the type of arrangement affects the performance of compressor while the earlier study with less damage showed no effect. This work quantifies the threshold at which damage becomes influential.

The choked mass flow rate shifts towards the right in case of higher damaged blades though no such behaviour was observed for blades with differing tip gaps wherein the choked mass flow rate remains largely unchanged. In case of blades with differing tip gaps, the zigzag arrangement of random tip gaps was found to give better performance than the single passage case of blade having average tip gap value. On the contrary, the zigzag arrangement of random damaged blades produced performance which was less than the single passage case of blade having mean damage value. Furthermore in case of sectoral damage for two rotors with same total damage, it was observed that rotor having more blades with smaller damage intensity fared better than the case with rotor having less number blades with larger damage intensity. Also, when increasing number of adjacent blades get damaged, there was an overall deterioration in the performance. It was also seen that the alternate arrangement of undamaged blade and damaged blade provided better pressure ratio, efficiency and stability compared to the case where all the damaged blades were concentrated in one half of the annulus.

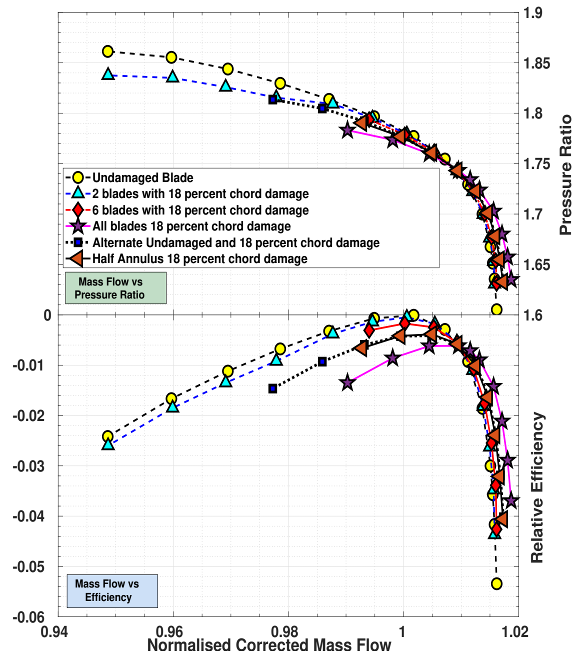

FIGURE 22. EFFECT OF NUMBER OF DAMAGED BLADES ON PERFORMANCE

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