Preventing Disasters: On-Site Frequency Analysis of Jet Fans

[1]V.Carstens, [2]P. Pospisil

[1]Consultant Aerodynamics Thermodynamics, Germany

[2]IP Engineering GmbH, Switzerland

ABSTRACT

Fractures in the blading of jet fans pose a serious risk to the traffic inside the tunnel. This concerns on the one hand the non-availibility of the fans as an important part of safety systems, and on the other hand the endangerment of vehicle occupants by fractions flying around. The precautionary measures used so far for supervision consist only of a regular measurement of the RMS values of vibrations at the fan housing, which provide insufficient information about previous damage to the blades. In this paper, an experimental method is presented with which cracks and already existing partial fractures in the fan blades as well as other anomalies in fan operation can be reliably detected. Although the procedure is more complex than the usual simple vibration measurements, it can be carried out within the scope of the regular maintenance and inspection of the tunnel ventilation equipment.

Keywords : jet fans, failure detection, forced vibrations, signal analysis, operational safety

1. INTRODUCTION

The reason for the investigations and the testing of a new measuring method was the accident of a jet fan in the 2.6 km long Flüelen tunnel in Switzerland. Unfortunately, the exact time of the failure could not be exactly determined (between January and April 2019), only the discovery of broken blades (Fig. 1) drew attention to the accident. Noticeable were the fracture areas at the blade foot. While approximately 80% of the fracture surface of one blade was corroded, which means that there must have been a partial fracture before, the other blades showed a violent fracture without corroded surface, which was caused either by the impact of the torn off blade or by the sudden imbalance of the fan impeller.

Fig 1: damaged jet fan: broken blades parts (left), blade foot (right)

Technically speaking, a jet fan is a turbomachine with only a single rotating stage, the fan wheel. To improve the aerodynamic efficiency, guide vanes can be installed in front or behind the rotor. The fan wheel (Fig. 2), consisting of hub and blades, is an elastic structure attached to an approximately rigid fixed point, namely the fan casing with drive motor and struts.

Every elastic structure - set into vibration by excitation - has an unchangeable pattern of natural frequencies with the associated eigenmodes, which can be understood as a structure-dynamic DNA profile that only changes if a certain material property, such as the stiffness of the structure, suddenly takes on different values. Cracks or partial fractures in blades or hubs reduce the stiffness of the structure, which in turn leads to reduced natural frequencies with correspondingly altered natural modes.

Fig 2: undamaged wheel of a jet fan

The principle of the novel measuring method, which has been tested on the object, is based on the change of the natural frequencies in case of damage. If the structure-dynamic parameters, especially the natural frequencies, are measured at regular intervals, starting with the factory condition of the fan wheel, the drop in these parameters provides a reliable indication of imminent blade breakage.

2. VIBRATIONS IN TURBOMACHINERY

Turbomachine vibrations are divided into two types, namely self-excited and forced vibrations. While self-excited vibrations (also known as "flutter") only occur in highly loaded turbomachine stages with slender blades at low natural frequencies, forced vibrations (also known as "forced response") are always present in a turbomachine. The forced vibrations that are important for the jet fan can have both structural and/or aerodynamic causes:

Forced excitation due to structural causes:

- unbalances, caused by uneven mass distribution
- damage of the rotor bearing, suspension vibrations

Forced excitation due to aerodynamic disturbances (Figure 3) :

- influence of the struts of the engine suspension on flow around the fan blades
- uneven spacing of the fan blades along the rotor hub
- incorrect inflow angle of individual blades
- stall at the leading edge of the blade during rotor run-up

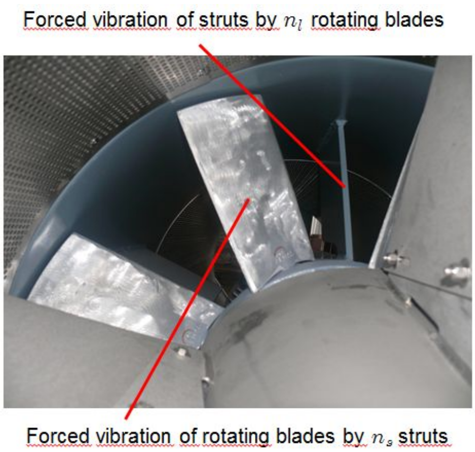

Fig 3: forced aerodynamic vibration in a jet fan

Decisive for the strength of the excitation, i.e. the amplitude of the forcibly moved elastic structure, is the frequency proximity of the excitation frequency with one of the natural frequencies of the fan wheel. This relationship can be illustrated in the Campbell diagram (Figure 4).

In the Campbell diagram, the natural frequencies of the blades are plotted against the frequency of the structural-dynamic or aerodynamic disturbance events. The frequency of the disturbance events increases linearly with the speed and is represented in the Campbell diagram by a straight line whose gradient is determined by the number of disturbance events per revolution (the so-called engine order, Fig. 4). Since the natural frequencies increase only slightly with increasing speed (increase in blade stiffness due to centrifugal forces), the straight lines of the excitation forces intersect them and cause resonances.

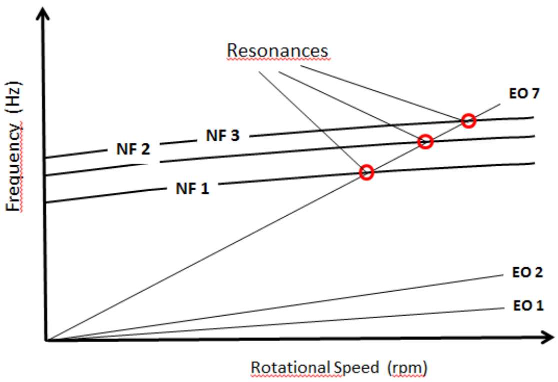

Fig 4: Campbell-Diagram (Principle)

3. THEORETICAL MODEL OF THE VIBRATIONS

If we consider an elastic structure as a continuum, we get the partial differential equations of continuum mechanics for it. However, if the structure is broken down into discrete masses between which stiffness and friction forces act, the following ordinary differential equation of structural dynamics applies to such a system

(1) $$M\ddot{\vec{x}} + D\dot{\vec{x}} + K\vec{x} = \vec{F_E}(t)$$

with the system variables mass matrix M, damping matrix D, stiffness matrix K and the excitation forces $\vec{F_E}(t)$. The vectors $\vec{x}$, $\dot{\vec{x}}$ and $\ddot{\vec{x}}$ denote deflection, speed and acceleration of the mass element. For simple structural dynamic models, the matrices M, D and K are small, for a calculation with finite elements they can take on very large dimensions. Is the right hand side of (1) zero, we have an eigenvalue problem whose eigenvalues are the squares of the natural frequencies while their solution vectors (eigenvectors) represent the natural modes. If forced excitation is present, the solution vector $\vec{x}$ contains the dynamic response function to the excitation vector $\vec{F_E}(t)$.

In order to get an overview of the natural frequencies and natural modes of a fan impeller, a simplified structural dynamic model with vanishing damping is considered, in which hub and blades are replaced by discrete masses connected to each other by springs (Figure 5). The corresponding hub masses (black) are connected with springs among each other as well as at the rigid motor (large black), while the blades (red) have a spring connection only with the hub masses. For each mass the dynamic equilibrium condition is then

(2) $$m_i \ddot{x}_i - k_i x_{i-1} + (k_i + k_{i+1}) x_i - k_{i+1} x_{i+1} = 0 \ , \ i = 1, ..., 2N$$

with the mass m_i and the spring stiffness k_i between the masses with indices i and $i+1$. A rotor with 10 blades is then represented by a system of 20 equations with 20x20 matrices M and K, whereby in particular the cyclic boundary condition for the movement of the hub masses has to be taken into account. The system resulting from equation (2) and the structure of the matrices M and K will not be shown here in detail, instead the frequencies and modes following from 2) will be described qualitatively.

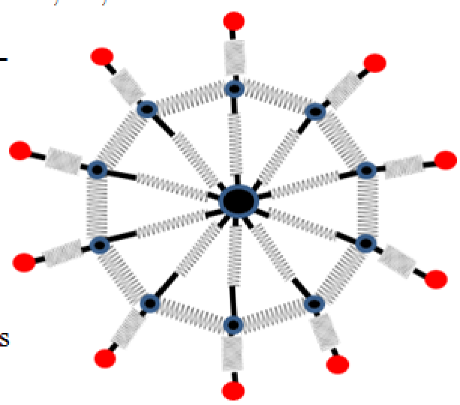

Fig5 Spring-Mass-Model

The spring-mass model shown in Figure 5 has 20 natural frequencies and 20 associated natural modes. Remarkable for a cyclic system is the occurrence of double natural fre-quencies, each of which is associated with a pair of eigenmodes characterized by clock-wise and counterclockwise traveling blade deflections, the deflections occurring with a constant phase difference angle between the blades (so-called Traveling Wave Modes). Also noteworthy are two rigid body modes, in which the hub and blade masses as a whole vibrate either in phase or in counter-phase.

4. FREQUENCY MEASUREMENTS

The natural frequencies of a structure are very sensitive to material fatigue as well as to damage in the form of small cracks or material chipping. In both cases, the reduction of natural frequencies is particularly noticeable at the lower frequency values. This property is the basis of the presented method of natural frequency measurement, which - repeated at regular intervals - reliably detects imminent blade breakage. For frequency measurement, the structure must be set into vibration. This is done either with the impact-hammer method or a forced excitation with a sliding sine wave. The dynamic response to the excitation is then measured with sensors whose output signal is proportional to the speed or acceleration of the vibrating surface.

Figure 6 shows an example of the sequence of the impact hammer method for measuring the natural frequencies. By striking the elastic structure with the impact hammer, this corres-ponds mathematically seen to the forced excitation with a delta function, all natural frequen-cies of the structure are excited equally. The middle graph of Fig. 6 showing the response signal in the time domain is discretized and subjected to a Fourier transformation, whereby a window function, selectable in form and width, is used for sliding over the digitized signals.

The result of the frequency analysis can be seen in the right graph of Fig. 6 and represents the frequency spectrum of the structure. The clearly recognizable peaks in the frequency spectrum are the natural frequencies.

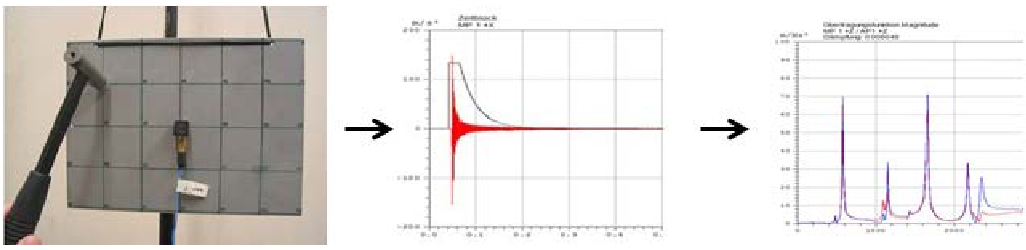

Fig 6: Frequency Measurement, Impact-Hammer-Method

Figure 7 shows the frequency analysis for an intact blade of a jet fan. The upper both graphs show the time course of the excitation (hammer impact, left) and the signal response (right), the lower left graph shows the Fourier transform of the signal response. For clarification, only the lower three natural frequencies are shown in the Fourier transform, where the lowest frequency is assigned to the rigid body vibration of hub and fan wheel taking place in phase, the other two are adjacent frequencies representing the pair of Traveling Wave Modes with the lowest natural frequency. Noteworthy for this pair is the observed beat of the signal in the time domain. The lower right graph shows the frequency response function (FRF), i.e. the ratio of response signal and excitation. It is clear that the frequency peaks are observed at exactly the same frequency values as in the graph of the Fourier transformation, only the absolute values of FFT and FRF are different.

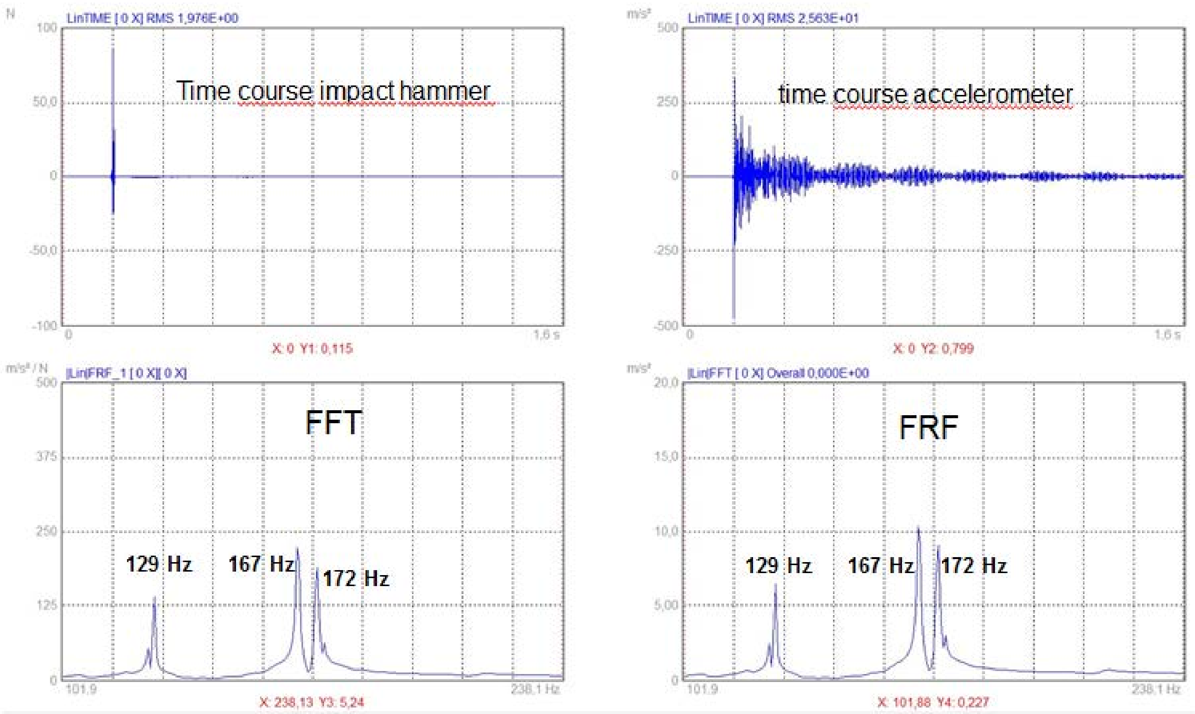

Fig 7: Frequency analysis of a blade, measured in situ

An interesting issue is the homogeneity of the fan blades with respect to their natural frequencies. One might expect that due to manufacturing defects or deviating mounting on the hub the natural frequencies show a spread between five and ten percent around an average value which means that deviations due to damage are more difficult to detect. However, this not the case as Table 1 demonstrates. The scattering around the average value of the natural frequencies is very small with a standard deviation within the accuracy of the Fourier transform.

Table 1: Natural frequencies of an in-situ measured fan rotor

Schaufel Nr.	EF1	EF2	EF3
1	129.3	168.1	172.5
2	-	-	-
3	129.3	168.1	171.8
4	128.7	166.9	171.8
5	128.1	166.8	172.5
6	129.3	168.1	172.5
7	128.7	166.8	172.5
8	129.3	167.5	171.8
9	128.7	167.5	172.5
10	128.7	166.8	172.5
MW +- SA	128.9 +- 0.42	167.4 +- 0.59	172.3 +- 0.35

Fortunately, a jet fan which had been replaced by a new one and was planned for scrapping was found to prove the reduction of frequencies by damaging a blade (red arrow in Fig 8). A first measurement confirmes that the natural frequencies of the undamaged blades where almost the same as depicted in Fig 7.

Afterwords one of the blades was partially cut (Fig 8) and the frequency analysis was repeated. The results of the undamaged and damaged blade are depicted in Fig 9 and Fig 10 and impressively demonstrate the usefulness of the method.

For the sake of simplicity Figures 9 and 10 show a comparison for the lowest three natural frequencies of of the fan rotor together with the frequency response functions (FRF).

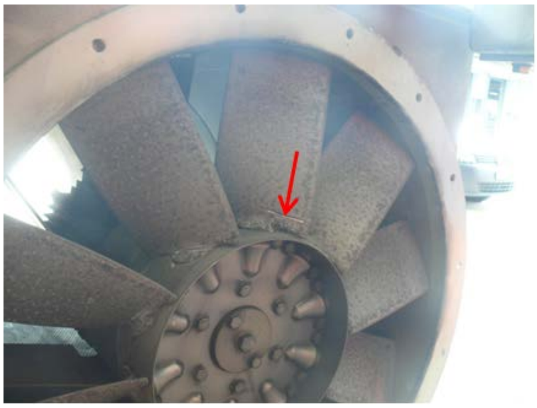

Fig 8: Damaged blade with a partial cut

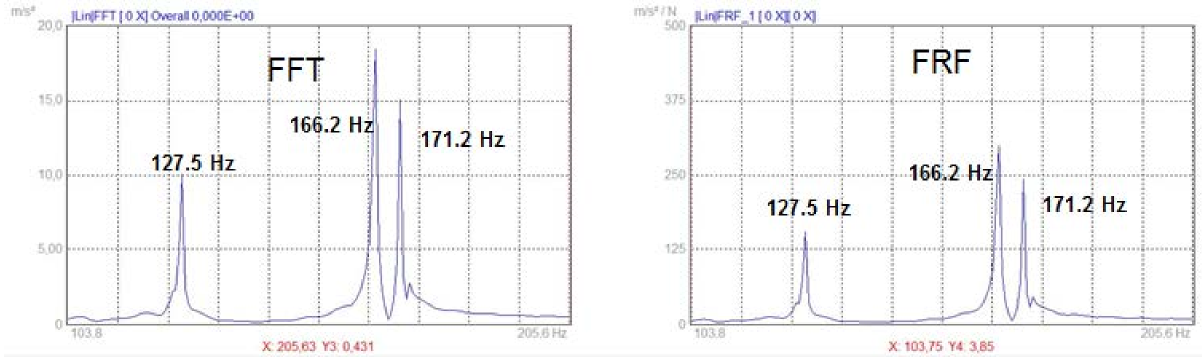

Fig 9: Frequency analysis of the undamaged blade

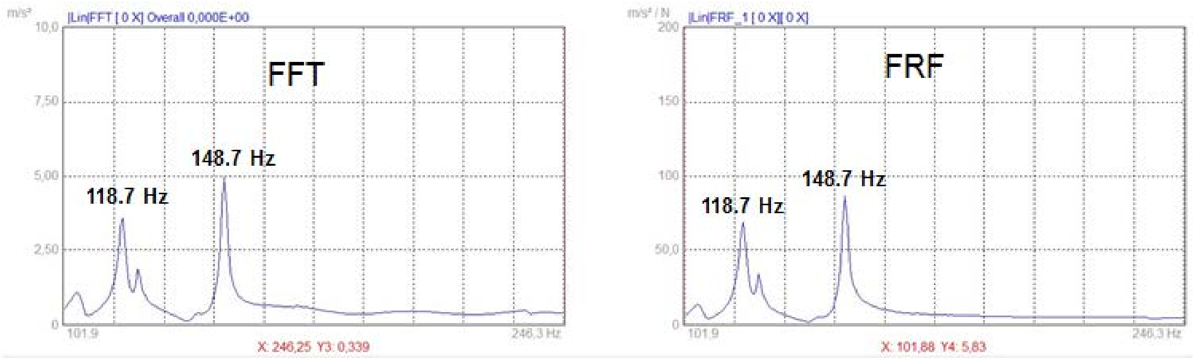

Fig 10: Frequency analysis of the damaged blade

As already mentioned, the lowest frequency is that of the rigid body motion of hub and fan wheel, while the other two closely adjacent frequencies belong to Traveling Wave Modes with almost the same frequency. The effect of the simulated crack is not only the significant reduction of the frequencies, but also the spreading of the paired modes and the fact, that the peak of the rigid body motion was always gone, i.e. that this mode was not excited by the the hammer stroke

5. FORCED RESPONSE MEASUREMENTS

The frequency measurements were completed by determining the response to an external excitation. In a fan rotor this always happens by the mutual aerodynamic interaction of the rotor with struts or guide vanes. The struts and/or guide vanes are fixed at the casing where the measuring signal is acquired. Unfortunately, only the aerodynamic excitation of the struts and/or guide vanes by the rotating blades can be measured but not the excitation of the blades by the struts and/or guide vanes, since in the latter case a complex transfer of the accelero-meter signals from the rotating to the stationary reference system ist necessary which can only be realized in an expensive test facility.

The frequency response measurements are performed by fixing an accelerometer at a suitable place on the casing of the jet fan. The response signals are recorded for a period of 30 or 60 seconds while the fan is running. A Fourier transform of the discretized signals then yields the typical aerodynamic excitation frequency which in the present case is the product of the rotational frequency and the number of fan blades (e.g. 50 Hz x 10 = 500 Hz). Figure 11 shows an example of a response measurement, where the peak near 500 Hz is clearly observable.

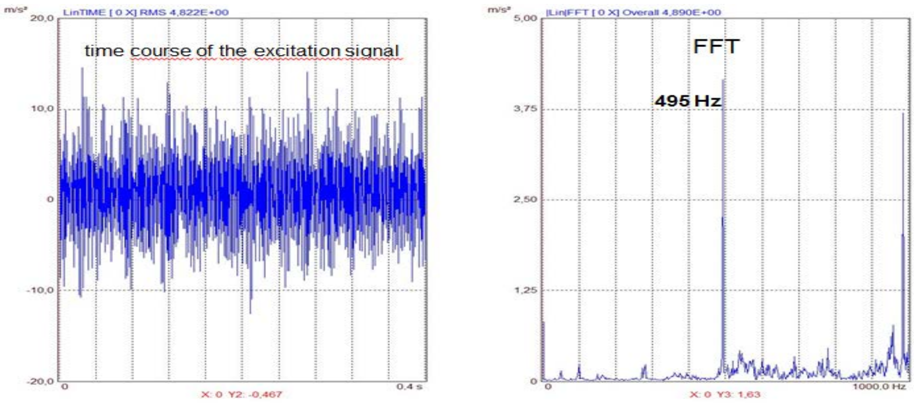

Fig 11: Aerodynamic forced response of a fan with 10 blades at normal speed

The most important task of forced excitation investigations is to determine resonances within the speed range. As already mentioned, these points cannot be determined by measurements, but have to be predicted by the Campbell-diagram. The investigated fan has 7 struts which excite the fan blades with a frequency which increases linearly at seven times the rotation frequency. If this straight line crosses the natural frequencies of the blades, resonance vibrations occur (Fig 12).

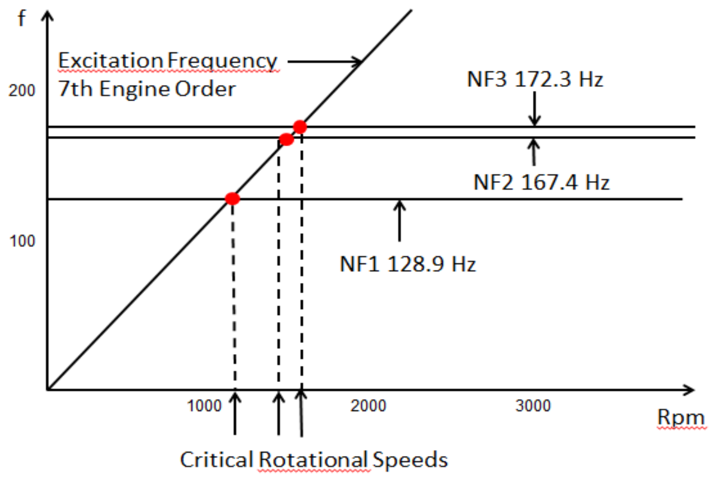

Fig 12: Campbell-diagram of investigated fan jet

6. SUMMARY

In this paper, an experimental method has been presented with which the failure of jet fans due to blade breakage can be prevented. It has been shown that early blade damage can be detected by regular monitoring of the natural frequencies of the fan impeller. In particular, this can prevent the dangerous tearing off of fan blades, which can lead to serious accidents in tunnels.

The measurements of natural frequencies have been performed by the impact hammer-method, which has been explained in detail in chapter 4.

In addition to the measurement of the natural frequencies, a method combining theory and measurement was dealt with, with which aerodynamically caused forced excitations can be predicted, which can lead to resonance vibrations. This methods leads to the prediction of resonance peaks with the aid of the Campbell diagram. Such can be used to avoid cririical speeds when using Variable Speed Drives.

It must be stressed that both the measurement of the natural frequencies and the resonance studies in the Campbell diagram go beyond the vibration measurements of the RMS values used so far, since the latter cannot detect imminent blade breakage.

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