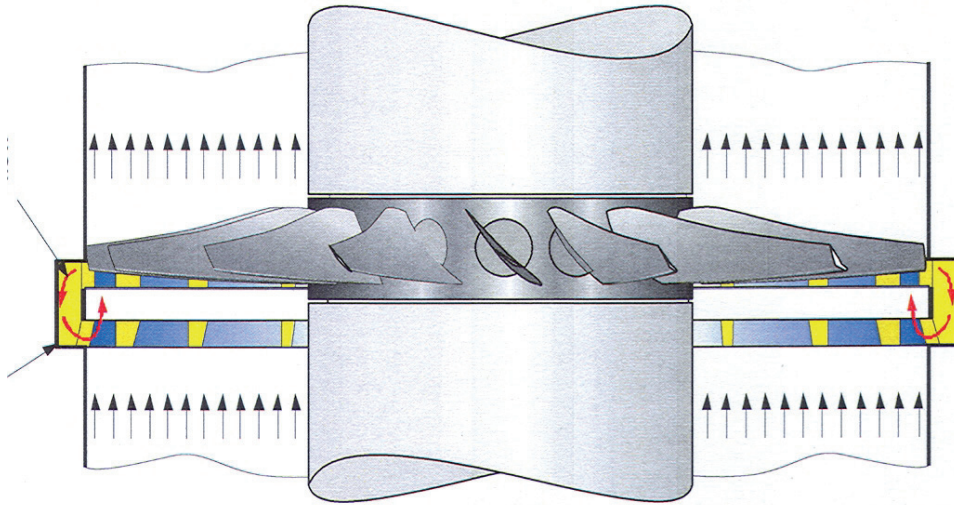


EUROVENT 1/11



FANS and SYSTEM STALL: PROBLEMS and SOLUTIONS





EUROVENT 1/11

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This document has been prepared by Eurovent WG 1 “Fans”, based on a draft written by its Chairman, Bill CORY. Important contributions have been provided by Alain Godichon (FlaktWoods), Professor Hans WITT, Henrik WITT (Witt & Sohn), Dario BRIVIO (Nicotra) and Ron MULHOLLAND (Howden). The text has been extensively discussed by all the other active members of the group. Special thanks go to Ron MULHOLLAND for the high quality drawings.



Alain GODICHON



Bill CORY



Ron MULHOLLAND



Hans WITT



Henrik WITT



Dario BRIVIO

CONTENTS

1. INTRODUCTION	2
2. PURPOSE	2
3. DEFINITIONS	2
4. TYPES of FAN IMPELLER STALL	4
5. ROTATING STALL	5
6. SURGE	9
7. STRESS and FLUCTUATING FORCES	11
8. NOISE and VIBRATION	11
9. INTERMITTENT STALL	12
10. THE PETERMANN PROBE	13
11. AVOIDING PERMANENT STALL IN FANS	14
12. SOME SOLUTIONS	18
13. CONCLUSIONS	22
Bibliography	22

1. INTRODUCTION

Fans and their systems have for many years been bedevilled by problems arising from instability. These have invariably been attributed to “stall”, although it must be recognised that this may be a blanket term encompassing a number of different but related phenomena.

Often, the first indication of operation in stall is an increase in fan noise. There may also be a reduced energy efficiency. In the worst cases premature failure can result, and has led to the introduction of ancillary devices to give the fan impeller some protection from the fluctuating forces, which lead to this unhappy state of affairs.

2. PURPOSE

It is the intention of this publication to describe these various phenomena, to suggest ways in which they may be detected and avoided or reduced in their severity, and to describe how any remaining effects can be evaluated for predicting the impeller life.

In this connection, it is important to note that in performance tests conducted with standardised airways, the stall may not always be apparent. Other means of detection may therefore be necessary.

3. DEFINITIONS

3.1 Stall

The term “stall” should strictly speaking be restricted by analogy to the situation found with aircraft wings where, as the angle of incidence of the airflow is increased, the lift force on the wing at first increases, but falls abruptly when a critical angle is exceeded (Figure 1). Similar effects occur across a fan impeller and in terms of the fan pressure there is a fall-off beyond the stalling incidence. This may be progressive or discontinuous as illustrated by Figure 2 and Figure 3. The interaction between the fan and system curves shows the likely instability.

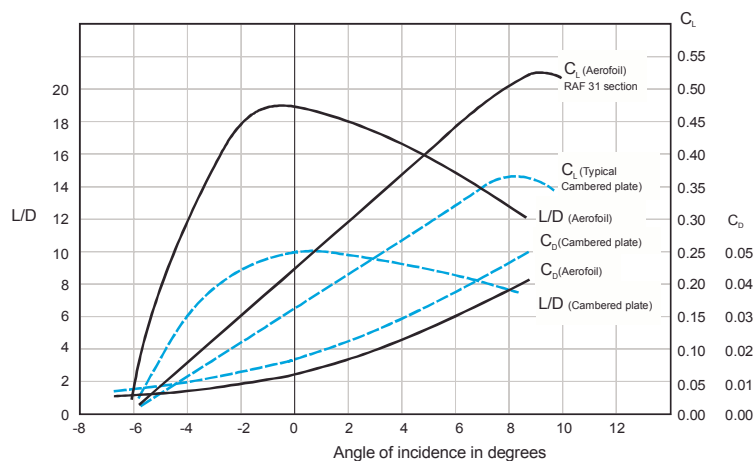


Figure 1: Lift and Drag Coefficients of a Typical Aerofoil and Cambered Plate

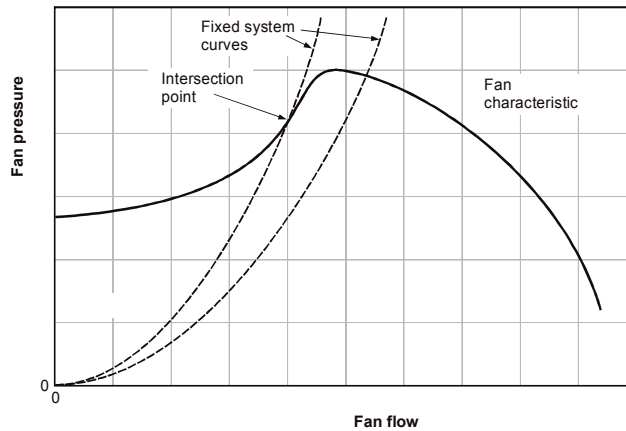


Figure 2: Progressive Stall

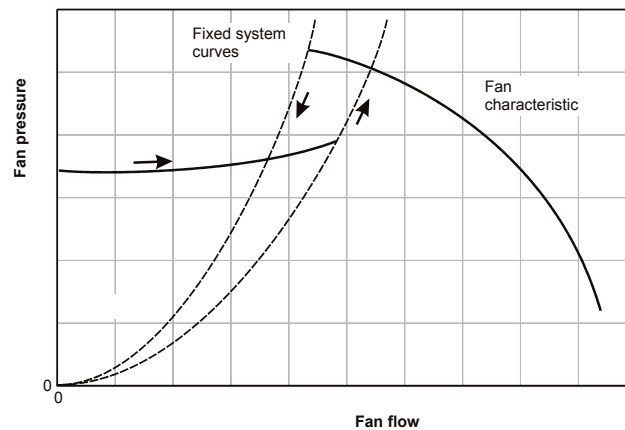


Figure 3: Discontinuous or Abrupt Stall

The case shown in Figure 3 can create applicational problems, as the actual duty point may significantly depend on the history of the working condition, the actual start-up cycle, and as an effect of temporary disturbances. This is not necessarily a cause for a “system instability”, i.e. a dynamic phenomenon leading to fluctuating or pulsating volume and pressure. It is true, however, that the system may switch between a normal working condition and a stall condition.

3.2 *Rotating Stall*

This occurs when a disturbance causes the flow to separate from one of the blades, which results in blocking of the flow through the corresponding blade cell. This in turn affects the flow angles in the blade cells either side to change, so that the following blade then tends to stall whilst the preceding blade becomes more stable. The stall cell eventually moves to the next passage and then the one after that, rotating around the impeller in the opposite direction to that of the rotation.

3.3 Surge

This is a violent instability of the complete fan and ducting system during which the airflow may reverse and recover at a frequency of a few cycles per second; it is most commonly found where the fan delivers into a large plenum chamber, or an extensive duct system having a large cubic capacity.

3.4 Fan/System Instability

This occurs where the fan and system curves intersect at more than one point, or the system changing due to variation in temperature, leading to the flow fluctuating between the various intersections.

The condition may also produce an unpredictable working condition of the fan, probably different from the original design working point, as the actual duty may depend on the way the system approaches the stable duty. This should not be confused with “instability”, as the resulting duty, although unexpected and unacceptable for many reasons, may well be perfectly stable.

4. TYPES of FAN IMPELLER STALL

From Section 3.1 it will be deduced that when the air cannot follow the blade surfaces, then separation takes place and the blade is stalled (see Figure 4). However, stall may be limited to just a part of the blade where the incidence exceeds the critical value. Rotating stall passes from one blade to the next. Pronounced separation leads to significant circulatory flow.

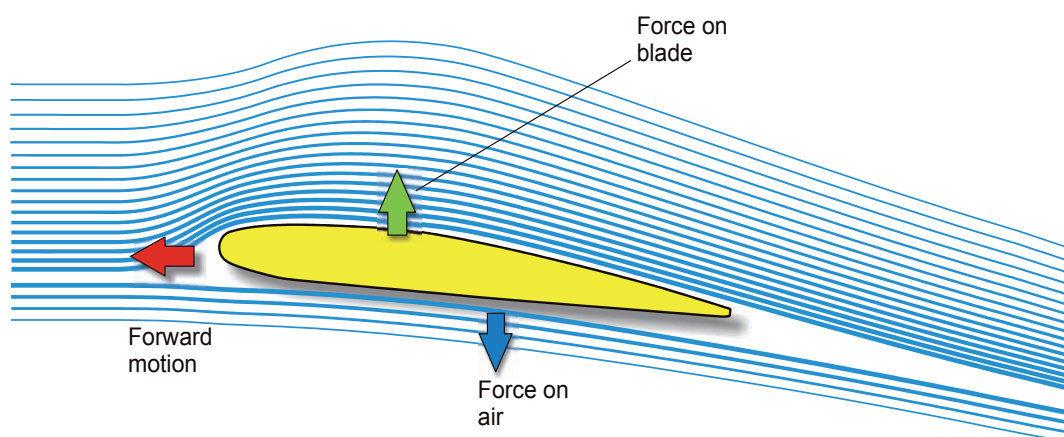


Figure 4a: Streamlined Flow Across an Aerofoil

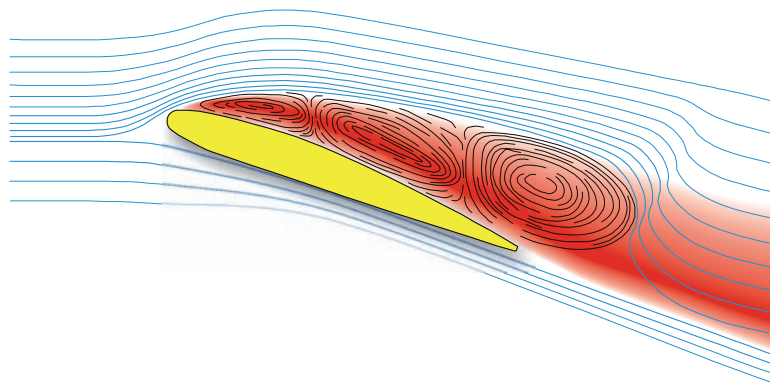


Figure 4b: Stall Flow Across an Aerofoil Blade

5. ROTATING STALL

It should be understood that the relative speed of the stall cell varies according to the point on the characteristic curve where there is a zone of discontinuity. The frequency of the stall disturbance ranges from virtually 100% of rotational frequency at the start to half rotational frequency at its most severe. The effects can however be experienced at above rotational frequency if two diametrically opposite stall cells develop when the most severe stall will be experienced at about four thirds rotational frequency.

Not all fans are subject to rotating stall and many fan manufacturers do not show it on their curves even when it is present. They draw a smooth curve through the test points and generally ignore it.

5.1 Rotating Stall in Backward Bladed Centrifugal Fans

Rotating stall is present in many designs of backward aerofoil bladed centrifugal fans, especially the wider designs (high specific speed) and/or those with a small number of blades. It can also be found in straight backward inclined bladed fans and backward curved bladed fans especially where these have a small inlet blade angle. Indeed these may even exhibit a negative stall region towards free outlet. (Figure 5)

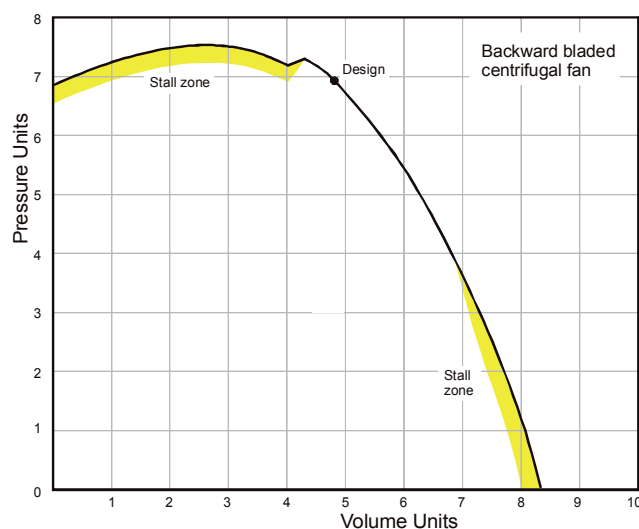


Figure 5: Backward Bladed Fan Characteristic Showing Possible Stall Zones

The onset of stall will occur on one blade at a time. The stall at one blade will then divert an increased airflow into the neighbouring blade passage, which will then consequently cause the next blade to stall. The blades will then stall one at a time until the entire impeller is in the stall condition.

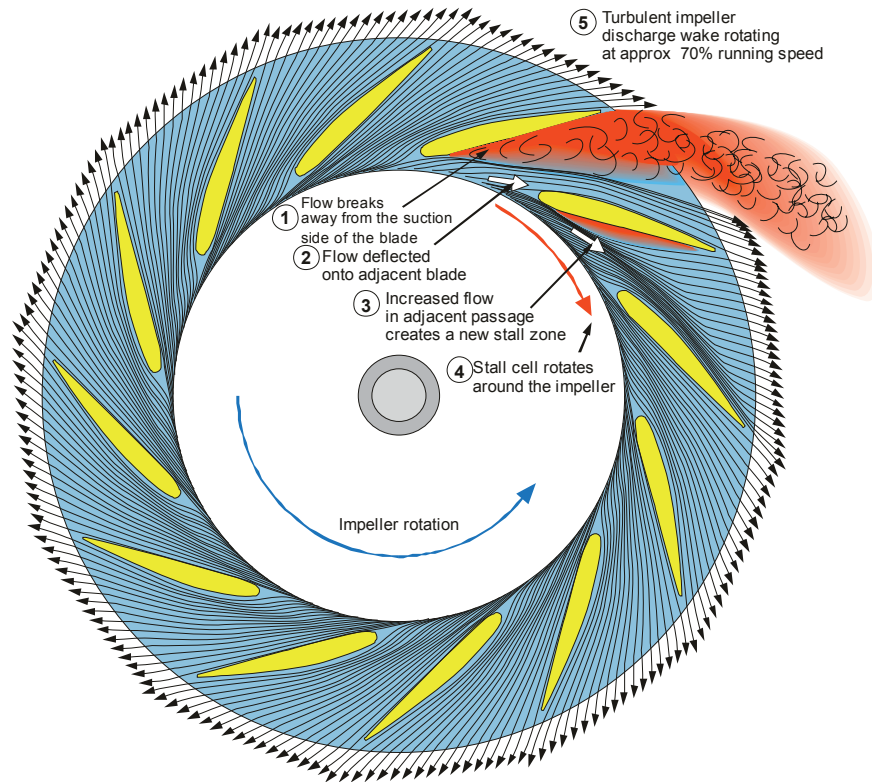


Figure 6: Rotating Stall in a Backward Inclined Aerofoil Impeller

The stall on a centrifugal fan is not as dramatic as the stall on an axial fan, due to the fact that a centrifugal fan does not rely on aerodynamic lift to produce pressure. The pressure remains fairly constant as the stall progresses and the volume flowrate decreases. The more severe aspect of a centrifugal fan stall tends to be the increased low frequency noise that is generated causing the casing, ductwork and impeller to vibrate. If large pressure pulsations are sustained then fatigue failure may occur in the impeller, casing and ductwork.

Although operating the fan in stall does not cause a significant drop in the pressure generated, operating a fan in stall for prolonged periods of time must be avoided.

5.2 ***Stall in Forward Curved Centrifugal Fans***

Forward curved centrifugal fans can also experience rotating stall, especially where the blades are circular arcs. Many stall cells usually develop at the same

time, leading to severe discontinuities in the pressure/flow characteristic, again rarely admitted to by the manufacturer who draws a smooth curve through too few test points and encourages operation at points to the left of peak pressure.

Some forward curved fans of the drum type (sirocco impellers) may be subject to different types of stall, as they share with low-pressure axial fans the long and thin shape of the blades, and with most backward curved centrifugals the interference between the impeller and the scroll case.

Under normal conditions, their behaviour at the stall is very similar to that of those axial fans having a low hub-to-tip diameter ratio and a low pitch angle, i.e. the abnormal flow extends progressively along the blade, often without producing any sharp “dent” along the fan pressure curve. This is particularly true for those impellers having a relatively large axial dimension (“wide” impellers).

Forward curved impellers of a rather narrower design, or those having sensitive scroll designs, may be more prone to another type of stall behaviour. This is, more similar to the rotating stall of the axial fans, but where the stall cell, although moving between the blades, is fixed with reference to the scroll case. The flow disturbance is partly due to the interference between the impeller flow and the spiral scroll case, and particularly that part which is closer to the blades, the cut-off edge. This kind of stall may arise more suddenly, and generate more pronounced pressure fluctuations, particularly when it is of intermittent nature.

As a matter of practice, forward curved fans having diameters up to 500 mm have been used for many years in very-low-pressure applications, as in some HVAC systems, taking little care to avoid stall conditions. This has seldom led to mechanical problems, mainly because of the relatively minor amplitude of the fluctuating stress induced by the stall on the structure of these impellers. This combined with the fact that, the impeller design was frequently used well below 70% of its maximum design speed. Thus the impeller had a considerable structural safety margin.

When the applications require a higher pressure and speed, both the structural safety margin of the commonly available forward curved impellers is reduced, and the amplitude of the pressure and stress fluctuations, acting on the parts of a stalled impeller increase. As a consequence, Forward Curved impellers used above 70% of their maximum speed should be selected more carefully and used in such a way to carefully avoid any long-lasting stall conditions.

Some forward curved fan designs may also show clear signs of inverse or low-pressure stall. They exhibit large pressure fluctuations when the fan is run below a critical value of the pressure coefficient, i.e. against too low a pressure loss. As a general rule, manufacturers do not publish the performance curve of the fan in this critical region, because the amplitude of the pressure fluctuations under these conditions may well be large enough to create mechanical problems both to the fan and to the connected ducts.

5.3 Stall in Axial Flow fans

Axial flow fans are much inclined to rotating stall. The stalled sectors may have a very low axial velocity and even a small negative velocity. In low hub to tip ratio fans the stall cell usually appears first at the periphery and as the flow is further reduced extends towards the hub (Figure 6). For high hub to tip ratio fans the stall cell(s) occupy the whole blade immediately on stall initiation. It should be noted that at low flow rates there is a centrifugal component to the flow and that there is often a recovery in pressure such that the dip is relatively small and may even be non-existent on fans with low pitch angle blades.

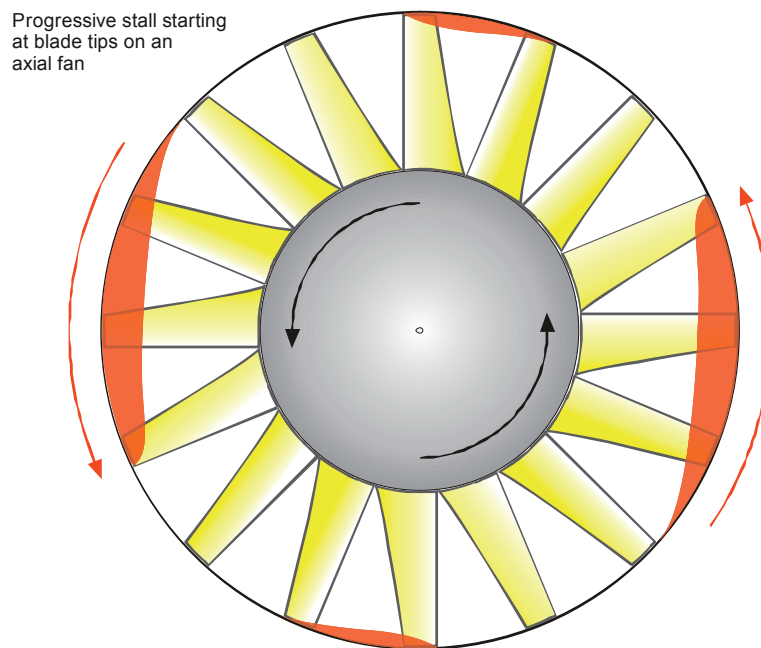


Figure 7: Stall in an Axial Flow Fan

5.4 Inlet Vortex and Rotating Flow Instability at Low Load Operation

Many fans are required to operate at varying flowrates. A typical way to reduce flow performance of a centrifugal fan is to use inlet vanes at the air inlet. The system provides a pre-whirl in the direction of fan rotation to modify the absolute flow velocity angle at the blade leading edge and make the relative flow velocity adapted to a reduced flow rate and a reduced pressure. As a consequence the fan efficiency remains acceptable with partial closure of the inlet vanes.

Nevertheless the system has some limits. When the inlet vanes are largely closed with a remaining opening angle of 30 to 45°, the high swirl is accompanied by an inlet vortex at the centre, which may create strong pressure pulsations. The frequency of the pulsation is usually the frequency of rotation.

One solution, which exists, consists of reducing the vortex at the central part of the inlet eye by additional stationary vanes (so called 'dorsal fins' see Figure 8).

These dorsal fins are the only known means to control this inlet vortex and they are considered as a mandatory feature to add to the inlet vanes when the fan power is significantly above 100 kW.

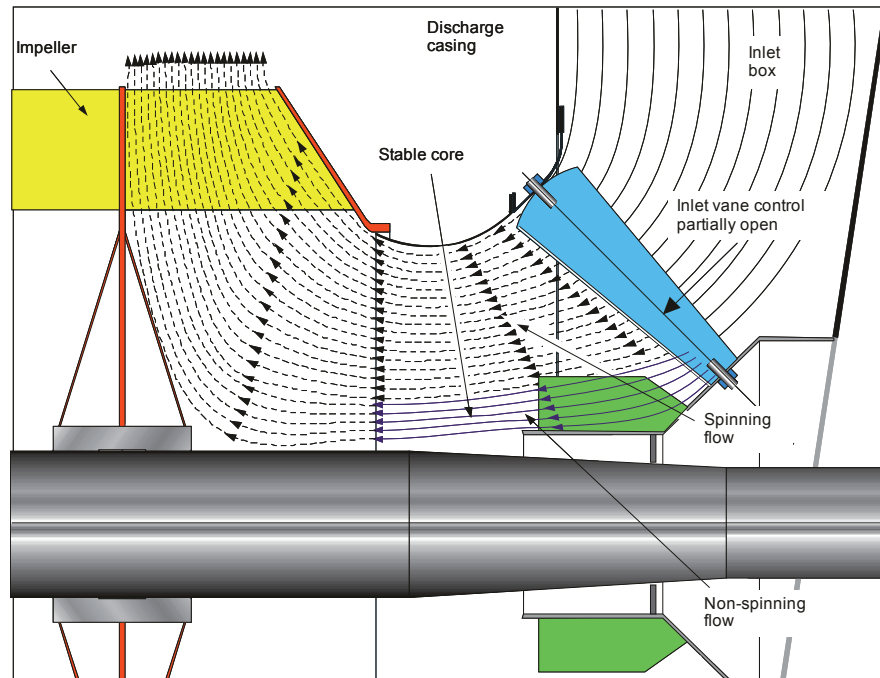


Figure 8: Dorsal Fins Fitted to Inlet Vane Control on a Centrifugal Fan

Even if we can prevent this major problem, flow instability with inlet vanes equipped with dorsal fins still exists for large power fans. The pre-whirl created is not sufficiently adapted to low flow operation and very commonly rotating stall appears. The pressure pulsations are observed at a frequency close to $2/3$ times the frequency of rotation and also at its harmonics. In a few cases, mechanical damage has been observed e.g., fatigue failures of the impeller or of the hub. More usually, duct vibrations and low frequency noise occur at an unacceptable level.

6. SURGE

From 3.3 we note that surge is a violent instability of a fan and system either having a plenum chamber or a ducting system having a substantial cubic capacity. It is particularly sensitive to parallel multi fan systems.

The pressure in this can only change slowly due to its large volume. If a small reduction in system pressure increases the flow from A to C, the plenum pressure may be greater than the pressure produced by the fan. This causes a greater resistance to the flow, which therefore tends to reduce. As the flow falls, the pressure delivered increases again until operation at A is restored. A small change of flow to point B causes the

pressure delivered by the fan to exceed the plenum chamber pressure, leading to an increase in the flow and again operation at A is restored.

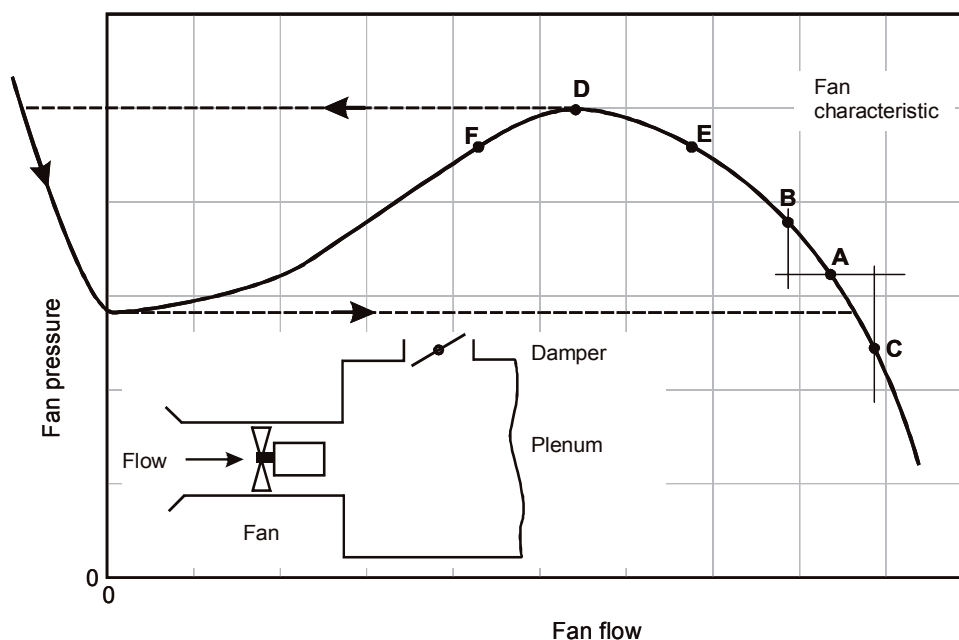


Figure 9: Stable and Unstable Flows

It is important to note that operation on the fan characteristic where the gradient of the pressure/flow curve is negative is inherently stable.

When operation close to the peak of the fan curve is considered (see Figure 9), it is clear that a perturbation of the flow from D to E will be stable in the same way that flow from A to C is stable. However, a change from D to F leaves the fan pressure lower than the plenum pressure, which therefore tends to a further reduction in flows. This process leads rapidly to a complete flow reversal with the fan operating momentarily on the negative part of its characteristic. The plenum discharges in both directions until its pressure has fallen significantly. The fan now starts to deliver again at the high flow end of its characteristic. Delivery to the plenum considerably exceeds the flow through the damper at entry to the ducting system and so the flow and pressure in the plenum build up again until the system is once more at point D and the cycle is repeated. This “cycling” will continue until the outlet damper is opened sufficiently, or the system losses beyond this damper are reduced to ensure that the operating point is on the falling (negative slope) part of the fan characteristic.

When the volume of the plenum is reduced, surge takes place at an increased frequency, as it takes a shorter time to empty and refill the plenum or duct volume.

If there is no plenum, then the damper opening will determine the operating point. Again the dangers of operating between D and F are apparent. If the system curve has a high fixed element (e.g. mixing boxes in a VAV system, or a fixed contrary wind at the system exit) then there is a greater chance of fan/system instability due to intersection

between the two curves at more than one operating point, see Figure 10. The area may also be subject to fan rotating stall and failure can often result.

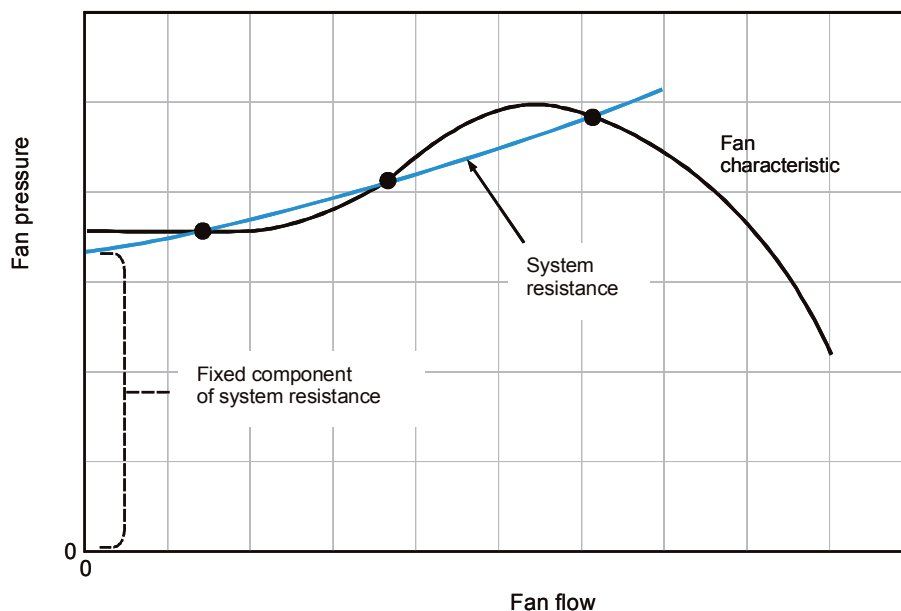


Figure 10: Fan / System Instability

7. STRESS and FLUCTUATING FORCES

All fan impellers will be subject to centrifugal forces during operation and thus the various elements will be stressed. In shrouded centrifugal impellers the blades are fixed at both ends such that fluctuating stresses due to aerodynamic effects are usually low. With axial flow impellers, however, the blades are cantilevered and only supported at the end adjacent to the hub. Any fluctuating forces are then important and can lead to failure due to fatigue. These fluctuating forces create fluctuating stresses, which vary according to the duty position on the fan characteristic. Those due to the phenomena already described (stall, rotating stall, surge etc.) are especially important. They are usually at a maximum just to the left of the first pressure peak and result in low cycle fatigue failure, due to low cycle excitation. It should be noted that the peak value of fluctuating stress reduces with decreasing pitch angle. This may occur due to a small number of large deflections or a large number of smaller deflections. Surface finish, porosity or surface damage are all important.

8. NOISE and VIBRATION

As stated in the Introduction, a fan operating in stall will easily be recognised by the increased levels of both noise and vibration. The fluctuating forces are efficient in their translation from force to airborne noise and structural vibration. It usually has characteristics of a pure tone imposed on the broad band background, but may also be imposed on a narrow band trace. The frequency of this noise and vibration is a tool to determine the likely cause(s).

9. INTERMITTENT STALL

It has been noted that stall can be the result of different phenomena. If the stall does not exist under normal service conditions i.e., when the system resistance is too high for the fan employed, it may still stall intermittently for other reasons, such as:

- Wind may cause an overpressure at the air outlet or under-pressure at the inlet.
- In many cases several fans are virtually working in series, for instance there can be a supply fan and an exhaust fan. If one of these fans is switched off, the other may go into stall.
- In many cases ventilation lines contain dampers, either fire dampers or volume flow-control dampers. Actuating such a damper may cause stall.
- Variable system resistance through filter clogging.
- Variable system resistance through change of parameters in the process.
- Variable air consumption of internal combustion machines in ventilated rooms.
- Parallel work of two or more fans outside the stable part of both the individual and sum characteristic.
- Change of pressure loss or system resistance due to local changes in temperature or density.
- Transient pressure waves as for instance caused by trains in narrow underground tubes.
- Transient pressure waves in road tunnels caused by passing lorries.
- Clogging of ducts, dampers or other components.
- Partial or total temporary closing of air inlets or outlets by for instance parked cars, deposited containers or other obstacles placed in front of the air inlets or outlets.
- Erosion of blades or dust build-up.

Whatever the reason for the stall, the system designer must foresee possible incidents and find effective counter measures. Dampers for instance should always be electrically interlocked with the fan motor. If no such simple methods give sufficient safety, one of the following protective procedures can be used:

- Fans with a non-stalling characteristic.
- Fans with a stalling characteristic can be protected by anti-stall rings.
- Protection of the fan through stall indicators based on increased airflow fluctuation.
- Protection of the fan based on change in the airflow direction as for instance Petermann probes.
- Protection of the fan based on the occurrence of current fluctuations.
- Protection of the fan by flow measurement, switching off of the fan at airflow below a given minimum value.
- Recirculation either inside or outside the fan.
- Increase the hub to tip ratio of an Axial Flow fan thereby improving its mechanical strength.

It should be noted that some of these solutions may not be applicable to fans having speed control.

10. PETERMANN PROBE: A DEVICE for DETECTING the STALL CONDITION of an AXIAL FLOW FAN

The pressure variations occurring in the stalling range express themselves by the creation of one or more zones in which the airflow through the vane channels is blocked or goes in the opposite direction.

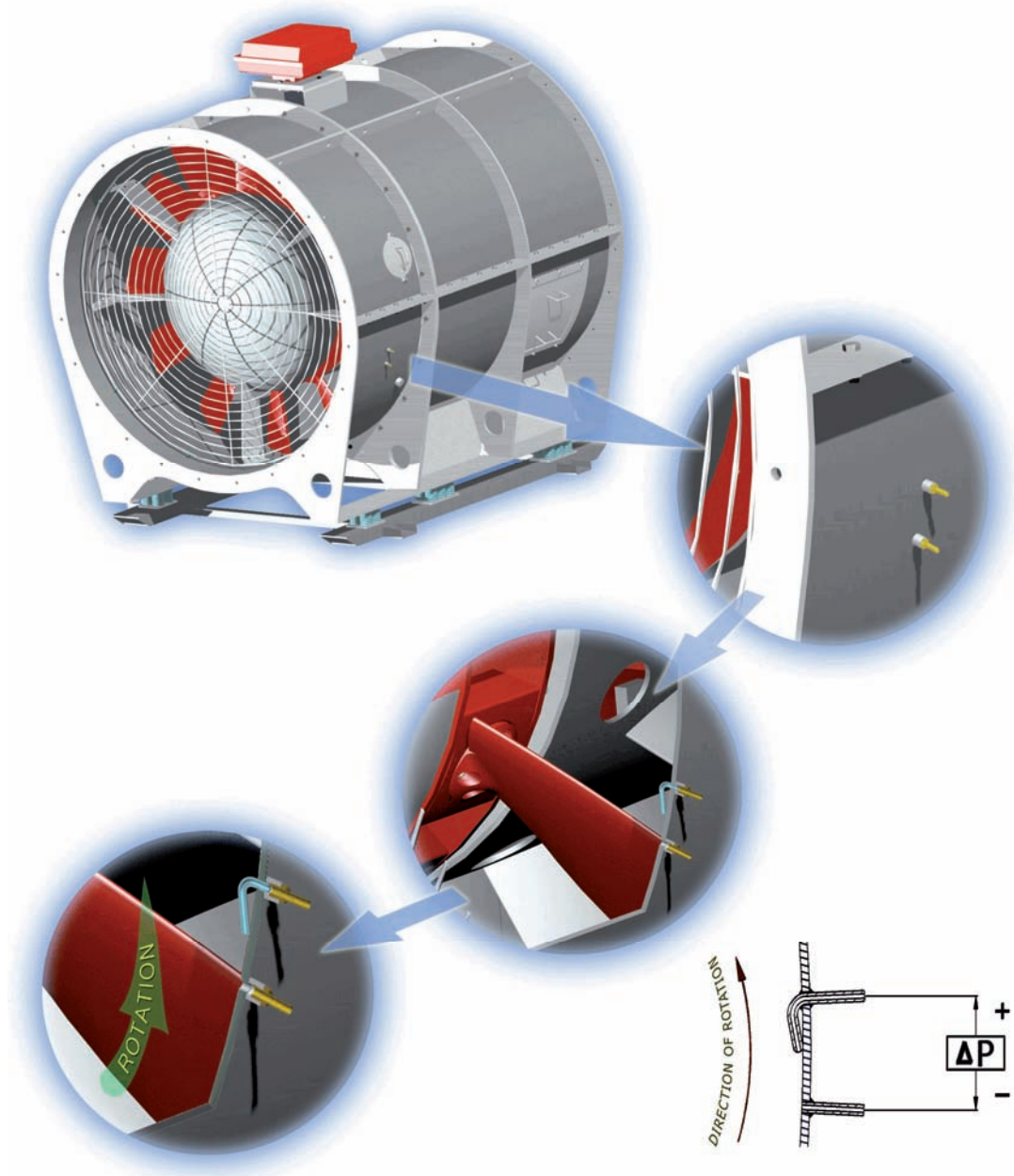


Figure 11: Arrangement of the Petermann Probe Fitted to an Axial Flow Fan

The Petermann probe uses this phenomenon to detect the stall condition of an axial flow fan by measuring the pressure difference between the total air pressure acting in a direction opposite to the direction of rotation of the fan impeller (by means of a hook-shaped tubular measuring probe) and a reference pressure corresponding substantially to the static pressure at the wall of the air duct in the same radial measuring plane immediately in front of the fan blades, upstream.

This pressure difference will be approximately zero in the stable working range of the fan, but increases considerably at a point corresponding to the inflection of the pressure difference characteristic when the fan enters into stall condition. Thus, by means of a simple differential pressure transmitter it is possible to know when the fan is in the stall condition.

In the following diagram, (Figure 12) the pressure difference is represented between both pressure probes (ΔP) as function of the air volume transported by the fan. In this curve it is possible to see the characteristic change at the junction between the stable working range and the unstable range in the fan characteristic.

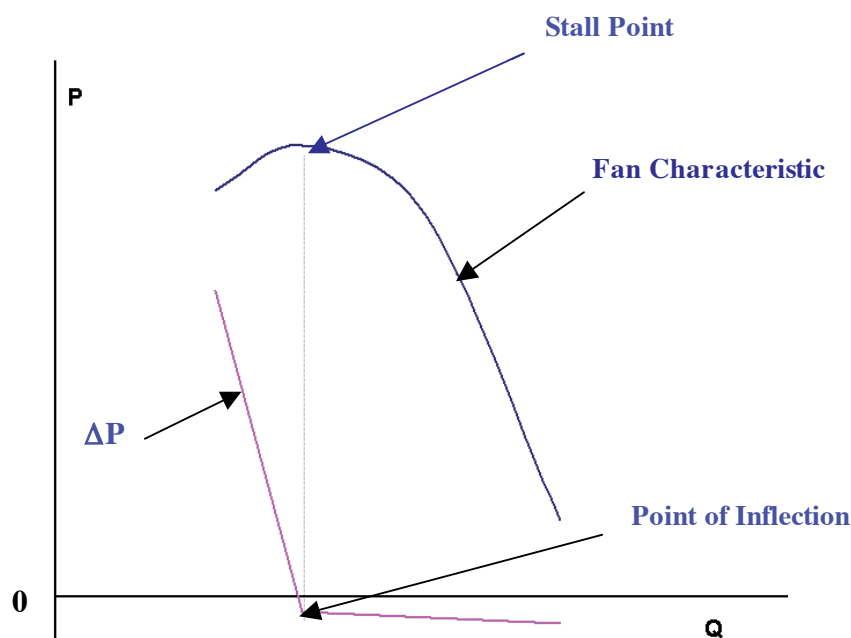


Figure 12: Differential Pressure of Petermann Probe versus Flowrate

11. AVOIDING PERMANENT STALL in FANS

Permanent stall is more often the result of an improperly selected or just improperly sized fan. When the system pressure loss has been correctly estimated, permanent fan stall can generally be prevented by the proper selection of the fan size in a range.

Nevertheless, in everyday practice, permanently stalled fans are far too common. While many of them will still be able to operate for a long time, without incurring any

serious problem, a number of them, and particularly those selected to operate closer to their mechanical stress limits, may be prone to mechanical problems, or create either mechanical or operations problems to other components of the system to which the fan is connected.

A small number of design incidents are known to be the origin of most of those incorrect fan installations. They can lead to permanent stall, and should be carefully avoided by the wise system designer.

11.1 Underestimated System Pressure Loss

The most frequent incident, leading to an improperly sized and permanently stalled fan, is a serious underestimate of the actual system pressure loss at design flow. When this occurs, the true system line will cross the fan pressure characteristic much to the left of the originally specified duty point, and, provided that the error is large enough, to the left of the stall line.

Under these circumstances, speed changes are useless to recover the fan from stall. When they are dictated by the will to regain the originally specified volume flow, they may even prove dangerous as, by increasing the fan speed and pressure, the change increases the average mechanical stress, as well as the amplitude of the pressure fluctuations, and of the resulting stress fluctuation; the final result may well be that of worsening the mechanical effects of the stalled working condition, up to the point of creating fatigue damage to the fan structure.

Selecting more forgiving fan designs, which have a wider margin between the best efficiency working condition and the stall line, may help to reduce the instances of occurrence of this problem, but there is a limited to what can be reasonably achieved, as true stall-free fan designs frequently pay either a cost or an efficiency penalty even when used at their most appropriate duty.

The most appropriate solution would obviously be the removal of the undesired and additional pressure loss sources. Unfortunately, this is not always achievable, and the increased pressure loss must be accepted, together with the consequent increase in system power consumption, and dealt with accepting a fan change, either shifting to a geometrically similar but smaller fan size, running faster, or changing fan design, either by changing blade angle, in the case of axial fans, or substituting a centrifugal forward curved impeller by an alternative backward curved impeller design where this is possible.

Sometimes air bleeding or recirculation may be a cheap and easy approach to solve the problem of permanent stall, but these kind of solutions may significantly reduce the energy efficiency of the system, and this alone should be enough to justify the preference for more rational approaches.

11.2 Fans Selected for Future Expansion

Sometimes a ventilation system is designed with future expansion in mind, e.g. for doubling the ventilated area by adding a second, identical branch to the air

distribution ducting system at a later stage. The fan may then be selected for the final duty rather than for the duty at start-up, with just half of the system connected. It is easy to understand that the temporary configuration of such a system, accepting half the volume flow at the same design pressure, would quite probably bring the duty point of a fan, properly sized for the final duty into its stall region.

The requirement for variable volume requirements along the system life could better be dealt with by using an aerodynamic device like a viable inlet guide vane device, which can alter the characteristic curve of a fan to provide the specified pressure at half the maximum design volume flow, without introducing the stall of the impeller, but these devices introduce a penalty on fan efficiency. It may be preferable to change the complete fan or its impeller. When these devices are not required for continuous system control, a better design approach may well be that of specifying the use of parallel fans and to begin, run or even install, just those required to cope with the requirements of the initial, reduced configuration of the system.

11.3 Selecting Fans for Minimum A-weighted Sound Power Level

The sound power produced by a fan changes in both overall level and frequency content, across the operating range of the fan. The noise produced in those working conditions, where the fan is running in stall, normally has a very high sound power content in the lower frequency bands, which are more heavily subject to the effect of the A-Weight filter, when a LwA overall sound power level is calculated.

As a result, any list of fans sorted in increasing order of LwA sound power levels may be dangerously biased toward fans running in the leftmost part of their operation region. If the sound power levels are properly predicted, and alternative fan designs and/or sizes, suitable for a specified duty, are sorted in the said way, quite frequently one or more of the fans showing the lowest LwA figures may well be fans running in or very close to the stall condition.

If the selected fan is not already in a stall condition at design duty, a very small margin between the expected duty point and the stall line may well mean that the smallest calculation mistake (or just a little filter clogging) can bring the actual duty within the stall region.

Selecting the fans for lowest un-weighted overall sound power level, Lw, may help to avoid the danger but the existing noise calculation models, as used by some fan manufacturers to predict noise at a specified duty, starting from experimental data, are often less reliable when calculating Lw ratings, than they are when predicting LwA levels, and so this approach must be treated with caution.

Hindsight is always required when sizing a fan for the lowest possible noise. The selection should be carried out without compromising too much of the stall margins.

The system designer should also bear in mind that, if what is really important is the noise level at the end of the duct system, then a fan providing a higher sound power level, because of a higher noise level in the high frequency bands, may well prove more quiet in practice, as high frequency noise, although higher at the source, may be absorbed much more easily along the ducts, so the approach of looking just for the lowest LwA figure is not a guarantee of an effective design of the system, concerning acoustic performance.

11.4 *Selecting Fan Size from a Specified Average Air Velocity at Fan Discharge*

Sometimes fans are sized according to a specified level of average air speed at the fan discharge. While specifying average air speed may be an effective approach when sizing air distribution ducts, this approach has little to do with the proper selection of fans.

Within a range of geometrically similar fans, the ideal i.e. best efficiency lowest noise and stall free compromise fan, has an average air speed at fan discharge, which can be demonstrated to increase proportionally with the square root of the specified fan pressure.

If the specified “design air speed at fan discharge” was established for a given pressure level, but is applied to a fan providing a significantly higher pressure, the result is the selection of an oversized fan, which may easily be large enough to run in a permanent stall condition.

Fan selection should be carried out with best efficiency and stall margin in mind first. Any problem of matching the air velocity at fan discharge with the specified air velocity in the ducts should be solved only with proper use of conveniently sized diffusers, rather than by compromising fan effectiveness.

11.5 *Aerodynamic System Interference*

Another design accident, which may lead to persistent fan stall, is not related with fan size selection, but rather with system design.

A number of fan designs are sensitive to airflow disturbance induced by system components, which are located immediately upstream or less frequently downstream of the fan.

Axial fans are known to be sensitive to airflow disturbances from the duct upstream, see Figure 13, but a number of single inlet centrifugal fan designs are also known to be sensitive to the same problem.

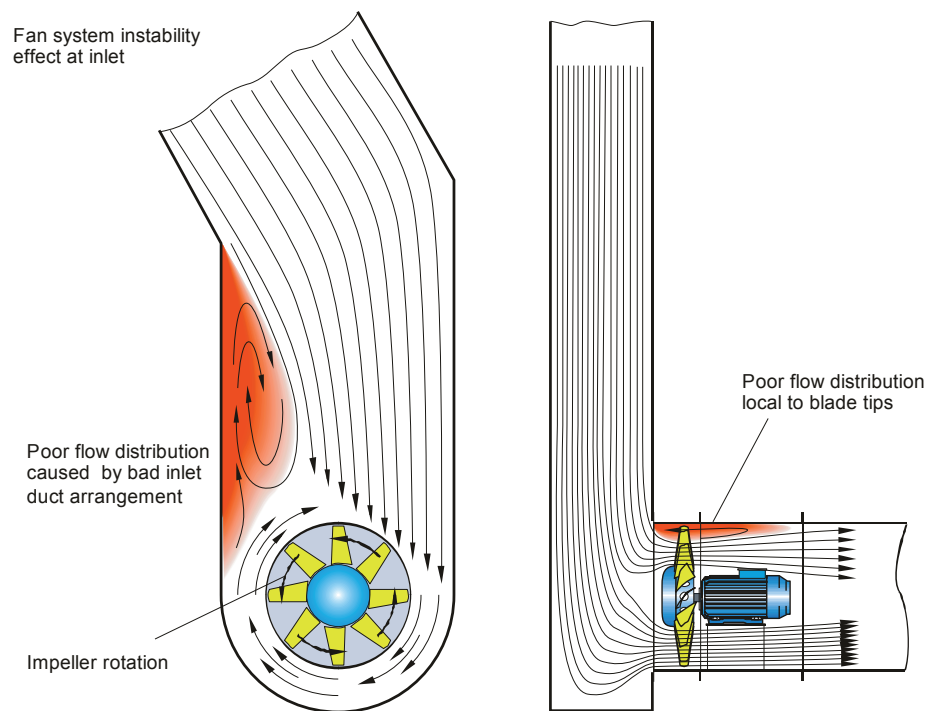


Figure 13: Poor Upstream Duct Components to an Axial Flow Fan Neglecting Swirl

A typical example is a plain duct elbow located immediately upstream of an axial fan, which creates a significant difference between the axial velocity entering the fan along the inner and outer parts of the elbow. The resulting cyclic change in local velocity direction, over the blades, can easily generate stall, in a fan and at a duty point, which, without the influence of the elbow, would run in a perfectly smooth way. Adding an inlet bellmouth between the duct and the fan inlet improves the situation considerably.

12. SOME SOLUTIONS

An obvious solution to all the problems encountered through fan stall is to provide a fan with a non-stalling characteristic. Ideally this would have a fan static pressure/flow curve continually rising to *Static Non Delivery*, i.e. zero flow.

For a centrifugal fan this is most likely to be achieved by a narrow or backward curved bladed fan with a tip impeller width less than 20% of the impeller diameter and an outlet blade angle of less than 40° . It is also desirable that the inlet blade angle should be not less than 30° . A log spiral blade form is ideal as this give a constant angle from the eye to the periphery.

In addition to the solution of designing a centrifugal fan with a non-surgings characteristic, one can also adapt either an external or an internal flow recirculation.

The external flow recirculation consists of connecting the fan discharge to the fan inlet at a proper location in order to maintain the fan operation at a minimum flow rate without surge. A throttling device may be used on this connecting duct to control the recirculated flow.

To be efficient, the system must be more active at a reduced flow and ideally not effective at all at design flow in order to maintain a good fan efficiency at design load.

The same principle applies to an internal flow recirculation: as an example, a modification of the inlet cone geometry and clearance can be adapted. (Figure 14a). Some positive results have also been observed with a cover plate geometry in two annular parts (see Figure 14b), the external one being separated from the internal one by a rather large clearance to allow more flow recirculation at a higher pressure and at a reduced delivery flowrate. This design does not allow for regulation, but is to a certain extent self-regulatory. When the main flow is low, the recirculatory flow is larger. Many parameters are relevant in this design and there is no general solution to be adapted on any fan type and geometry: model tests are usually required to make this kind of feature adapted to the need without excessive fan efficiency decrease. A good design exhibits a pressure-flow curve without any surge area.

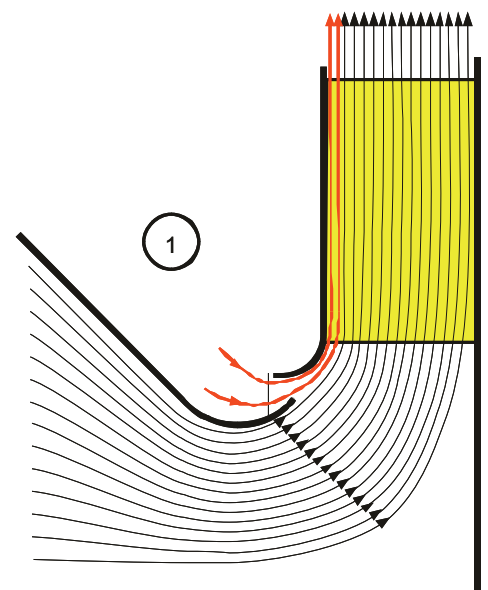


Figure 14a: Internal Flow Recirculation Obtained with a Large Clearance between the Inlet Cone and the Impeller

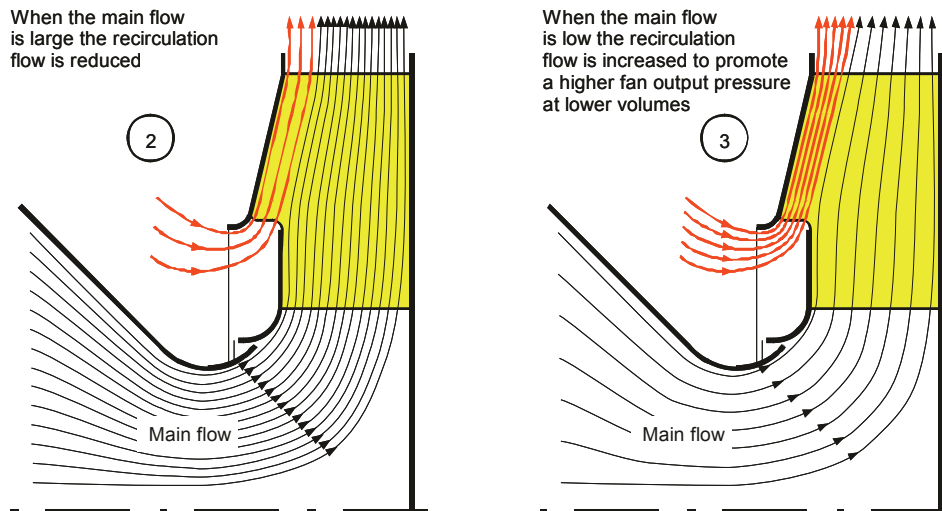


Figure 14b: Internal Flow Recirculation Obtained with an Impeller Shroud in two Parts

For axial flow fans a low hub to tip ratio is desirable and for most aerofoil sections a pitch angle of 16° or less is indicated.

These designs invariably lead to fans of large diameter and a much greater price. The solutions given in section 4 have therefore been receiving considerable attention over the last few years. For axial flow fans, the second solution suggested i.e. anti-stall rings is of particular interest. It has, for example, been incorporated in the main fans for the Channel Tunnel, which are subject to high transient pressures due to the piston effect of the trains in a relatively close fitting tunnel.

With the introduction of alloys having a greater resistance to fatigue and lower hub to tip ratio (down to 0.25) in axial flow fans a greater tolerance to stalled operation has been achieved. Nevertheless, it should be avoided if at all possible. Anti-stall rings have been installed to a number of different designs as shown in Figure 15.

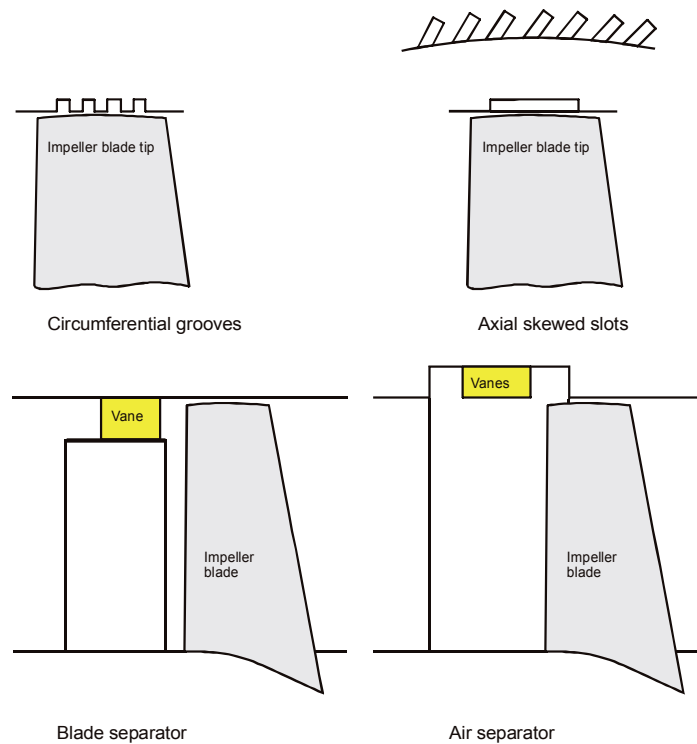


Figure 15: Some Different Designs of Anti-Stall Rings

However, the design of Ivanov has been the most successful and is shown in Figure 16. Now that the patent has expired, it is offered by a number of companies and a typical example of a slightly modified design is shown in Figure 17. It should be noted that the size of the recess appears to be critical if efficiency is to be maintained and that some increase in absorbed power is found at the low flow rates.

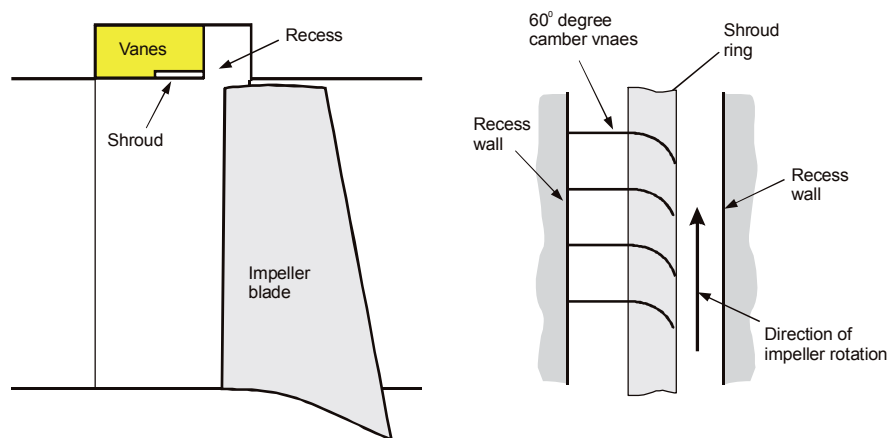


Figure 16: Ivanov Design of Anti-Stall Rings

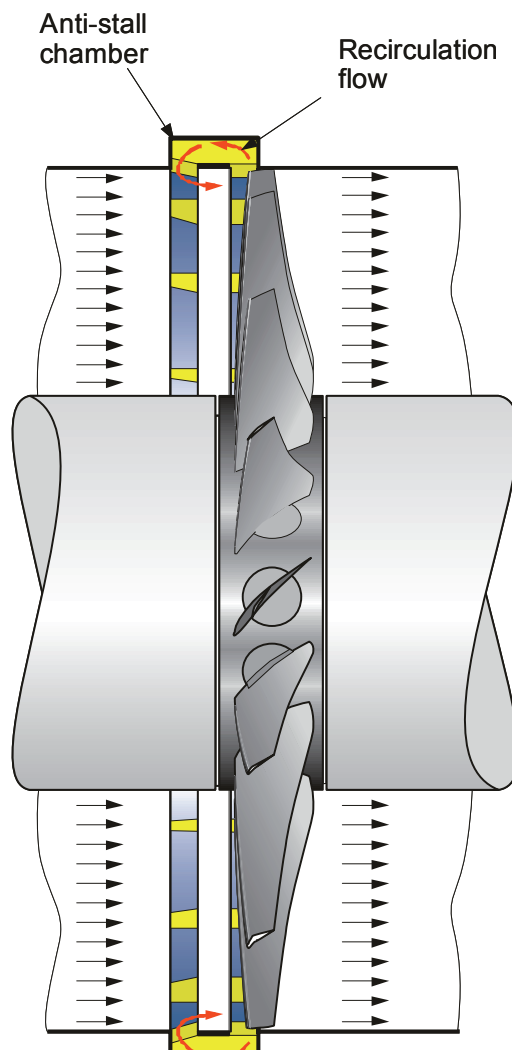


Figure 17: Axial Fan with Anti-Stall Chamber

13. CONCLUSIONS

Many cases of “stall” could be avoided if a better calculation of system resistance was made by the system designer. Equally the fan manufacturer should look to more careful fan selection, providing non-stalling designs wherever possible, or selecting well away from the stall region. Only then should anti-stall devices be considered. These modifications inevitably increase the first cost of the fan, but the reward is more reliable operation and less likelihood of failure.

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62, boulevard de Sébastopol
75003 Paris, France
Tel. 33 1 49 96 69 80
Fax. 33 1 49 96 45 10
E-mail: s.becirspahic@eurovent-certification.com

National Associations

SPAIN

AFEC

Francisco Silvela, 69-1°C
E-28028 MADRID
Tel. 34/91/4027383
Fax: 34/91/4017927
E-mail: afec@afec.es
Contact: Mr José María Ortiz

ANEFRYC

Principe de Vergara, 72
28006 MADRID, Spain
Tel. 34/91/4113271
Fax: 34/91/5637105
E-mail: anefryc@anefryc.com
Contact:
Mr Carlos Barnils Deulonder

BELGIUM

AGORIA

80 bd Reyerslaan
B-1030 BRUSSELS
Tel. 32/2/706 79 62
Fax: 32/2/706 79 66
E-mail: felix.vaneyken@agoria.be
Contact: Mr Felix Van Eyken

GERMANY

FV ALT im VDMA

Postfach 71 08 64
D-60498 FRANKFURT/MAIN
Tel. 49/69/6603 1227
Fax: 49/69/6603 1218
E-mail: thomas.schraeder@vdma.org
Contact: Dr Thomas Schröder

NORWAY

NVEF

Postboks 7174 Majorstua
N-0307 OSLO
Tel : 47/2/3087751
Fax: 47/2/3087752
E-mail:
mats.eriksson@nvef.no
Contact: Mr Mats Eriksson

FINLAND

FAMBSI

Eteläranta 10, P.O. Box 10
FIN-00131 HELSINKI
Tel.: 358 919231
Fax: 358 9624 452
E-mail: ilkka.salo@techind.fi
Contact: Mr Ilkka Salo

NETHERLANDS

FKL

Postbus 190
NL-2700 AD ZOETERMEER
Tel : 31/79/353 1124
Fax: 31/79/353 13 65
E-mail: fk&l@fme.nl
Contact: Ms Femke Teeling

FRANCE

UNICLIMA

F-92038 PARIS LA DEFENSE
CEDEX
Tel : 33/1/47 17 62 92
Fax: 33/1/47 17 64 27
E-mail: uniclimate@uniclima.org
Contact: Mr Antoine

DENMARK

DANSK VENTILATION

Wichmandsgade 19,
Postbox 199
5100 ODENSE C
Tel : 45 6614 1536
E-mail: email@nki.dk
Contact:
Mr Alex Rasmussen

SWEDEN

KTG

P.O. Box 55 10
SE-11485 STOCKHOLM
Tel. 46/8/7820800
Fax: 46/8/6603378
E-mail: anders.ostergren@vi.se
Contact: Mr Anders Östergren

ITALY

ANIMA

Via Angelo Scarsellini, 13
I - 20161 Milano
Italian Refrigeration
Equipment Manufacturers
Association
Contact: Mrs Myriam Poli
Tel : 39/02/45418500
Fax: 39/02/45418545
E-mail: poli@anima-it.com

SLOVENIA

HVAC Cluster / Grozd KGH

Vojkova 4
SI-5280 Idrija
Tel: +386 5 3734 120
Fax: +386 5 3734 142
E-mail: info@hvac-cluster.com
or
milos.sturm@hvac-cluster.com
Contact: Mr Milos Sturm

CO.AER

Via Angelo Scarsellini, 13
I - 20161 Milano
Contact: Mr Giampiero Colli
Tel : 39/02/45418556
Fax: 39/02/45418545
E-mail: colli@anima-it.com

TURKEY

ISKID

Yeni Halyolu Cad. Cayiryolu Sok
Bay Plaza No: 5 Kat: 3
Icerenköy 34752 Kadiköy/
ISTANBUL
Tel. +90/216-469 44 96
Fax: +90/216-469 44 95
E-mail: iskid@iskid.org.tr
Contact: Mr Tunç Korun

SWEDVENT

P.O. Box 47 103
SE-10074 STOCKHOLM
Tel. 46/8/762 75 00
Fax: 46/8/668 11 80
E-mail: magnus.everitt@svenskventilation.se
Contact: Mr Magnus Everitt