Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45187

THE IMPACT OF AN ANTI-STALL STABILISATION RING ON INDUSTRIAL FAN PERFORMANCE: IMPLICATIONS FOR FAN SELECTION

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ABSTRACT

This paper describes work aimed at establishing the ability of an industrial fan to operate in the event of stall, with and without an anti-stall "stabilisation ring" fitted to the fan casing. The research establishes the aerodynamic characteristics of a 2.24 meter diameter 190 kW fan with and without the stabilisation ring. It also measures the fan's peak stress in both stable and stalled conditions, thus facilitating discussion of the implications of the experimental results for industrial fan design methodology. The paper then presents three different strategies for industrial fan selection in applications where the selected fan will most likely stall. The first strategy selects a fan with a low blade angle that is "none stalling". The second strategy selects a fan with a high pressure developing capability. The third strategy selects a fan with a fitted stabilisation ring. System designers each have their favoured industrial fan selection strategy. However, all three strategies can produce a system design within which an industrial fan performs reliably in-service. The paper considers the advantages and disadvantages of each fan selection strategy and concludes with a brief assessment of the strengths and weaknesses of each.

KEYWORDS

Anti-stall device, stall suppression, fan performance stabilisation, fan selection strategy, fan mechanical failure.

INTRODUCTION

Aerodynamic instability when throttling flow rate constrains the operational envelope of fans, blowers and compressors. To avoid instability, aerodynamicists must provide adequate stability (stall) margin to accommodate inlet distortions, degradation due to wear, throttle transients, and other factors that reduce fan, blower and compressor stability from the original design base-line. Assuring adequate stall margin is a critical consideration during the design process.

Aerodynamic stall results in an increase in alternating aerodynamic loads. Industrial fan designers classically produce a mechanical design that can withstand the alternating loads imposed on the fan blades associated with stall, and therefore mechanical failure in the event that stall is not instantaneous. Aluminium is both low cost and light weight, and consequently the industrial fan designers' preferred choice of blade material. A weakness of aluminium as a structural material is its propensity to fail in fatigue. As such, fan blades that do not instantaneously fail as a fan stalls routinely fail in fatigue as a consequence of the higher alternating induced blade stress when the fan operates in a stalled condition.

This paper studies the performance of a class of anti-stall devices. First reported in the literature by Bard [1], the studied anti-stall system comprises a "stabilisation ring" around the casing immediately up-stream from the fan blades. The stabilisation ring modifies the fans' aerodynamic characteristic as it produces a continuously rising fan characteristic back to zero flow.

This paper starts with a brief literature review relating to fan, blower and compressor stall before moving on to review the anti-stall concepts that other scholars have developed in their attempts to improve axial decelerating turbomachinery aerodynamic stability. The paper then presents the methodology industrial fan designers classically use to ensure that a fan does not suffer a mechanical failure during normal operation. The paper then examines industrial fan performance with and without a fitted stabilisation ring as the fan is driven into stall. Placing strain gauges in the location of the fan blades' peak stress establishes the mechanical impact of driving a fan into stall with and without a fitted stabilisation ring. The paper then moves on to consider the implications of the strain gauge results for fan design, identifying the circumstances under which a fan fitted with a stabilisation ring can operate in the stalled condition without risk of a mechanical failure. The paper concludes with a brief assessment of the alternatives to a stabilisation ring, and their impact on fan performance, capital and through-life cost.

CONTROL OF STALL

The phenomenon of stall

Scholars have examined the detection and analysis of different forms of aerodynamic instability since the 1950's. According to Gravdahl & Egeland [2], two main types of aerodynamic flow instability exist in compressors: (i) 'rotating stall' (in which regions of reversed flow occur locally); and (ii) 'surge' (which is characterised by periodic backflow over the entire annulus involving violent oscillations in the compression system).

The first of these, 'rotating stall', is a mechanism by which the rotor adapts to a reduction in flow rate, which results in circumferentially non-uniform flow patterns rotating in the annulus. Researchers have studied the problem of rotating stall in axial flow compressors in multi-stage machines (Greitzer [3], Day & Cumpsty [4] and Moore [5]). The earlier work of Emmons *et al.* [6] was one of the first attempts to describe the mechanism underlying the propagation of rotating stall. In reviewing the evolution of rotating stall, Cumpsty [7] noted that a drop in overall performance can occur as either a 'progressive stall' or an 'abrupt stall'. Engineers usually associate the former with a part-span stall which results in a small performance reduction, whereas they associate the latter with a full-span stall and a large reduction in performance. Notably, the part-span rotating stall typically occurs in single blade rows [7] and usually leads to more complex disturbances in single-rotor or stage machines than in multi-stage compressors [5].

The family of industrial fans under scrutiny have been the subject of recent experimental investigation (Bianchi *et al.* [8], Sheard *et al.* [9]). The investigation focused on the stall modes, identifying a rotating stall inception mechanism driven by circumferentially localised pressure disturbances confined to the blade passage's tip region. Localisation of the disturbances in the blade tip region supports the hypothesis of a causal link between tip clearance flow and stall inception.

Some scholars have focused on the physics underlying the tip clearance flow related mechanisms that can lead to the formation of pressure disturbances. Among them, Koch & Smith [10] and Saathoff & Stark [11] have observed experimentally that a fan reaches the limit of its pressure developing capability when the interface between the incoming flow and tip clearance vortex region lined-up with the leading edge plane at the blade tip. Numerical simulation has revealed two more mechanisms related to the onset of tip blockage growth (Khalid *et al.* [12] and Vo *et al.* [13]), namely: (i.) the backflow of tip clearance fluid at the trailing edge impinging on the blade pressure side of adjacent blades and (ii.) the spillage of tip clearance fluid ahead of the blade leading edge below the blade tip into the next blade passage.

All forms of instability place increased mechanical stress on the rotors which can, in turn, eventually lead to mechanical failure. Strain gauge measurements on axial compressors (Rippl [14]) have reported bending stress in vanes exceeding stable operation by a factor of five under 'rotating stall' conditions. This increase in bending stress leads to the blades' rapid fatigue failure. In contrast, a 'surge' can lead to an increase in bending stress sufficient to cause a mechanical failure during the surge event itself.

Scholars who have studied instability in fans and compressors have suggested that some features of the tip flow of both are directly responsible for the generation of short wave-length disturbances (also called 'spikes' or 'pips') that cause the inception of localised part-span stall cells (Camp & Day [15], Deppe *et al.* [16], Bright *et al.* [17]). Researchers have correlated this inception mechanism with a modification of the acoustic emissions from the casing region. Recently they have proposed a stall warning concept based on the acoustic signal visualisation (Sheard *et al.* [18]).

HISTORICAL OVERVIEW OF ANTI-STALL CONCEPTS

Given the potentially catastrophic consequences of a stall event, there is an incentive for developing technologies that can extend the stable operating range of axially decelerating turbomachinery without undue performance degradation. In 2007, Hathaway [19] systematically reviewed techniques and design concepts to improve the stall-free operating margin or to suppress a stall event.

Hathaway noted the earliest proposed techniques from the 1950's that had successfully extended the axial compressor's stable operating range. In 1950 on behalf of Rolls-Royce Ltd Wilde [20], and in 1955 Turner [21] on behalf of Power Jets Ltd filed patents. These concepts were both based on the treatment of the casing end-walls motivated by a desire to control the boundary layer development by combining rear air bleeding and front re-injection [20] or the use of holes and slots as a method of promoting turbulence, and in so doing, energising the end-wall flow [21].

Griffin & Smith [22] conducted the first systematic experimental campaign on so-called "porous end-walls" in compressor rotors during the 1960's at NASA. Their work demonstrated improved stall margins in cascade tunnels irrespective to the air blowing/bleeding. In a similar vein, scholars studied casing treatments, specifically holes, slots and grooves with and without plena in the 1970's (Bailey & Voit [23], Wisler & Hilvers [24] and Prince *et al.* [25]). They found their effectiveness primarily associated with delaying the onset of stall in tip-limited blade rows.

Takata & Tsukuda [26] conducted detailed measurements within casing slots, and found that they achieved their anti-stall effect as a consequence of periodic pumping of the flow within the slots into the main stream. Moreover, Greitzer *et al*'s [27] investigations demonstrated that the end-wall treatments were effective mostly in high-solidity blade rows prone to wall-stall, but not in low-solidity rotor affected by blade-stall.

In the 1990's, researchers proposed concepts to exploit the potential benefits of flow bleeding (from the stalled region) and blowing (into the clean inflow), and in so doing, revisited earlier stall inception concepts. Most notably, Koff *et al.* [28], Nolcheff [29], Kahlid [30] and Glemedov *et al.* [31] patented different variants of re-circulating casing treatment. A common theme with the different re-circulating casing treatments was the provision of a path through the casing for low momentum fluid re-circulating upstream from the blade tip leading edge.

More recently, Hathaway [19] observed that the most significant advances in anti-stall devices have resulted as a consequence of insight into the flow mechanism that researchers associate with three specific technologies: first, circumferential grooves; second, tip injection control technology; and third, stage re-circulation devices. Fan and blower designers most favour the stage re-circulation devices.

PASSIVE CONTROL BY STABILISATION RINGS

Since the early 1960's scholars have endeavoured to develop stage re-circulation devices tailored to the pressure rise and volume flow rate ranges typical of industrial fans, Ivanov [32]. The concept is of an annular "slit" in the casing upstream of the blades (labelled "a" in Figure 1) that stabilises fan performance as it approaches stall.

The slit enabled air to recirculate (through the slit formed between the annular casing labelled 2 and ring labelled 3, Figure 1). A set of static vanes (labelled 7, Figure 1) then re-directed the re-circulating flow in an axial direction as it turns back and re-enters the fan impeller (labelled 4 and 5, Figure 1). As the fan approaches stall, the slit and static vanes provide a path for low momentum flow to recirculate. In practice, this stabilises fan performance.

Karlsson & Holmkvist [33] subsequently developed and enhanced Ivanov's [32] patent by incorporating static vanes into the casing, Figure 2. Then Bard [1] named the vanes embedded within the fan casing a "stabilisation ring". Miyake & Inaba [34] further developed and patented the original concept proposing the use of air-separators based on an open circumferential cavity facing the rotor-blade tips, which Yamaguchi *et al.* [35] further developed. Similarly, Kang *et al.* [36] optimised casing recess geometries and their relative position to the blade rows. Despite subsequent developments to the concept, the industrial fan community has mostly adopted Karlsson & Holmkvist's [33] configuration.

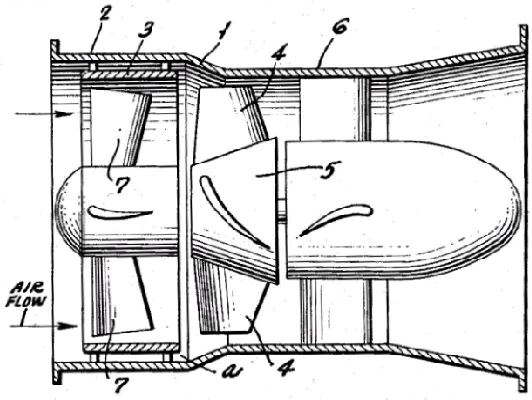


Figure 1 The proposed "blower arrangement", from Ivanov [32].

In practice the concept has proven highly effective, with the stabilisation ring guide vanes removing the momentum component both radially and circumferentially and reinjects the flow in the axial direction. The flow through the stabilisation ring vanes is turned such that it exits the vanes upstream from the impeller, reenergised and flowing in an axial direction.

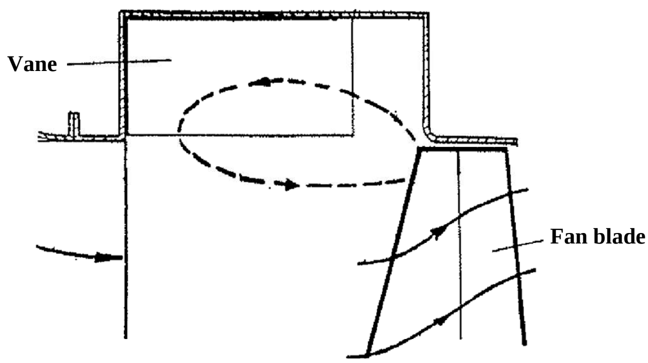


Figure 2 The proposed "stabilisation ring" arrangement adapted from Karlsson & Holmkvist [33] by Bard [1].

The effect of the stabilisation ring on the fan characteristic is to eliminate the sharp drop in its pressure developing capability, which engineers classically associate with fan stall. The primary characteristic of an industrial fan fitted with a stabilisation ring is continuously rising pressure back to zero flow. It was this modification in the fan characteristic that led to industrial fan designers widely embracing the use of stabilisation rings.

A continuously rising characteristic facilitates multiple fan operation in parallel. As a fans speed falls, its pressure developing capability also falls. During a fan's starting and stopping transient, its pressure developing capability will be below what other fans generate when operating in parallel. As a consequence, a fan in parallel operation will inevitably drive transiently into stall each time it starts or stops. During the 1980's, variable speed drives were not widely available.

Therefore varying the speed of all fans in a parallel installation was not practical, making it inevitable that individual fans would have to start and stop, whilst others ran at full speed.

The ability of the stabilisation ring to facilitate the starting and stopping of individual fans when in parallel operation was critically important. Application of the stabilisation ring largely eliminated in-service mechanical failure in industrial fan parallel operation.

INDUSTRIAL FAN DESIGN METHODOLOGY

The term 'fatigue' refers to the phenomenon whereby virtually all materials will break under numerous stress repetitions that are not sufficient to produce an immediate rupture in the first instance. In this regard, fan blades are subject to fatigue stress induced by: (i) the mean force arising from rotation and aerodynamic loading and (ii) the alternating force produced by variations in lift as the fan rotates. This combination of mean and alternating forces on the blade result in mean and alternating stress in the blade, and making them inherently susceptible to fatigue.

The endurance limit corresponding to any given range of stress variation has been the subject of extensive study (reviewed by amongst others Young [37]), as has the ability of a wide range of materials to withstand different combinations of mean and alternating stress. Industrial fan blades are usually manufactured from aluminium and the ability of aluminium to resist fatigue for a fixed alternating stress reduces as mean stress increases. When researchers study material test data for various levels of mean and alternating stress, they derive a relationship known as the Gerber Line (See Figure 3).

Gerber [38] himself derived this line and proposed a parabolic relationship between alternating stress and mean stress in iron structures. The maximum alternating stress level σ for any mean stress in the material, up to the tensile strength of the material, is given by the expression:

$$\pm\sigma = \pm\sigma_0 \left\{ 1 - \left(\frac{\sigma_m}{\sigma_t} \right)^2 \right\}$$

σ_0 = Alternating stress level that constitutes the fatigue limit of the material with zero mean stress.
σ_t = Tensile strength of the material.
σ_m = Mean stress in the material.

The ability of a given aluminium alloy to resist the effect of mean and alternating stress is dependent on the maximum defect size in the material samples. The larger the defect, the lower the level of mean and alternating stress required to induce fatigue failure. Industrial fan manufacturers, therefore, first experimentally establish the relationship between mean and alternating stress for a given defect size, and then undertake X-ray examination of all rotating components to ensure that the maximum defect size is below that on which they established the Gerber Line. Engineers usually analyse a new fan blade design using finite-element methodology to establish the location of the peak mean and alternating stress.

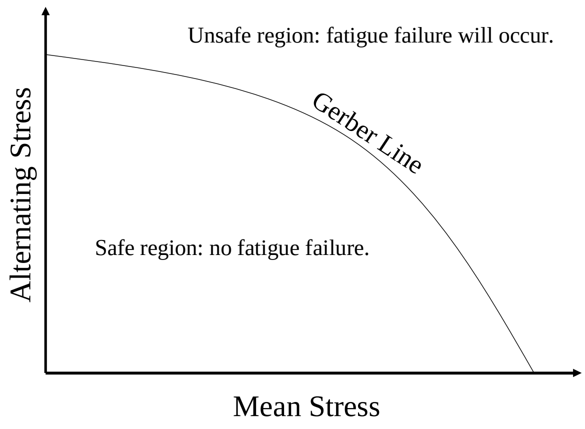


Figure 3 The curve of best fit through material test data is referred to as the Gerber Line.

If the peak mean / alternating stress point is below the Gerber Line, the fan blade should not fail due to fatigue. However, in practice, there is some uncertainty about the location of all Gerber Lines as they are derived from experimental data. Additionally, the ability to calculate mean and alternating stress levels is imperfect as a consequence of assumptions during the modelling process. Therefore, in practice, industrial fan designers classically choose to design fans with a safety factor of two (See Figure 4).

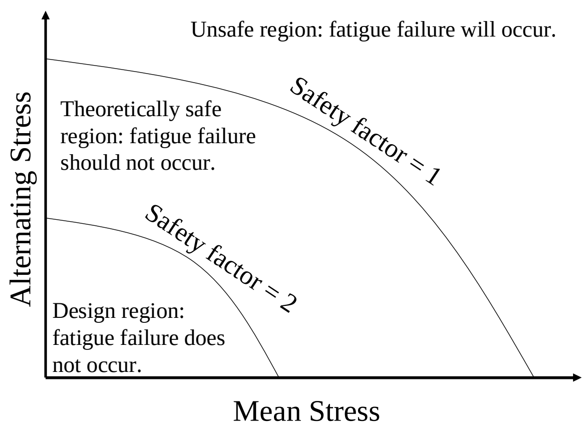


Figure 4 The Gerber Line associated with material test data, and the Gerber line with a safety factor of two.

Fan design with a safety factor of two is conservative and generally results in reliable in-service operation. However, if the fan stalls, alternating stress levels induced within the fan blades as a consequence of the aerodynamic loads associated with operating in the stalled condition will increase. Although the alternating stress levels associated with operating in a stalled condition do not classically result in an instantaneous mechanical failure, they are high enough to result in the safety factor falling on a Gerber Line with a safety factor of less than one, and consequently, a fatigue failure will follow.

THE TEST FACILITY

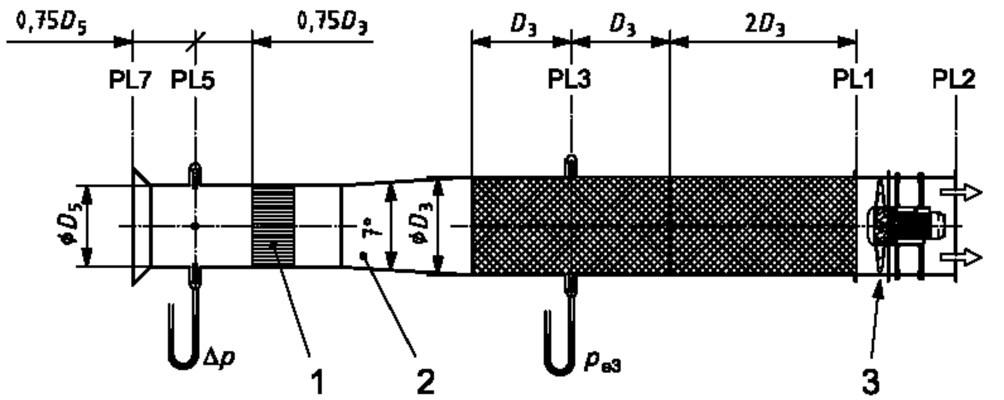
In order to establish the likelihood of mechanical failure of an industrial fan in stall, with and without a fitted stabilisation ring, the researchers selected a fan typical of industrial applications as Table 1 below illustrates.

Table 1: Fan data

Nominal speed	980 rpm
Tip speed	115 m/s
Nominal pressure coefficient, Ψ_{nom}	0.189
Nominal flow coefficient, Φ_{nom}	0.220
Duty point efficiency, η_{tot}	0.69
Tip diameter	2,240 mm
Blade height	720 mm
Blade chord at the tip	163 mm
Tip stagger angle	70°
Tip gap (% of fan diameter)	0.45%
Blade count	16
Tip solidity	0.37

The researchers operated the fan in a custom-built casing made from cast and machined steel rings. The blade sections, of modified ARA-D type, were single-parameter airfoils originally designed for propeller applications. The impeller had a mechanism for varying the blade-pitch angle to enable the researchers to study fan performance over a range of flow and pressure. The researchers used a 222 kW, direct coupled-induction 400-volt (AC), 3-phase motor to drive the rotor at a constant speed of 980 rpm. Manufactured as part of a contract for the London Underground, the fan for the current study, Figure 5, is designed for both routine operation at ambient temperature providing cooling air and once only emergency operation for two hours at 300^0C.

Figure 5 Casing for the 2.24 meter fan with the stabilisation ring (top) undergoing final inspection and the fan installed in a test duct system (bottom).



Key

1 flow straightener (cell type shown)

2 transition section — the straight section with $\alpha = 0$ and length $\geqslant 3D$ can also be used, i.e. $D_5 = D_3$ test installation

3 test fan (tube-axial type shown)

Figure 6 Experimental arrangement utilised for fan testing, one of the allowable test methods in ISO 5801: 2007, page 157.

The manufacturers certified the fan for once only emergency operation for two hours at 300^0C in accordance with the Euro Norm EN 12101-3 [39] and International Standard ISO 21927-3 [40]. The researchers installed the fan in a ducted test system, Figure 6. They aerodynamically optimised the bell-mouth's shape to provide uniform and unseparated flow into the fan. They measured the fan performance according to the International Standard ISO 5801:2007 [41].

By throttling flow downstream from the fan rotor, the aerodynamic instabilities of interest were induced in the fan. During the flow / pressure throttling, the fan remained in rotating stall without going into surge, irrespective of the rotor speed. The rotor aerodynamic load and the plenum geometry ensured that the system could not develop a counter-pressure high enough to induce a surge.

EXPERIMENTAL RESULTS

The researchers developed a solid model of the test fan's blade using a CAD package, and they used a finite element analysis programme for structural analysis. They applied centrifugal force and bending moments (due to the design radial work distribution) using Sheard *et al's* [42] original method using nodal forces in the finite element analysis boundary conditions in order to calculate blade stress.

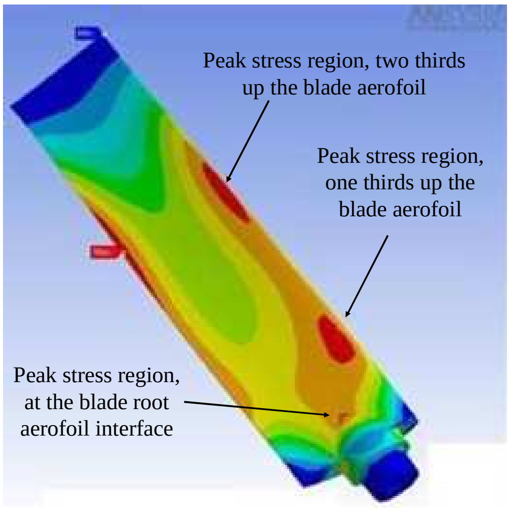


Figure 7 An example of finite element analysis predicted stress levels in a typical fan blade, from Sheard *et al.* [42].

The predicted stress field, Figure 7, illustrate peak stress levels at approximately one third and two thirds blade aerofoil length, plus at the blend between the root section and aerofoil. The peak stress levels at one third and two thirds blade height occur as a consequence of Sheard *et al's* [42] blade design methodology that switches from one optimisation technique to another at one third and two thirds blade height. The peak stress level at the blend between the root section and aerofoil occurs as a consequence of the relatively stiff root section blending into a relatively flexible aerofoil.

When operating a fan blade design using Sheard *et al's* [42] methodology in stall long enough for a blade fatigue failure to occur, the failure can take place at either the root aerofoil interface, one third or two thirds along the aerofoil. The top third of the aerofoil and the top two thirds of the aerofoil each have a separate and distinct first bending natural frequency. If the stalled fan flow aerodynamic forces happen to excite one of these natural frequencies, then there will be a mechanical failure at either one third or two thirds of the aerofoil, depending on which natural frequency is excited.

In practice a failure at one third or two thirds or the aerofoil is unusual. A stalled fan flow generates turbulence across a broad spectrum, exciting all natural frequencies, including the aerofoil first bending mode. The first bending mode classically results in the aerofoil failing mechanically at the root aerofoil interface, Figure 8.

Figure 8 An example of a fan blade with a mechanical failure of the blade at the root aerofoil interface.

In practice over 90% of the in-service blade failures that occur when a fan operates in stall for an extended period of time are blade root aerofoil failures. The industry widely regards this root aerofoil failure as the "classical" failure mode that it associates with operating a fan in stall. As previously mentioned, Sheard *et al's* [42] design methodology results in three high stress locations on the blade, any one susceptible to peak stress when operating a fan in stall.

The authors applied strain gauges to three blades in the three locations that they predicted as the blades' high stress regions. Application of multiple strain gauges to numerous blades enabled the authors to experimentally determine the actual highest stress location, as well as the impact of manufacturing tolerances from blade to blade. The variation in strain gauge output from blade to blade at nominally the same location on different blades was 2 – 3%. This variation constitutes a combination of errors associated with gauge calibration, uncertainly in gauge location plus blade to blade variation of blade geometry.

Using data from a typical strain gauge located at the highest stress position on one blade, the researchers established fan performance with and without a fitted stabilisation ring as Figure 9 illustrates. When throttling the fan without a fitted stabilisation ring, pressure rises until it reaches a peak, and then falls as the fan stalls. This is the classical fan characteristic. In the study, the blade's peak alternating stress increased from 2.27 MPa (Point A, Figure 9) to 3.53 MPa as the researchers throttled the fan. As the fan stalled, peak alternating stress increased to 16.00 MPa (Point B, Figure 9). An increase in alternating stress from 2.27 MPa to 16.00 MPa is a 7.05 time increase.

When the researchers throttled the fan with a stabilisation ring, pressure rose continuously with no evidence of a reduction in pressure developing capability as the fan passed through the point at which it stalled without a stabilisation ring (See Figure 9). The fan characteristic is remarkable in that the pressure rises so smoothly that it is barely possible to identify the onset of stall from the fan's flow / pressure characteristic. However, in studying the strain gauge data, it is apparent that the initial alternating stress level is 2.13 MPa (Point C, Figure 9), and remains lower then a fan without a stabilisation ring until the onset of stall. As the fan fitted with a stabilisation ring approaches stall, there is a single point (at 60 m^3/s) where the data from the fan without a stabilisation ring measures lower stress than the fan with a stabilisation ring. Alternating stress in the fan with stabilisation ring goes on to peak at 4.60 MPa (Point D, Figure 9).

The increase in alternating stress from 2.13 MPa to 4.60 MPa is a 2.16 time increase in alternating stress. Compared to the 7.05 increase in alternating stress with the same fan when not fitted with a stabilisation ring, an increase of 2.16 is small; however, it is potentially significant given that industrial fan designers classically use a safety factor of two when mechanically designing the fan.

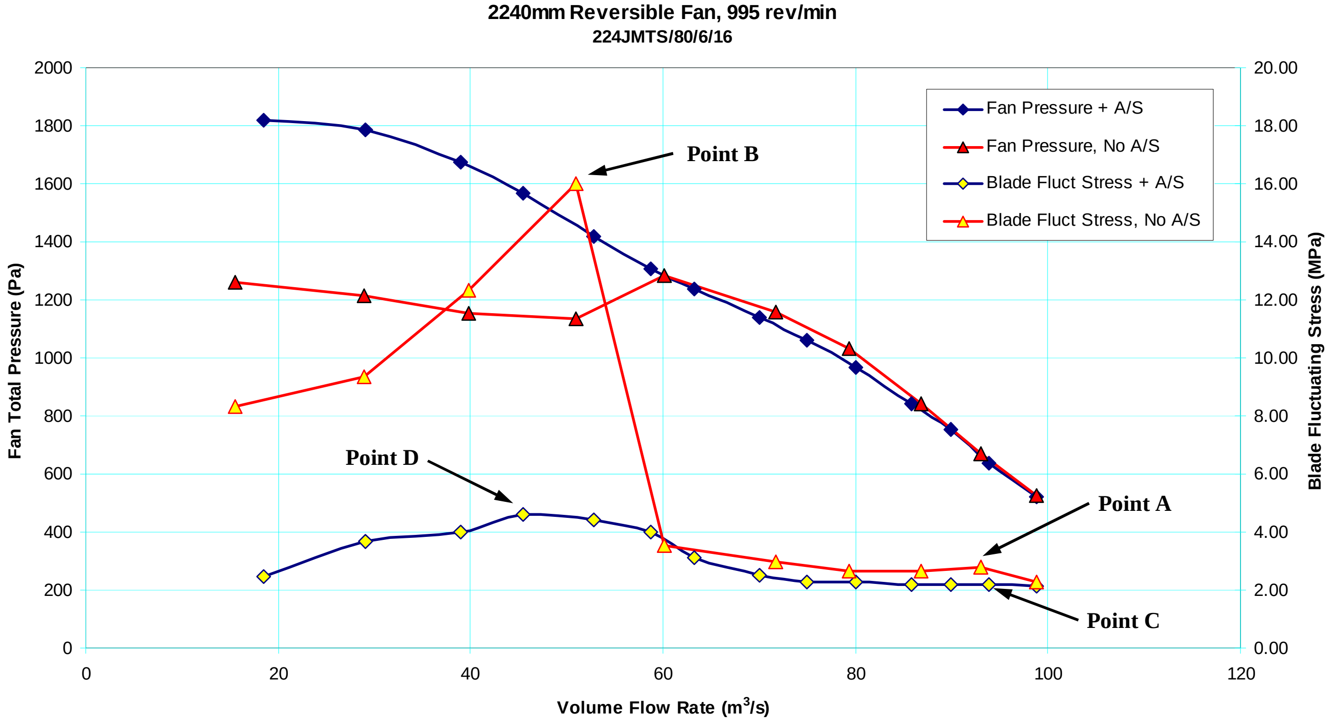


Figure 9 Stall characteristics of the test fan with and without a fitted stabilisation ring.

STRUCTURAL ANALYSIS

The authors assessed the significance of the alternating stress results both with and without a fitted stabilisation ring within the context of impact on the fan's mechanical ability to operate in the aerodynamically stalled condition. The manufacturers designed the fan that the authors used in this study with predicted direct and alternating stress levels that would fall on a Gerber Line calculated with a safety factor of two. The actual direct and measured alternating stress levels for the fan with no fitted stabilisation ring resulted in the combination of direct and alternating stress actually falling on a Gerber Line slightly below two (point A in Figure 10). This result gives confidence in the accuracy and conservatism of the manufacturers' mechanical design methodology.

Throttling the fan until stall without a stabilisation ring resulted in a 7.05 time increase in alternating stress. When plotted (point B in Figure 10), the peak alternating stress with the fan's operating in stall, the combination of direct and alternating stress is significantly beyond the Gerber Line with a safety factor of one. From this we may conclude that if this fan operated in the stalled condition for an extended period of time, it would suffer a fatigue related failure.

The actual fatigue failure mechanism observed in-service is a classical combination of low and high cycle fatigue. Operation of the fan in the stalled condition induces a low cycle fatigue crack localised around a defect in the blade, following which this crack propagates under the action of high cycle fatigue. This failure mechanism is significant in that if a fan operates briefly in stall, it may be enough to initiate a low cycle fatigue crack, but not actually propagate that crack far enough for a mechanical failure to occur during the stall event itself. Once created, however, the crack propagates under the action of high cycle fatigue during routine fan operation in a non-stalled condition. Mechanical failure, therefore, classically occurs some time after the fan operates in stall, and therefore, the fan apparently fails during routine operation. However, the stall event initiated the failure.

The researchers plotted the predicted direct and measured alternating stress results for the test fan with the fitted stabilisation ring as Figure 11 illustrates.

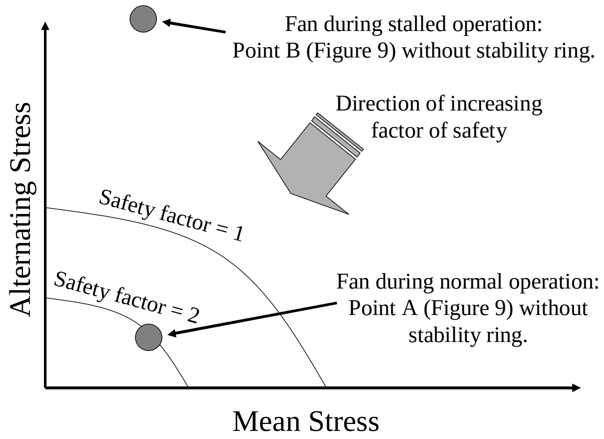


Figure 10 Direct and experimentally measured alternating stress during normal and stalled operation for the test fan with no stabilisation ring.

The reduction in alternating stress during non-stalled operation resulted in the fan operating with a slightly larger safety factor, slightly further into the safe operating region, Point C in Figure 11. In stalled operation the alternating stress increased by a factor of 2.16 and in so doing lifts the direct / alternating stress points, Point D in Figure 11, into a Gerber Line with a safety factor of approximately 1.5. As a safety factor of 1.5 is greater than one, the mechanical design of the tested fan can tolerate a 2.16 increase in alternating stress, and therefore, the risk of a fatigue induced mechanical fan failure is low.

The experimental results are significant in that they provide insight into a likely reason for industrial fans' in-service failure. In this study, the alternating stress level with fan operation in stall with a stabilisation ring resulted in a 2.16 increase in alternating stress. This increase is significant within the context of a smoothly rising fan characteristic that provided little indication that alternating stress had increased.

The smoothly rising fan characteristic resulted in Bard [1] claiming that "unstable performance due to stalling is completely eliminated." As Bard [1] conducted a purely aerodynamic programme, making no calculation of steady stress or measurement of alternating stress, we may assume that when Bard [1] referred to "unstable performance" he was referring to unstable aerodynamic performance. However, the claim may have resulted in some fan designers assuming that mechanical stress would also remain stable. It is incorrect to assume that mechanical stress will remain stable as a fan moved from its aerodynamically stable to stalled condition as Figure 11 indicates. Alternating stress rises by a factor of 2.16, significant enough for the designer to take this into account during fan mechanical design, development and validation.

STRATEGY FOR FAN SELECTION

Industrial fans have a multitude of applications, and therefore any generalisation about the correct strategy for fan selection is challenging.

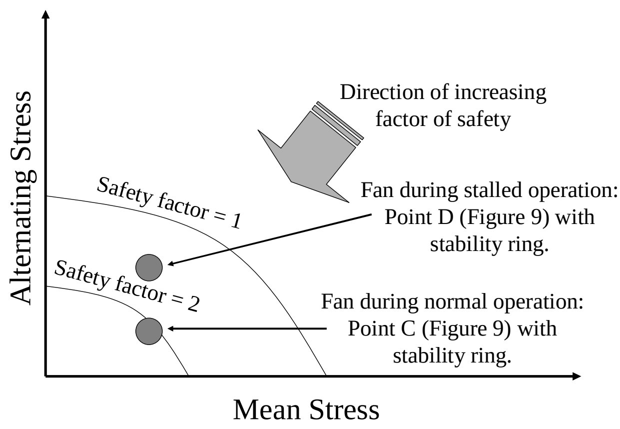


Figure 11 Direct and experimentally measured alternating stress during normal and stalled operation for the test fan with a stabilisation ring.

There is, however, a specific instance where fan stall should form the basis of a strategy for fan selection: tunnel ventilation fans for mass transit systems. The demand for new mass-transit systems generally, and metro systems in urban areas specifically, has increased rapidly over the last two decades and continues today. However, recent changes in market requirements for tunnel ventilation fans present fan designers with a challenge. The proposed tunnels for the next generation of mass transit systems are longer than the historic norm, and the trains that run in them are to run faster than the historic norm. These two factor result in mass transit systems requiring higher pressure ventilation fans (as the tunnels are longer) with the capability of operating under the influence of larger pressure pulses (as the trains are running faster).

A third factor increasing the magnitude of pressure pulses in metro systems is the trend towards the use of platform screen doors as Figure 12 illustrates. Platform screen doors at metro stations screen the platform from the train. They are a relatively new addition to metro systems, and are today in wide use in

Asia and Europe. In 1998, the Tung Chung Line and Airport Express saw the earliest operations of platform screen doors in Hong Kong. In 2000, the MTR Corporation began a six year programme to add 2,960 pairs of platform screen doors at 30 underground stations on the Hong Kong Kwun Tong Line, Tsuen Wan Line, and Island Line. In South America the São Paulo Metro currently has three stations with platform screen doors: Sacomã station (Line 2 - Green) opened on January 30, 2010 and the São Paulo Metro Faria Lima and Paulista stations (Line 4 - Yellow) opened on May 25, 2010. All future São Paulo Metro stations will have platform screen doors, and the transit authority plans to install them in the old stations as well.

Passenger safety is driving the adoption of platform screen doors. By separating the platform from the train, platform screen doors prevent the travelling public either accidentally or deliberately falling into the path of oncoming trains. As a consequence of their positive impact on passenger safety, it is likely that an increasing proportion of new and refurbished metro stations will include platform screen doors.

Figure 12 Platform screen doors, fitted to platforms during the recent extension of the London underground Jubilee Line.

Historically, a pressure of 1,200 Pa with pressure pulses of 300 Pa has been typical in tunnel ventilation system application. Today a pressure of 1,500 Pa with pressure pulses of 500 Pa is typical. Increasing pressure and pressure pulse size increase the importance of tunnel ventilation fan selections that either avoid or manage the effect of fan stall.

The tunnel ventilation system application drives the need to avoid or manage the effect of fan stall. A modern mass transit system in a busy urban area can have 500 trains a day passing each tunnel ventilation shaft. Consequently, the fans in those ventilation shafts are subjected to 500 pressure pulses a day, and therefore are potentially driven into stall each time. With a typical in-service life of 20 years, the probability of incorrectly selected fans for the application suffering a fatigue induced mechanical failure becomes high.

Industrial fan designers have classically utilised one of three approaches during the selection of tunnel ventilation fans that must operate in the presence of pressure pulses:

- Select a fan with a low blade angle (alternatively referred to as a high blade stagger angle) that is “non-stalling”, such that as the fan is driven out of its normal operating range, mechanical stress increases within manageable limits.
- Select a fan with a high enough pressure developing capability to operate with a pressure pulse without stalling.
- Select a fan with a stabilisation ring, such that as the fan is driven out of its normal operating range, mechanical stress increases within the limit of the mechanical design.

There is technically a fourth fan selection strategy: to select a fan that can accommodate the pressure pulse without stalling, and then fit a stabilisation ring to protect the fan in the event of an unforeseen event that drives the fan into stall. The most conservative tunnel system designers utilise this fourth strategy, and it is this strategy that the London Transport favours when specifying fans for the on-going upgrade of the London Underground. For the purpose this study’s analysis, the authors will regard the fourth strategy as a combination of the second and third strategy.

All three fan selection strategies are valid, and ventilation system designers have used them for tunnel ventilation system design. The Delhi Metro Rail Corporation (DMRC) favoured the first strategy, selection of a fan with a non-stalling blade stagger angle, for the recent Delhi Metro extension. At high blade stagger angles, industrial fans classically exhibit a continually rising characteristic. Although not the subject of this study, the authors have previously studied the increase in alternating stress levels with operating a fan with a non-stalling blade stagger angle as Figure 13 shows.

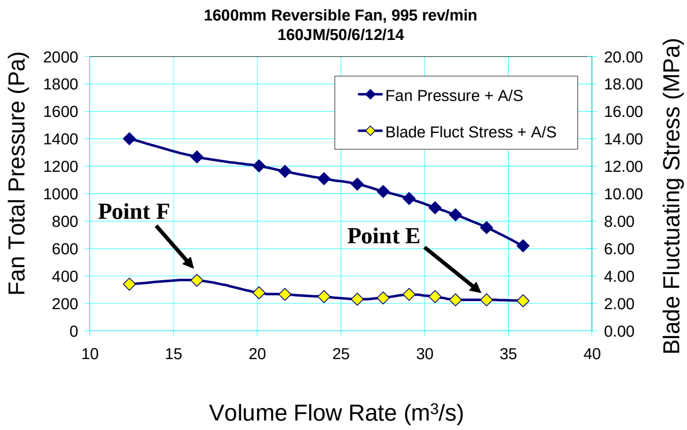


Figure 13 Measured fan performance and alternating stress during normal and stalled operation for a fan with a non-stalling blade stagger angle.

Alternating stress in a fan blade with a non-stalling blade stagger angle when operated in the unstable region of the fan characteristic does increase compared to the same fan operating in the stable region. Due to the non-stalling blade stagger angle, alternating stress typically rises by less than that of a stalling blade stagger angle and stabilisation ring. An example of the alternating stress for a fan with a non-stalling blade stagger angle illustrates that alternating stress is 2.2 MPa when the fan operates in the stable region of its characteristic, Point E, Figure 13, rising to a peak value of 3.6 MPa in the unstable region, Point F, Figure 13. The increase in alternating stress from stable to unstable region is equivalent to a factor of 1.6, approximately 25% lower than the factor of 2.16 with a stalling blade stagger angle and a stabilisation ring (Figure 9). As such, design methodology developed for a fan with stalling blade stagger angle and stabilisation ring will be conservative when applied to a fan with a non-stalling blade angle. The second strategy, a fan with a high enough pressure developing capability to operate with a pressure pulse without stalling, works well with smaller pressure pulses. However, as the size of the pressure pulse increases, the operating point of the fan moves further from the optimum resulting in a less efficient fan selection. Despite the reservation about fan efficiency, this strategy avoids the fan stalling completely in the event of foreseen events. Tunnel ventilation system designers who are confident in their ability to predict the conditions under which the fans will operate during all tunnel ventilation system scenarios favour this second strategy. The third strategy, a fan with a fitted stabilisation ring, works well with larger pressure pulses, allowing fan selection close to its peak aerodynamic efficiency, and then effectively managing the mechanical consequences of driving into stall under the influence of pressure pulses. Tunnel ventilation system designers who favour this strategy argue that it is not possible to guarantee that the fan will never drive into stall, therefore a stabilisation ring that provides mechanical protection in the event of aerodynamic stall is prudent. The tunnel ventilation system designers who favour this approach cite the possibility that occasionally two trains might pass close to a ventilation shaft, effectively doubling the size of the pressure pulse. The Pittsburgh Metro Corporation favours this strategy for the forthcoming extension to the Pittsburgh Metro.

We can compare the three fan selection strategies via a consideration of both initial capital cost and through-life cost. To facilitate comparison of the three strategies, the authors have made assumptions typical of a present day urban area metro system as Table 2 illustrates.

Table 2: Factors impacting on fan capital and through life cost

Design Point Pressure	1,500 Pa
Pressure Pulse	500 Pa
Design Point Flow	85 m^3/s
Fan Type	Reversible, 300^0C 2 hours
Running hours per year	4,400 (12 hours a day)
Cost of electricity	0.04 £ per kW/Hour
Cost of capital	8%
Period of assessment	10 years

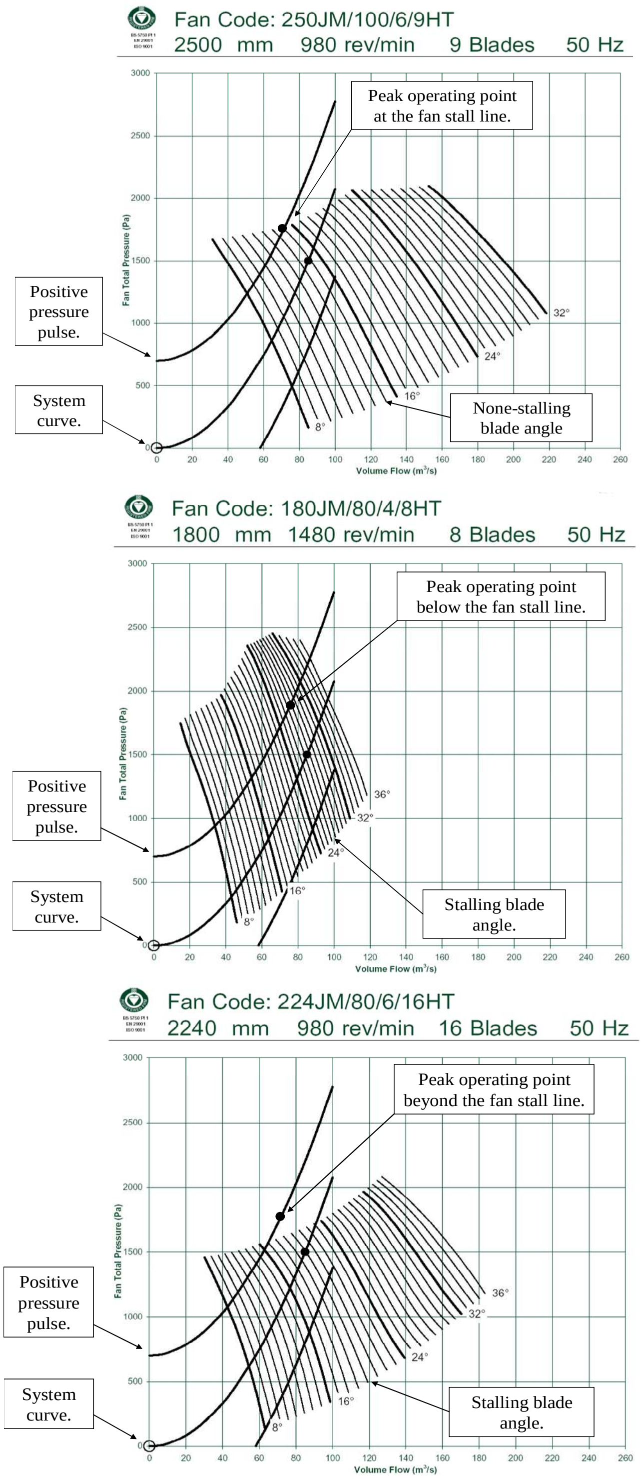


Figure 14 Optimum fan selections for a common duty point, associated with each of the three fan selection strategies: non-stalling blade stagger angle (top), high pressure capability (middle) and stabilisation ring (bottom).

Table 3: Capital cost and ten year through life cost of each selection strategy

	Fan Diameter (m)	*Fan Efficiency*	*Fan investment cost (£)*	*Motor power (kW)*	*Electricity cost/year (£)*	*Electricity cost (10 yrs) (£)*	*Total Fan and running costs (£)*	*Running cost as a percentage of total cost*
Strategy 1	2.5	71%	28,500	185	32,412	217,488	245,988	88%
Strategy 2	1.8	66%	23,000	214	37,493	251,582	274,982	92%
Strategy 3	2.24	69%	32,000	190	33,288	223,366	255,366	87%

When considering the required pressure, flow and size of the pressure pulse, the first strategy, a fan with a non-stalling blade stagger angle, results in a fan of 2.50 meter diameter with a design point efficiency of 71% (See Figure 14a). The second strategy, a fan that can accommodate the pressure pulse, results in a fan of 1.80 meter diameter with a design point efficiency of 66% (See Figure 14b). The third strategy, use of a stabilisation ring, results in a fan of 2.24 meter diameter with a design point efficiency of 69% (See Figure 14c). We can combine the capital cost of each fan selection strategy with the Table 2 assumptions to calculate the through-life cost of each strategy (See Table 3). We define the initial capital cost in Table 3 as the initial cost of purchasing and installing a tunnel ventilation fan. Table 3 defines the operating cost as the cost of fan purchase and installation plus electricity costs over ten years, based on Table 2 assumptions. To facilitate a direct comparison between initial capital cost and operating cost, Table 3 presents the net present value of operating costs.

As previously mentioned, different tunnel ventilation system designers favour different fan selection strategies. However, all three strategies have reliable in-service records. In this example, the largest fan (Strategy One) has the lowest operating cost over ten years, despite not having the lowest initial cost, making this selection strategy attractive to those seeking the lowest cost of ownership.

In this example, the smallest fan (Strategy Two) has the highest operating cost over ten years, despite having the lowest initial cost, making this selection strategy apparently unattractive to those seeking lowest cost of ownership. However, if one accounts for the cost of excavating an underground plant room, the cost of building a plant room for a 2.5 meter diameter fan (Strategy One) may be significantly higher than the cost of building a plant room for a 1.8 meter diameter fan (Strategy Two), therefore making the second selection strategy attractive.

In this example, the medium size fan (Strategy Three) has the highest initial cost, reflecting the manufacturer's cost of a fan casing with a stabilisation ring. Operating costs fall between costs for Strategy One and Two, reflecting the efficiency of the medium sized fan that is between those of the largest and smallest.

Stabilisation rings can reduce the efficiency of a fan, due to the recirculating flow in the blade tip region. For fans intended for use at ambient temperature only, the reduction in efficiency can be significant.

In this study, the authors have assumed that the fans associated with each strategy are designed for once-only emergency operation at 300^0C. The use of aluminium blade results in the blades expanding more rapidly then the steel casing as temperature rises. Consequently, the blade tip to casing clearance has to increase at ambient temperature to prevent it closing at high temperature. The increase in blade tip to casing gap from typically 0.25% of fan diameter (for ambient duty only) to 0.45% (for once-only 300^0C emergency duty) reduces fan efficiency. In practice, the stabilisation ring does not reduce efficiency further when a fan has a large tip gap to facilitate high temperature operation. Consequently, the fans that the authors used in this study confirm the general rule of thumb that larger fans are more efficient for a fixed duty point.

SUMMARY & CONCLUSIONS

The current study has illustrated that for the specific fan studied a stabilisation ring does reduce the mechanical impact of a fan driving into stall, but does not eliminate the impact. Alternating stress levels in the fan blade rise by a factor of 2.16 when the studied fan drove into stall with a fitted stabilisation ring.

The authors conducted the current study on a single fan, and it is therefore not possible to generalise the findings to all fan types. A responsible fan manufacturer will, therefore, empirically establish the specific response of each new fan design when fitted with a stabilisation ring and driven into stall.

Despite the limitations of the current study, the seven fold increase in alternating stress the authors observed in the tested fan without the stabilisation ring fitted is consistent with Rippl' [14] observations when he measured a five fold increase in axial compressors during a rotating stall event.

Despite the difficulty in generalising results of the reported research, it is possible to observe that if the fan is to operate reliably, the mechanical design must account for the increase in alternating stress when the fan stalls; not doing so could result in the fan's fatigue related mechanical failure.

When accounting for the increase in alternating stress when a fan stalls, and when assessing if that increase is acceptable, the fan designer must make an assumption about the maximum defect size in the fan blades and hub. Therefore, responsible fan manufacturers 100% X-Ray inspect all fan blades and hubs to verify that they do not exceed the fan designer's assumptions regarding maximum defect size.

The research reported in this paper enables the authors to identify three issues that fan designers must consider during the design, development and validation of a new fan design:

1. alternating stress level more than doubled as the tested fan drove into its unstable operating range despite the presence of a stabilisation ring;
2. defect size in blades and hub must be checked for every rotating component to ensure it is lower than the maximum assumed by the designer; and
3. one cannot generalise results from a single fan test; all fans that routinely drive into stall must be empirically tested to validate that actual peak alternating stress is in line with the designer's assumptions.

Last, the authors conclude that the three fan selection strategies classically used by tunnel ventilation system designers each have specific advantages and disadvantages. The optimum fan selection strategy in a specific application will, therefore, depend on the impact of fan diameter on plant room cost, and the relative importance of fan initial cost and fan operating cost.

ACKNOWLEDGEMENTS
The authors conducted this research in the context of contract FW-DMA09-11 between Fläkt Woods Ltd and the Dipartimento di Ingegneria Meccanica e Aerospaziale, "Sapienza" University of Rome.

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