PAPER NO: TP03-05

CATEGORY: VIBRATION

COOLING TECHNOLOGY INSTITUTE

VIBRATION CONTROL: NEW FAN BLADE TIP REDUCES PULSATION

HENK VAN DER SPEK
HOWDEN COOLING FANS

Presented at the 2003 Cooling Technology Institute Annual Conference

Vibration Control; New Fan Blade Tip Reduces Pulsation

Henk F. van der Spek
Technical Manager, Howden Cooling Fans
vdspek@howden.nl

Introduction

Vibration control is a key factor for success in mechanical draft air-cooled installations. This topic has become more important over the last decade because the construction of air-cooled installations has fundamentally changed. This change is due on one hand to the introduction of new construction materials such as fiberglass reinforced polyesters and on the other hand to the pressure of cost savings. It has resulted in technical designs that have progressively been "optimized". One of the most uncertain aspects in this process has turned out to be the dynamic characteristic of the installation or more simply said, its vibration behavior. This paper aims to explain the nature of the different vibration modes and will then focus on the vibration of the fan ring or fan stack, which is caused by the passing of the fan blades. An important point of discussion is the application of vibration standards. These need to be clear, realistic, and practical.

Scope of vibration modes

When dealing with vibrations, the control parameters are the vibration frequency and the vibration mode. In fact, the fan generates vibration in mechanical draft air-cooled installation. The main vibration modes are the following:

1. Vibrations in the plane of the fan at the fan rotation frequency
2. Rocking and jumping modes of the motor+fan combination
3. Vibration of the fan stack at blade passing frequency

Vibrations in the plane of the fan at the fan rotation frequency

Vibration at the rotation frequency of the fan immediately indicates imbalance of the fan. This means that if vibration at the rotation frequency of the fan is observed, imbalance should always be checked first. Both the origin of the imbalance and its remedy are clear. Imbalance is caused by an unequal or an asymmetric mass distribution and it must be corrected by changing the mass distribution by adding, repositioning, or removing mass. Typically, the most practical way is by adding mass. It must be emphasized that besides unequal mass distribution of the impeller itself, alignment errors and installation deviations may also result in similar effects. For cooling fans, ISO standard 1940 is useful. It defines the circumferential speed of the center of gravity as the ruling parameter. In the industry, a value of 6.3 mm/s is generally accepted. This concerns the so-called static imbalance.
One should be aware that imbalance manifests itself in two ways. A so-called static imbalance, which is caused by an unequal mass distribution with respect to the axis of rotation, is well known. Less known is the situation of an equal mass distribution around the axis of rotation, but distributed asymmetrically with respect to the plane of rotation.

The result is so-called dynamic imbalance. Unfavorable fan blade tracking can result in this dynamic imbalance.

Rocking and jumping modes of the motor+fan combination

The rocking and jumping modes of the motor-fan combination are the most complex because they appear at different frequencies and with different modes. These vibration modes occur particularly in air-cooled installations, which are built with support structures of limited stiffness. The origin of these modes is the interaction of the passing fan blades with support structures. When a fan blade passes the fan bridge, it experiences a reaction pressure pulse from that bridge. If the fan has an odd number of blades, this mostly results in a rocking vibration mode that can be either in line with the fan bridge or perpendicular to that fan bridge. With an even blade number, there is a trend to jumping excitation.

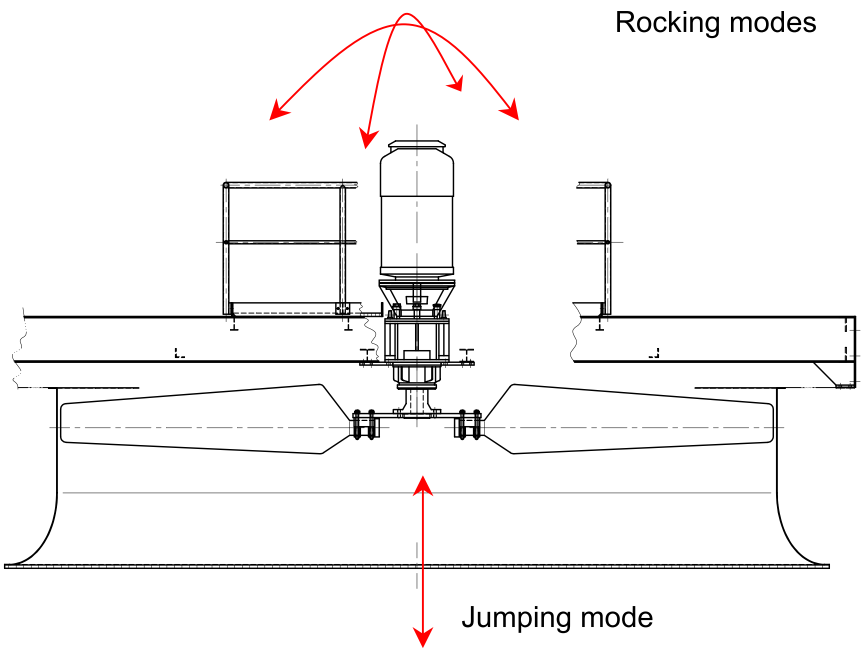

Figure 1: Vibration modes of in-line gear unit motors

All those excitations are not that strong. Nevertheless, they can lead to problems where support elements have limited stiffness. For instance, if the natural frequencies of the vibration modes correspond or get close to the blade passing frequency, resonance can be the result, even for the double and multiple values of this blade passing frequency. Limited support stiffness may arise for instance from any of the following:

- In-line gearbox-motor units without additional motor support
- Plate supports for gearboxes instead of structural ones
- Insufficiently structured fan bridges

These excitations are most known from 3 and 4 blade fan applications and have made those applications less popular despite their economic attractiveness.

Vibration of the fan stack at blade passing frequency

A difficult vibration phenomenon is the vibration of the skin of the fan ring by the passing of the blade tip of the fan. As long as the blade tip does not touch the fan ring, which has to be avoided, the pulsation on the fan ring is not caused by the impeller itself, but indirectly by the pressure field around the blade tip. This pressure field around the tip of each fan blade moves with each blade of the impeller around the fan ring and results, at a fixed position on the fan ring, in a pulse every time a blade is passing. This means that the frequency of the pulsation is equal to the blade passing frequency. In order to estimate its magnitude and to reduce the pulsation, the nature of the pressure field must be well understood.

Pressure field around blade tip

This pressure distribution around the blade tip of an axial flow fan is ruled by the following two phenomenons:

1. Lift
2. Surface velocity pressure at the tip

1. Lift is the useful pressure difference over the blade airfoil, which forces the air to flow along the heat exchanging flow resistance in an air-cooled installation. In fact, it is this same phenomenon, which keeps us flying in airplanes. The pressure difference over the airfoil is caused by the difference in air velocity on either sides of the airfoil. The characteristic shape of the airfoil is the key to generation of this velocity difference. Figure 2 illustrates this phenomenon for an airplane wing section.

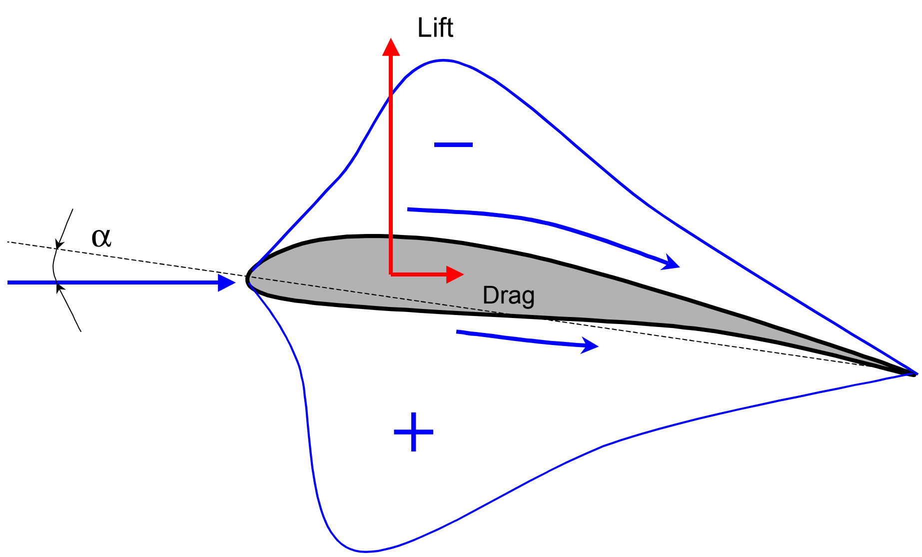

Figure 2: Lift and drag on an airplane wing section

The theoretical mathematical value for the lift L is:

$L = 0.5 \times \rho \times vt^2 \times Cl$		[Pa]	{1}
L	= Lift (pressure drop)	[Pa]	
ρ	= Air density	[kg/m^3]	
vt	= blade tip speed	[m/s]	
Cl	= lift coefficient (0-1.6)	[-]	

The value of the lift coefficient Cl depends strongly on the so-called angle of attack α between the blade core line and the undisturbed air velocity. Transforming this flow situation from an airplane wing to an axial flow fan, it becomes clear that in an axial flow fan the flow and pressure distribution is more complicated than for an airplane wing. The reason is dual. First, the air velocity "perceived" by the airfoil is the result of the addition of two velocity vectors, the blade's circumferential velocity and the axial air velocity. Second, the value of both these two velocities vary along the radius of the fan blade. The circumferential velocity of the fan blade varies exactly with the blade radius in a linear fashion. However the variance of the axial air velocity is more complicated and not easy to predict. Figure 3 illustrates the pressure field around the airfoil of a fan blade tip.

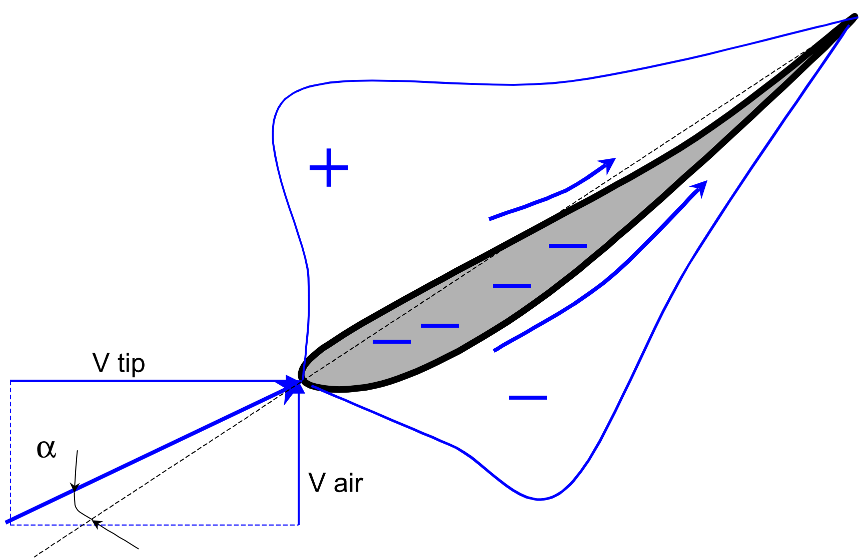

Figure 3: Pressure distribution around the airfoil at the tip of the fan blade

2. Surface velocity pressure, the second pressure component, in the pressure field around the blade tip is similar in nature to the lift phenomenon. Due to the relatively high speed at the blade tip surface, an under pressure is generated. Its theoretical value is:

$$ptip = 0.5 \times \rho \times vt^2 \quad [Pa] \quad \{2\}$$

ptip = surface velocity pressure at the blade tip [Pa]

By multiplying this value with the surface A of the blade tip, the pressure obtains the characteristic of a force F, directed inward.

$$F = ptip \times A \quad [N] \quad \{3\}$$

F = Force by pulse [N]
A = Tip surface [m^2]

It is this pressure distribution defined by the lift and the surface velocity pressure which moves around with each fan blade tip that is then experienced as a pulse at a fixed position on the fan ring.

Verification

In order to verify this model and to quantify these pulsations, a full scale, 32'-0" diameter test facility was built in Hengelo, The Netherlands. It consists of a standard fan ring, fully functional axial flow fans of various types, and a variable speed drive. The airflow direction is downwards. In this way the annular gap and rough ground surface simulates the flow resistance of an air-cooled installation. See figure 4.

Figure 4: 32ft Test facility for blade tip to fan ring pulsation simulation

The pressure distribution field around the fan blade tips as it is "felt" by the fan ring can be accurately mapped. This is achieved by high-speed data acquisition and measuring sequentially the pressure in real-time at several points traversing the blade cross-section. See figure 5.

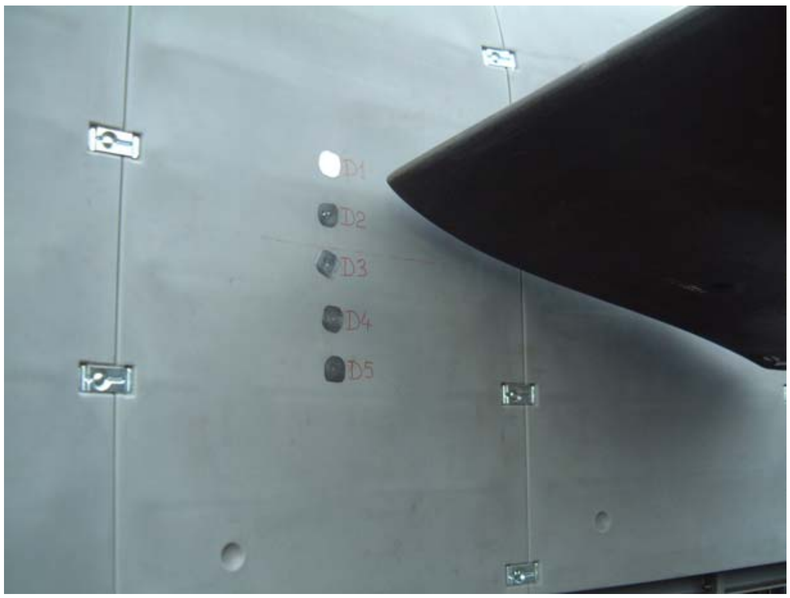

Figure 5: Measuring probes for the real-time pressure readings

The theoretically expected pressure distribution, based on the model of the two pressure field components of lift and surface velocity pressure, is shown in figure 6.

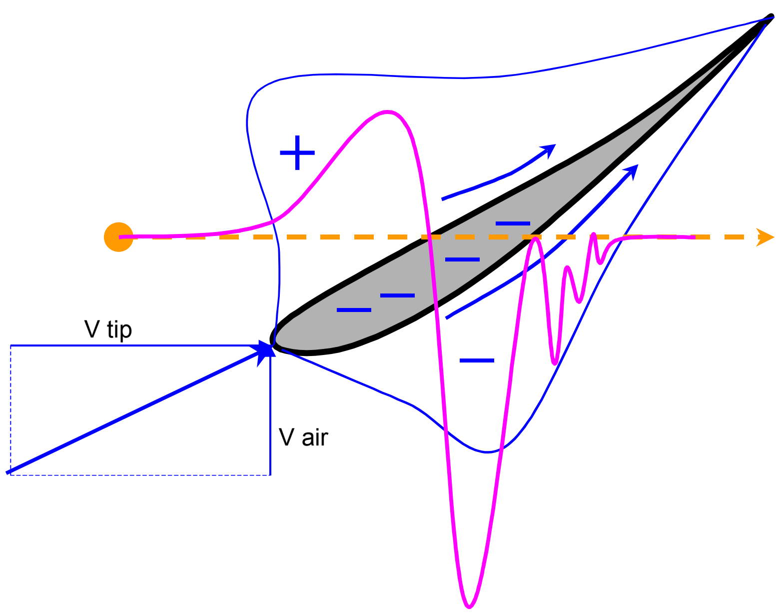

Figure 6: Expected pressure distribution

Figure 7 shows the real reading. There is a remarkable correspondence between the real pressure pattern and the theoretical one. The separate influence of the lift and of the surface pressure can be clearly observed.

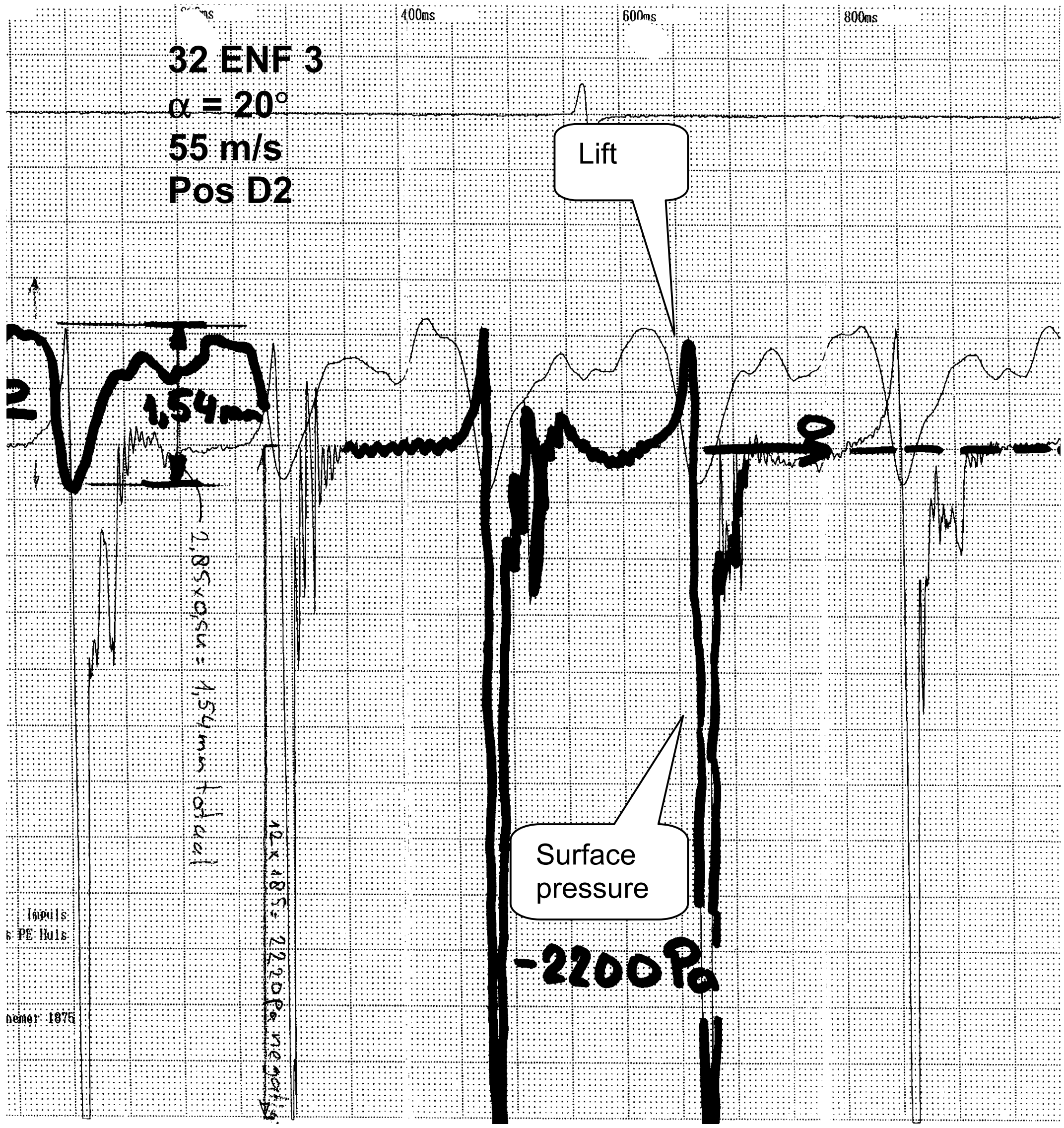

Figure 7: Real-time pressure distribution reading on the fan ring

Strategy for reduction of pulsation

Now that the nature of the pressure distribution is clear, the challenge is to reduce the pulsation without losing fan performance. The pulsation is clearly caused by two phenomenons; the lift, one that is fundamentally needed and indispensable, and a second one that is not useful. It is, in fact, only disturbing. This is the surface velocity pressure at the blade tip. The research strategy for improvement must therefore be to maintain lift and to reduce the effect of the surface velocity pressure. Since the magnitude of the pulse is proportional to the surface of the blade tip (formula {3}), reduction of that surface while maintaining the shape and effect of an airfoil was investigated. The solution was found in an airfoil shaped device on the blade tip, which follows the spherical line of the profile. See figure 8.

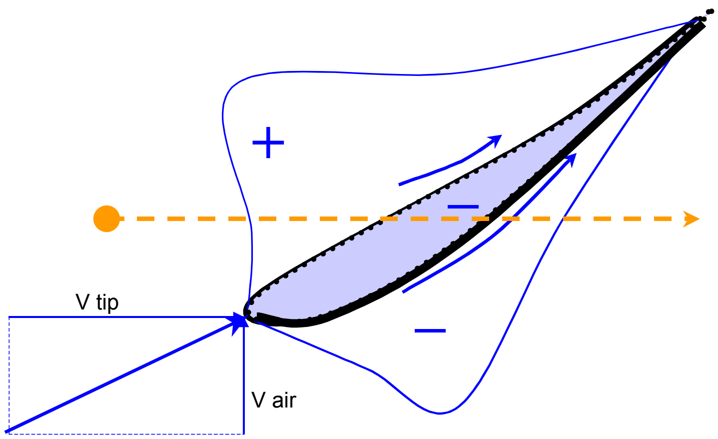

Figure 8: Proposal for adaptation of the blade tip with a so-called Aerotip

An actual sample of a fan with an example of the Aerotip is shown in figure 9.

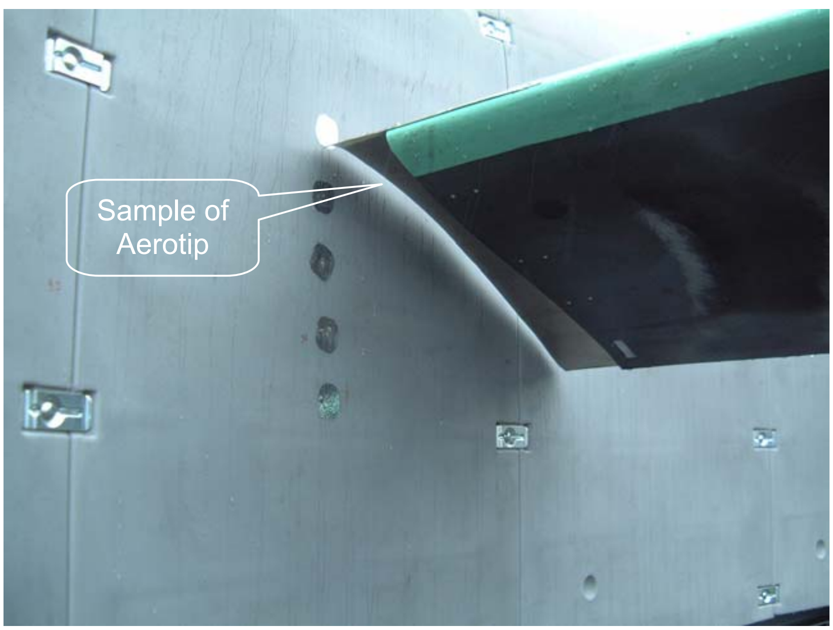

Figure 9: Example of an Aerotip application in the test rig

Figure 10 compares the new pressure distribution and the reference one of figure 7. The change in the pressure distribution by the modification with the Aerotip is remarkable. The pulse caused by the surface velocity pressure of the blade tip has been minimized,

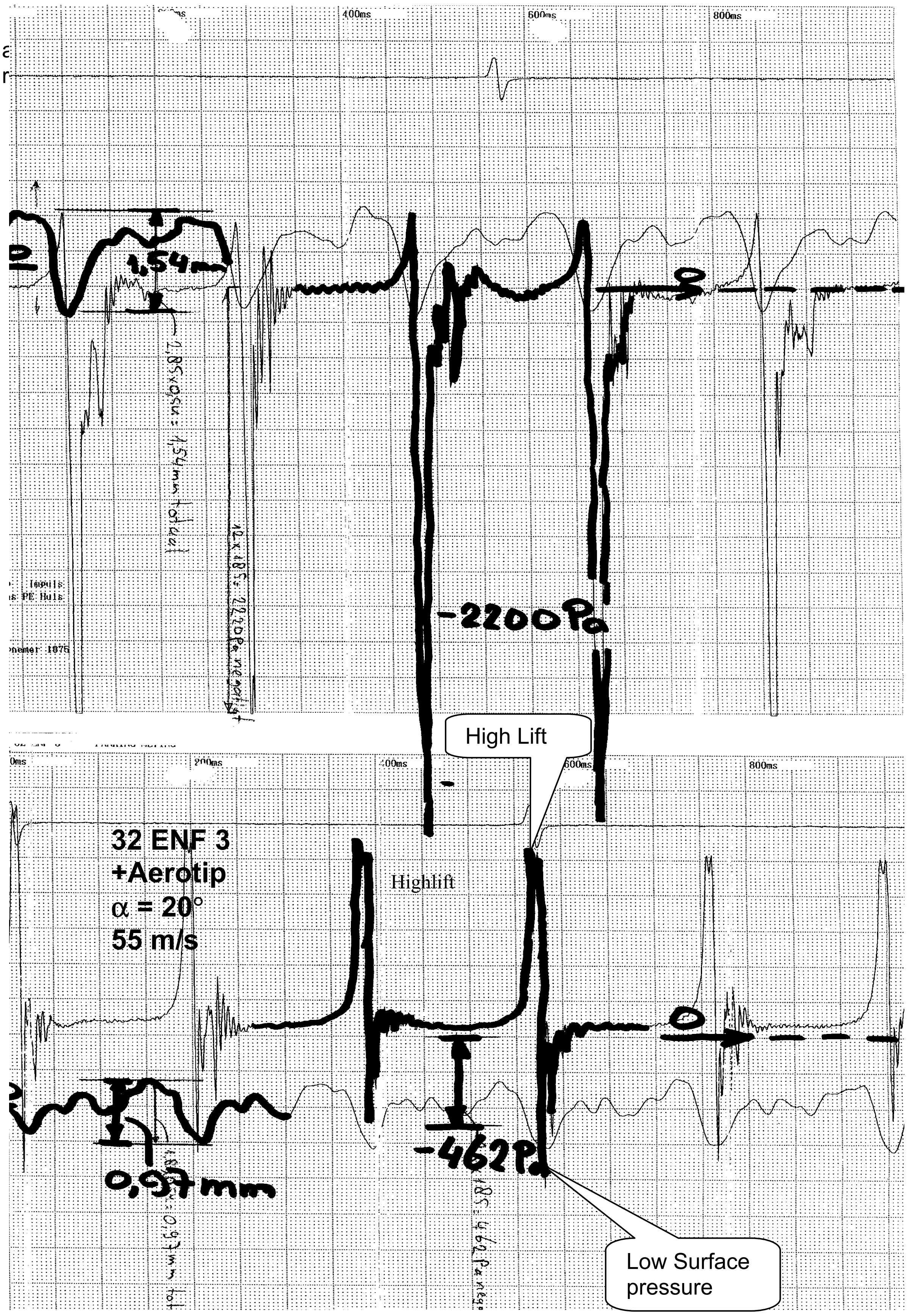
400ms
600ms
800ms
1,54mm
-2200Pa
High Lift
200ms
400ms
600ms
800ms
32 ENF 3
+Aerotip
α = 20°
55 m/s
Highlift
0,97mm
-462Pa
Low Surface pressure

Figure 10: Comparative pulse distributions for standard fan and fan with Aerotip

Reduction of fan ring pulse and vibration of the cooler structure

Pressure distribution is not the correct variable for monitoring vibration. In fact the pulsation itself only provides the mechanical excitation, which is transmitted to the fan ring and is then absorbed by the structure as vibration. Vibration is expressed as an effective velocity Vrms or as a peak-to-peak displacement. Its magnitude depends on the level of the transmitted pulsation, and also on the stiffness and the damping characteristics of the fan ring and of the supporting structure.
The pressure distribution, which was found above, suggests that the application of the Aerotip has a favorable influence on the vibration level. This was checked by measurement of vibration and real-time displacement on the fan ring of the test facility. The results are very encouraging. The vibration levels of the traditional fan are 50 % higher than the new fan concept with Aerotip. This phenomenon was measured for all common blade chord widths and was found to apply to them all. See figure 11.

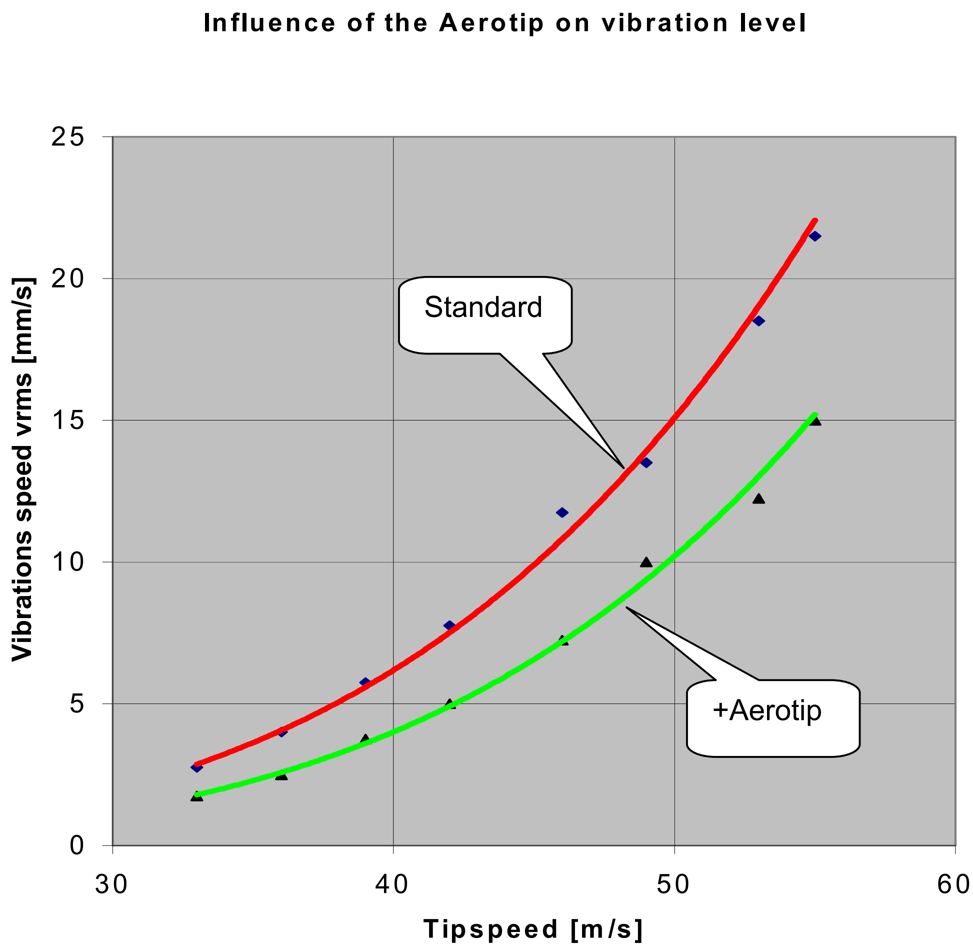

Figure 11: Vibration level on the fan ring as a function of the fan rotation speed. It is clear that the standard fan has a 50% higher vibration level than the fan with Aerotip

Additional benefits

The research for the Aerotip was initiated by the wish to reduce the vibration of the fan ring and supporting structure. This aim was achieved as previously demonstrated. However, the comparative pressure distribution readings indicate an additional advantage of the Aerotip; an increase of the lift generated at the blade tip. Since this is where the blade is turning at its highest velocity, a significant fan performance improvement can be expected.

Comparative readings were taken of pressure and efficiency versus air flow for conventional fans and fans with Aerotips. The results were favorable:

- maximum pressure levels have increased by up to 10%,
- fan efficiency has increased up to 5 % higher
- noise levels were 1 to 2 dB(A) lower.

See figure 12.

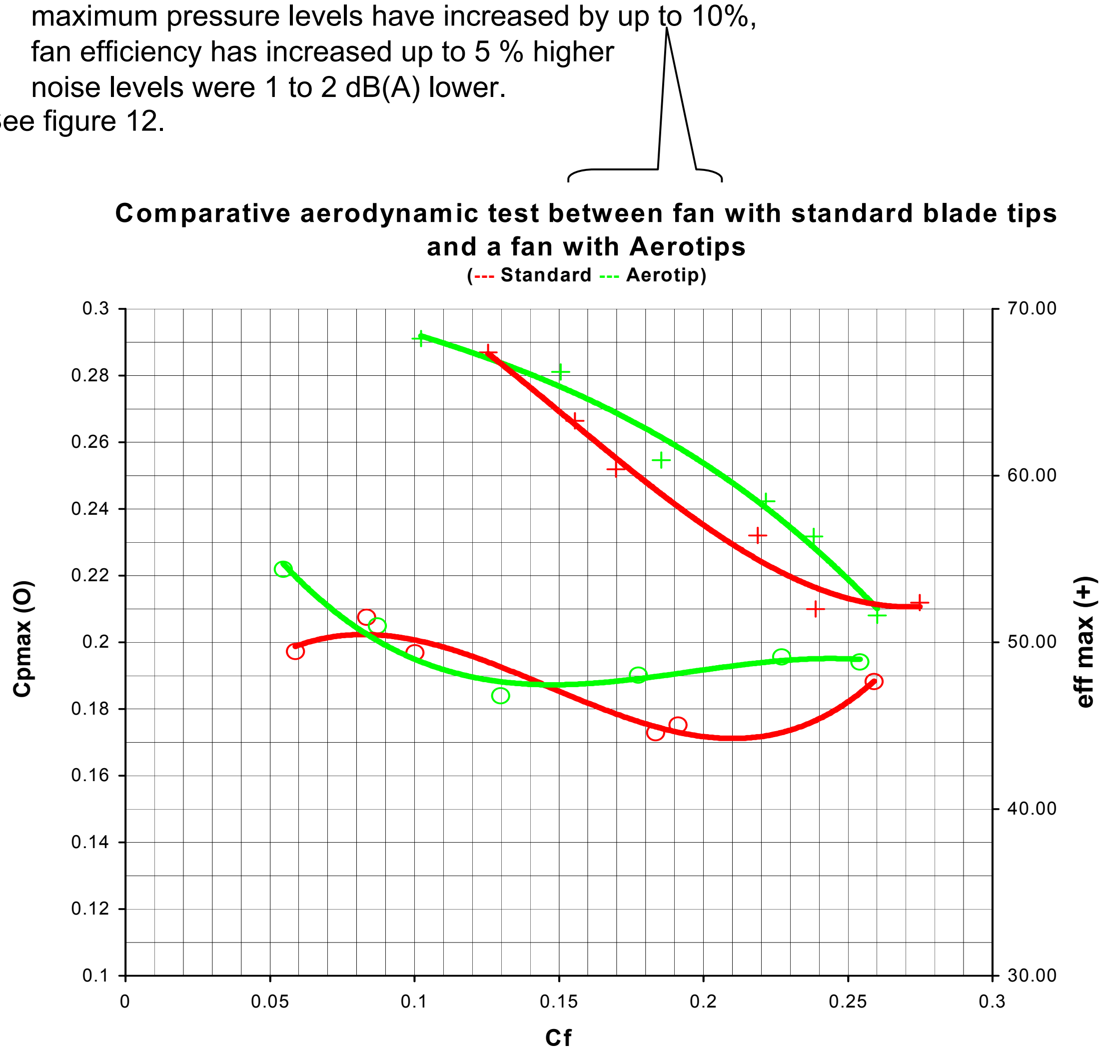

Figure 12: Comparative performance of fan pressure and fan efficiency versus airflow for conventional fans and fans equipped with Aerotips

Vibration codes and standards for cooling applications

Present national and international standards applying to vibration of rotating machinery are frequently not suitable for application in typical cooling installations. A common sense approach to applying these vibration standards is essential. Consideration should be given to extending the above standards to suit the particularities of cooling installations.
Important acceptance criteria for such standards are:
- direct applicability to air-cooled systems
- simple verification
- world-wide recognition

Particular vibration-related features of air-cooled installations are:
- large diameters
- low rotation speeds
- relatively soft foundations
- continuous operation

These features are quite extreme for general rotating equipment diagnostics. For instance, a general guideline by IRD Mechanalysis focuses too little on the particular features of air-cooled installations. Also, the CTI Operation Manual chapter 10 can be confusing with respect to the definition of the vibration parameters of amplitude and peak-to-peak displacement. In addition, the definition is not extensive enough for different types of foundations for drive motors and gearboxes.

Considering all these points, ISO standard 10816 –1 is remarkably distinguishing because it focuses clearly on the different characteristics of machine vibration.
It classifies machines into their basic concepts. For instance, its class IV ("*Large prime movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurements*") corresponds quite well with air-cooled installations. This is particularly so in combination with its application zone B: *"Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation."* An extra benefit of the ISO standard is, of course, its world-wide recognition.

Conclusion

Vibration phenomenons in forced draft air-cooled installations have to be analyzed on the basis of understanding of the excitations by the fan. The step-by-step analysis of the pulsation on the fan ring and structure has confirmed this and has provided the tools to significantly reduce the vibration level by application of an Aerotip. An additional benefit of the Aerotip is the increase of the pressure level by up to 10 percent and a fan efficiency increase of up to 5 percent.