A self-adaptive resonant device and its use for noise control in turbo-prop aircraft

Mats Gustavsson
A2 Research AB, Sweden, *Mats@A2Research.se*

ABSTRACT

Tonal noise is a common reason for annoyance and reduced comfort. Even if the level of the tone is relatively low, human hearing is such that tones are standing out. In many situations the tonal frequency will vary for different operation conditions. A typical situation is the noise in turbo-prop aircraft, where two or more propeller speeds are used to get better fuel economy and better performance than using a single propeller speed for the complete flight envelope. Before the SAAB 2000 was introducing Active Noise Control, in 1994, the most efficient noise control treatment for propeller noise was Dynamic Vibration Absorbers (DVAs). A major advantage of active noise control systems is the ability to adapt to a frequency shift, which a traditional passive DVA cannot, and thereby the effect of the DVA will essentially only be achieved at one of the operating propeller speeds. Present paper introduce a device that can adapt to more than one excitation frequency and is hence very suitable for application to propeller aircraft and other similar situations with tonal noise. Numerical simulation results and experimental data are presented, and application is made to a Vibro-Acoustic model of a turbo-prop aircraft.

1. INTRODUCTION

The noise inside a propeller aircraft is a typical situation where tonal noise dominates the subjective impression and needs to be controlled to meet the market requirements in terms of comfort. With the higher efficiency achieved by propellers compared to jet propulsion for short flights, say up to around 1500km, modern propeller aircraft is a better alternative with respect to the environmental impact. Use of Contra-Rotating propeller systems, recently being studied by e.g AIRBUS [1] could be an alternative also for longer flights and at flight speeds comparable to those of jet aircraft.

Devices capable of adapting a resonance frequency by means of preload [2], or adjustment of a suspension [3] are well known. Common for both these two concepts is the need for a controller, and at least one sensor to identify the dominant vibration frequency, and one actuator (or "mechanism") to adjust the device to get a favorable resonance frequency. The devices studied in this paper are Self-Adaptive and adapts as a result of the applied excitation only.

2. THE ADVANTAGES OF A SELF-ADAPTIVE DEVICE

A device that is adapting itself depending on the dominant excitation frequency is clearly advantageous for vibration control. The obvious advantage is the maximum effect of having the device operating at the tuning frequency not only for one operating condition but for two. Further, the potential disadvantage of having DVAs installed for operating condition different from the tuning frequency, is also avoided. In theory the installed mass could be as low as only 50% for a dual-frequency Self-Adaptive DVA compared to having DVAs for two operating frequencies.

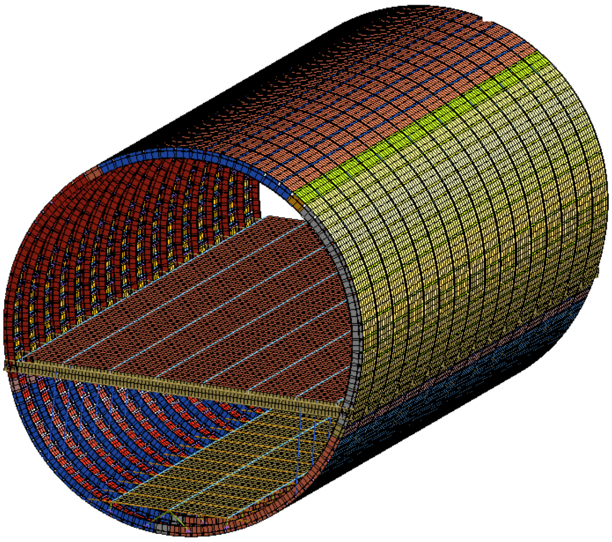
There may also be other applications of a self-tuning devices such as in energy harvesting [4], but in this paper the focus is on control of noise and vibrations.

3. APPLICATION OF SELF-ADAPTIVE DYNAMIC VIBRATION ABSORBERS TO REDUCE NOISE IN TURBO-PROP AIRCRAFT

Aircraft with propellers driven by a gas turbine are called “Turbo-Prop Aircraft” since the power is generated by turbines, but the propulsion is mainly from the propellers. Examples of turbo-prop aircraft are ATR-42/72, the Bombardier Dash8 family, and the SAAB 340/2000.

Figure 1 Turbo-prop Aircraft (ATR-72 Left and SAAB 340/2000 Right).

The propellers normally operate at a rotational speed in the range 600-1200rpm and with the typical 6 blade propellers this gives a “Blade Passage Frequency” of 60-120 Hz. At most of the locations in the passenger cabin the noise is dominated by this low frequency tonal noise. The noise levels inside turbo-prop aircraft is commonly predicted and analyzed by software based on the Finite Element Method, FEM, solving the responses of the Vibro-Acoustic system with the excitation provided by Aero-Acoustic analyses for the propeller noise sources. As an illustration of the benefit of a Self-Adaptive DVA a schematic aircraft fuselage section, Figure 2, is analyzed by Vibro-Acoustic FEM computations.



The Aircraft Section:
Length: 3.6m
Diameter: 3.0m
Weight: 860 kg

Acoustic Volume
(not shown):
The volume above the

Figure 2 FE-Model of a schematic Turbo-Prop Aircraft structure.

This fuselage section is 3.6 meter long, has a diameter of 3.0 meter and a weight of 860 kg. No interior nor seats are included, but the weight is adjusted to better represent a furnished aircraft, and the damping of the structure is set to a Loss Factor of 8%, which also better represent a furnished aircraft. An acoustic model, with thermal insulation at the cabin-fuselage boundary, is giving the acoustic pressures in the passenger cabin. Evaluation of the Sound Pressure Level, SPL, is made at 10 249 locations approximately at the level of “Seated Earhight”. The Noise Control for the aircraft model is assumed to be DVAs with the three options:

A. Self-Adaptive to match the excitation frequency

B. Standard DVA tuned to the lower excitation frequency 100 Hz

C. Standard DVA tuned to the higher excitation frequency 120 Hz

Each DVA is having a dynamic mass of 0.5 kg and a loss factor of 4% (Q=25). The dynamic force of the three different versions of DVA, as a function of the operating frequency, are given below (Figure 3).

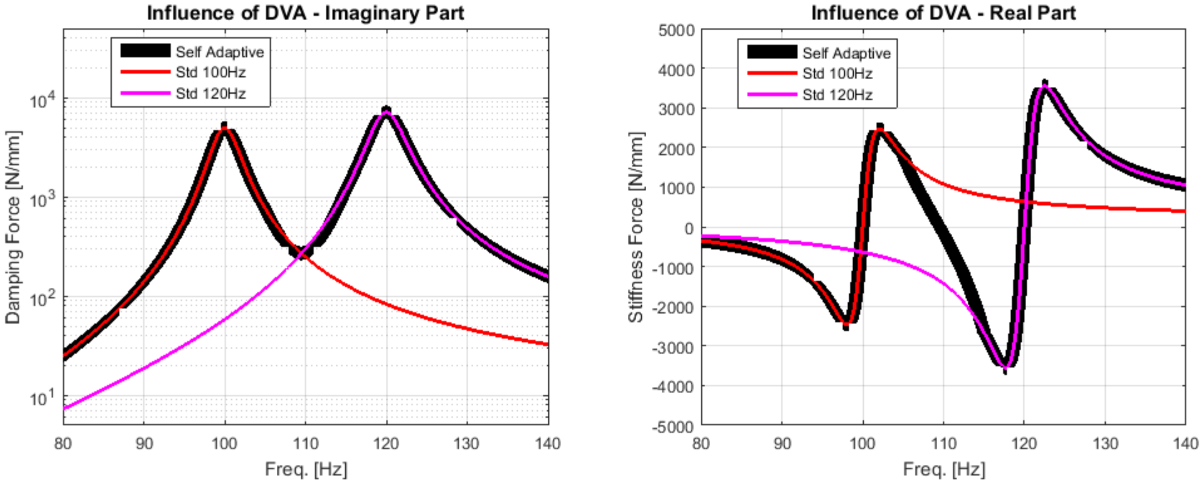


Figure 3 The dynamic force for the three different DVA types.

Optimization is made to find good locations for 10, 20, 40 and 60 DVAs, i.e. for an installed weight of 5 kg, 10 kg, 20 kg, and 30 kg on the 860 kg fuselage structure. As the adaptive DVA will shift its resonance frequency depending on the excitation frequency it will be correctly tuned for operation at both 100 Hz and 120 Hz. This give the results marked “SA-100 Hz” and “SA-120 Hz” in Figure 4 below.

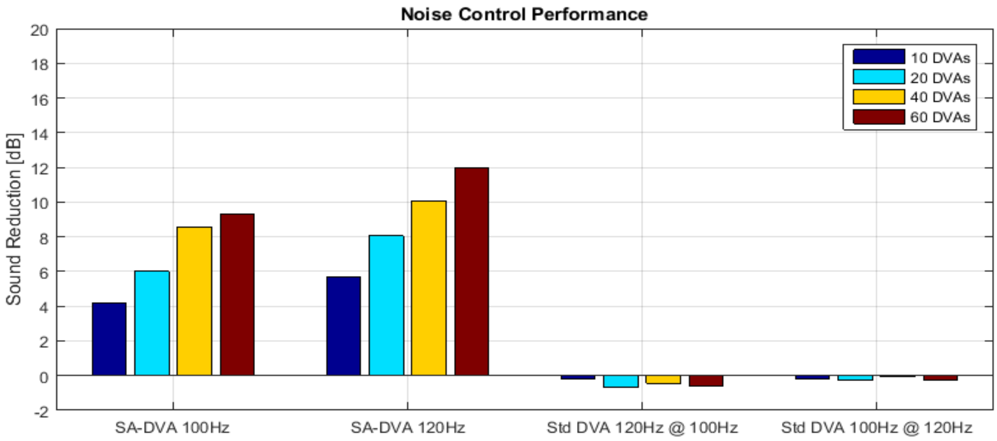


Figure 4 Noise control performance for DVA installation alternatives.

Obviously the noise control performance, “Sound Reduction” as given in Figure 4, is increasing with the number of installed DVAs. The improvement is about 2dB for each increase of DVAs, i.e. from 10 to 20, from 20 to 40, and from 40 to 60 for both 100 Hz and 120 Hz. Without adaptation, an installation of DVAs tuned for 120 Hz will give very low effect at 100 Hz operation, the bars identified with “Std-120 Hz @ 100 Hz”. The results are similar for DVAs tuned to 120 Hz and operation at 100 Hz. It is noticeable that the standard DVAs are increasing the noise level when operated at the non-tuned frequency, compared to the case with no DVAs, although by a small amount.

This means that any of the installations suggested and using a Standard single frequency DVA would only give noise reduction for excitation at its tuning frequency and a degradation at the other operating condition, giving a different dominant excitation frequency. These results are typical, but there are of course other DVA installation configurations that can give different result than shown here. However, the advantage of Self-Adaptive DVA is an undoubted fact even if other optimization conditions are used.

4. THE SELF-ADAPTIVE DVA

The Self-Adaptive Dynamic Vibration Absorber overcomes the need for a controller, sensor and actuator by the use of a self-adaptive mechanism. Several design alternatives are suggested to achieve this unique feature. Some of them are disclosed in [5] and [6] and are reproduced below (Figure 5).

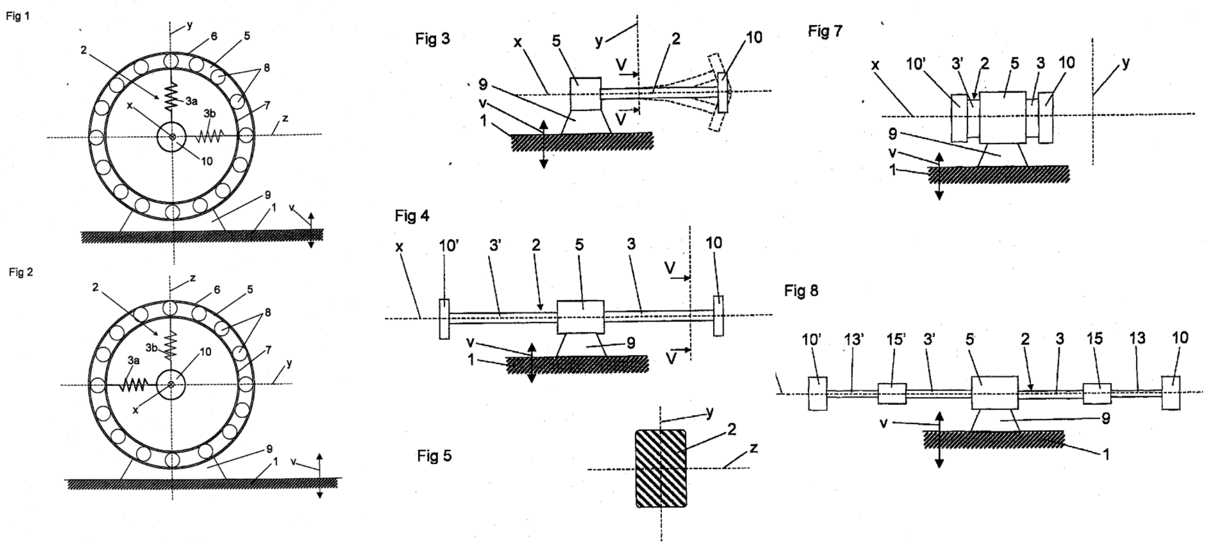


Figure 5 Design alternatives for Self-Adaptive Dynamic Vibration Absorber

Common for all design alternatives is the rotational bearing installed to allow for rotation around an axis perpendicular to the direction of operation for the vibration absorber. In the basic design there will be two primary resonances as a result from the different bending stiffness depending on the rotational position of the elastic element (5 in the figures above). Adding more rotational bearings to separate parts of the elastic element allow for combinations of bending stiffness for segments of the elastic element, and for each new bearing the number of fundamental resonances increases by a factor 2.

The prototypes, of which some are shown in Figure 6, are made to verify the concept and quantify the physical properties controlling the behavior of the devices for various conditions, such as the excitation frequency and excitation level,

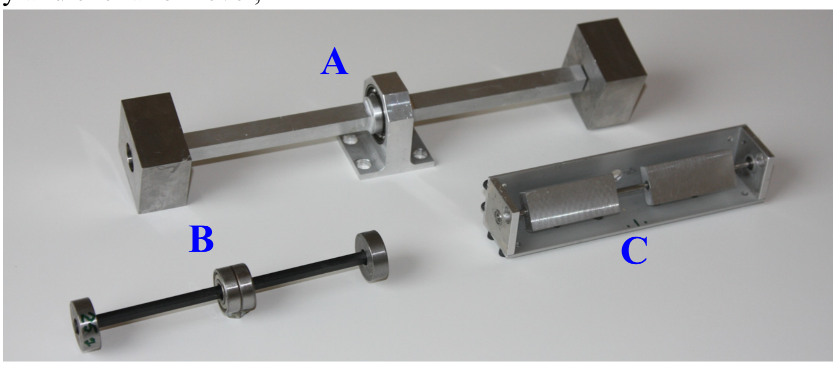


Figure 6 Self-Adaptive DVA prototypes. “A”: Large, “all-metal”, low damping version. “B”: Plastic/Fibre Glass elastic section. “C”: Prototype having different rotational inertia and circular elastic element.

Validation tests of device “B” has proven the adaptation takes place, and that the devices are “locking” the position when adapted to the desired rotational position. Concept “A” was made in a customer project and cannot be tested with the test setup currently available due to the large mass and very low damping. Concept “C” is a first prototype for using variation in rotational inertia and suffers from imprecise manufacturing, but show the shift in frequency predicted for this concept.

5. PROTOTYPE MEASUREMENTS

Experimental testing was made using one of the prototypes, more specifically the prototype marked with a “B” in Figure 6. A non-contact position sensor, KAMAN KD 2446-5CM, was used to measure the vibration of one of the two dynamic masses. A diagonal starting position is used, and observing the device it is found that the rotation is starting almost immediately when the base vibration is applied, and the final rotational location is reached after about 0.6-1.0 seconds. The vibration sequences for the base vibration, blue curve, and the dynamics mass, red curve, are given by Figure 7.

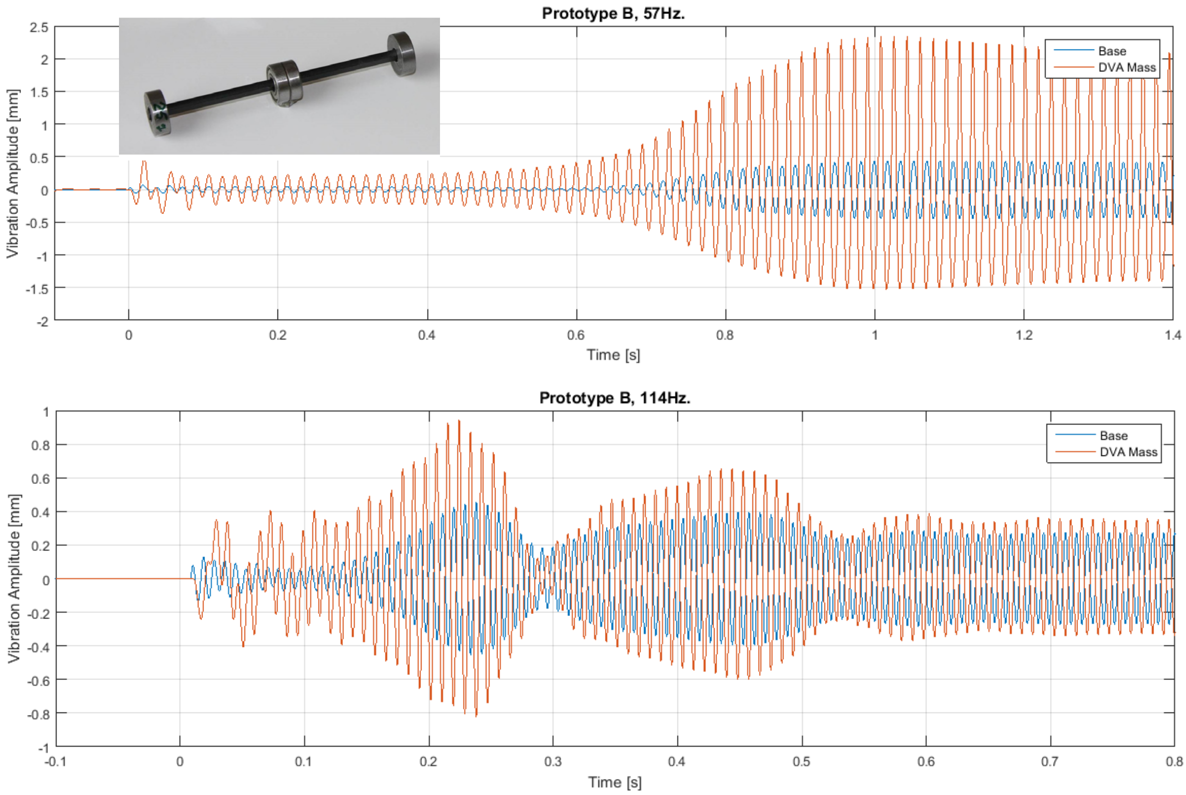


Figure 7 Prototype test results. The green cure show the vibration of the base (shaker) and the red curve is the vibration at one of the dynamic masses.

The vibration amplitudes of the non-contact sensor are difficult to calibrate as they vary with measurement distance, but the relative levels are showing a Steady State condition is reached after about 1.0 second for the 65 Hz excitation, and after ~0.6 second for the 130 Hz excitation. When the Steady State condition is reached the rotational position is stable and to bring it away from this position requires a force/moment. Touches by a finger on one of the dynamic masses to make it rotate results is a quick return to the rotary position at Steady State.

However, a limitation in the current test setup is the controllability of the excitation. A rather small shaker is used, and hence the driving level is strongly influenced by the DVA in the frequency rage of the DVA resonances. More extensive and detailed experimental testing is required to fully understand the characteristics of the prototypes and the general possibilities and limitations of the Self-DVA concepts suggested.

6. SIMULATIONS USING HYPERWORKS

In order to make some basic simulations a numerical model was made using ALTAIR/HyperWorks 14.0. The model is a cantilever beam, with may be considered as a symmetric part of a typical design alternative as given in Figure 8.

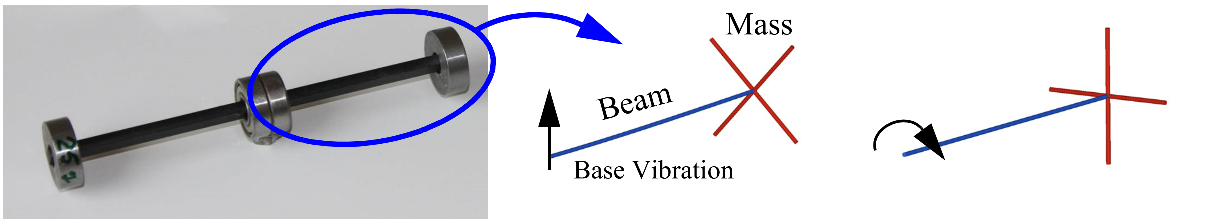


Figure 8 The model used for the numerical simulations.

The properties for the model are give by Table 1

Table 1:

Property:	Beam height	Beam width	Beam E	Mass (at Tip)	Resonance "Flat"	Resonance "Standing"
Value:	8mm	4mm	20GPa	25 gram	65 Hz	130 Hz

The elastic beam section, blue in Figure 8, is divided in eight (8) elements. The red cross is made by four (4) 15mm beams with high stiffness. At the free end of each beam a 25/4 gram concentrated mass is located. All analyses are started with the beam oriented so that the beam cross section is 45degrees from both the main axes of the elastic beam. The beam is oriented "flat" when rotated -45degrees, and "standing" when rotated +45degrees. With a prescribed base vibration at the two resonance frequencies of the beam, 65 Hz and 130 Hz respectively, the dynamic mass of the DVA is getting the response shown in Figure 9.

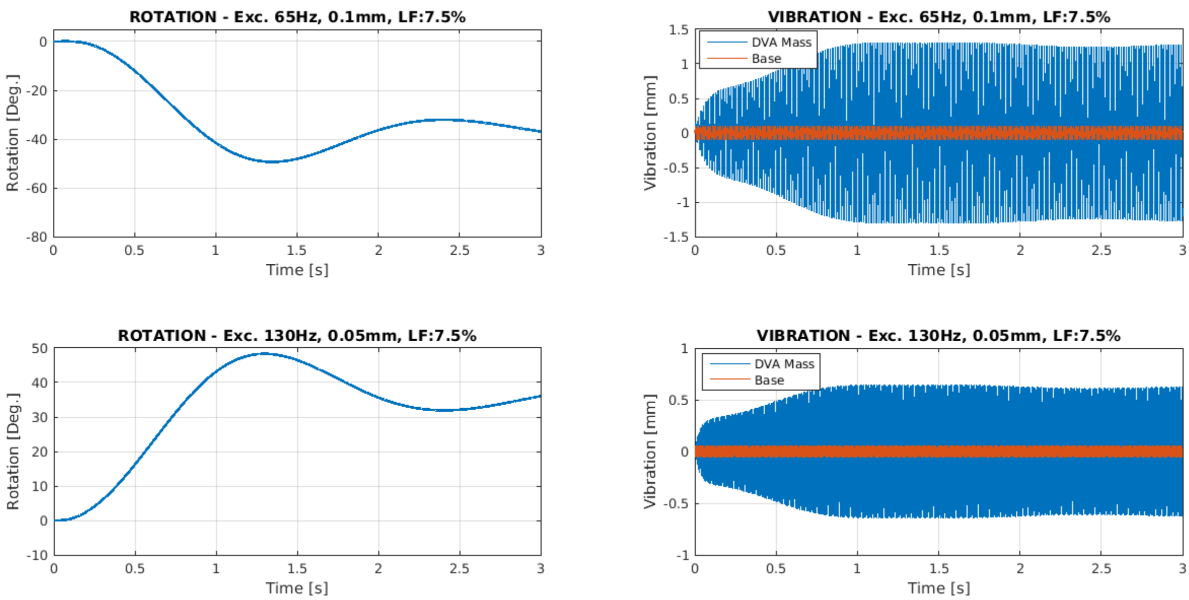


Figure 9 SA-DVA response for excitation at 65 Hz, 0.1mm [Peak], upper graphs, and for 130 Hz, 0.05mm [Peak], lower graphs.

The adaptation to a final rotational position is not completed within the shown 3 seconds analysis time, but it is clear in both cases the system is going towards convergence. Already after about 0.8 seconds the vibration amplitude of the DVA mass is reaching a value close to the final. The applied level of the base excitation is scaled by the frequency to give equal vibration velocity, and hence the 130 Hz excitation displacement level is 50% (0.05mm) of the displacement level at 65 Hz (0.1mm). The damping specified is a 7.5% loss factor, giving a DVA mass vibration amplitude of ~13 times the base excitation. For 65 Hz this means a peak amplitude of ~1.3mm, or ~0.5m/s. For 130 Hz excitation the base vibration is set to 0.05mm, giving a DVA mass vibration amplitude of ~0.65mm (and the velocity ~0.5m/s).

An interesting observation is that the DVA mass is getting high vibration amplitude already after the initial build up phase always found when an excitation is switched on. In fact the vibration amplitude reaches ~50% of the final, maximum, value after just ~0.2seconds. This means a significant noise control performance is achieved already after this short time, even if the adaptation to the desired rotational position has just started.

7. SUMMARY AND CONCLUSIONS

Experimental tests and numerical simulations show the basic function of a Self-Adaptive device. Adaptation to a desired state is achieved experimentally using a prototype, and in the numerical simulations. In order to fully understand the physics, and thereby get detailed design guidelines, more analyses and testing are required.

Application of adaptive Dynamic Vibration Absorber to an aircraft structure show the benefit of an adaptive device compared to a non-adaptive device.

REFERENCES

1. "*Airbus Determines Open-Rotor Airliner is Technically Feasible*" Aviation Daily, Mar 20,2014.
2. Breitbach E, *Adaptive Vibration Damper* European Patent, EP1528281
3. Bailey D. et. al., *Adaptively Tuned Elastomeric Vibration Absorber* European Patent, EP0922877
4. L. M. Miller, P. Pillatsch, E. Halvorsen, P. K. Wright, E. M. Yeatman, and A. S. Holmes, *Experimental passive self-tuning behavior of a beam resonator with sliding proof mass J. of Sound and Vibration*, 09/2013
5. Gustavsson M, *A device for reducing Vibrations and Sounds* Swedish Patent, SE528267, Approved Oct. 10, 2006 (European Patent Pending)
6. Gustavsson M, *A device for reducing Vibrations and Sounds* Swedish Patent, SE528384, Approved Oct. 31, 2006 (European Patent Pending)
7. Gustavsson M, *A Vibration Absorbing Device for Reducing Vibrations and Sounds in a Structure* Swedish Patent, SE538004, Approved Feb. 9, 2016 (European Patent Pending)