KAUNAS UNIVERSITY OF TECHNOLOGY

DARIUS VAINILAVIČIUS

DEVELOPMENT OF METHODS AND INSTRUMENTS FOR PRECISE ROTARY SYSTEMS DIAGNOSTIC PARAMETERS MEASUREMENT

Summary of Doctoral Dissertation
Technological Sciences, Measurement Engineering (10T)

2016, Kaunas

This doctoral dissertation was prepared in Kaunas University of Technology, Faculty of Electrical and Electronic Engineering, Department of Electronic Engineering in 2011 – 2016.

Scientific Supervisor:
Prof. Dr Habil. Vygantas Stasys AUGUTIS (Kaunas University of Technology, Technological Sciences, Measurement Engineering – 10T)

English Language Editor:
UAB "Synergium"

Lithuanian Language Editor:
Rita Malikėnienė
Publishing house "Technologija"

Dissertation Defence Board of the Measurement Engineering Science Field:
Prof. Dr Liudas MAŽEIKA (Kaunas University of Technology, Technological Sciences, Measurement Engineering -10T) – **chairman**;
Prof. Dr Mindaugas JUREVIČIUS (Vilnius Gediminas Technical University, Technological Sciences, Measurement Engineering -10T);
Prof. habil. Dr Vladimir T. MICHENYA **(**Belarusian National Technical University, Technological Sciences, Mechanical Engineering -09T);
Prof. Dr Renaldas RAIŠUTIS (Kaunas University of Technology, Technological Sciences, Measurement Engineering -10T);
Dr. Egidijus ŽUKAUSKAS (Kaunas University of Technology, Technological Sciences, Measurement Engineering -10T).

The official defence of the dissertation will be held at 1 p.m. on 13th October, 2016 at the public meeting of the Dissertation Defence Board of Measurement Engineering Science Field in Dissertation Defence Hall at Kaunas University of Technology.

Address: K. Donelaičio St. 73 – 403, 44249 Kaunas, Lithuania.
Tel. No. (+370) 37 300 042; fax. (+370) 37 324 144; e-mail doktorantura@ktu.lt.

A summary of this doctoral dissertation was sent on 13th September, 2016.

The doctoral dissertation is available on the internet http://ktu.edu and at the library of Kaunas University of Technology (K. Donelaičio St. 20, 44239 Kaunas, Lithuania).

KAUNO TECHNOLOGIJOS UNIVERSITETAS

DARIUS VAINILAVIČIUS

PRECIZINIŲ ROTORINIŲ SISTEMŲ DIAGNOSTINIŲ PARAMETRŲ MATAVIMO METODŲ IR PRIEMONIŲ SUKŪRIMAS

Daktaro disertacijos santrauka
Technologijos mokslai, matavimų inžinerija (10T)

2016, Kaunas

Disertacija rengta 2011-2016 metais Kauno technologijos universitete, Elektros ir elektronikos fakultete, Elektronikos inžinerijos katedroje.

Mokslinis vadovas:
prof. habil. dr. Vygantas Stasys Augutis (Kauno Technologijos universitetas, technologijos mokslai, matavimų inžinerija – 10T)

Anglų kalbos redaktorius:
UAB "Synergium"

Lietuvių kalbos redaktorė:
Rita Malikėnienė
Leidykla "Technologija"

Matavimų inžinerijos mokslo krypties disertacijos gynimo taryba:

Prof. dr. Liudas MAŽEIKA (Kauno Technologijos universitetas, technologijos mokslai, matavimų inžinerija – 10T) – **pirmininkas**;
Prof. dr. Mindaugas JUREVIČIUS (Vilniaus Gedimino technikos universitetas, technologijos mokslai, matavimų inžinerija – 10T);
Prof. habil. dr. Vladimir T. MICHENYA **(**Baltarusijos nacionalinis technikos universitetas, technologijos mokslai, mechanikos inžinerija – 09T);
Prof. dr. Renaldas RAIŠUTIS (Kauno technologijos universitetas, technologijos mokslai, matavimų inžinerija – 10T);
Dr. Egidijus ŽUKAUSKAS (Kauno Technologijos universitetas, technologijos mokslai, matavimų inžinerija – 10T).

Disertacija bus ginama viešame Matavimų inžinerijos mokslo krypties disertacijos gynimo tarybos posėdyje, 2016 m. spalio 13 d. 13 val. Kauno technologijos universiteto Disertacijų gynimo salėje.

Adresas: K. Donelaičio g. 73 – 403, 44249 Kaunas, Lietuva.
Tel. (370) 37 300 042; faks. (370) 37 324 144; el. paštas doktorantura@ktu.lt.

Disertacijos santrauka išsiųsta 2016 rugsėjo 13 d.
Su disertacija galima susipažinti internetinėje svetainėje http://ktu.edu ir Kauno technologijos universiteto bibliotekoje (K. Donelaičio g. 20, 44239 Kaunas).

1. INTRODUCTION

Work relevance

A rotor is an element of a machine, which rotates about a fixed axis. Rotary systems are widely used in industry. There are different types of systems and they have various purposes. From systems without aerodynamic elements: electric motors, generators, shaft systems, to systems with aerodynamic elements: compressors, wind turbines plants, industrial fans, also hydrodynamic systems, such as propellers, gas and liquids pumps.

A rotor converts received energy into work, however an amount of the energy is converted into vibrations, which can be caused by an undesirable movement of the rotor's central axis. These vibrations transfer to other components and structures causing damage and reducing the life of these components. Vibrations over time can lead to various consequences (1):

- fatigue, which can cause damage;
- friction, which can lead to component wear or additional acoustic noise;
- lower efficiency;
- production of defective products.

The diagnostics of rotary systems is an important field. Its objectives are monitoring, recording, processing and analysing of work parameters. Its aim is not only to detect parameter alterations, but also to predict them.

The main task of rotor diagnostics is to detect early deviation of the work parameters and offer solutions to bring these parameters back to normal values. The main defect of a rotor system can cause secondary defects. For example, rotor imbalance can accelerate wear of support bearings, so it is necessary to remove the main defect and prevent secondary defects from occurring. A deviation from the parameters must be detected as early as possible and the source defect named.

Diagnostics are also relevant to precise rotary systems. Due to the specific construction of these systems, small changes of working parameters can cause noticeable vibration changes. For example, in this work we have investigated rotor balancing of the autogyro. The adjustment of the blade attack angle by 0.044° leads to the reduction of particular vibrations by 44%. The diagnostics of these systems due to specific construction and work regimes requires unique measurement solutions. This work is dedicated to the investigation of three rotor systems. Specific measurement systems are proposed for the measurement of the diagnostic parameters of these rotor systems.

The objective and tasks of the work

The objective of the dissertation is to develop methods and instruments for the measurement of the diagnostic parameters for specific rotary systems, asses their metrological characteristics, and implement practical applications.

Tasks of the work:

- overview of related scientific work in Lithuania and abroad;
- development of a measurement system to correspond to the objectives of diagnostics;
- assessment of metrological characteristics of developed systems;
- measurement of diagnostic parameters of selected specific rotary systems;
- proposition of new algorithms for measurement data processing.

Scientific novelty of the work

- Proposition of measurement system structures for diagnostic parameters measurement for specific rotors systems and assessment of their metrological characteristics;
- proposition of processing algorithms for diagnostic signals;
- presentation of options for complex blade balancing;
- investigation of aerodynamic characteristics of autogyro blades impact on second harmonic vibration.

Practical application

The proposed system structures were implemented and the specific measurement systems were developed. These systems were used to research specific objects with rotary systems. The results obtained are presented in this work.

The wireless system for the measurement of an autogyro rotor's imbalance parameters was developed and is actively used in *UAB Birdmanas* autogyro fleet.

Approbation of results

The main scientific results have been published in five articles: two publications in journals, which are indexed in Thomson Reuters List, 1 article in a journal, which is in the list of international database publications approved by the Research Council of Lithuania, and 2 articles in conference proceedings.

The results were also presented at three international conferences in Lithuania, Poland and Italy.

Presented for defence

1. Proposed structures of measurement systems and data processing algorithms.
2. Results and conclusions obtained with developed systems.

Structure and volume of the dissertation

The doctoral dissertation consists of an Introduction, 3 main chapters, conclusions, a list of references and a list of scientific publications. The entire content is presented in 92 pages featuring 64 figures, 26 tables, 24 mathematical expressions and 89 references.

2. THE SYSTEM FOR COMPLEX BALANCING AND IMBALANCE PARAMETERS MEASUREMENT

2.1. The measurement system

We have designed a system (Fig. 1) that enables safe control of the mass and aerodynamic imbalance and measurement of various parameters. These parameters reflect the imbalance effect.

The system consists of a 2 blade fan with various sensors, a data acquisition system, a personal computer with software for data processing and visualizing.

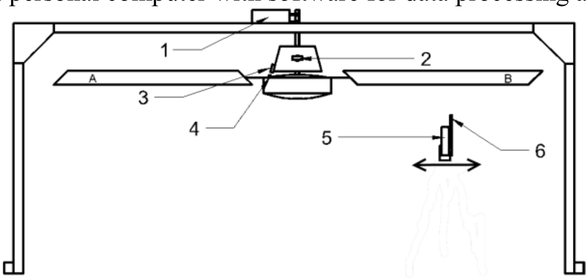


Fig. 1. The structure of the measurement system. 1 – lift force sensor, 2 – accelerator, 3 – magnetic field sensor, 4 – magnet, 5 – contactless displacement sensor, 6 – dynamic pressure sensor.

The fan is powered by an electric motor. The length of the blades is 680 mm. The rotation speed of the fan is selected from 3 pre-set speeds; 2, 3 and 4 Hz.

The measurement system is capable of measuring various parameters of the rotating blades:

- lift force that is generated by the blades;
- the fan's vibrations;
- dynamic blades pressure;
- blades profile;
- rotation speed;

The fan is mounted on its support frame through the force sensor, this configuration enables the measurement of the fan's weight. From the measurement results at different rotation speeds, the lift force can be calculated.

The fan's vibrations are measured by using 2 axis accelerometers that are mounted on the static element of the fan. A pressure sensor is used to measure dynamic blade pressure. A laser triangular sensor is used for the blade profile scan.

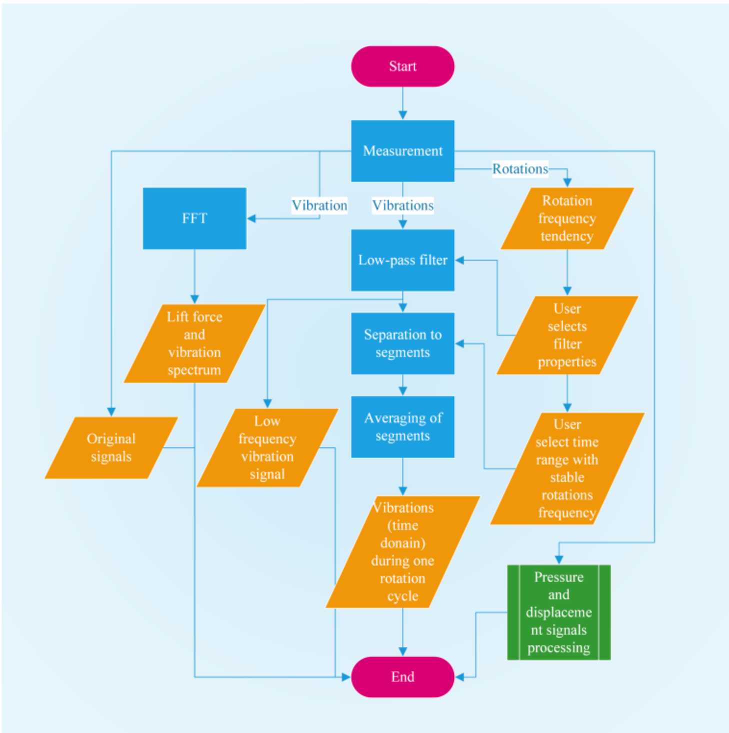


Fig. 2. Simplified signal processing algorithm

The magnetic field sensor and 5 x 5 x 5 mm neodymium magnet are used as a marker for acquiring the rotation speed of the blades. Another purpose of this sensor is to synchronize the start of the rotation cycle with other measurement channels. This synchronization is important for further measured data processing and analysis. It allows the data between rotations to be averaged and thus minimizing random measurement errors. It is also vital for acquiring phase information of fan vibrations and separating individual blade pressure, profile from primary measurement data.

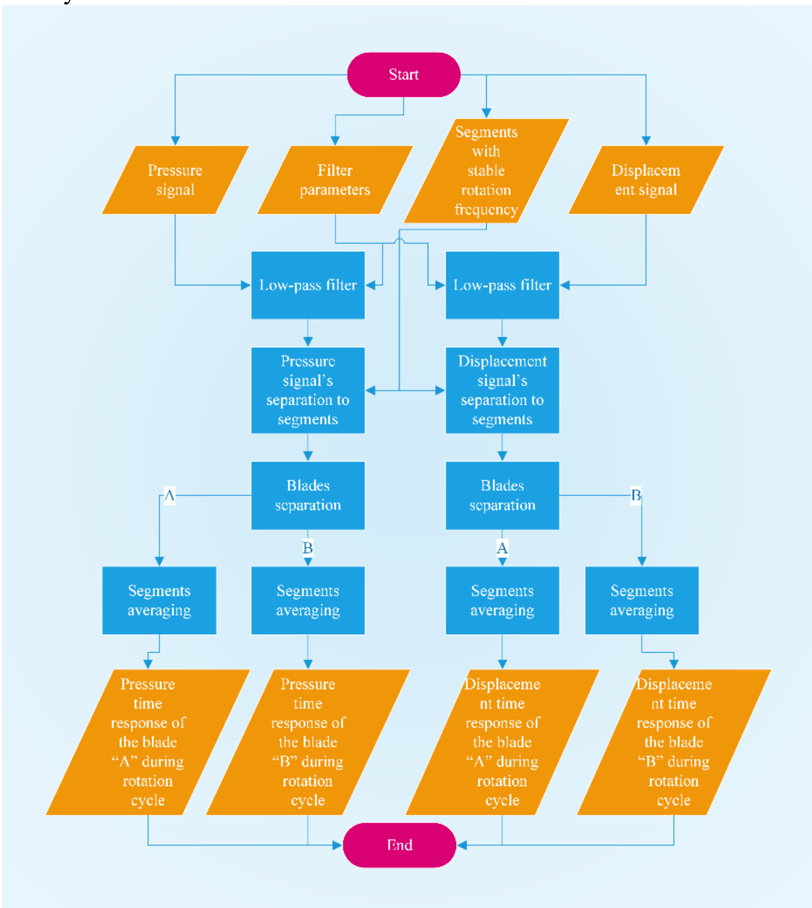


Fig. 3. Pressure and displacement signal processing algorithms

After a measurement, the recorded data were processed using Matlab 2013b (Fig. 2.). The FFT analysis of the vibration signal and lift force signal reveals the main frequency components. The rotation speed trend helps to select the measurement interval with a stable rotation speed. The vibration signal is passed through a low pass filter the parameters of which are adapted to the extract frequency equal to the rotation frequency. The signal of this particular frequency reveals information about the rotor's mass imbalance. This signal is decomposed to segments, each is related to the individual fan revolution. The random measurement error is minimized by averaging segments. Processing was also used on this algorithm based signal in the experiments that are presented in the next chapters.

Similar signal processing (Fig. 3) is performed with the displacement and pressure signals, but additional measurement results are related not only to the individual rotation, but also to the individual blade.

2.2. Blade profile measurement

The aim of the blade profile measurements is to evaluate the fan blades' angle of attack alteration at various rotation speeds, particularly if both blades behave in a similar way while the rotation speed is changing.

Measurements were performed using a laser triangular displacement sensor with the laser light directed at the bottom surface of the blades (Fig. 4.). The laser displacement sensor was fixed on a movable tripod. The blade's profile scan was performed by moving it along the x-axis from the tip of the blade to the centre of the fan with steps of 50 mm. The y axis represents width of the blade. The z axis shows the distance from the displacement sensor to the surface of the blade.

Measurements at 10 points along the x axis were taken at the 3 pre-set rotation speeds; 2, 3 and 4 Hz. Measurement data were acquired and saved during several rotations.

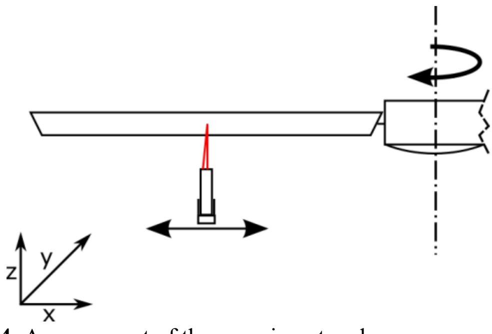
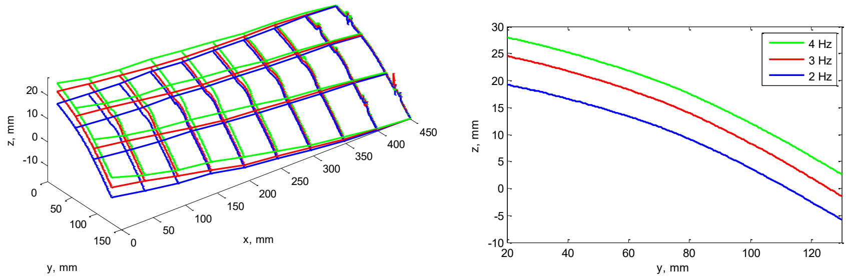


Fig. 4. Arrangement of the experiment and measurement axis

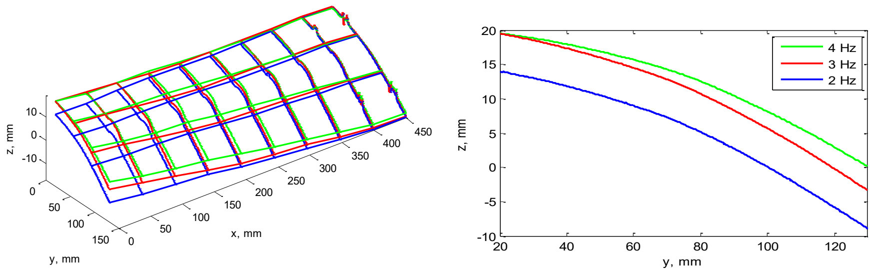
The 3D line grid represents the results of the measurements for easy comparison between scans. The 2D lines represent the blade profile scan at the furthest section from the centre of the rotor where the peak of blade bending can be found.

Fig. 5 and Fig. 6 show the blade profiles at rotation speeds equal to; 2, 3 and 4 Hz. It shows how blades “A” and “B” tend to bend upwards as rotation speeds increase.



a) b)

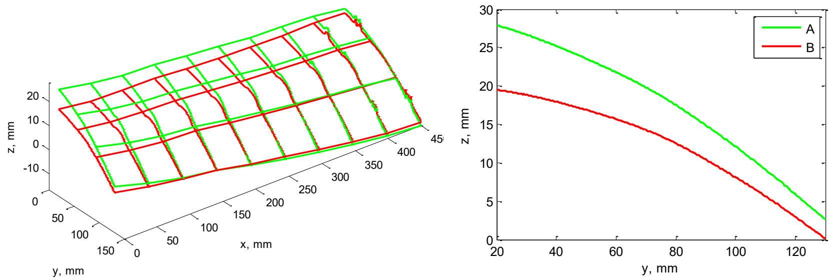
Fig. 5. a) Blade “A” profile scan at 3 rotation frequencies. b) View from left, plane cut at x = 0 mm.



a) b)

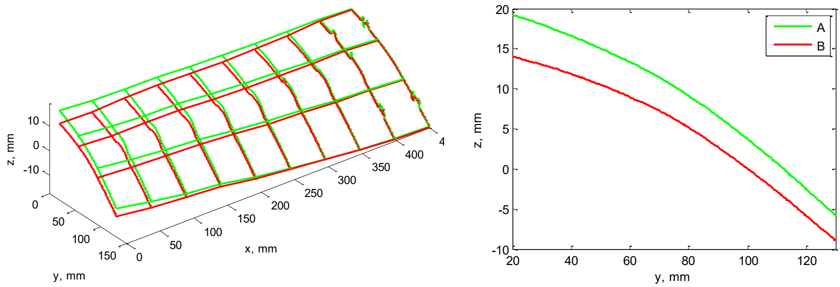
Fig. 6. a) Blade “B” profile scan at 3 rotation frequencies b) View from left, plane cut at x = 0 mm.

Fig. 7 and Fig. 8 compare the differences between “A” and “B” blade profiles when rotation speeds are 2 and 4 Hz.



a) b)

Fig. 7. a) Blade profile scans at 4 Hz rotation speed. b) View from left, plane cut at x = 0 mm.



a) b)

Fig. 8. a) Blade profile scans at 2 Hz rotation speed. b) View from left, plane cut at x = 0 mm.

The results show that not only do the blades profiles not match, but they have different rigidity and bend unequally while the rotation speed is rising. Blade "A" vertical displacement is larger than blade "B" while rotation speeds increase. This means that the blades do not generate equal lift force at the lowest speed and this difference increases alongside rotation speed.

2.3. Blade pressure measurement

The aim of this measurement was to find a link between blade profiles and their generated dynamic pressure. A pressure scan was performed along the x axis with 50 mm steps from the blade tip to the centre of the fan at rotations speeds: 2, 3 and 4 Hz.

The pressure scan results for individual blades at different rotation speeds are presented in Fig. 9. The pressure magnitude increases with rotation speed for each blade, however blade “A” magnitude is larger due to a steeper blade angle.

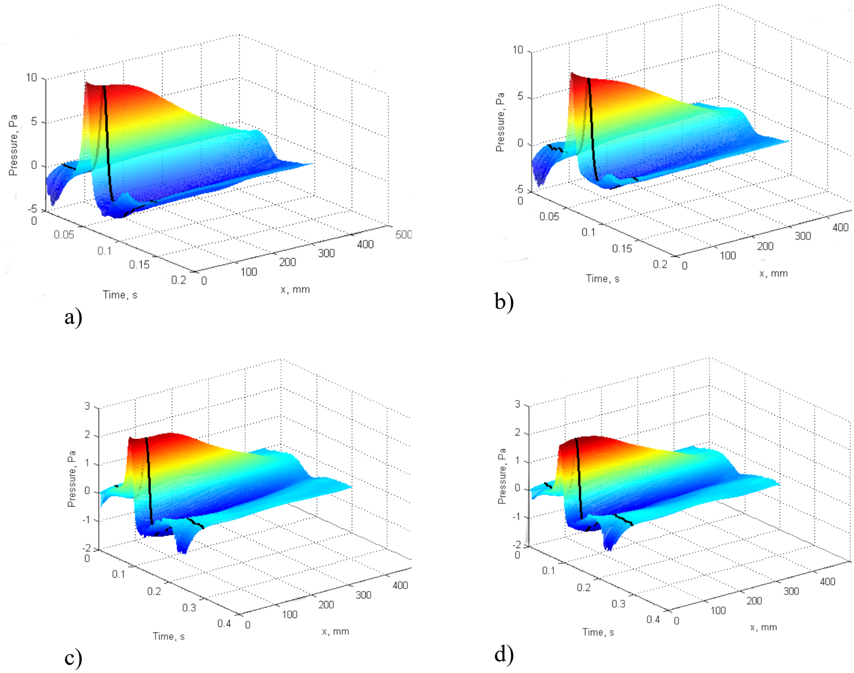
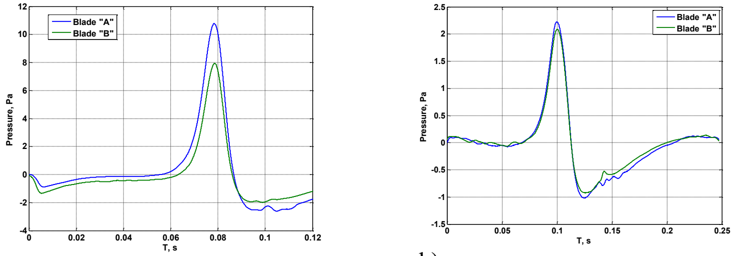


Fig. 9. Blade pressure scan: a), b) f = 4 Hz, c), d) f = 2 Hz. a), c) blade “A”, b), d) blade “B”.

This final result of the pressure scan was produced by processing the data in following order: during several fan rotations, recorded measurement data subsequently assigned to individual blades and averaged to reduce random measurement errors, likewise processing blade profile data; 3D pressure scan combined from individual measurement points at x direction and final graph smoothed for better presentation by interpolating data between existing points in x axis.

Pressure difference between blades is visualized better as plane cuts in Fig. 10.



a) b)

Fig. 10. Cut plane from blades pressure scan at x = 50 mm. a) f = 4 Hz, b) f = 2 Hz

A pressure difference between top and bottom blade surfaces generates lift. The difference between blade pressures indicates that the blades do not create even lift force, and aerodynamic imbalance occurs.

Fig. 11 represents measurement points with maximum pressure along the blade's length.

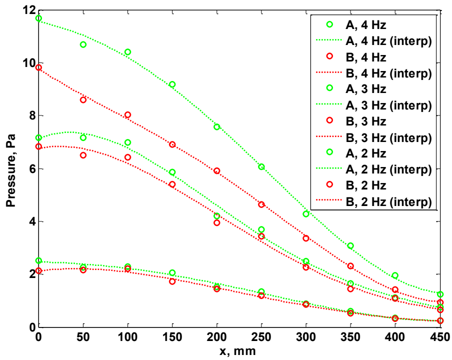


Fig. 11. Pressure curve peaks along the blade's length

Results show that the pressure increases toward the outer end of the blade. This can be explained by the higher speed of the air mass, which creates a greater pressure difference.

2.4. Interaction between mechanical and aerodynamic imbalance

A mechanical and an aerodynamic imbalance causes unwanted torque that leads to vibration of the rotor. Consequently, the presumption follows that force torques in the opposite direction would counterbalance and minimize vibrations of the rotor. Condition for minimal vibrations:

$$M_1 = M_2; \quad (1)$$

where M_1 – torque created by aerodynamic imbalance, M_2 – torque created by mechanical imbalance.

The lift force difference between the first and the second blade creates torque M_1. The external weight creates torque M_2. The torques act around the point of the fan's consolidation. The torque is expressed further and visualized in figure Fig**. 12**.

$$M_1 = F_l \cdot R_l; \quad (2)$$

where F_l is the difference between the lift force of blade "A" and blade "B", R_l – distance between centre of lift force and centre of the fan.

F_l is measured using a load cell transducer. $F_l = 0.84$ N at $f = 3.8$ Hz; $F_l = 0.37$ N at $f = 3.2$ Hz. Typically, the centre of the lift force is located at 75% of the blade length. For the fan used in the measurements $R_l = 0.51$ m.

$$M_2 = F_m \cdot L, \quad (3)$$

$$F_m = m \cdot (2\pi f)^2 \cdot R_m; \quad (4)$$

where F_m is the centrifugal force created by the external mass m, L - distance between the consolidation point and centrifugal force vector, f – rotation speed of the blades. R_m is the distance between the external weight and centre of the rotor.

The mass of the weight used during the experiment was 0.014 kg. $L = 0.2$ m, R_m varies from -0.68 to 0.68 m.

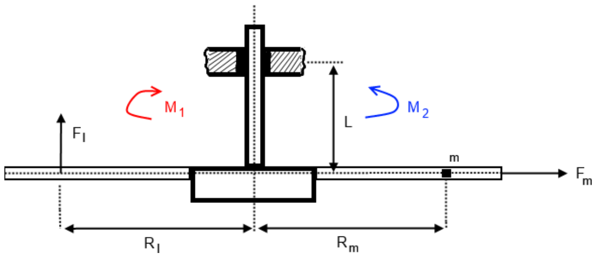


Fig. 12. Mechanical model of the system

Based on the formulas (1-4) a theoretic model was created and the calculations performed (Fig. 14) which confirmed the presumptions that the torque created by the aerodynamic and mechanical imbalance counterbalance at particular conditions, consequently the experiment was performed.

The external weight was moved from one blade edge to the opposite during the experiment. It was placed at 8 fixed locations (Fig**.** 13) on the blades and the rotor. Locations were selected to reflect specific geometric points based on the structure of the fan.

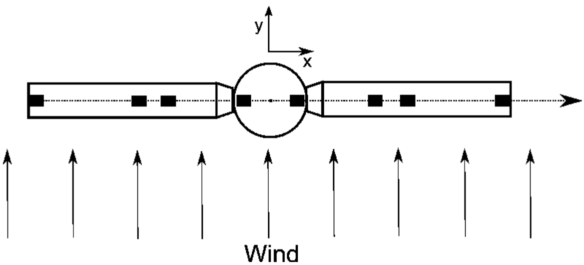


Fig. 13. Positions of external weight, that creates mechanical imbalance.

The blades were rotated at 2 different speeds for each weight position. Vibration measurement results show that there is a point where the level of vibration is minimal, but the location of that point changes with rotation speeds.

The results of the measurements (Fig. 14b) were compared with the theoretical calculations (Fig. 14a) which shows the torque difference minimum near the same position of the weight as during the experiment.

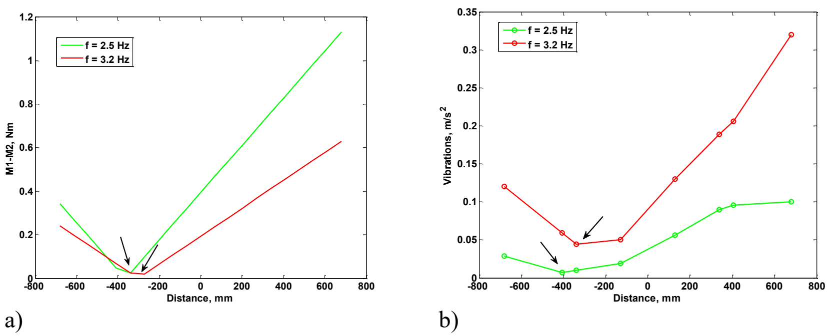


Fig. 14. a) Lift force difference versus external weigh location (model). b) Vibration level versus external weigh location, arrows pinpoint at minimal values.

The imbalance experiment was repeated, introducing a new factor that impacts the balance of the rotor – wind. This airflow should increase the aerodynamic imbalance of the blades due to the blade geometry not being identical.

Fig. 13 demonstrates the direction of airflow. In Fig. 15 the presented measurement results show that the wind also influences the location of the weight, where vibrations are minimal. The point where vibrations are minimal moved by one position to the opposite side when an external airflow was present.

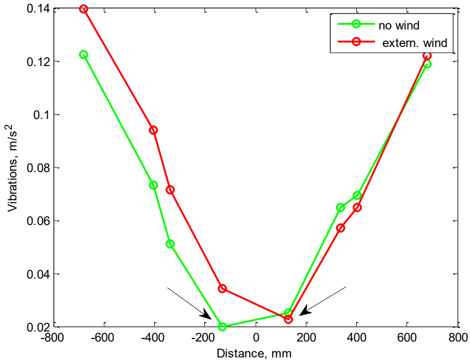


Fig. 15. Vibration level versus external weigh location with external wind source, arrows pinpoint minimal values.

The frequency spectrum of the vibrations is presented in Fig**.** 16, where the fan rotation frequency is 3.2 Hz. It shows that the spectrum component with a double frequency of rotation increases significantly more than the component that is equal to the rotation frequency when an external airflow impacts the fan. Vibrations in y direction are more sensitive to airflow impact, because of the airflow direction in respect of the fan and sensors position.

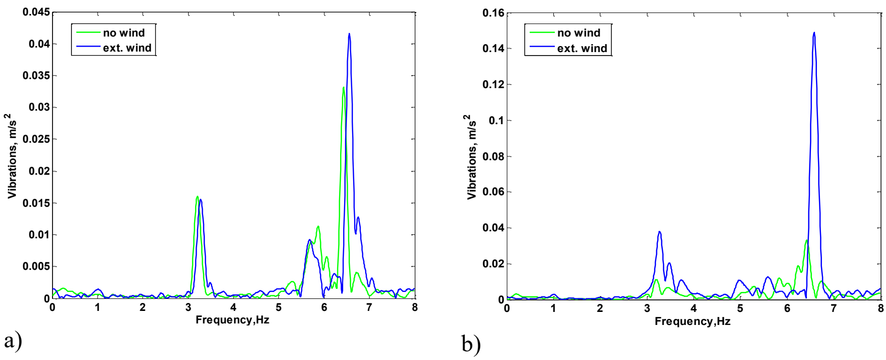


Fig. 16. Frequency spectrum of fan vibrations. a) Vibrations in x direction, b) Vibrations in y direction.

2.5. Conclusions

The comparison of blade geometry at different rotation speeds shows that blades tend to change geometry but not at an equal rate. This show the quality of the blades and its supporting mechanism.

The forced mechanical imbalance may compensate for the aerodynamic imbalance and minimize system vibrations, however the best location for the weigh placement moves due to the changed rotation speed and the introduction of an external wind, so this way of compensating aerodynamic imbalance is not suitable for rotating systems that work outside of constant conditions.

The double rotation frequency component of the vibration spectrum is sensitive to aerodynamic forces that occur during blade rotation when an external airflow impacts the fan.

3. THE SYSTEM FOR MEASUREMENT OF AUTOGYRO'S ROTOR'S IMBALANCE PARAMETERS

In aviation, rotary systems are the driving force not only in the air, but also on the ground. The reliability of these systems are directly related with the passengers' safety.

3.1. The measurement system and data processing algorithms

The designed measurement system was used to acquire vibration information during the balancing of the Calidus and MTOsport autogyros. The system (Fig. 17) consists of the measurement module, main module and tablet with dedicated software for signals processing.

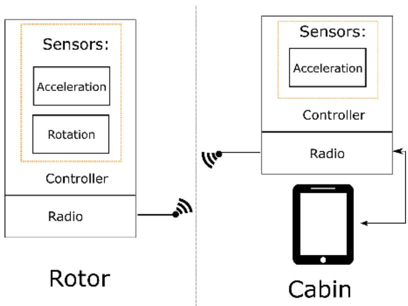


Fig. 17. The structure of the measurement system

The main tasks for the system for the measurement of the autogyro's imbalance consists of:

- recording of vibration and rotation signals;
- transmitting of signals from the measurement module to the main module;
- analysis and visualization of acquired signals.

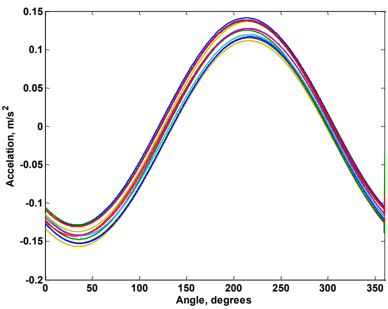
Before starting the measurements the user can select the measurement duration and range, sampling frequency and accelerometer's internal filter parameters. After the user selects to start the measurements, the measurement command is transmitted via radio to the measurement module. It starts the vibration and rotations measurements within the selected parameters. After the measurements are finished, the data is transmitted to the main module. Further data processing is automated for user convenience.

During the measurement a data array with timestamps is formed from the outputs that are generated by the rotation sensor. Timestamps relate to the start and finish of every rotor rotation. From these timestamps, the main rotation frequency is found. Measurement segments, where the rotation frequency is equal to the main rotation frequency with ±5% error, are extracted. Previous processes provide measurement results with a stable rotation frequency.

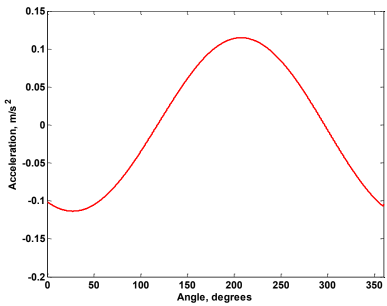
The known rotation frequency and FFT analysis provide the automatic calculation of magnitudes of the vibration components that are equal to the rotation frequency and double rotation frequency. These magnitudes, vibrations frequency spectrum and RMS vibration amplitude are presented to the user.

Vibration signals are passed through the low pass filter. The parameters of the low pass filter are related to the rotation frequency of the rotor. To isolate the signal that represents rotation frequency vibrations f, the low pass filter has the following properties: $fpass = 1.1f$, $fstop = 1.5f$. For the extraction of the double rotation frequency signal, the low pass filter properties are: $fpass = 2.2f$, $fstop = 3.0f$.

In the last step of signals processing, vibration signals are split into segments (Fig. 18.) that are related to each of the measured rotor rotations. Segments with a magnitude or phase standing out from the majority are rejected as invalid. Remaining segments are averaged to minimize the random measurement error.



a)



b)

Fig. 18. Vibration segments before a) and after b) averaging.

3.2. Analysis of autogyro rotor balancing quality

The measurement system was used to acquire vibration data during the rotor balancing of the Calidus (as shown in figure Fig. 19) and MTOsport autogyros.

Fig. 19. Calidus Autogyro and sensors locations

Vibration data were recorded on the ground, while the rotor was spinning and during consistent flight. Vibration sensors were mounted near the centre of the rotor and in the cabin. The purpose of these measurements was to obtain vibration magnitude and phase variations after performing various rotor adjustments.

Calidus autogyro' rotor balancing procedure starts with regular maintenance tasks that include an inspection of the rotor and the blades for damage and component wear. The Calidus autogyro' rotor hub and head assembly are shown in Fig. 20. (2). During regular intervals (based on flight hours), the blades with the rotor hub are dismounted from the rotor head. In further steps, the blades are aligned with the rotor hub, and non-reusable parts are replaced with new ones.

There are three methods for autogyro rotor balancing:

- introducing the external weight inside the blade;
- changing the angle of attack of both blades;
- changing the rotor hub position in relation to the rotor head.

Unlike most common helicopters, an autogyro has a 2 blade rotor instead of 4, consequently there are only two balancing weight placement positions instead of 4. It limits the number of weight mass and position combinations that are available for rotor balancing.

In the Calidus autogyro the external weights for blade balancing are placed inside the blade. Due to centrifugal force the weights glide to outer end of the blade. During the balancing procedure that we performed, we used washers that weigh 0.7 – 1.4 g. Calidus' blade radius is 4.22 m. During flight, the blades rotate at a velocity of around 360 rpm. Presumably, the washer weighing 1 g is at the end of the blade during flight conditions, from these values the calculated centrifugal force is 6.0 N. At the given blade radius and rotation speed any weight placed inside the blade outer end will be accelerated by 611 g.

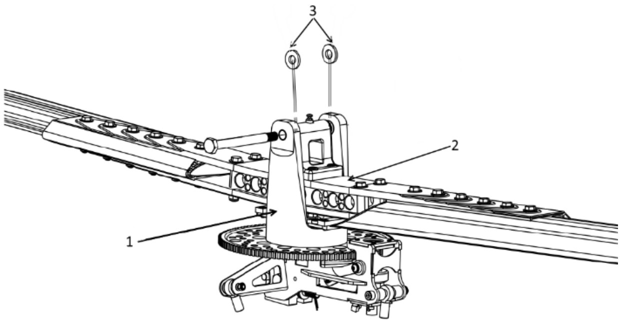


Fig. 20. Rotor hub and head assembly. 1 – rotor head, 2 – rotor hub, 3- dim washers. Taken from "Autogyro MT-03 Gyroplane maintenance manual".

Due to the simplicity of the autogyro blades and rotor construction, independent adjustment of the angle of attack for each blade is impossible. The method for altering the blade attack angle increases one blade angle; however, the other blade angle is lowered. This angle alteration is performed by dismounting the rotor hub and blades from the rotor head. A thick metal sheet is added between the rotor hub upper part and rest of the body as shown in figure Fig. 21.

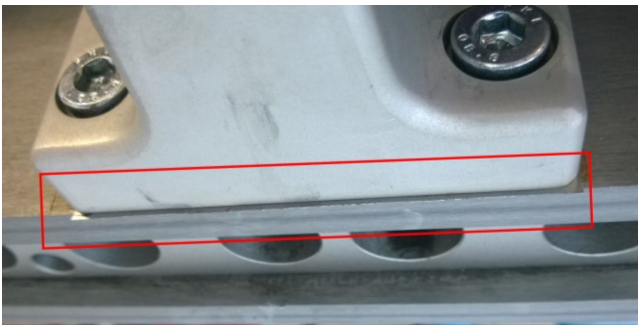


Fig. 21. Metal sheet (0.05 mm) between the upper part of the rotor hub and the rest of the body. This method is used to change the blade angle of attack.

The last adjustment that is used to balance the rotor is the rotor hub displacement in relation to the rotor head. The rotor hub can be displaced perpendicular to the autogyro's length. This displacement is maintained by selecting the width of the dim washers (Fig. 20), which intermediate between the rotor hub and head. During our balancing experiments, the maximum displacement of rotor hub was 0.1 mm.

3.2.1. The experiment and measurement results

Vibration measurements were acquired from three autogyros: Calidus (LY-BAR, LY-BBG) and MTOsport LY-MTO. Vibration data were recorded during steady flight at a speed of 100 km/h. Results that included vibration magnitudes and phases are presented in the table and in polar charts.

Recorded signals from the vibration sensors were separated by individual rotor rotations sensed with the rotation sensor. Signals were averaged and passed through the low pass filter. Vibration magnitudes were calculated from the frequency spectrum, and phase from the averaged and filtered signals.

To assess the phase measuring accuracy, the measuring system was tested using a controlled vibration source. The measured vibration phase was compared with the reference measurement (Fig. 22.). The results reveal that the phase lag is less than 6° degrees in the frequency range that is used for rotor balance analysis.

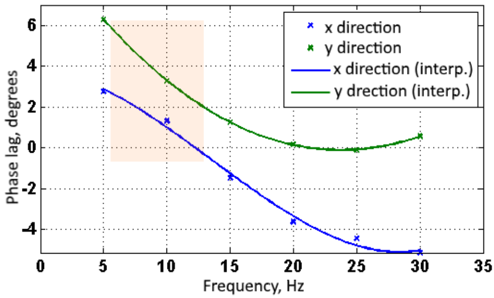


Fig. 22. Measurement system's phase lag dependency from signal frequency. 6-12 Hz frequency range is marked.

The rotor hub and head element adjustments were made taking into consideration the previous measurement results and the pilot's knowledge from previous balancing procedures.

Measurements were obtained during the initial flight and after various adjustments, that included rotor hub displacement and balancing weight introduction:

Initial flight;

- Rotor hub displaced by 0.1 mm to the left (relative to initial position at the 1st flight);
- 1.7 g weight added to 2 blades;
- Weight removed, rotor hub displaced by 0.05 mm to the left (relative to the initial position at the 1st flight);
- 1.7 g weight added to 1 blade;
- Weight removed, rotor hub displaced by 0.1 mm to the left (relative to initial position at the 1st flight).

The results of the measurements are presented in Table **1**.

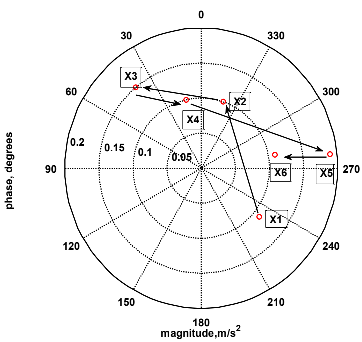
Table 1. Measurement results for Calidus autogyro LY-BBG

Flight No.	**Vibrations (rotor), X direction**		**Vibrations (rotor), Y direction**		**Vibrations (cabin), Z direction**		**Vibrations (cabin), Z direction (2 rev)**
	Magni-tude, m/s^2	**phase, degrees**	**Magnitude, m/s^2**	**phase, degrees**	**Magni-tude, m/s^2**	**phase, degrees**	**Magni-tude, m/s^2**
1	0.11	231	0.08	89	0.08	0	0.95
2	0.1	341	0.1	133	0.1	74	0.63
3	0.15	40	0.18	72	0.03	33	0.75
4	0.10	13	0.12	98	0.09	61	0.66
5	0.19	276	0.19	158	0.17	75	0.76
6	0.11	280	0.11	125	0.11	75	0.70

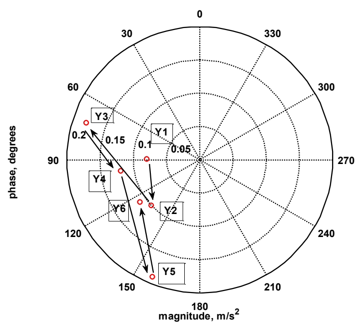
The numbers in the first column represents the magnitudes of the vibrations with a frequency that are equal to the rotor rotation frequency. The vibration in the Z-axis are presented for the rotor rotation frequency and double rotation frequency.

The vibration magnitude and phase variations also are presented in polar charts Fig. 23. Individually for every channel. These charts present the vibration and phase movement after various rotor and blade adjustments.

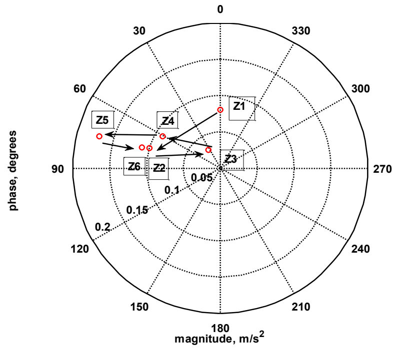
The phase in these charts increases anticlockwise, in the same manner as the actual rotorcraft blades rotates. 0° point presents the condition when the first blade is parallel to the autogyro body and points in the same direction as the nose of the autogyro.



a)



b)



c)

Fig. 23. Calidus autogyro LY-BBG vibrations polar chart: a) vibrations in x axis; b) vibrations in y axis; c) vibrations in x axis.

The results show that the lowest vibrations were on flights numbered 2 and 6 when compared to the initial conditions and the rotor hub was moved by 0.1 mm to the left. Although the vibration levels at the x and y axis near the rotor remained the same, the double rotating frequency vibrations at the z axis measured in the cabin lowered ~ 30% compared to the measurements taken on the first flight.

Lower double rotation frequency vibrations were not only detected by our measurement system, but the pilot also felt greater flight comfort.

Similar rotor balancing was performed with the Calidus autogyro LY-BAR:

- Initial flight;
- Angle of attack altered;
- Weight 1.7 g. added to the second blade
- Weight left, rotor hub displaced by 0.05 mm to the left (relative to initial position at 1 flight);

Vibrations results are presented in Table **2**. Calidus autogyro LY-BAR vibrations were minimized after second adjustment, when weight was added into the first blade.

Table 2. Measurement results for Calidus autogyro LY-BAR

Flight No.	**Vibrations (rotor), X direction**		**Vibrations (rotor), Y direction**		**Vibrations (cabin), Z direction**		**Vibrations (cabin), Z direction (2 rev)**
	Magnitude, m/s^2	**phase, degrees**	**Magnitude, m/s^2**	**phase, degrees**	**Magnitude, m/s^2**	**phase, degrees**	**Magnitude, m/s^2**
1	0.45	352	0.42	330	0.09	283	0.68

Flight No.	Vibrations (rotor), X direction		Vibrations (rotor), Y direction		Vibrations (cabin), Z direction		Vibrations (cabin), Z direction (2 rev)
	Magnitude, m/s^2	phase, degrees	Magnitude, m/s^2	phase, degrees	Magnitude, m/s^2	phase, degrees	Magnitude, m/s^2
2	0.40	352	0.41	332	0.05	318	0.73
3	0.03	53	0.10	348	0.05	349	0.90
4	0.14	143	0.07	157	0.03	332	0.67

During one flight the vibration levels were measured on the MTOsport autogyro. Among the autogyros, this aircraft had the lowest vibration levels that were felt by the pilot.

Table 3. Measurement results for the MTOsport autogyro

Vibrations (rotor), X direction magnitude, m/s^2	Vibrations (rotor), Y direction magnitude, m/s^2	Vibrations (cabin), Z direction magnitude, m/s^2	Vibrations (cabin), Z direction (2 rev) magnitude, m/s^2
0.25	0.1	0.06	0.337

Fig. 24 presents the frequency spectrum of the autogyro vibrations when the vibration sensor is mounted in the cabin.

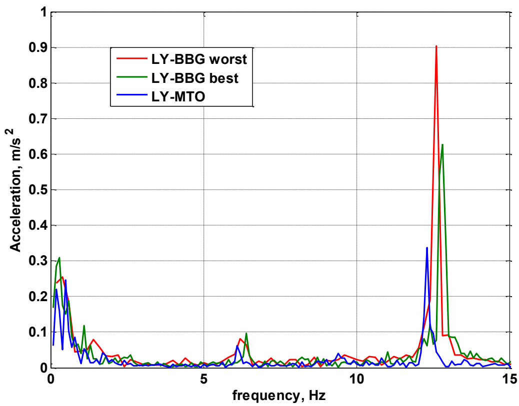


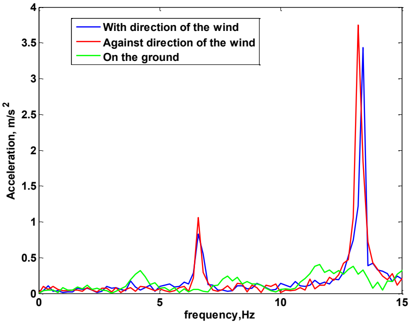
Fig. 24. Autogyro vibration spectrum from sensor in the cabin.

Although the levels of rotor vibration were similar to the minimal vibrations in the Calidus autogyros, according to the pilot he felt minimal vibration in this autogyro. In addition, the frequency spectrum calculated from the vibration measurements in the cabin showed that the component of double rotation frequency is minimal.

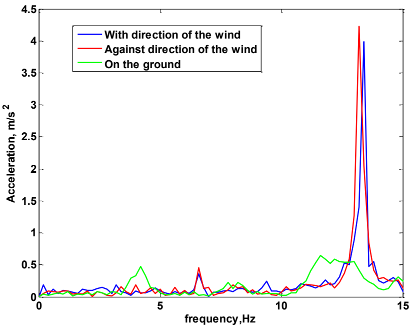
Measurement results show that the low frequency region is the dominant component of the double rotation frequency (12 Hz.). This component is reduced during the autogyro balancing procedure, and the remaining components are not affected by balancing.

3.3. Wind influence to autogyro vibrations

Flight conditions also influence the vibrations in rotorcraft. We have performed experiments with a different autogyro model, the ELA07S. The next figures will illustrate how the vibrations are dependent on the flight conditions. Fig. 25 presents the spectrum of vibrations while the autogyro is on the ground, during the flight at a steady speed of 120 km/h with the wind, and against the wind direction.



a)



b)

Fig. 25. Autogyro ELA07S vibration spectrum: a) Sensor position near the rotor, x direction; b) Sensor position near the rotor, y direction.

While the autogyro was on the ground the rotor was rotated at 250 rpm (4.2 Hz). During flight, the rotor speed reached 400 rpm (6.6 Hz). Figure 25 shows that comparing the measurement data acquired on the ground and during flight, the level of vibration increases, but the amount of increase is individual for every component of the frequency spectrum. Components that present double rotation frequency increase to 3.5 – 4.0 m/s^2 although on the ground this component was indistinguishable.

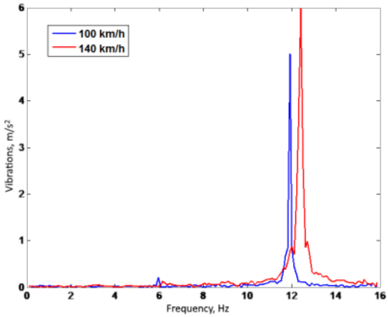
The results of the measurements show that during flight against the direction of the wind the aerodynamic resistance of the blades increases and that condition influences the vibrations of the rotor.

3.4. Analysis of flight speed influence on vibration signals

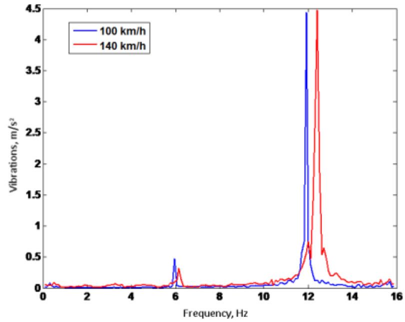
The autogyro rotor's vibrations are caused by a mass and aerodynamic imbalance. The measurements were performed at different flight speeds, 100 and 150 km/h, to investigate the influence of the aerodynamic forces on the levels of vibration of the rotor.

The first analysis of vibrations in the 0 – 800 Hz range showed that at higher speeds the RMS value of the vibrations increase about 50%, this increase was noticeable not only at the rotor, but also in the cabin.

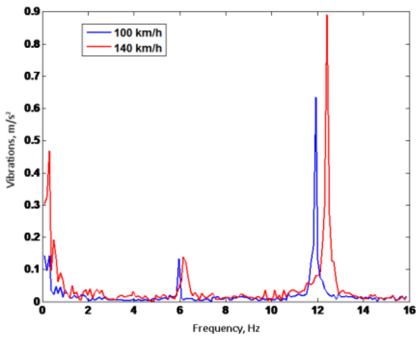
The analysis of the low frequency spectrum component was performed. In cases where the rotor has an imbalance (Fig. 26), the vibration component with a frequency that is equal to the rotation frequency is several times lower than the component of the double rotation frequency. After the rotor is balanced (Fig. 27) the rotation frequency component is lowered 5 times and is not noticeable in the plot.



a)

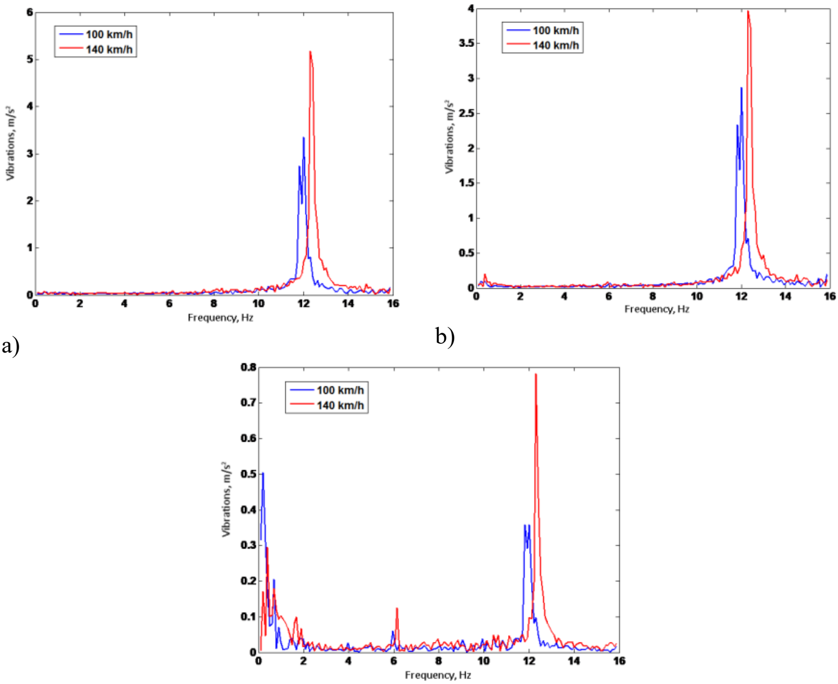


b)



c)

Fig. 26. Vibration frequency spectrum of rotor before balancing: a) rotor's vibrations in x direction; b) rotor's vibration in y direction; c) cabin's vibrations in z direction.



a) b) c)

Fig. 27. Vibration frequency spectrum of rotor after balancing: a) rotor's vibrations in x direction; b) rotor's vibration in y direction; c) cabin's vibrations in z direction.

In the case of 150 km/h flight, vibrations increase noticeable. The relative increase is about the same percentage for the components of single and double rotation speed. However, in absolute value increase the vibrations increase more significantly for the component with double rotation speed.

3.5. Conclusions

The results of the measurement show that discomfort in the cabin is caused by the double rotation frequency vibrations. The levels of these vibrations are not always related with the rotor vibrations. Consequently, it can be challenging to minimize the rotor vibrations for minimal components stress and cabin vibrations to maintain an acceptable comfort level for the pilot and passengers.

The differences in the vibration spectrum measurements during rotor rotation on the ground and during flight suggest that the component of double rotation spectrum frequency is caused by an increased air stream through the blades and it is sensitive to aerodynamic imbalance.

4. THE ASSESSMENT OF PAPER FOLDING MACHINE WORKING RINGS QUALITY WITH DISPLACEMENT MEASUREMENT SYSTEM

The proper functioning of rotor systems provides not only efficient work and low levels of vibrations, but in some cases the production of high quality products (3). One type of specific rotor system is a paper folding machine. Its work elements are shafts with rubber segments, which shape paper products. If these segments do not comply with the manufacture requirements, production defects occur – segments do not fold the paper products on the correct fold mark causing deviations.

The specific measurement system was created in order to assess the condition of the segments and to detect incorrect displacements. It allows the measurement of the displacement and surface irregularity of the segments during normal working conditions. A laser displacement sensor was selected for this task, because it is capable of measuring the displacement of individual segments of any selected shaft. Furthermore, the segments are made of rubber, so eddy current, or capacitive displacement sensors are unsuitable.

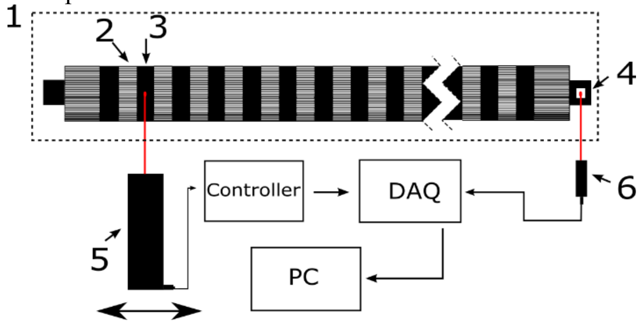


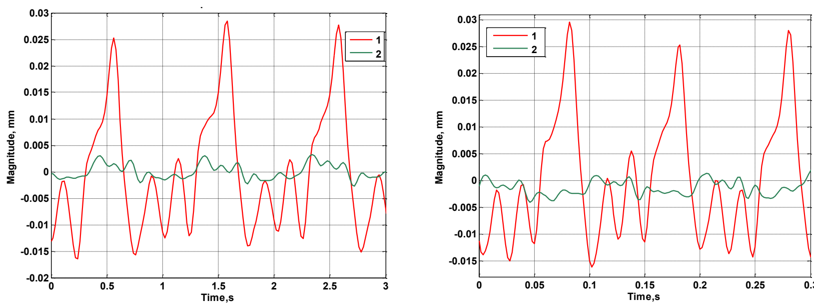
Fig. 28. The structure of the measurement system: 1 – folding roller; 2 – segment of the folding roller; 3 – working ring; 4 – reflector; 5 – laser measurement head; 6 – laser throttle sensor; control unit - laser head controller, DAQ - data acquisition unit, PC - personal computer with signal processing software.

The measurement system consists of a laser displacement sensor and its controller, reflective sensor for rotation tracking, data acquisition unit and personal computer with custom software for data processing, analysis and visualization.

The laser beam of the displacement sensor was directed to measure the roller's working rings. The laser beam from the reflection sensor was directed to the reflector that was glued to the roller. The reflective sensor helps to relate the measured displacements to rotations and extract the displacement signals for particular rotations. This way the signals can be averaged, random measurement errors minimized and repetitive displacement tendency exposed.

4.1. Measurement results

The measurements were performed at the printing house and displacements were recorded for a "Herzog & Heymann" paper folding machine. The main task was to asset the working rings displacement differences between old and renewed rollers. The first comparison (Fig. 29) shows low frequency displacement measurements that were obtained on 16 ring positions.

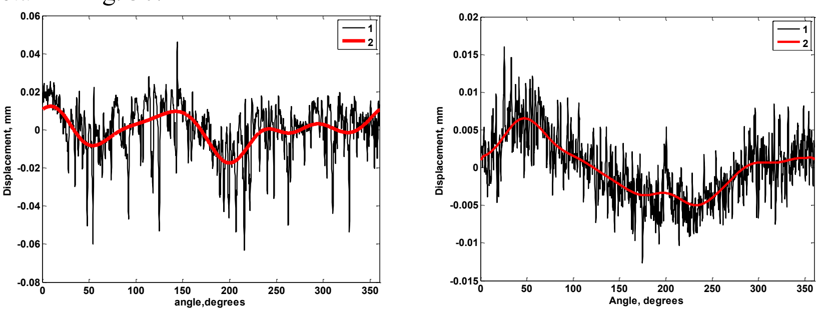


a) b)

Fig. 29. Results of measurements of displacement of segments of the rollers: a) at 9 m/min velocity; b) at 90 m/min velocity; 1- before renewal, 2- after renewal.

The measurement results show that before the rollers were renewed the peak-to-peak displacement at 9 m/min was 35µm, and 90 m/min – 12 µm. As it was expected, after the renovation of the rollers the lower peak-to-peak displacements were 9 m/min – 5 µm, and 90 m/min – 3µm.

The differences between the old and renewed rollers ae presented in more detail in Fig. 30.



a) b)

Fig. 30. Roller working ring displacements during one revolution: a) before renewal; b) after renewal; 1 – obtained signal; 2 – signal after low pass (*Fpass = 30 Hz, Fstop = 150 Hz)* filtering.

The results show the displacement that is present during one roller revolution at 90 m/min. The displacements reflect the roller working ring's irregularities and roughness. It is shown that the old roller not only has a larger low frequency displacement, but also more uneven surface, compared to the renewed roller.

The calculated displacement frequency spectrum (Fig. 31) also reveals the measurable difference between rollers.

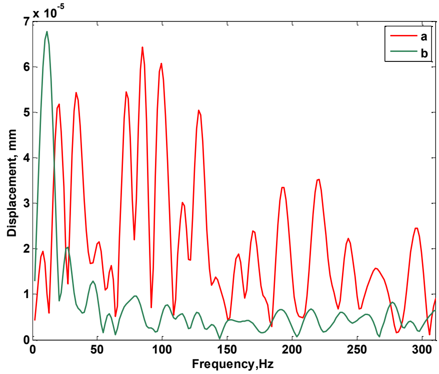


Fig. 31. The spectrum of working ring displacement. a) not renewed roller b) renewed roller.

The old roller's spectrum has several noticeable frequency components. They represent the working rings roughness. Meanwhile, the renewed roller's spectrum has only rotating frequency spectrum component.

The comparison of all the rollers rings is presented in Fig. 32. The average old roller's ring displacement is 19.9 µm, deviation – 7.3 µm. The average displacement of renewed rings is 9 µm, deviation – 2.5 µm.

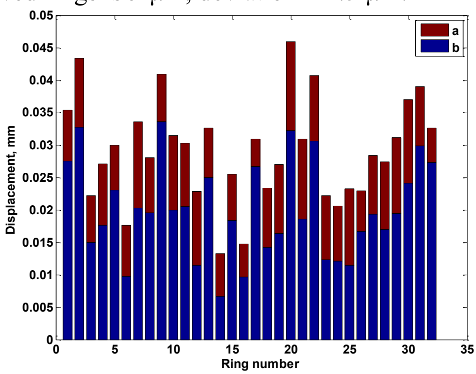


Fig. 32. Displacement magnitudes of the segments of rollers: a – before renewal, b – after renewal

After renewal all the segments have a lower displacement.

4.2. Displacement measurement error analysis

The uncertainty analysis (method "A") was performed to ensure the accuracy of the displacement measurement. The set of measurements was obtained during the measurement of a known thickness paper strip. The paper strip was glued to the surface of the rotor (Fig. 33).

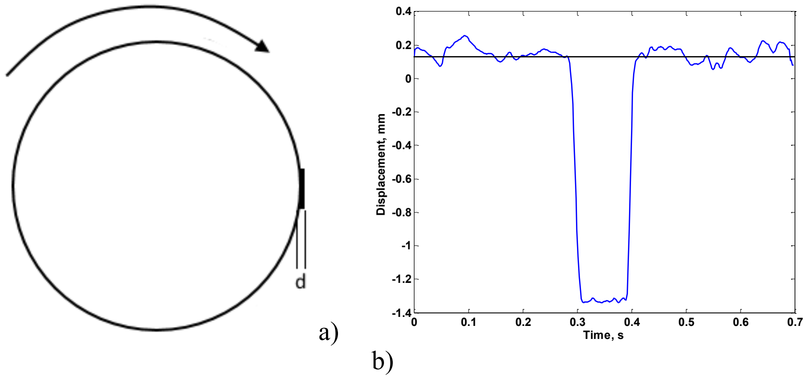


Fig. 33. The investigation of accuracy of displacement measurement: a) the rotating rotor with known width paper strip; b) Obtained signal after processing.

The measurement uncertainty U_k was calculated:

$$U_k = 2\sqrt{(S_x)^2 + \left(\frac{\Delta d}{\sqrt{3}}\right)^2} = 2\sqrt{0.5231^2 + \left(\frac{3}{\sqrt{3}}\right)^2} = \pm 3.6186\ \mu\text{m},$$

where S_x – standard deviation of set of measurements, Δd – paper strip width error.

The calculated uncertainty is enough to provide accurate working ring displacement errors.

4.3. Conclusions

The suggested measurement system was tested on the paper folding machine working elements. The results showed that the accuracy of the suggested system is enough to detect defective rollers.

The results of the measurements showed that the rollers before and after renewal can be distinguished not only by low frequency displacement, but also by surface roughness.

The standard displacement deviation of the ring's displacement is 3 times larger for not renewed rollers. It suggests that it may lead to an uneven pull force between rollers, that may damage the paper product.

5. CONCLUSIONS

1. Measurement systems, including those based on wireless technology, were developed for the diagnostics of specific rotary systems. These systems implement the proposed data processing algorithms and ensure that measured values (acceleration, pressure, displacement, force and rotation speed) have met the metrological requirements.
2. The results of experiments performed on complex balance system revealed that at constant conditions (speed of rotation, constant external air flow) mechanical imbalances can compensate for aerodynamic imbalances and thereby reduce system vibrations.
3. Acquiring blade profiles by scanning their surface and measurement of the generated pressure, allows blade geometry variations to be captured, which causes aerodynamic imbalance.
4. Autogyro experiments performed on the ground and during flight revealed that during flight, as opposed to on the ground, vibrations in the cabin dominate with a frequency that is equal to double the rotation frequency vibrations. Double rotations frequency vibrations are several times larger in magnitude than rotation frequency vibrations.
5. The performed experiments revealed that both autogyro's and fan's vibration components of the double rotations frequency reflect the blade's aerodynamic properties.
6. The methods and instruments of contactless displacement measurements were suggested and validated. They allow the assessment of the paper folding machine's segment working ring quality with sufficient uncertainty.

LIST OF SCIENTIFIC PUBLICATIONS ON THE TOPIC OF DISSERTATION

In international journals, which are included in Thomson Reuters List

1. Augutis, Stasys Vygantas; Vainilavičius, Darius; Malcius, Marius. Dynamic compensation of rotating blades imbalance // Journal of Vibroengineering / Vibromechanika, Lithuanian Academy of Sciences, Kaunas University of Technology, Vilnius Gediminas Technical University. Kaunas: Vibroengineering. ISSN 1392-8716. 2015, vol. 17, iss. 2, p. 949-956
2. Kibirkštis, Edmundas; Augutis, Stasys Vygantas; Vainilavičius, Darius; Miliūnas, Valdas; Pauliukaitis, Darius; Ragulskis, Liutauras. Effect of dynamic regime of rollers of pocket folding machine to quality of printing products // Journal of Vibroengineering / Vibromechanika, Lithuanian Academy of Sciences, Kaunas University of Technology, Vilnius Gediminas Technical University. Kaunas: Vibroengineering. ISSN 1392-8716. 2015, vol. 17, iss. 6, p. 2869-2880

In list of international databases publications approved by the Research Council of Lithuania

1. Ragulskis, Kazimieras; Ragulskis, Liutauras; Kibirkštis, Edmundas; Augutis, Stasys Vygantas; Vainilavičius, Darius; Miliūnas, Valdas; Pauliukaitis, Darius. Measurement of vibrations of rotating elements in a folding machine // Journal of measurements in engineering / Lithuanian Academy of Sciences, Kaunas University of Technology, Vilnius Gediminas Technical University. Kaunas: Vibroengineering. ISSN 2335-2124. 2015, Vol. 3, Iss. 1, p. 9-16

In proceedings of scientific conferences

1. Vainilavičius, Darius; Augutis, Stasys Vygantas; Malcius, Marius; Bezaras, Aurimas. Analysis of autogyro rotor balancing and vibration // Metrology for Aerospace (MetroAeroSpace), 4-5 June 2015, IEEE. Piscataway: IEEE, 2015. p. 416-420. DOI: 10.1109/MetroAeroSpace.2015.7180693.

2. Vainilavičius, Darius; Augutis, Stasys Vygantas; Malcius, Marius; Bezaras, Aurimas. Measurement system for analysis of autogyro rotor imbalance // Vibroengineering procedia : international conference Vibroengineering - 2014, Katowice, Poland, 13-15 October, 2014. Kaunas: JVE International. ISSN 2345-0533. 2014, vol. 3, p. 5-8.

Reports in science conferences

1. D. Vainilavičius, V. Augutis, M. Malcius, A. Bezaras. Research of autogyro blades balance. International conference Vibroengineering 2012. (Kaunas, Lithuanian, October,2012).

References

1. MUSZYNSKA, A. *Rotordynamics.* Boca Raton: Taylor & Francis Group, 2005 ISBN 0-8247-2399-6.

2. *Autogyro MT-03 Gyroplane Maintenance Manual.* [viewed 2015-12-28]. Available from:http://www.gyrokopter.se/files/101107/mt-03maintenancemanual.pdf.

3. KIBIRKŠTIS, E., et al. Effect of Dynamic Regime of Rollers of Pocket Folding Machine to Quality of Printing Products. *Journal of Vibroengineering*, 2015, vol. 17, No. 6. pp. 2869-2880 ISSN 1392-8716.

Information about the author of the dissertation

Darius Vainilavičius was born on 11th September, 1986 in Kaunas, Lithuanian.

2005: graduated from Kaunas Salomėja Neris high school.

2005-2009: studies at Kaunas University of Technology, Faculty of Telecommunications and Electronics; Bachelor of Science in Electronics engineering.

2009-2011: studies at Kaunas University of Technology, Faculty of Telecommunications and Electronics; Master of Science in Measurement engineering.

2011-2016: Doctoral studies in a field of Measurement engineering at Kaunas University of Technology.

For contacts: darius.vainilavicius@gmail.com

REZIUMĖ

Darbo aktualumas

Rotorius – variklio ar mašinos sukamoji dalis, įtvirtinta ir sukasi apie nejudamąją ašį. Rotorinių sistemų dažnai pasitaiko pramonėje. Jos yra įvairių tipų ir paskirčių, pradedant sistemomis be aerodinaminių elementų – tai elektros varikliai, generatoriai, velenų sistemos, baigiant sistemomis su aerodinaminiais elementais – tai kompresoriai, vėjo jėgainės, pramoniniai ventiliatoriai, turbinos, bei hidrodinaminėmis sistemomis, tokiomis kaip laivų sraigtai, skysčių ir dujų pompos.

Rotorinės sistemos dažnos ir aviacijoje. Šios sistemos čia yra orlaivių varomoji jėga tiek žemėje, tiek ore. Tokių sistemų darbo patikimumo rodikliai tiesiogiai siejasi su keleivių saugumu.

Rotorius suteiktą energiją konvertuoja į naudingą darbą. Tačiau dalis energijos virsta virpesiais, kuriuos sukelia nepageidaujamas centrinės rotoriaus ašies judėjimas. Rotoriaus sukelti virpesiai perduodami ir kitiems mašinos komponentams ir struktūroms. Taip sukeliami pažeidimai ir sumažėja pažeidžiamų elementų darbo resursai. Virpesiai, nepaisant to, ar jų lygis viršija leidžiamąjį, per tam tikrą laiką gali sukelti įvairių pasekmių:

- nuovargį, kuris per laiką sukelia pažeidimus;
- trintį, kuri sukelia nusidėvėjimą, o šis – pažeidimus arba papildomą triukšmą dėl padidėjusių atstumų tarp mašinos elementų;
- sumažėjusį efektyvumą;
- defektuotų gaminių gamybą.

Rotorinių sistemų diagnostika – svarbi diagnostikos šaka, kurios uždaviniai yra šie: rotorinių sistemų tiesioginių ir netiesioginių darbo parametrų stebėjimas, kaupimas, apdorojimas, analizė, siekiant ne tik fiksuoti sistemos darbo parametrų nuokrypius, bet ir juos prognozuoti.

Pagrindinis diagnostikos tikslas – aptikti ankstyvus rotorinės sistemos darbo parametrų nuokrypius ir įvardyti, kokiu būdu jie gali būti pašalinami. Pagrindinė mašinos veikimo sutrikimo priežastis gali sukelti antrinius pažeidimus. Pavyzdžiui, rotoriaus masės disbalansas sukelia greitesnį guolių dėvėjimąsi, todėl būtina identifikuoti ir pašalinti pagrindinę sutrikimo priežastį, nes kitaip sutrikimas pasikartos. Nuokrypiai turi būti aptinkami kuo anksčiau, nurodant pagrindinę blogo veikimo priežastį.

Diagnostika aktuali ir precizinėms speficinėms rotorinėms sistemoms. Preciziškumas šiose sistemose pasireiškia tuo, jog dėl specifinės konstrukcijos itin maži darbo parametrų pakeitimai gali sukelti pastebimus virpesių pokyčius. Tarkime, šiame darbe tirto malūnsparnio rotoriaus balansavimo metu menčių atakos kampas buvo keistas tik 0,044° kampo, tačiau tam tikro tipo virpesiai sumažėjo 44 %. Dėl specifinės tokių rotorinių sistemų konstrukcijos ir darbo režimo ypatumo tokių sistemų diagnostika reikalauja unikalių matavimo sprendimų. Šiame darbe nagrinėjamos trys rotorinės sistemos. Šių sistemų diagnostiniams parametrams matuoti pasiūlytos specifinės matavimo sistemos.

Darbo tikslas ir uždaviniai

Darbo tikslas – sukurti matavimo metodus ir priemones specifinių rotorinių sistemų diagnostiniams parametrams matuoti, įvertinti jų metrologines charakteristikas ir pritaikyti jas praktikoje.

Darbo uždaviniai:

- atlikti užsienyje ir Lietuvoje vykdytų susijusių tyrimų apžvalgą;
- sukurti matavimo sistemas, atitinkančias diagnostikos objektus;
- atlikti sukurtų matavimo sistemų metrologinių charakteristikų tyrimus;
- išmatuoti pasirinktų specifinių rotorinių sistemų diagnostikos parametrus;
- pasiūlyti naujus matavimo rezultatų apdorojimo algoritmus.

Mokslinis naujumas

- Pasiūlytos matavimo sistemų struktūros specifinių rotorinių sistemų diagnostiniams parametrams matuoti ir atlikta jų metrologinių charakteristikų analizė.
- Pasiūlyti diagnostinių signalų apdorojimų algoritmai.
- Parodytos kompleksinio menčių balansavimo galimybės.
- Ištirta malūnsparnio menčių aerodinamikos įtaka antrai vibracijų harmonikai.

Praktinė nauda

Įgyvendinant pasiūlytas struktūras ir signalų apdorojimo algoritmus, buvo sukurtos specifinės matavimo sistemos. Šios sistemos buvo naudojamos specifiniams objektams su rotorinėmis sistemomis tirti. Gauti tyrimų rezultatai pateikiami šiame darbe. Sukurta bevielė malūnsparnių rotoriaus disbalanso parametrų matavimo sistema naudojama UAB „Birdmanas“ malūnsparnių parke.

Darbo aprobavimas

Pagrindiniai darbo rezultatai aprobuoti penkiomis publikacijomis: dviem publikacijomis Mokslinės informacijos instituto (IS) patvirtintuose sąrašo leidiniuose, viena publikacija – Lietuvos mokslo tarybos patvirtinto sąrašo tarptautinėse duomenų bazėse referuojamuose leidiniuose ir dviem straipsniais – konferencijų straipsnių rinkiniuose. Darbo rezultatai pristatyti trijose tarptautinėse konferencijose Lietuvoje, Lenkijoje, Italijoje.

Gynimui pateikiama

1. Pasiūlytos matavimo sistemų struktūros ir duomenų apdorojimo algoritmai.
2. Su sukurtomis matavimo sistemomis gauti tyrimų rezultatai ir jų išvados.

Disertacijos struktūra bei apimtis

Disertaciją sudaro įvadas, trys skyriai, bendrosios išvados, literatūros sąrašas, autoriaus publikacijų disertacijos tema sąrašas. Darbo apimtis – 92 puslapiai, kuriuose pateikiami 64 paveikslai, 26 lentelės, 24 matematinės išraiškos ir 89 pozicijų literatūros šaltinių sąrašas.

Bendrosios išvados

1. Sukurtos matavimo sistemos, taip pat ir grindžiamos bevielėmis technologijomis, skirtos specifinių rotorinių sistemų diagnostikai, naudoja pasiūlytus duomenų apdorojimo algoritmus ir užtikrina matuojamų dydžių – pagreičio, poslinkio, slėgio, jėgos bei sukimosi dažnio – matavimo reikiamus metrologinius parametrus.
2. Eksperimentai su kompleksinio balansavimo sistema parodė, kad pastoviomis sąlygomis (sukimosi greitis, pastovus išorinis oro srautas) mechaninis disbalansas gali kompensuoti aerodinaminį disbalansą, taip sumažindamas sistemos virpesius.
3. Dirbančių menčių skenavimas, matuojant jų profilį ir sukeliamą dinaminį slėgį, leidžia fiksuoti geometrinius menčių pakitimus, kurie sukelia aerodinaminį disbalansą.
4. Atlikti malūnsparnio vibracijų matavimai skrendant ir ant žemės parodė, kad skrendant, skirtingai negu ant žemės, malūnsparnio kabinoje vyrauja virpesiai, kurių dažnis sutampa su dvigubu sukimosi dažniu. Dvigubo dažnio virpesių lygis yra šešis kartus didesnis už sukimosi dažnio virpesius. Šių virpesių lygis rotoriui balansuojant yra nesusijęs su paties rotoriaus virpesiais.

5. Eksperimentai parodė, kad tiek malūnsparnio, tiek ventiliatoriaus rotoriaus virpesių dvigubo sukimosi dažnio dedamoji rodo menčių aerodinamines savybes.
   Pasiūlyta ir aprobuota bekontakčio poslinkio matavimo metodika ir įranga leidžia su pakankama neapibrėžtimi įvertinti popieriaus lankstymo mašinos velenų segmentų darbinių žiedų kokybę.

Trumpa informacija apie autorių

Darius Vainilavičius gimė 1986 m. rugsėjo 11 d., Kaune.

1993-2005 metais mokėsi Kauno Salomėjos Neries vidurinėje mokykloje (nuo 2010 m. Kauno Gedimino sporto ir sveikatinimo gimnazija).

2005-2009 metais studijavo Kauno technologijos universiteto Telekomunikacijų ir elektronikos fakultete ir įgijo elektronikos inžinerijos bakalauro kvalifikacinį laipsnį.

2009-2011 metais studijavo Kauno technologijos universiteto Telekomunikacijų ir elektronikos fakultete ir įgijo matavimo inžinerijos magistro kvalifikacinį laipsnį.

2011-2016 metais studijavo Kauno technologijos universiteto Elektros ir elektronikos fakultete matavimo inžinerijos mokslo krypties doktorantūroje.

El. pašto adresas: darius.vainilavicius@gmail.com

UDK 531.7 + 534-6 + 62-187.4] (043.3)

SL344. 2016-09-05, 2,5 leidyb. apsk. l. Tiražas 50 egz. Užsakymas 315.
Išleido Kauno technologijos universitetas, K. Donelaičio g. 73, 44249 Kaunas
Spausdino leidyklos „Technologija“ spaustuvė, Studentų g. 54, 51424 Kaunas