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Application of Piezoelectric Materials for Aircraft Propeller Blades Vibration Damping

Alaa M. Morad[a], Aly Elzahaby[b], S. Abdallah[c], M. Kamel[d] and Mohamed K. Khalil[e]

Abstract— Vibrations of turbomachinery blades are critical to jet engine durability and performance. Active vibration control using piezoelectric sensors and actuators have recently been emerged as a practical and promising technology. This paper deals with a new vibration damping technique for an Unmanned Aerial Vehicle (UAV) propeller (BIELA 24 in diameter and 12 in pitch) utilizing piezoelectric transducers (sensors and actuators). The propeller blades are one of the main sources of turboprop engine vibration. The damping performance depends on the type and location of the piezoelectric transducers with respect to the mode shape of the blade mechanical strain. Numerical simulations are carried out for the propeller blades. A finite element modal analysis of non-rotating propeller without piezoelectric transducers is built in ANSYS-Workbench 15.0, where the numerical results are compared to the experimental measured modal data for verification. The numerical results are in very good agreement with the experimental measured data. A numerical model of the propeller without and with piezoelectric transducers is built in ANSYS. The results indicate the feasibility of using piezoelectric transducers as a smart material for vibration suppression in turboprop engines by applying these transducers to propeller blades at the first mode high modal strain areas.

Index Terms—Aircraft propeller, ANSYS simulation, Gas turbine engine, Piezoelectric materials, Smart materials, Vibration control

1 INTRODUCTION

SMART materials are materials that respond to their environments in a timely manner, this definition can be expanded to materials that receive, transmit, or process a stimulus and respond by producing a useful effect that may include a signal that the materials are acting upon it [1]. Piezoelectric materials as smart materials have been extensively used as sensors and actuators for vibration control because of their ideal properties: light weight, high bandwidths, efficient energy conversion and easy integration [2]. Piezoelectric materials present a new approach to turbomachinery blade vibration suppression. They can be adapted to the creation of damping/ or control elements [3]. Recent studies considered active and passive systems to damp vibrations in turbomachinery; such systems adopt piezoelectric transducers that tested experimentally and numerically simulated.

Cross and Fleeter [4] studied experimentally vibration suppression of an airfoil in the stator row of a compressor. An array of lead zirconate titanate (PZT) piezoelectric elements was bonded to the thin low-aspect-ratio airfoils and tuned electrical circuits were connected. The airfoil was excited in a resonant chord-wise bending mode. Experimental data demonstrate that shunted piezoelectrics have significant damping capability and could be practical mean for the elimination or minimization of gas turbine fan and compressor blading flow-induced vibrations.

[a] *Corresponding author is currently pursuing PhD degree program in Military Technical College (MTC), Egypt, alaa_morad@yahoo.com, alaa.morad@uc.edu.*
[b] *Professor, Depart of Mechanical Power Engineering, Tanta University, Tanta, Egypt, elzahaby47@gmail.com.*
[c] *Professor, Department of Aerospace Engineering and Engineering Mechanics, University of Cincinnati, Cincinnati, Ohio 45221-0070, USA, shaaban.abdallah@uc.edu.*
[d] *Doctor, Military Technical College (MTC), Cairo, Egypt, kamelema_1971@hotmail.com*
[e] *Doctor, Military Technical College (MTC), Cairo, Egypt, khilo99@yahoo.com*

Yu et al. [5], [6], [7] examined different techniques for reducing vibrations of mistuned bladed disks. The authors combined several discrete beams into a bladed disk model; adjacent beams were coupled by springs. Each beam was equipped with a piezoelectric patch, which was connected to an electric network. This network was designed to create a new electro-mechanical wave/energy channel which sustains the energy propagation throughout the structure and it is found that the network is effective for suppressing vibration in mistuned bladed-disk systems.

Livet et al. [3], [8] studied vibration suppression for a simplified blade model using a piezoelectric patch of PZT material placed on top of the blade. The authors used an Euler-Bernoulli beam model to calculate the size and location of piezo elements and they presented a finite element approach to the problem. Theoretical results were verified experimentally, whereby active and passive control techniques were compared. The authors concluded that a significant reduction of vibration is possible, with both active and passive shunting.

Sebastian et al. [9] used numerical and experimental investigations on a rigidly clamped simplified compressor blade. An optimization process incorporating electromechanical finite element calculations determined the optimal position of the piezo damper in regard to the mode shape of interest. By applying the computed and measured Frequency Response Functions (FRF), the damping performance with and without piezo-damper are compared and referred to the measured material damping. Their results show that the passive nature of piezo damper, which does not require any power supply, allows for long-term operation without any service.

Kauffman and Lesieutre [10] used analytical simulations and experimental data applied to turbomachinery. They used semi active approach by switching the piezoelectric electrical boundary conditions applied to a titanium flat plate with surface-mounted piezoelectric material patches.

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They concluded that resonance frequency detuning can reduce vibration by altering the structural stiffness to detune it.

Zhou and Zuo [11] used a self-powered piezoelectric PZT film applied on flexible cantilever beam for vibration damping. They used theoretical analysis and simulation, the simulation results show that under a random excitation, the realized "control force" can track the desired values very well, the vibration of the beam can be effectively attenuated, and the accumulated harvested energy is more than the consumed energy in the system.

This paper deals with a new vibration damping approach utilizing piezoelectric transducers applied to a real propeller blades used for small Unmanned Aerial Vehicle (UAV). The propeller used is BIELA 24 in diameter and 12 in pitch that is part of a turboprop engine. First, numerical simulations are carried out in ANSYS-Workbench 15.0 for the propeller without piezoelectric transducers, where the numerical results are compared to the experimental measured modal data for verification. Numerical simulations are also carried out by applying piezoelectric transducers passively at first mode high modal strain area of the propeller blades to damp vibrations of the propeller. The piezoelectric transducers are simulated by using ACT (Application Customization Toolkit, Piezo extension) software, which is a new tool that is integrated in ANSYS-Workbench 15.0.

2 Experimental Work

Although high fidelity computer simulations can include many details and provide accurate estimates, it is necessary to perform experiments to tune a model as well as to verify numerical models. In this work, numerical results are compared with experimental results. A 3-blade composite fixed pitch propeller type BIELA with diameter 24 in and pitch angle 12 in as shown in Fig. 1 designed and fabricated for small Unmanned Aerial Vehicles (UAV's) is used for experimental modal testing. The first series of tests are conducted by using a small hammer. The second series of tests are conducted by using a shaker contacted at one of the propeller blade tip.

Fig. 1. Three-blade propeller in real shape

2.1 Propeller Modal Analysis Using Hammer

The series of tests are conducted on a suspended propeller from the hub exited by a small hammer. These tests are carried out in Structural Dynamics Research Laboratory (SDRL), College of Engineering and Applied Science, University of Cincinnati (UC), USA. Using X–Modal computer program, small hammer and three accelerometers each accelerometer attached to the propeller blade tip as shown in Fig. 2. X-Modal is an experimental modal analysis research software package that the University of Cincinnati, Structural Dynamics Research Lab (UC-SDRL) has, and continues to develop in conjunction with a group of sponsoring companies. The primary function of this software package is to provide a flexible environment for acquiring and analyzing data acquired for the purpose of experimentally determining the modal parameters of a structure in research situations. X-modal includes an embedded data acquisition capability; this module is referred to as the virtual data acquisition (VACQ) module, this data acquisition module has been developed by the (UC-SDRL).

Three ceramic shear piezoelectric accelerometers manufactured by PCB Piezoelectronics are used. The used hammer is ICP (Integrated Circuit Piezoelectric) impulse force test small impact Hammer manufactured by PCB Piezoelectronics. The propeller is drown in X-Modal program using 64 points, 21 points in each blade and 1 point at the center of the hub as shown in Fig. 3, these points have a corresponding points that drawn on the propeller itself. The experimental modal analysis of the propeller was conducted by using the hammer method. Then the dynamic characteristic of structure is described by the natural frequencies and corresponding vibration modes.

Fig. 2. Experiment lay out

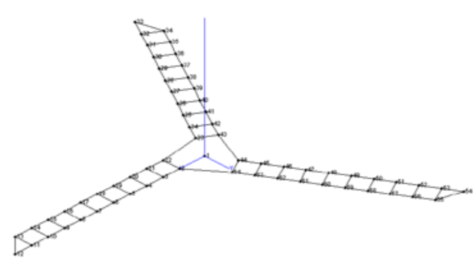

Fig. 3. Propeller shape in LMS program using 64 points

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2.1.1 Propeller Natural Frequencies and Mode Shapes for Hammer Tests

In this experiment, the modal testing is carried out by using the hammer to hit the propeller three times in each of 63 points, combined with the multiple input and output of 3 accelerometers, each accelerometer attached to propeller blade tip. This type of modal testing is known as Frequency Response Function (FRF) method, which measures the input excitation and output response simultaneously, the (FRF) is shown in Fig. 4 and the Propeller mode shapes are shown in Fig. 5.

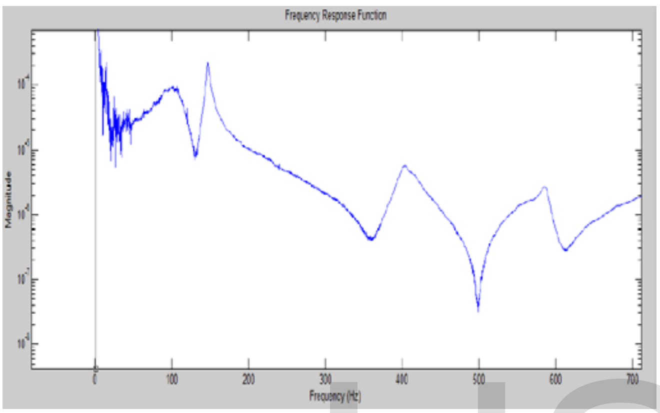

Fig. 4. Frequency Response Function (FRF) of the propeller

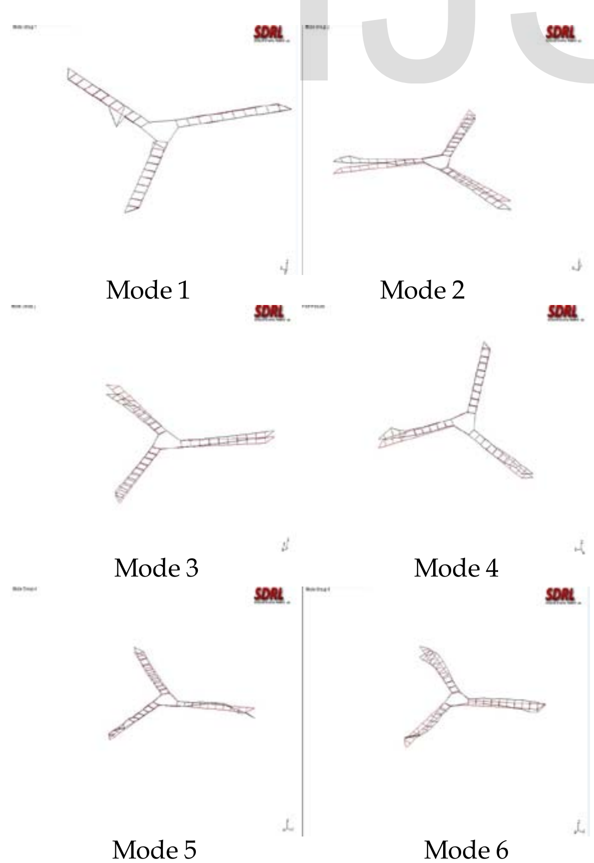

Fig. 5. Experimental propeller mode shapes using hammer

2.2 Propeller Modal Analysis Using Shaker System

The same propeller that used for the previous tests is used for shaker system tests. Experimental modal testing is carried out in dynamic laboratory at Air Research Center (ARC), Cairo, Egypt. Using shaker system for modal testing to validate the previous modal testing using hammer.

Using LMS SCADAS lab which consists of PC computer, SCADAS III data acquisition system, signal amplifier, shaker, force transducer and 21 accelerometers as shown in Fig. 6Fig. 1 The propeller is clamped at the propeller hub and attached to a shaker at the tip of one propeller blade which is vibrated accordingly. The power amplifier of the shaker provides the necessary power levels through the operating frequency range so that the shaker produces a controlled level of force.

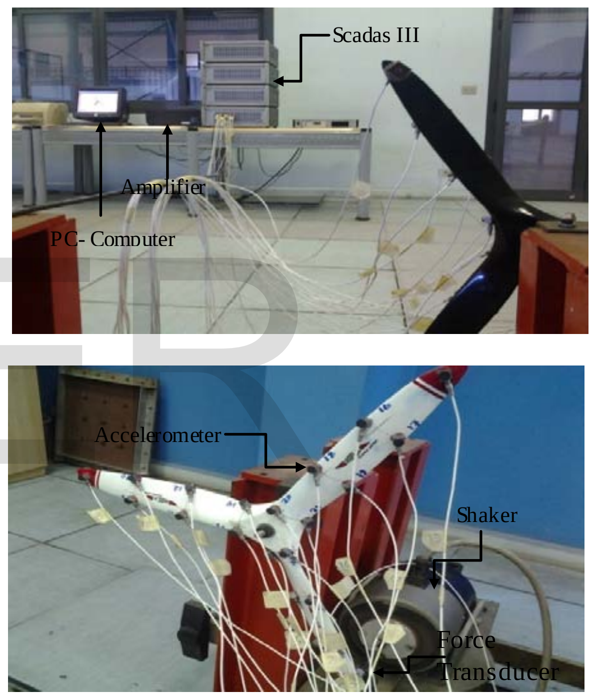

Fig. 6. Experiment lay out

A vibration generating system is used to vibrate the propeller. It consists of three separate parts: the shaker (ET-139 manufactured by Labworks Inc), the power amplifier and the control module in data acquisition system (SCADAS III). Vibration is controlled using a digital signal processing module in Scadas III.

A ceramic shear accelerometer with Integrated Electronics Piezo Electric (IEPE) output type is used, this type featuring a built-in preamplifier. Scadas III, totally integrated system concept, is the product of LMS Difa Instruments. LMS program require the 3-D shape in coordinates points, the propeller drown in an INVENTOR program to take the coordinates of 21 points in 3-D, 7 point on each blade to plot in LMS program as shown in Fig. 7. The propeller first is fixed at the hub, an accelerometer is put at each corresponding point at the propeller. Using 21 accelerometers attached to the propeller by wax and their

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wires connected to SCADAS III in 21 channel, then a shaker is connected to the amplifier which take signal from SCADAS III, the shaker contacted at the tip of a propeller blade through force transducer.

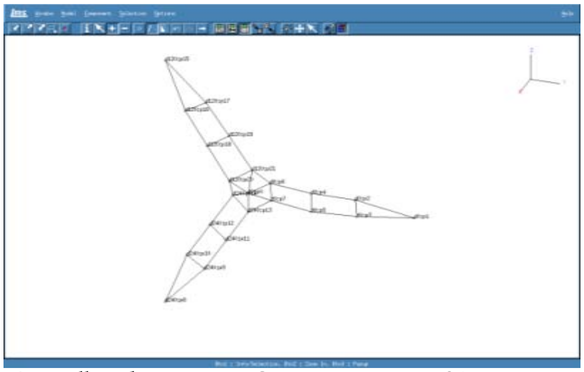

Fig. 7. Propeller shape in LMS program using 21 points

2.2.1 Propeller Natural Frequencies and Mode Shapes for Shaker Tests

At or near the natural frequency of a mode, the overall vibration shape ("operating deflection shape") of a structure will tend to be dominated by the mode shape of the resonance. The type of modal testing known as Frequency Response Function (FRF) method is applied. Modal testing of the propeller is carried out at frequency range from 0 to 1000 Hz, the Frequency Response Function (FRF) is shown in Fig. 8 and the Propeller mode shapes are shown in Fig.9.

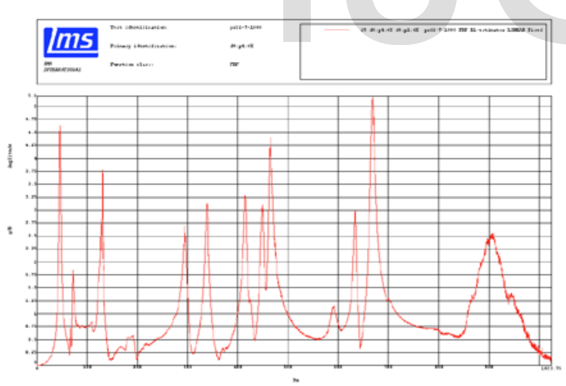

Fig. 8 Frequency Response Function (FRF) of the propeller

The corresponding natural frequencies for different mode numbers for both experiments by hammer and shaker tests are shown in Table 1. These results verify both experiments.

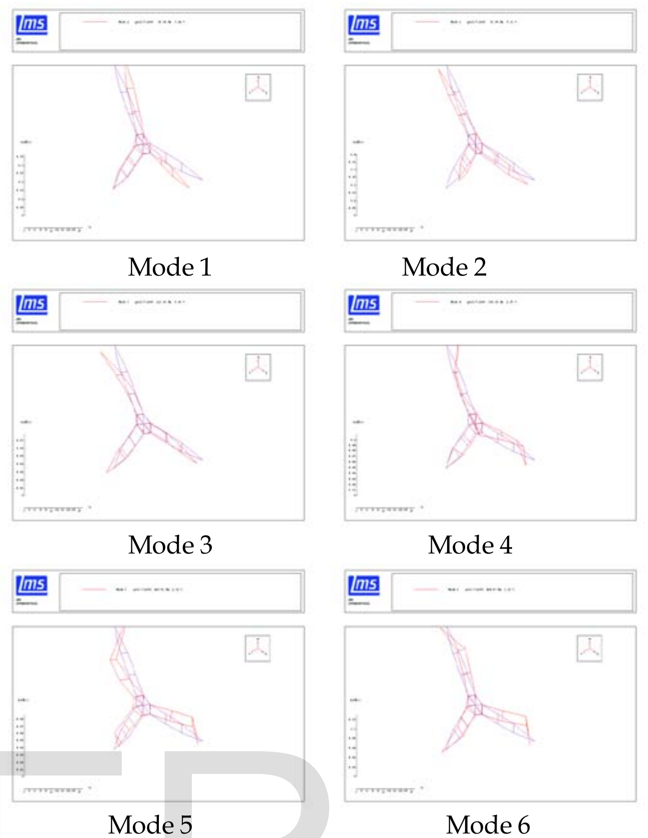

Fig. 9. Experimental propeller mode shapes using shaker

TABLE 1
NATURAL FREQUENCIES AND MODE NUMBERS FOR HAMMER TESTS AND SHAKER TESTS

Mode Number	Frequency (Hz) Hammer	Frequency (Hz) (Shaker)
1	46.12	45.56
2	68.45	68.95
3	71.12	71.32
4	150.64	151.02
5	185.21	184.36
6	210.94	211.63

3 PROPELLER SIMULATIONS IN ANSYS

ANSYS-15 workbench requires a 3-D shape for the propeller; the propeller is measured by using Coordinate Measuring Machine (CMM) Model: BRT APEX 1220 from Mitutoyo America Corporation as shown in Fig. 10. This machine uses a vision probe which makes it possible to measure fine contours, thin and elastic work pieces. After measuring the propeller, it is drawn using AUTO CAD program, its shape imported to ANSYS-15 workbench in 3-D.

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Fig. 10. CMM measuring the propeller in 3-D

In order to apply modal analysis within ANSYS-Workbench 15.0, first in ANSYS-Structure, the propeller is modeled and meshed by using 13611 mesh elements and 25374 nodes as shown in Fig. 11. A mesh convergence study is conducted to ensure that the cell-based smoothed finite element technique is stable and provides accurate results with the increase in the number of degrees of freedom. The mesh was refined by increasing the node number of the structure. Then, modal analysis is chosen; the properties of the material and the boundary conditions of the propeller are applied. Subsequently, calculations are executed.

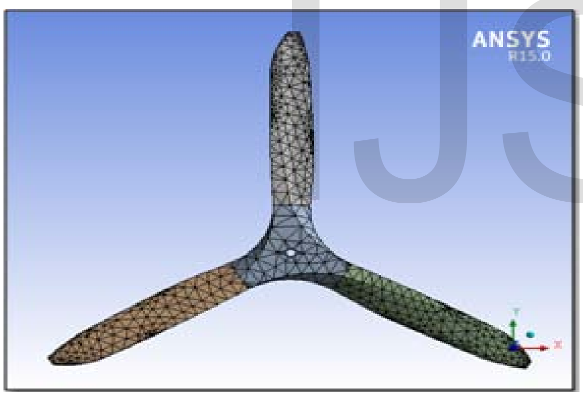

Fig. 11. Mesh configuration of the propeller in ANSYS

3.1 Finite Element Analysis for the Propeller without Piezoelectric Transducers

A finite element analysis of the propeller without piezoelectric transducers is developed and analyzed for natural frequency and mode shape determination. Fig. 12 shows the propeller mode shapes and modal displacement contours at zero rotating speed. The results comparison between experimental data and the numerical simulations results in ANSYS for natural frequencies and mode numbers are shown in table 2. These results verify the simulations in ANSYS.

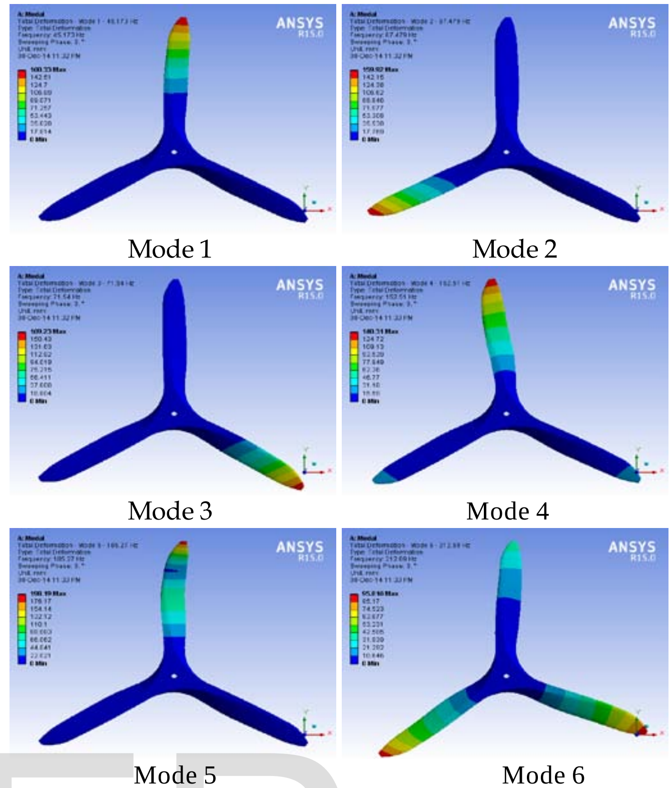

Fig. 12. Modal displacement contours for the propeller without piezoelectric transducers.

TABLE 2
EXPERIMENTAL AND ANSYS RESULTS VIBRATION MODES AND NATURAL FREQUENCIES

Mode Number	Frequency (Hz) Hammer	Frequency (Hz) Shaker	Frequency (Hz) ANSYS
1	46.12	45.56	45.173
2	68.45	68.95	67.479
3	71.12	71.32	71.54
4	150.64	151.02	152.51
5	185.21	184.36	185.27
6	210.94	211.63	212.69

In order to determine the high modal strain area of the propeller for the best location of the piezoelectric patches, equivalent elastic stain for first mode shape is determined using Finite Element (FE) simulation as shown in Fig. 13. The piezoelectric transducers are placed at the first mode maximum modal strain location. Piezoelectric transducers bonded to one side of the blade (front side), a perfect adhesive bonding between transducers and blades is assumed.

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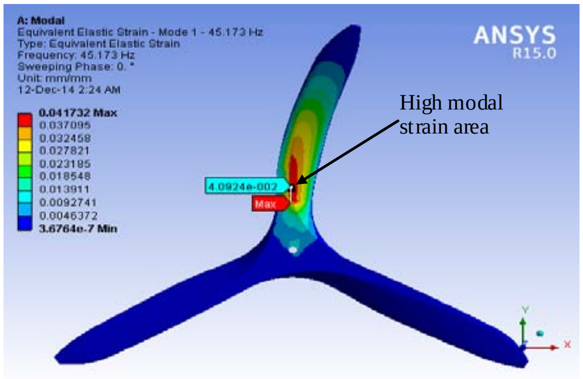

Fig. 13. Maximum modal strain area for the first mode shape from propeller equivalent elastic strain

3.2 Finite Element Simulation in ANSYS for the propeller with Piezoelectric Transducers

3.2.1 Piezoelectric Constitutive Relationship in ANSYS

Conversion of material properties of piezoelectric materials for use in ANSYS Multi physics is challenging because of differences between manufacturer-supplied data and the format required by ANSYS. This section provides a framework of finite element (FE) modeling techniques for shunt circuit-fed piezoelectric vibration damping analyses. A piezoelectric model requires permittivity (or dielectric constants), the piezoelectric matrix, and the elastic coefficient matrix to be specified as material properties. The constitutive relationship usually given by manufacturers or published reports is in the following form:

$$\{S\} = [S^E]\{T\} + [d]\{E\} \quad (1)$$

$$\{D\} = [d]^t\{T\} + [\varepsilon^T]\{E\} \quad (2)$$

ANSYS requires data in the following form:

$$\{T\} = [C^E]\{S\} - [e]\{E\} \quad (3)$$

$$\{D\} = [e]^t\{S\} + [\varepsilon^S]\{E\} \quad (4)$$

Where: {S}=Strain vector (six components x, y, z, xy, yz, xz), {D}= Electric displacement vector (three components x, y, z), {T}= Stress vector (six components x, y, z, xy, yz, xz), {E}=Electric field vector (three components x, y, z), $[C^E]$=Stiffness matrix evaluated at constant electric field, i.e. short circuit, [e]=Piezoelectric matrix relating stress/electric field, $[e]^t$=Piezoelectric matrix relating stress/electric field (transposed), $[\varepsilon^S]$=Dielectric matrix evaluated at constant strains, i.e. mechanically clamped, [d]=Piezoelectric matrix relating strain/electric field, $[d]^t$=Piezoelectric matrix relating strain/electric field (transposed) and $[\varepsilon^T]$=Dielectric matrix evaluated at constant stress.

In order to convert the manufacturer's data presented in the form of (1) & (2) to ANSYS notation ((3) & (4)), (1) needs to be based on stress rather than strain. Equation (1) can be rearranged as:

$$\{T\} = [S^E]^{-1}\{S\} - [S^E]^{-1}[d]\{E\} \quad (5)$$

Since (2) relates electric displacement to strain rather than stress, (3) can then be plugged back into (2) to yield:

$$\{D\} = [d]^t[S^E]^{-1}\{S\} + [\varepsilon^T] - [d]^t[S^E]^{-1}[d]\{E\} \quad (6)$$

Upon comparison of (5) & (6) with (3) & (4), one can obtain the relationship between manufacturer-supplied data and ANSYS-required values as follows:

$$[C^E] = [S^E]^{-1} \quad (7)$$

$$[\varepsilon^S] = [\varepsilon^T] - [d]^t[S^E]^{-1}[d] \quad (8)$$

$$[e] = [S^E]^{-1}[d] \quad (9)$$

These equations form the basis of the conversion routines for converting manufacturer's data to ANSYS data for stiffness matrices, dielectric constants, and piezoelectric constants [12].

3.2.2 Finite Element Analysis

To simulate piezoelectric transducers in ANSYS workbench-15.0, Application Customization Toolkit (ACT) Piezo extension R 15.0 software is used which is a tool with piezoelectric capabilities integrated in ANSYS-Workbench. ACT allows customization in ANSYS-Structure, application that replaces command snippets with interactive objects, create customized loads, results and ability to connect a third party solver in a standard ANSYS Workbench process. The poling direction for the piezoelectric transducers is applied in the vertical direction through the thickness of the propeller blades at Z direction. Taking into account high modal strain area of the first mode shape (for best damping) for each propeller blade, the optimal placement for the piezoelectric transducers is determined. Using one shunted piezoelectric transducer for each propeller blade, the chosen piezoelectric transducers are PZT-5A type Macro Fiber Composites (MFC) which include sensor and actuator, the material properties are shown in table 3.

TABLE 3
PIEZOELECTRIC PROPERTIES FOR PZT-5A USED FOR FINITE ELEMENT MODEL.

PZT-5	Elastic properties	Piezoelectric coefficients (10^{-12}m/v)		Electric permittivity	
E_{11} (GPa)	77	d_{31}	-130	$\varepsilon_{11}/\varepsilon_0$	1300
E_{22} (GPa)	77	d_{32}	-130	$\varepsilon_{22}/\varepsilon_0$	1300
E_{33} (GPa)	77	d_{33}	330	$\varepsilon_{33}/\varepsilon_0$	1300
G_{23} (GPa)	29.6	d_{24}	327		
G_{13} (GPa)	29.6	d_{15}	327		
G_{12} (GPa)	29.6				
γ_{12}	0.3				
γ_{12}	0.3				
γ_{12}	0.3				

Electric permittivity of air $\varepsilon_0 = 8.85 \times 10^{-12}$, *mass density* $\rho = 7700\ kg/m^3$

A finite element analysis for the propeller with piezoelectric transducers is developed and analyzed to obtain its natural frequencies and mode shapes. Fig. 14 shows mode shapes for the propeller with piezoelectric transducers and table 4 shows propeller vibration modes and corresponding natural frequencies without and with using piezoelectric transducers. These comparison results show that the natural frequencies for the same mode shape increase with using piezoelectric transducers for non-rotating propeller.

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TABLE 4
VIBRATION MODES AND NATURAL FREQUENCIES FOR THE PROPELLER WITHOUT AND WITH PIEZOELECTRIC TRANSDUCERS

Mode Number	Frequency (Hz) without piezoelectric transducers	Frequency (Hz) with piezoelectric transducers
1	45.173	53.107
2	67.479	69.809
3	71.54	73.605
4	152.51	166.84
5	185.27	213.92
6	212.69	220.98

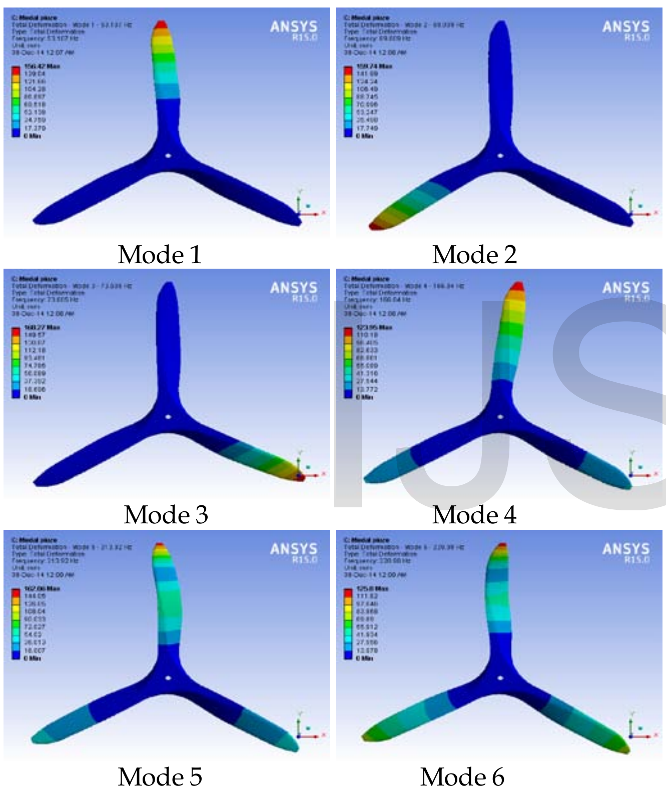

Fig. 14. Modal displacement contours for the propeller with piezoelectric transducers

The equivalent elastic strain of first mode for non-rotating propeller with piezoelectric transducers is shown in Fig. 15, this figure shows that the maximum value of equivalent elastic strain with piezoelectric transducers is 3.0372×10^{-2}, while the maximum value of equivalent elastic strain without piezoelectric transducers is 4.0924×10^{-2}as shown in Fig. 13. It is concluded that the maximum value of equivalent elastic strain decreases by 24.6% with using piezoelectric transducers for non-rotating propeller.

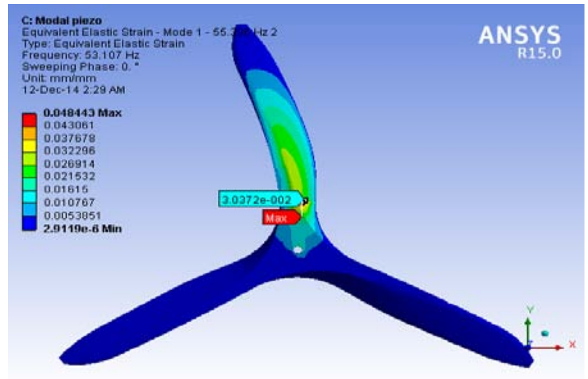

Fig. 15. Equivalent elastic strain for the propeller with piezoelectric transducers

To get Frequency Response Function (FRF) in ANSYS-15 Workbench. A mode superposition harmonic analysis with a unit forcing function is conducted. Fig. 16 and Fig. 17 show the FRF for non-rotating propeller without and with piezoelectric transducers respectively. These figures show that the harmonic amplitudes are reduced in the first three modes with using piezoelectric transducers. Table 5 shows comparison amplitudes from FRF for non rotating propeller without and with piezoelectric transducers. This table indicates amplitude reduction of 90.55 % at the first mode, while the reduction reaches 62.40 % at the third mode.

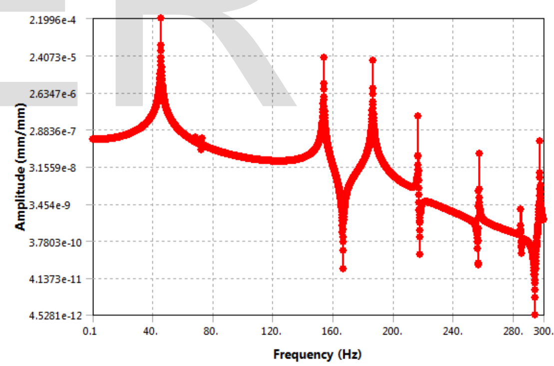

Fig. 16. FRF for the propeller without piezoelectric transducers

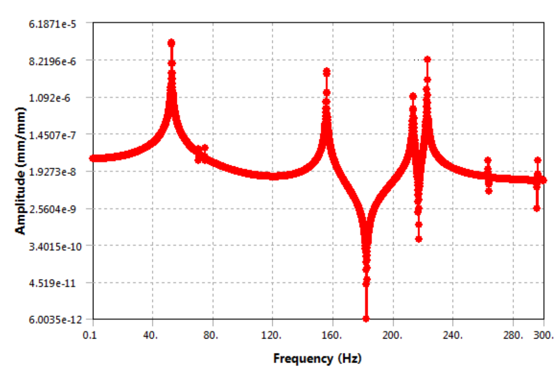

Fig. 17. FRF for the propeller with piezoelectric transducers

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TABLE 5
COMPARISON AMPLITUDES FROM FRF FOR THE PROPELLER WITHOUT AND WITH PIEZOELECTRIC TRANSDUCERS

Mode No.	Amplitude without piezoelectric transducers (mm/mm)	Amplitude with piezoelectric transducers (mm/mm)	Reduction Percent
1	2.2×10^{-4}	2.08×10^{-5}	90.55 %
2	1.82×10^{-7}	6.32×10^{-8}	65.27 %
3	1.75×10^{-7}	6.58×10^{-8}	62.40 %

4 CONCLUSIONS

In this paper experimental modal testing is carried out for UAV propeller, then series of numerical simulations using ANSYS-15.0 Workbench are conducted for the propeller to damp vibrations by applying piezoelectric transducers. Experimental modal testing of the propeller without piezoelectric transducers shows excellent agreement with the numerical simulation with deviation about 1-2 %, this result verifies the numerical simulations. Numerical results are obtained for non-rotating propeller without and with piezoelectric transducers. Shunted piezoelectric transducers are applied at the first mode high modal strain area for each blade. The comparison of the natural frequencies without and with piezoelectric transducers show that using piezoelectric transducers increase natural frequencies. In addition using piezoelectric transducers reduce the maximum value of the elastic strain for the first mode and also reduce the amplitudes for FRF in the first three modes. These results show that the vibrations induced in the propeller are damped by applying piezoelectric transducers at the first mode maximum modal strain locations of the propeller blades. Therefore, piezoelectric transducers can be used for vibration damping in turboprop engines by applying these transducers to propeller blades.

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