CFD ANALYSIS OF UNSTEADY FLOW IN NON-UNIFORMLY DISTRIBUTED FAN BLADES

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SUMMARY

The objective of the present study is to compare results from a Ffowcs Williams-Hawkings (FW-H) Steady numerical solver to experimental results acquired during a CETIM campaign. The aim of the CETIM campaign was to reduce tonal noise of fans with non-uniformly distributed fan blades. The present paper describes Computational Fluid Dynamics (CFD) calculations to study complex phenomena related to unsteady flow and aeroacoustics noise in non-uniformly distributed fan blades. The studied phenomena are interactions and perturbations induced by the rotating blades motion and their impact on the aeroacoustics behavior of the fan. Numerical simulation has been carried out using Siemens STARCCM+ software. Investigations of the flow variables provided by CFD calculations were used as inputs in Ffowcs Williams-Hawkings equation to calculate sound pressure levels for related frequencies. These describe noise source terms that are determined by numerical simulations. Appropriate boundary conditions have been applied to limit-reflection of acoustic waves, and, to provide inflow and outflow of the aerodynamic field. At the end of the work, numerical simulation is compared to experimental measurements and relationship between surface pressure fluctuations and the field noise signals is highlighted.

INTRODUCTION

Advanced aerodynamic and aeroacoustics multiphysics modeling and simulation are becoming a major topic in the modern aerospace and automobile industry and even more in the design of HVAC (Heating, Ventilation and Air-conditioning) and cooling fan industries. Acoustic noise generated by a flow can be created through different mechanics but is ultimately due to fluctuations in the flow. Aeroacoustics computation of airborne noise requires highly accurate numerical approaches to deal with the complexity of phenomena involved, such as turbulent flow over solid bodies, high-speed turbulent shear flows, structural vibration that is induced by fluid flow, turbulent combustion, and laminar instabilities. Moreover, the requirements in terms of time and space resolution for the aeroacoustics as well as the identification and calculation of aeroacoustics sources make the computation of aerodynamic generated noise often extremely time demanding. Therefore, we evaluate a simplified numerical approach that is less expensive, light, fast, and suited for rotating fans: the Ffowcs Williams-Hawkings (FW-H) Steady Aeroacoustics Model available in STARCCM+.

CONTEXT OF THE STUDY

Fans on mobile machinery are often the dominant external noise source. During a working group at CETIM in autumn 2016, manufacturers expressed the wish to know acoustic performances of fans with disordered blades (non-uniformly distributed fan blades). The aim for manufacturers of mobile machines is to reduce the tonal noise level of the machines and improve the sound quality.

A CETIM study [1] aiming at testing fans with disordered blades on an excavator was carried out from 2017 to 2019 using ISO 3744 standard [2]. The objective was to compare their performances to standard fans. Figure 1 shows a schematic description of microphone locations:

- 6 microphones (black dots) arranged within a radius of 10 m (after ISO 6395 standard [4]).
- 3 microphones (blue dots) placed 3 meters in front, left and right of the gate of the fan.
- 1 microphone (red dot) at ear position in the cabin.

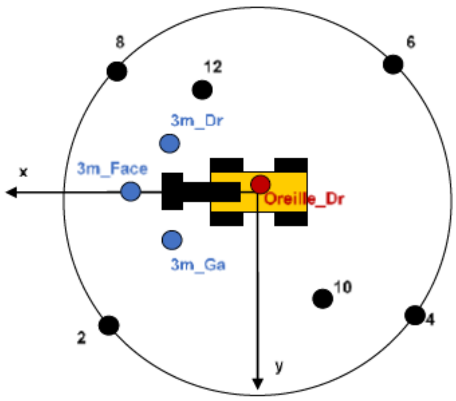

Figure 1 : Schematic description of microphones location

In this paper we focus on recordings of the 3 microphones placed at 3 meters (blue dots in Figure 1). Tests carried out in February 2019 on regular fan (uniformly distributed fan blades) and disordered blades (see Figure 2) resulted in series of acoustic spectral plots averaged over the 3 microphones with A-Weighted Sound Pressure Level (see Figure 3):

- for the originally mounted fan in series (see “Orig” label in solid blue line in Figure 3).
- for the uniformly distributed blades (see label “Reg” in red solid line in Figure 3)
- for the disordered blades (see label “VPD1”, “VPD2” in green solid lines in Figure 3)

Figure 2 : On the left photo of the disordered blade (“VDP1”), on the right the regular fan (uniformly distributed blades, "Reg")

Figure 3 shows results of the test carried at 2 489 rpm fan speed (2 150 rpm motor regime) with fans composed of 8 blades (see Figure 2). Main conclusion of CETIM study [1] is that disordered blades have little impact on the overall A-weighted level but allow nevertheless the tonal noise associated with blade passage frequency (around 333 Hz in Figure 3) to be reduced.

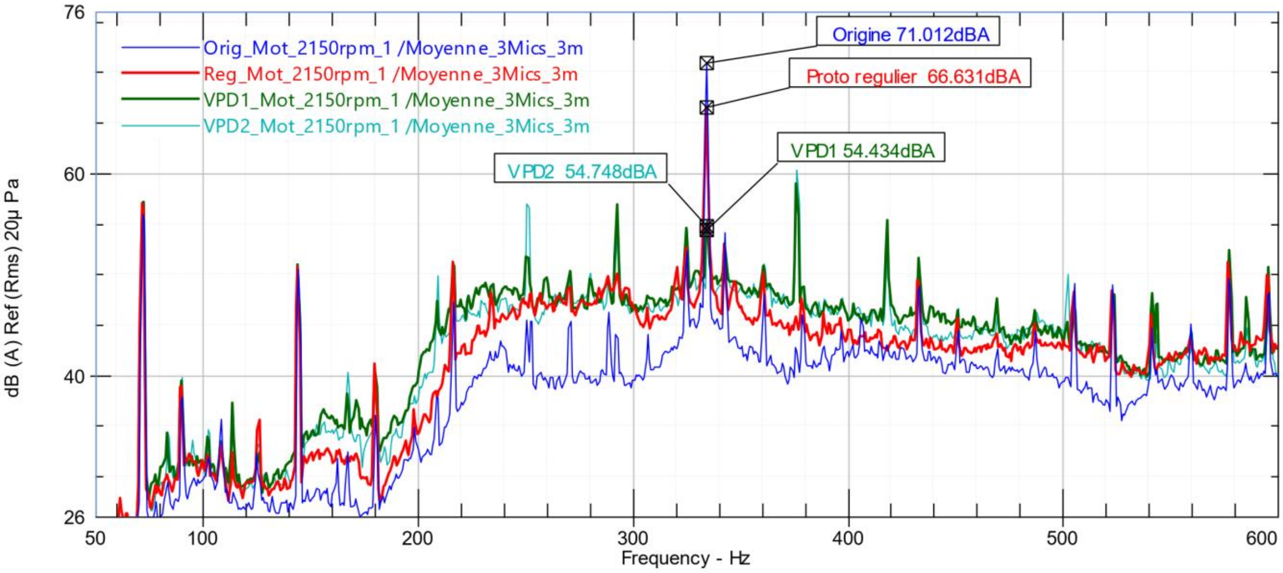

Figure 3: Mean acoustic pressure from 3 microphones at 3meters for the original fan (blue line), the uniformly distributed fan (red line) and disordered blades (green lines) [1]. Fan speed ~2500 tr/min

AEROACOUSTIC MODEL

When predicting far-field acoustics there are two main strategies that can be used:

1. Direct Noise Simulation of the sound propagation
2. Acoustic Integral Formulation

The Ffowcs Williams Hawkings (FW-H) model [3] is an acoustic integral approach. The model calculates far-field acoustics radiated from near-field flow data taken from CFD simulation.

In this paper we present a specific variant of the FW-H model available in the SIMCENTER STARCCM+ software. This model uses the FW-H acoustic analogy applied to cases where an unsteady motion problem with associated unsteady flow can be converted to a steady-state RANS problem using moving reference frames. The applications area for this model are propellers and fans and has been applied previously by, [7], [8], [9].

This model looks particularly at two sound generation mechanism. First is « Thickness surface noise » than can be thought of as caused by the rotor blade displacing the air. This contribution is dependent only on the shape and motion of the blade. The second mechanism is the « Loading noise » that is aerodynamic adverse effect due to the acceleration of force distribution of the air around the fan blades. Although they are other sound generating mechanism listed by Brentner and Farassat [5], these two sound generation contributors are considered the major one in the absence of any installation effects.

The FW-H Steady model is based on a convergent (see Figure 5) steady-state RANS approach for aerodynamic computation of fan blade loads for a multiple rotating reference frames with the FW-H Acoustic Analogy Model. It is a three-dimensional simulation with impermeable FW-H blade surfaces and three FW-H receivers located at 3 meters from the fan similarly to the previously described test setup (see Figure 1). It aims to reproduce environments of the fan blades in the excavator machine (see Figure 6). The unsteady motion problem related to the rotation of the blade is converted to a steady-state RANS problem by imposing a rotating frame of reference on a static mesh. The rotating reference frame is applied to the region of mesh containing the fan blades. The remainder of the mesh is static. Applying a moving reference frame to a region generates a constant grid flux in the appropriate conservation equations. This grid flux is calculated based on the properties of the reference frame rather than the local motion of cell vertices (see Figure 7). The thickness and loading noise from relations of the FW-H acoustics model are computed from steady-state computation of the pressure field (see Figure 8).

The steady formulation of the FW-H acoustics model allows prediction of an unsteady pressure time signal from a steady simulation by defining an artificial unsteady time-step. The solver provides an approximation of the results that you can expect from a steady state simulation. It implements the advanced time approach, to account for the time delay between the emission time and reception time. The resulting transient data can be used in a Fast Fourier Transform to plot the sound pressure level across the selected frequencies to obtain noise prediction at receiver location comparable to test spectral plot (see Figure 3).

RANS TURBULENCE MODEL

RANS turbulence models provide closure relations for the Reynolds-Averaged Navier-Stokes (RANS) equations. The $k-\omega$ SST model (Wilcox [10], Menter [11]) turbulence model is a two-equation model that solves transport equations for the turbulent kinetic energy and the specific dissipation rate to determine the turbulent eddy viscosity. It provides a better prediction of flow separation than most RANS models and accounts for its good behavior in adverse pressure gradients. It accounts for the transport of the principal shear stress in adverse pressure gradient boundary layers. It is the most commonly used model in the industry given its high accuracy to expense ratio.

NUMERICAL SCHEMES

The coupled solver solves the conservation equations for continuity, momentum, energy equations, in a coupled manner. The velocity field is obtained from the momentum equations. From the continuity equation, the pressure is calculated, and the density is evaluated from the equation of state, i.e., ideal gas law in this study. The discretization scheme used for computing the convection flux and the turbulence model is the second-order upwind convection scheme.

HARDWARE & SOFTWARE SPECIFCATIONS

Numerical calculations are carried on a DELL Tower precision 7920 with an Intel Xeon Platinum 8280 with a total 28 cores with the double precision version of SIMCENTER STARCCM+ 2021.3.1 software. Computation time necessary for a single parallel calculation of an acoustic spectral plot is approximately 3 hours on a mesh of roughly of 4.1 million cells (see Figure 4).

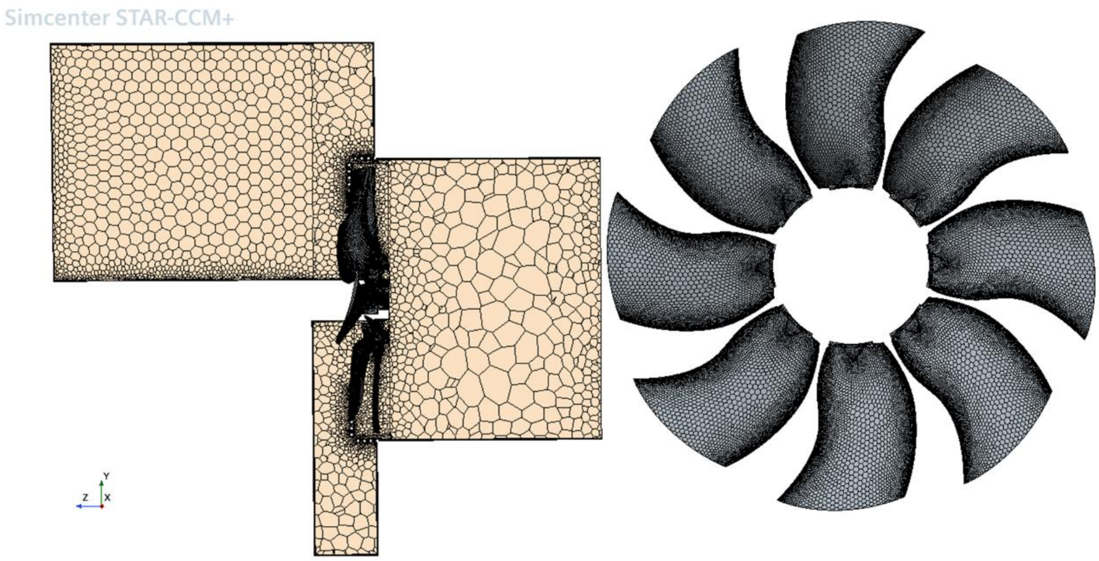

Figure 4: illustration of the polygonal mesh used in this study

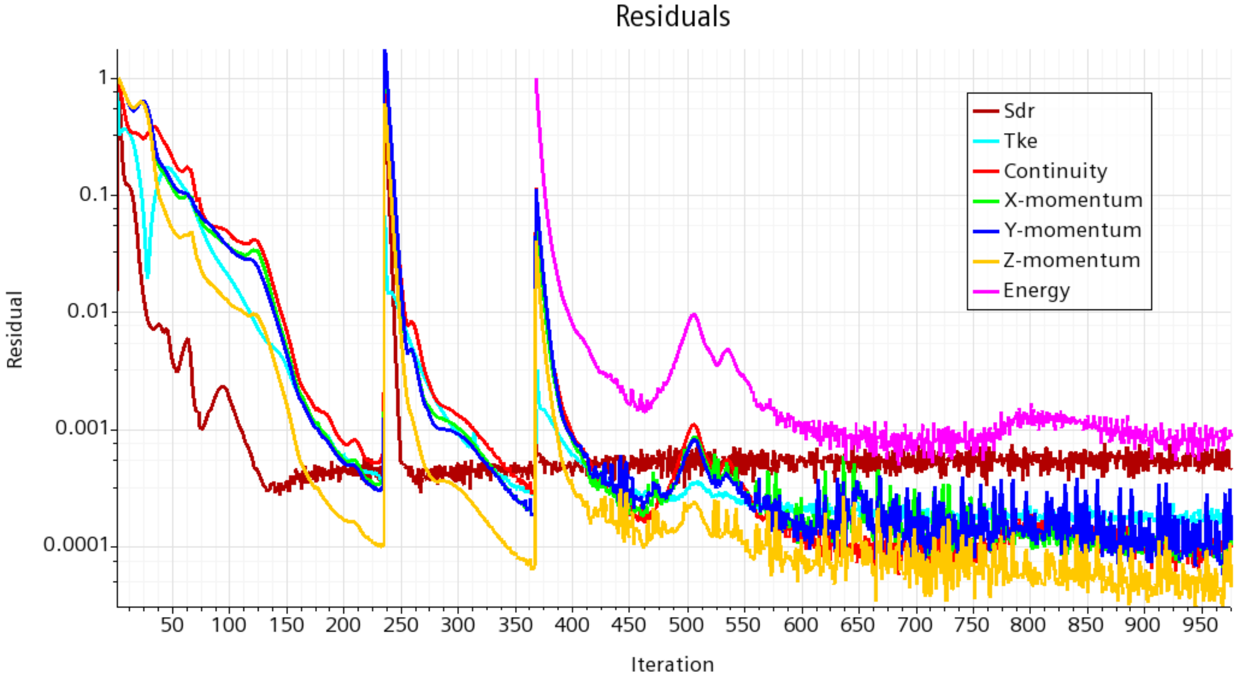

Figure 5: Convergence of the steady-state RANS residuals under 0.001

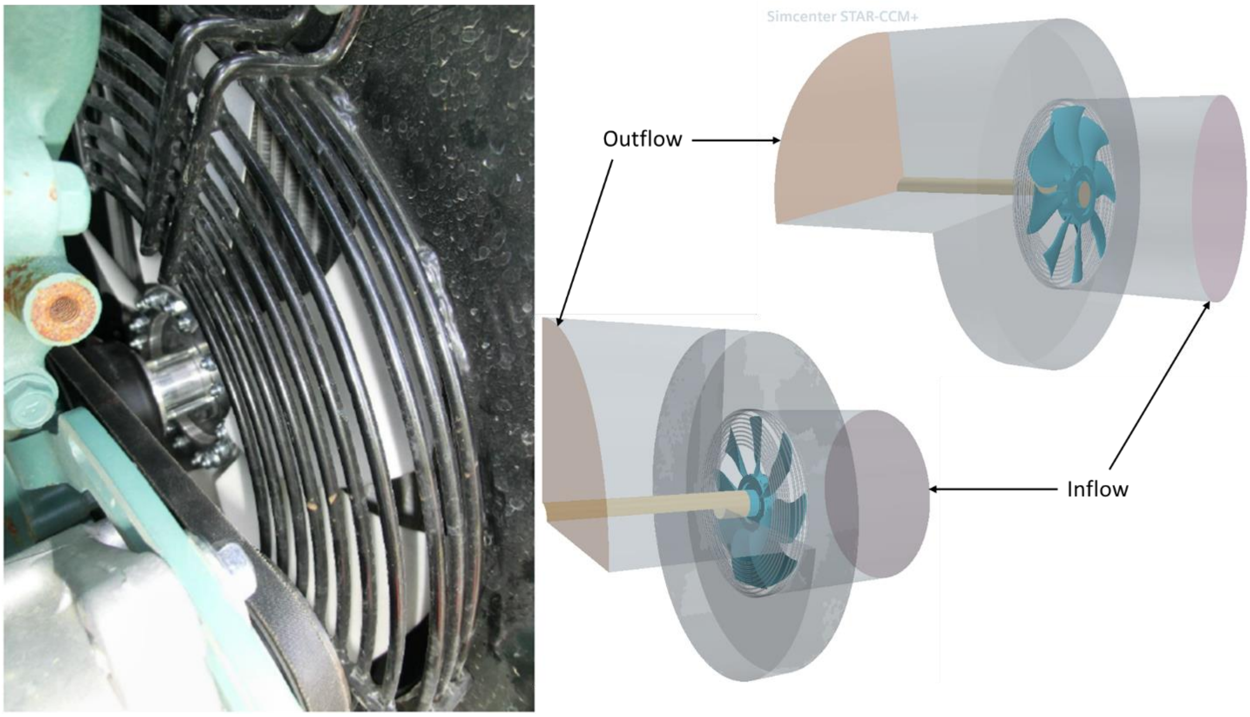

Figure 6: 3D environment of the fan, (left) photo of the actual environment, (right) idealized geometry of the fluid volume

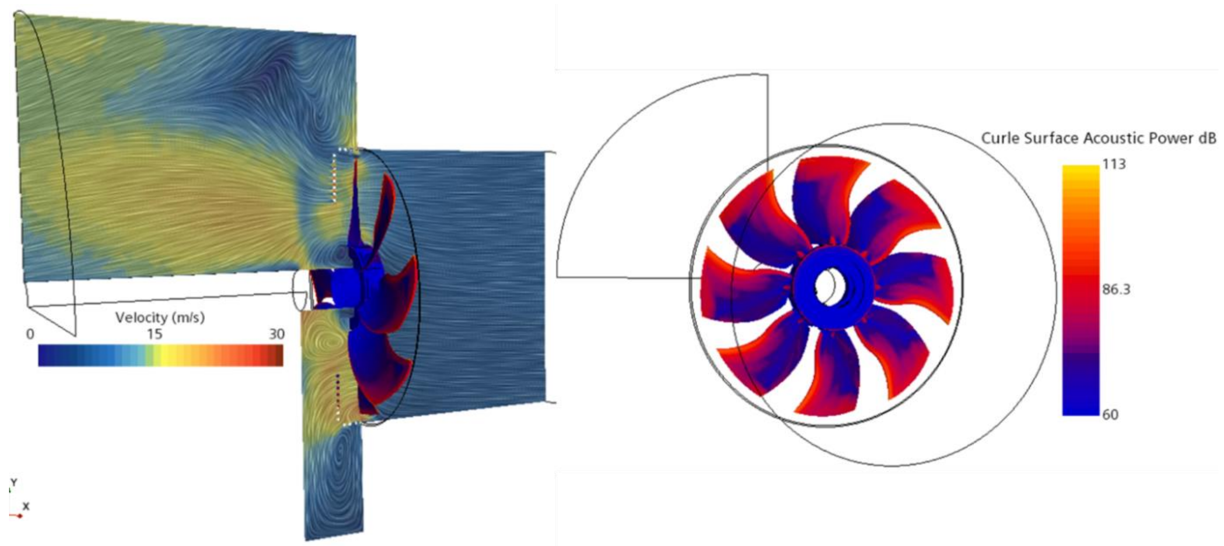

Figure 7: Vertical cut plane shows velocity magnitude, the color on the blades shows Curle Surface acoustic Power

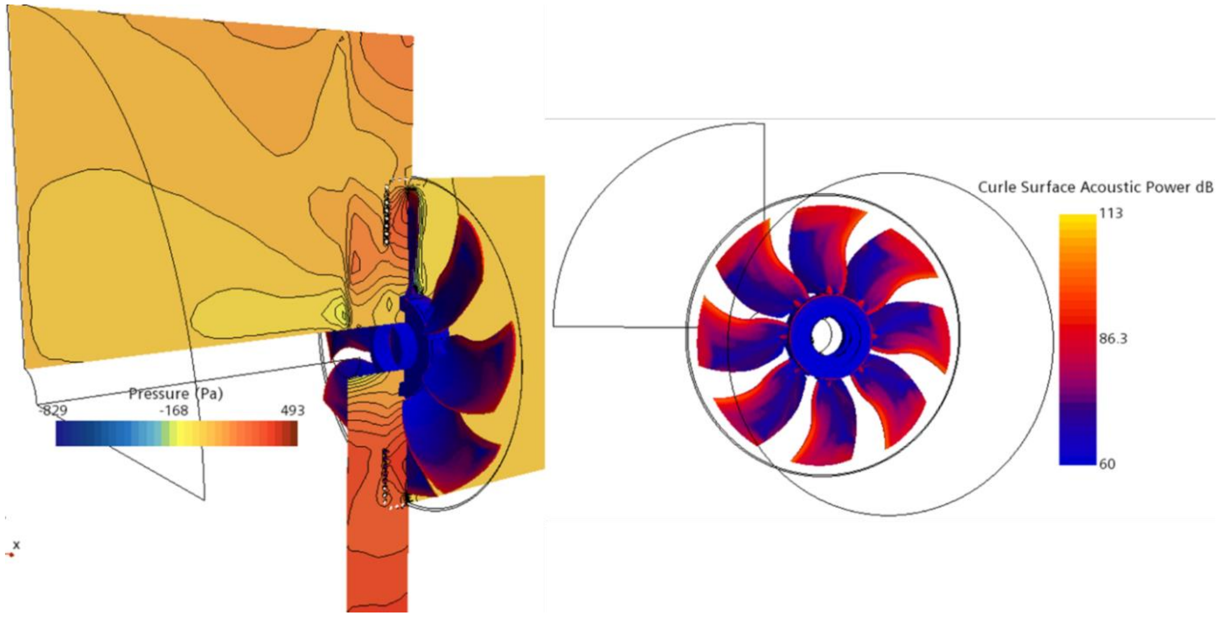

Figure 8: Vertical cut plane shows the relative pressure field, the color on the blades shows Curle surface acoustic power

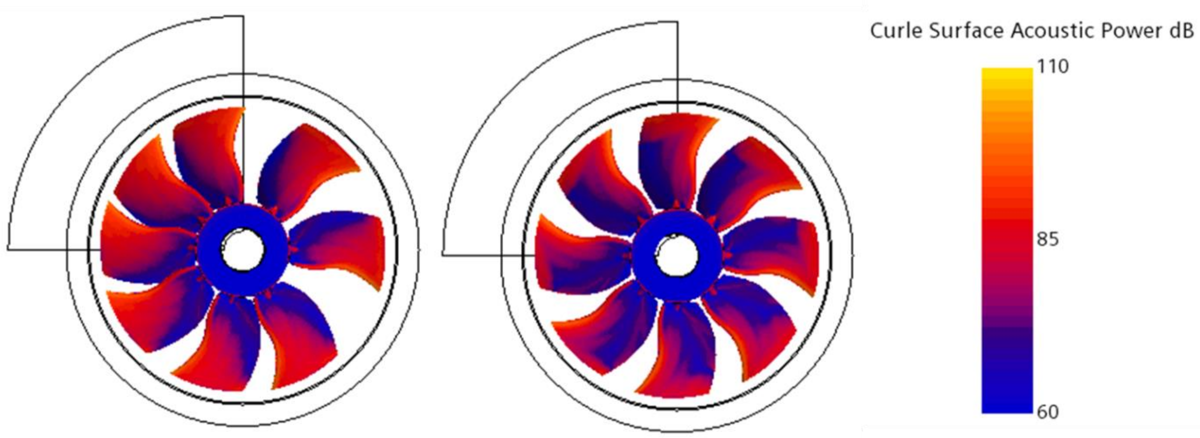

Figure 9: Curle surface acoustic power of the disordered blades ("VDP1") on the left, on the right for regular fan (uniformly distributed blades, "Reg")

ACOUSTIC RESULTS

In this section we compare spectral analysis plot extracted from the previously described FW-H model to previously described experimental results (see Figure 3). Figure 10 and Figure 11 shows comparison of spectral plots between 200 and 1 000 Hz of the sound pressure level in dB(A) between numerical simulation (blue solid lines) and experimental results (yellow solid lines) for the regular fan, and the disordered blades respectively.

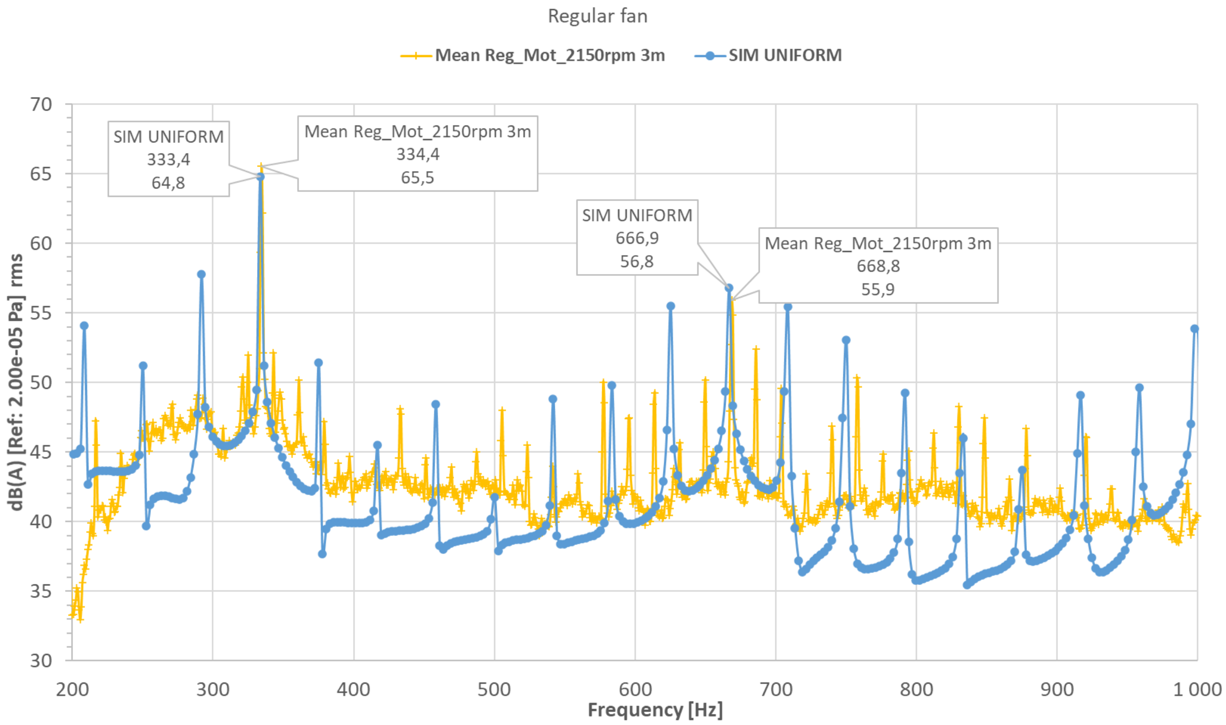

Figure 10: comparison of noise spectrum between 200 and 1 000 Hz between the numerical simulation (blue solid line) and experimental results (yellow solid line) for the regular fan (uniformly distributed blades)

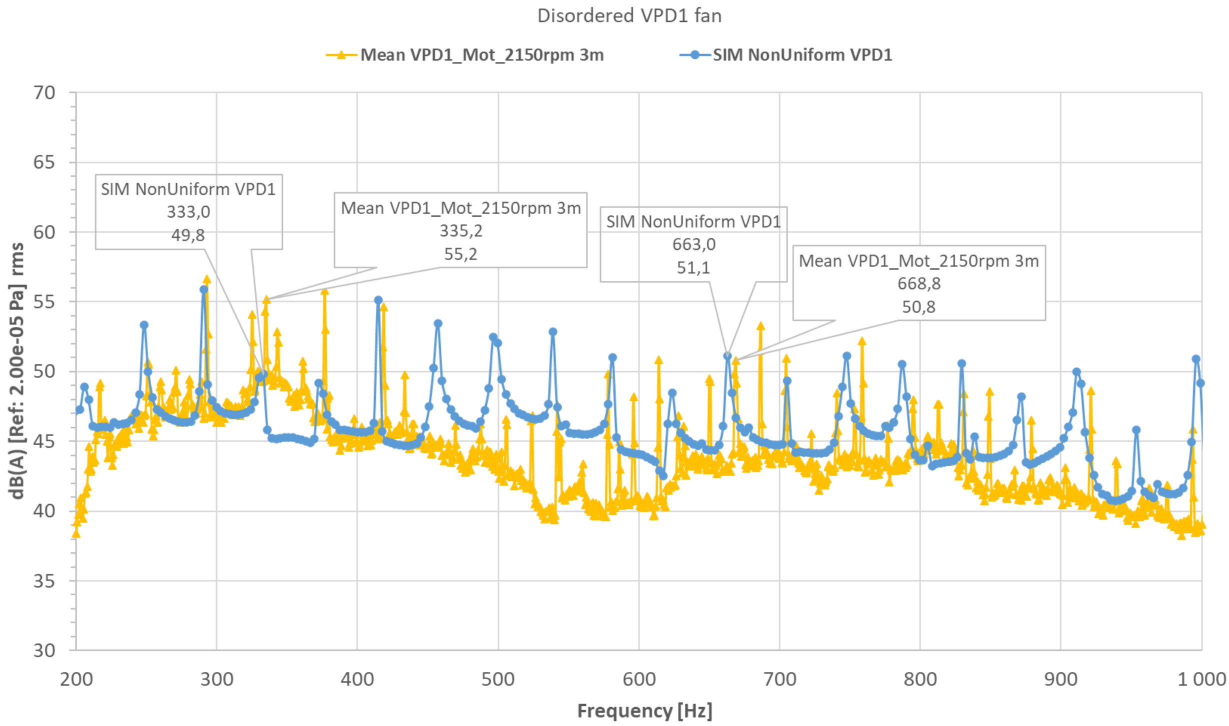

Figure 11: comparison of noise spectrum between 200 and 1 000 Hz between the numerical simulation (blue solid line) and experimental results (yellow solid line) for the disordered blades (non-uniformly distributed blades).

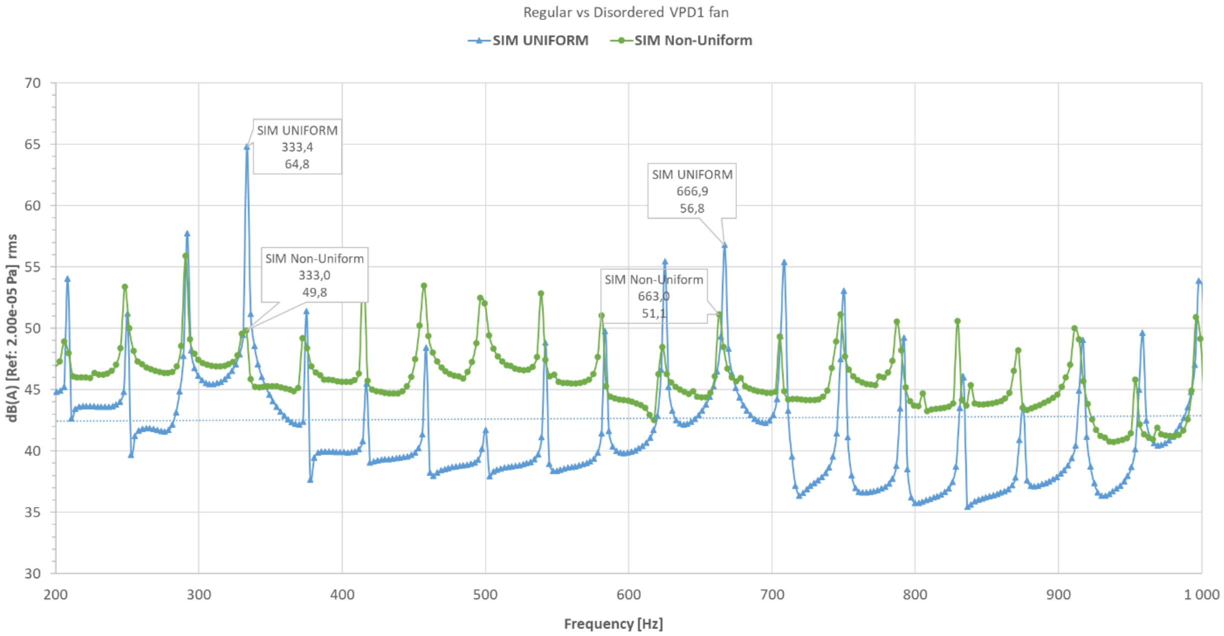

Figure 12: comparison of noise spectrum between 200 and 1000 Hz between the numerical simulation for the regular fan (blue solid line) for the disordered blades (green solid line).

DISCUSSIONS

Figure 10 shows a good agreement between numerical results and experimental measurements for the regular fan (uniformly distributed blades). This is especially true for the tonal noise level associated with blade passage frequency (around 333 Hz) and its first harmonic (around 666 Hz) that show agreement within less than 1 dB(A). The broad band noise is slightly underestimated by the numerical simulation compared to experimental measurements within 10 dB(A).

Figure 11 shows more differences between numerical results and experimental measurements for the disordered blades than for the regular fan (Figure 10). The tonal noise level associated with blade passage frequency is almost inexistent compared to other frequencies and is underestimated by around 15 dB(A) compared to test measurements. The first harmonic cannot be easily identified neither in the numerical results nor in the test measurements but still show good agreement with around 5 dB(A) difference. The broad band noise is slightly overestimated by the numerical simulation compared to experimental measurements within 5 dB(A).

When numerical results are compared between the regular fan and the disordered blades in Figure 12, it clearly shows a reduction of the tonal noise associated with blade passage frequency (and its first harmonic for the disordered blades compared to regular fan. It also shows a slight increase of the broad band noise for the disordered blades compared to the regular fan. It is important to note that same observations can be made in the Figure 3 for the test measurements when comparing regular fan to disordered blades.

OUTLOOK

In the present study we compared results from a Ffowcs Williams-Hawkings (FW-H) Steady numerical solver (in STARCCM+) to experimental results acquired during a CETIM campaign [1]. This numerical model is a simplified numerical approach that is inexpensive, lighter (roughly 4 million cells mesh) and faster (3 hours of calculation time with a 28 cores machine) compared to direct noise calculations. Nevertheless, this model gives remarkably good agreement to test measurement. It succeeds to predict tonal noise reduction associated with blade passage frequency of disordered blades (non-uniformly distributed blades) compared to regular fan (uniformly distributed blades) as observed during a CETIM test campaign. It successfully reproduces tonal noise level of the regular fan within 1 dB(A).

These preliminary results are very encouraging and open the scope to fast numerical prediction of fan noise. Nevertheless, it still requires an extensive sensitivity study, on mesh type and size, on numerical scheme, on turbulence model, on fan geometry and rotation speed to be confirmed. This sensitivity study will be carried out in the coming month with comparison to new additional test measurements and new disordered blades.

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