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Investigations of Sound Induced by Grass-Cutting Blades

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Abstract: Sound induced by a pair of counter-rotating grass-cutting blades in a cutting deck is studied in this study. Experimental sound pressure results are obtained using a sound pressure transducer and a data acquisition system. To explain the extraordinarily high sound pressure levels at several frequencies, a 3-dimensional finite element model for the cutting blades is developed to determine the dynamic natural frequencies including the effect of spinning. It was found that, during operations, the blades experienced several modes of resonant vibrations in the interested frequency range, which contributes to the very high noise levels at several frequencies. To reduce vibration of the cutting blades due to multi-frequency oscillatory dynamic pressure field, the effect of application of viscoelastic foil is investigated.

Key words: Fast Fourier transform, grass cutting blade, resonance, rotation, sound, vibration

INTRODUCTION

In the lawn care industry, the multifunctional cutting blades are designed to cut grass, draw air into the cutting chamber and propel the air and grass clipping mixture into a collector. During operations, the cutting blades rotate at desired speeds for continuous grass-cutting and suction of air into the cutting system from the circumferential gap just below the cutting pan. The cutting blade tip is so designed that it also generates a strong upward airflow for lifting and transporting the air-grass clipping mixtures towards the pan-tunnel juncture. Oscillating airflows, blade vibrations and rotating unbalance associated with rotations of blades inside a cutting chamber are important sources of noise. A multi-functional double-blade deck, designed by a Canadian lawn care equipment manufacturer is shown in Fig. 1. To generate sufficient airflow and produce balanced aerodynamic lift forces from a grass-cutting blade. The blade is designed to have 2 identical cutting edges and aerodynamic profiles at both ends sketched in Fig. 2.

The multi-spindle rotary mowing equipment is primarily used in commercial sectors specializing in maintenance of road sides, sport fields and golf courses. These professional lawn mowers are typically loud and large. The cutting system is the main source of noise emission, where the noise is generated by the internal combustion (IC) engine, rotary blades and power transmission. Noise reduction may be achieved by replacing the IC engine with an electric motor. This

Fig. 1: Underside view of the cutting pan with a pair of counter-rotating blades

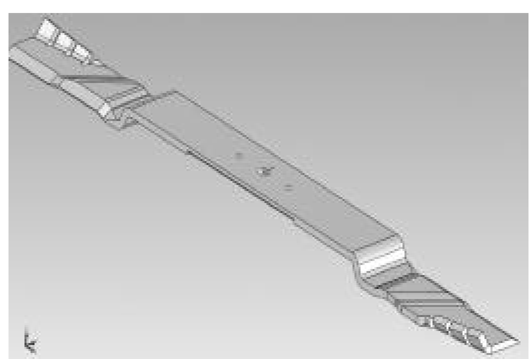

Fig. 2: The solid model for a serrated grass-cutting blade

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approach works only for single blade lawnmowers designed for use in the vicinity of an electrical outlet. To cut grass in a large area, multi-spindle rotary mowers are usually powered by IC engines. A practical approach in noise reduction lies with in the design of blades. This is difficult to achieve because most of the lawnmower noise depends on the rotational speed and aerodynamics of the blade, which in turn control the performance of the lawnmower (quality of cut, generation of airflow and energy consumption).

To market the lawn care equipment in the European Union, the lawncare equipment manufacturers must ensure that their products comply with the European Noise Directive (2000/14/EC) and "EU 84/538/EEC (1984), which limits the allowable noise level. Noise level limits are based on cutting width. The Directive requires that manufacturers meet these limits and include statistical uncertainties, then declare their noise levels and have these results certified by an independent third party agency. There are two major challenges with this Directive (Drutowski and Fetzer, 2005). First by taking statistical uncertainties into account, this creates a limit lower than the published limit. Second is to lower the sound power level while still maintaining acceptable performance. For rotary lawnmowers, a dominant noise source is the cutting deck assembly, which is required to lift and cut grass, then dispose of the clippings. To guarantee a good quality of cut appearance, especially in golf courses and sports fields, the blades are required to turn at a very high tip speed of about 100 m s^{-1} to generate the necessary airflow. For lawnmowers, noise levels and unit performance are intrinsically connected. Consequently, very significant reduction of noise in large lawnmowers is impractical. However, due to rising concerns of noise pollution, many countries defined noise limits on lawnmowers.

Although, 3 dimensional oscillatory flows in a single-, double- and triple-blade lawnmower deck were studied experimental and numerically in references (Abbasian *et al*., 2007; Chon and Amano, 2004; Hagen *et al*., 2002), no systematic studies of sound produced by the lawnmower are available in the literature. This study presents some recent work in this area.

MEASUREMENTS OF SOUND PRESSURE LEVELS

The experimental setup is pictured and illustrated in Fig. 2. The entire setup consists of a 10 HP AC motor, 2 belt drives, a cutting deck and a discharge tunnel, a microphone, a data acquisition system and a personal computer. Through the main belt drive, the motion from the motor is transferred to the first cutting blade axle. The secondary cross-over belt drive is used to transfer motion from the first blade axle to the second blade axle.

Table 1: Measurements of sound pressure level from a 1/3 Octave dBA meter

Band centre frequency (Hz)	Sound pressure level (dBA)	Band centre frequency (Hz)	Sound pressure level (dBA)
10	-9.4	500	98.3
12.5	-0.8	630	88.8
16	5.9	800	90.7
20	15.9	1000	91.4
25	20.1	1250	94.4
31.5	29.3	1600	94.6
40	37.1	2000	93.3
50	35.9	2500	91.4
63	42.7	3150	89.5
80	73.5	4000	86.5
100	70.5	5000	84.0
125	63.0	6300	81.6
160	86.7	8000	77.8
200	84.3	10000	73.5
250	79.8	12500	69.7
315	92.0	16000	65.3
400	101.8	20000	58.9

The motor runs a nominal speed of 3535 rpm and transfers the electrical power to the blade axles through a pair of belt drives. As the blade rotates at 2640 rpm or 44 Hz, it generates two high pressure regions in the vicinity of the blade tips, which travel at a speed of approximately 100 m s^{-1}. When resolving the traveling aerodynamic forces in the radial, tangential and axial directions, the oscillatory axial components of the aerodynamic force or the lift force tend to excite the out-of-plane lateral vibrations in the blade. Simulation results given in Abbasian *et al*. (2007) show that the lift force has a large steady component and oscillatory harmonic compositions. The oscillatory lift force has a dominant frequency of 2f, where f is the blade rotational speed. However, harmonic compositions at multiples of 2f, i.e., 4f, 6f, etc., are also significant.

To quantify the levels of noise generated by the cutting system, an A-scale, 1/3 octave sound pressure level (SPL) meter is used. The A-weighting network is almost exclusively used in dealing with human response to noise from the considerations of both hearing damage and annoyance (Irwin and Graf, 1979). The meter was held two meters away from the left side of the cutting pan and one meter above the ground. A 30 sec test was performed during steady-state running condition of the cutting system. Based on the measurements Table 1, the SPL is very low for band center frequencies below 125 Hz. The highest SPL (101.8 dBA) occurs at the centre frequency of 400 Hz; the second highest SPL (98 dBA) occurs at the centre frequency of 500 Hz. For frequencies above 6300 Hz, the noise levels are low again. From the aerodynamic studies and measurements for a cutting deck consisting of 2 counter rotating blades, the following excitation frequencies are known to exist: 88 Hz, 2×88 Hz, 3×88 Hz, etc. From the measurement, the highest sound pressure level occurs at the center frequency of 400 Hz. The 400 Hz band permits passage of sound of frequencies between 356 Hz and 449 Hz. The aerodynamic pressure pulsation at 440 Hz or 10f (the 10^{th} order frequency when normalized to the blade rotational speed) is the cause of the very high sound level. Similarly the second highest SPL at 500 Hz band centre frequency is associated with the aerodynamic excitation frequency of 528 Hz or 12f (the 12^{th} order frequency).

MEASUREMENTS OF SOUND PRESSURE BY MEANS OF MICROPHONE

To measure the instantaneous sound pressure generated from the cutting system, a pre-polarized condenser microphone (B and K 4188) is used. According to the manufacturer's specifications, the uncertainty of the microphone with 95% confidence level is 0.35%. The analog sound signal is sampled and digitized using a data acquisition system. The time domain data are analyzed using the Visual Designer software or written into Excel files for further processing.

The main source of sound for the experimental setup comes from the operation of the cutting system, namely, the oscillatory airflows induced by the two counter-rotating grass-cutting blades, structure-borne vibrations of components induced by the oscillatory flows and rotational unbalance, the motor and the belt drives (Fig. 3). The identifiable background noise in the laboratory is from a power supply and a computer fan, which is extremely low in magnitudes and very different in frequencies from the sound emitted from the cutting system. When investigating the blade-induced sound, the laboratory background noise is negligible.

Calibration is required to relate the voltage output from the microphone-based data acquisition system to instant sound pressure. For this purpose, a sound level calibrator (B and K 4230) is used to produce a 1000 Hz sinusoidal sound at 94 dB. The following well-known equation defines the relationship between the sound pressure level (SPL) in decibels (dB) and the root mean square (RMS) value of sound pressure SPL dB = $20 \log_{10} p\backslash p^0$, where p is the RMS value of sound pressure; p_0 is the standard reference sound pressure, taken to be 20 μPa. For a sinusoidal sound of a known SPL in dB, the RMS value of the sound pressure may be determined from $p = p^0\, 10^{\text{SPL dB}\backslash 20}$. for SPL dB = 94, the RMS value of

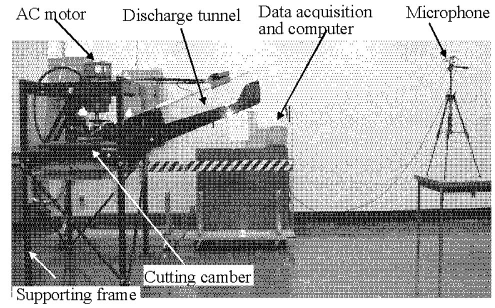

Fig. 3: The experimental setup

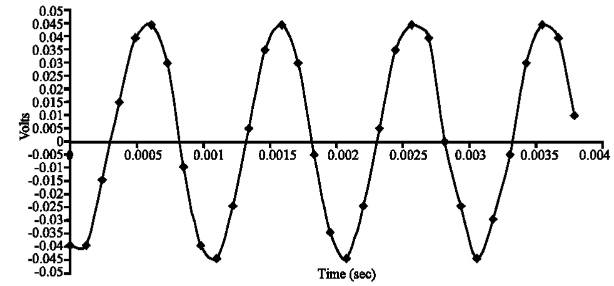

Fig. 4: The electrical signal measured from the sound level calibrator in time domain

pressure of the corresponding sinusoidal sound is 1.00237 Pa. The peak pressure of the same sound is 1.41757 Pa. The calibration sound signal was recorded using a pre-set amplification factor and a sample rate of 8192 Hz. Use of 8192 Hz or 2^{13} Hz is for the convenience in the subsequent fast Fourier transform (FFT) of actual sound measurements. This setting is kept for use for all experiments carried out in this study. A portion of the electrical signal in time domain is shown Fig. 4. From the direct measurement, the peak voltage is 0.044 Volt. Therefore, the microphone-based data acquisition system has a sensitivity of 32.2175 Pa/Volt.

To capture the sound directly emitted from the cutting deck, the microphone was placed at 3.35 m away from the center of the cutting pan and 1.70 m from the ground. The microphone was also slanted to an angle so that the effect of airflow is eliminated. The analog sound signal was recorded using the calibrated microphone and data acquisition system. To discern accurately sounds in the 1000 Hz rang, the analog sound signal was sampled at a frequency 8192 Hz. The recorded sound pressure in time domain is shown in Fig. 5a. To determine the frequency composition, an FFT was conducted. The results are shown in Fig. 5b. It can be seen clearly that in the 1000 Hz range, the sound pressure has a discrete spectrum. The discrete frequencies are 88 Hz and multiples of 88 Hz.

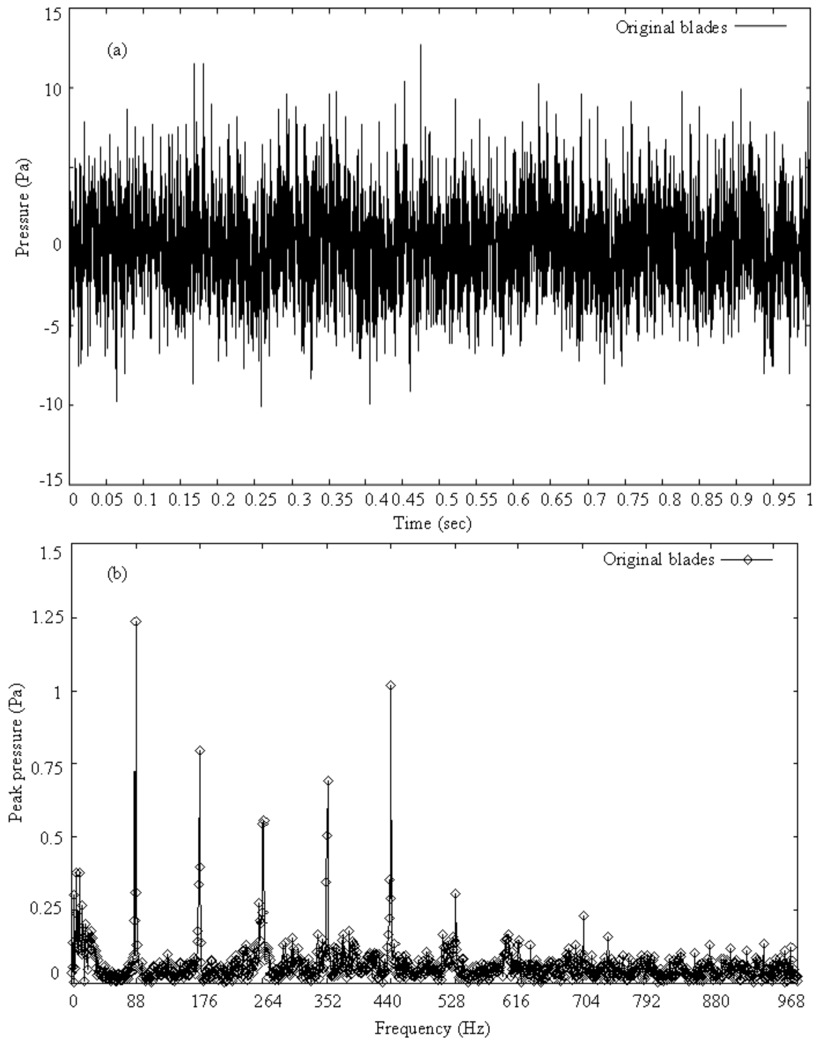

Fig. 5: Measured pressure of sound from the cutting deck (a) time domain, (b) frequency domain

These frequencies are associated with the oscillatory airflows generated by the 2 counter-rotating blades. It is interesting to notice that the measured sound pressure has significant amplitude at 440 Hz, which is comparable with that at 88 Hz. This is consistent with the SPL of 101.8 dBA @ 400 Hz, recorded using a 1/3 octave sound meter. However, from the measurements of the oscillatory pressure taken for the same setup in the air intake region (Abbasian *et al*., 2007), the magnitude of the oscillatory pressure at 440 Hz is less than 10% of its magnitude at 88 Hz. The high noise level and the anticipated weak pressure fluctuation at 440 Hz, seems to indicate that the presence of resonance of some component in the cutting system. An examination of all components in the cutting system quickly points quickly to the 'flexible' cutting blades. This prompted the undertaking of studies of sound and vibration associated with the rotation and vibration of the grass-cutting blades.

FREE VIBRATION ANALYSIS OF BLADES CONSIDERING THE ROTATIONAL EFFECTS

In a way similar to rotation of a beam (Yoo and Shih, 1998), the blade rotation produces a distributed centrifugal force in the radial direction. This centrifugal force field induces a tensile radial stress, which plays a stiffening role in the blade flexural bending motion. The stiffening effect is usually significant for the lower vibration modes. Since, it is impossible to measure the actual dynamical natural frequencies of the rotating blades, we decided to obtain the dynamic natural frequencies of the rotating blades by means of a finite element analysis. The finite element model is validated against the experimental data for the at-rest natural frequencies.

To experimentally determine the at-rest natural frequencies of a blade made of steel, an impact hammer, an accelerometer and a data acquisition system are used. A

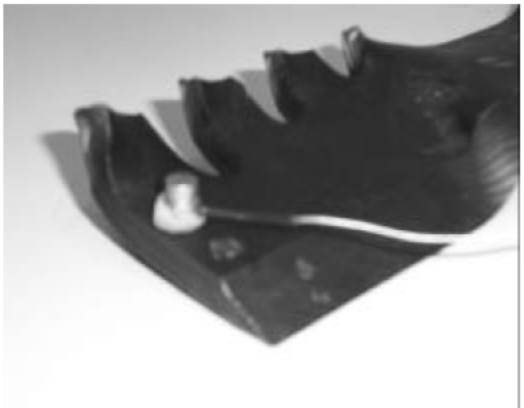

Fig. 6: Accelerometer mounted on serrated blade

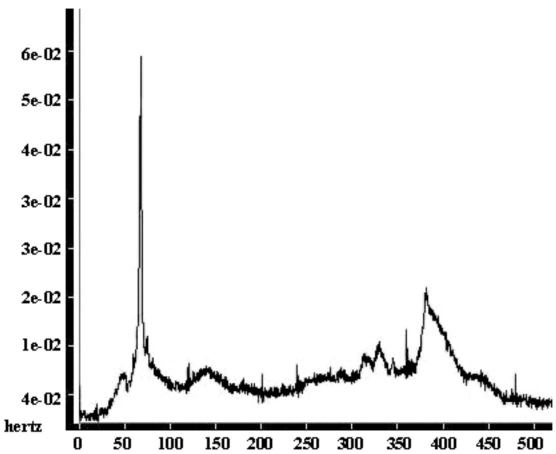

Fig. 7: Fourier transform of measured blade tip free vibration

Fig. 8: The finite element mesh of a serrated cutting blade

blade is struck with the impact hammer on the side where the accelerometer is mounted, its motion is recorded (Fig. 6). The fast Fourier transform (FFT) of the blade tip response is plotted in Fig. 7. There are 2 natural frequencies at 68 Hz (first mode) and 380 Hz (second mode), respectively. Because of the rotational symmetry of the blade, the other half exhibits the same response

Fig. 9: Blade mounted to the bottom plate with a clear gap between them

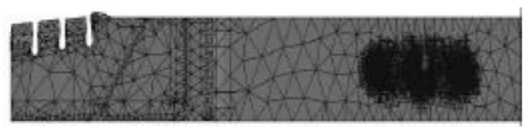

Fig. 10: Constraints of degrees of freedom on finite element nodes in the area of tight contact

characteristics. Each of the two cutting blade has a pair of repeated natural frequencies at 68 and 380 Hz, respectively.

A finite element model for free vibration analysis of a cutting blade was created using ANSYS, a commercially available general purpose finite element analysis package. To capture the complex geometry of the blade, the SOLID92 finite element (a 3-D and10-node tetrahedral structural solid element) was used. The finite element model, shown in Fig. 8, has 9, 068 elements and 18,747 nodes. The material properties used for the steel blade are Young's modulus 207 GPa, Poisson's ratio 0.3 and density 7700 kg m^{-3}.

Fastening of the cutting blades onto the lawnmower is done through a clamping system where the blade is set between a round shaft with a T-shaped flange and a bottom plate shown in Fig. 9. The blade is securely fit between the two pieces of steel plates by a threaded bolt in the middle. The plate is flush with the blade and no gaps are present. The dimensions of the T-shaped flange are 4.5 cm by 9.8 cm. In the actual setup, the bottom plate does not make full contact with the blade. To model the boundary conditions correctly, nodal displacements are constrained only for those nodes located in the region of the T-shaped clamp on the top and bottom surfaces of the blade as shown in Fig. 10.

To compute the at-rest natural frequencies, the spin rate is set to zero in the computer model. To compute the dynamic natural frequencies, a static stress analysis is first performed to determine the initial stress field induced by the blade rotation at 44 Hz. The subsequent modal analysis is then performed by including stress-stiffening effects. The first ten at-rest and dynamic natural frequencies are given in Table 2.

An examination of numerical and experimental data in Table 2 indicates that there is excellent agreement in the at-rest natural frequencies for the first 4 modes between the experimental data and the computational results. The maximum error is about 2%. It is therefore, concluded

Table 2: Comparison of Static and Dynamic Frequencies

Modes	At-rest natural frequencies (Hz) Experimental	Computed	Dynamic natural frequencies (Hz) Computed	Theoretical stiffening effect (%)
1	68	69.76	86.27	23.7
2	68	70.38	86.86	23.4
3	380	379.44	380.37	0.2
4	380	380.03	380.97	0.2
5	-	396.83	412.52	4.0
6	-	399.77	415.44	3.9
7	-	666.38	669.77	0.5
8	-	666.46	669.87	0.5
9	-	1023.7	1038.0	1.4
10	-	1030.7	1045.0	1.4

that the computational models give reliable prediction of at-rest natural frequencies. The first at-rest natural frequencies and their corresponding mode shapes are shown in Fig. 11. Among the first 10 modes, 8 of them are associated with the out-of-plane bending deformations; and two of them are associated with torsional deformations.

After the computer model is validated by comparing the results of the at-rest natural frequencies of the blade, the dynamic frequencies given in Table 2 are considered credible and used in the subsequent discussions. According to the simulation results, the increase in natural frequencies due to the rotational effect can be as high as 24% for the first 2 vibration frequencies.

To determine the dynamic (resonant or non-resonant) state of the blades for multiple natural frequencies and multiple excitation frequencies, a resonance identification chart is prepared and plotted in Fig. 12. On the resonance line, an excitation frequency coincides with a natural frequency. To account for the discrepancies between computed dynamic natural frequencies and the actual dynamic natural frequencies and the deviation of side band frequencies due to damping, a ±5% uncertainty is assumed. From the resonance chart, at 88 and 1056 Hz, the blade is nominally at the state of resonance; at 352 Hz, 440 and 704 Hz, the

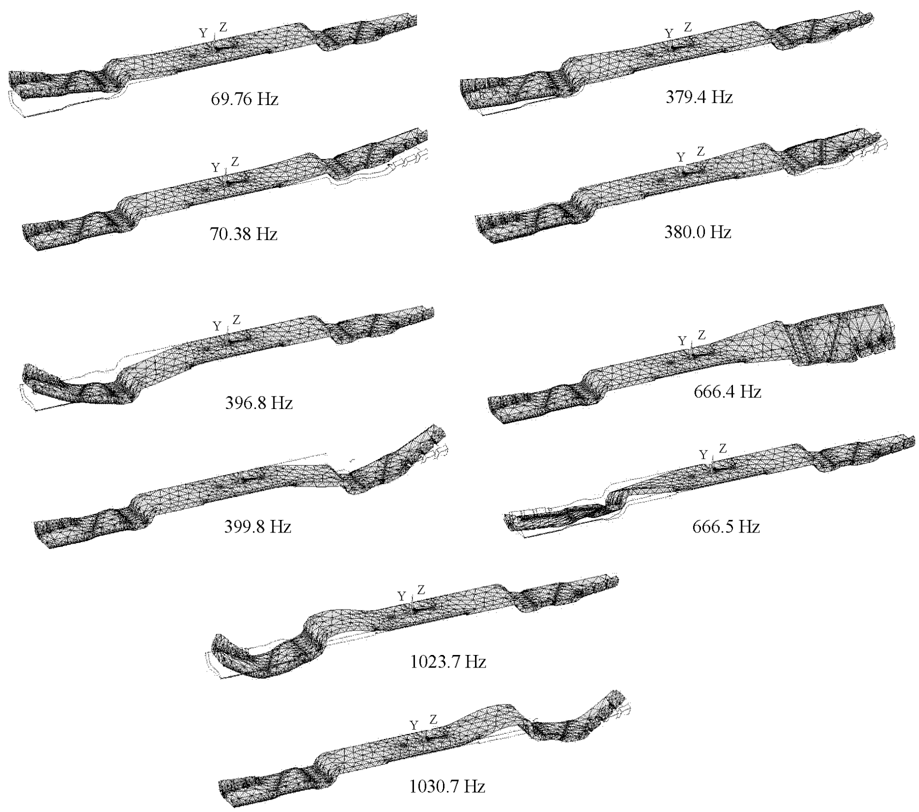

Fig. 11: The first ten at-rest natural frequencies and mode shapes of the cutting blade

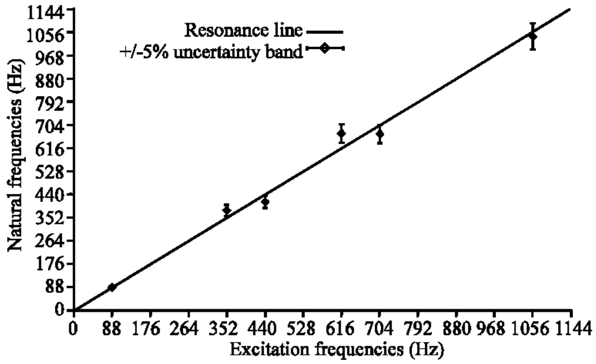

Fig. 12: Resonance identification chart for multiple excitation and natural frequencies

blade is very close to resonances. It can therefore, be concluded that the cutting blades experience several modes of resonant vibrations during operations. This explains why the sound pressure level reaches 101.8 dBA in the 400 Hz centre frequency band while the aerodynamic excitation magnitudes are actually very small.

BLADE DESIGN AND SOUND REDUCTION MEASURES

The cutting blades under investigations were designed to rotate at 2640 rpm or 44 Hz for good quality of cut, generate adequate airflows for normal and aggressive cutting, compact grass-clippings effectively with the fins, have the adequate strength for moderate impact with foreign objects on its path and have the fundamental natural frequency above the operating speed. Any modification to the material and geometry of the blade must be certified and tested for the above five main criteria.

Noise was not a design criterion in the lawncare industry until 2000 when the European Union Noise Directive became effective. Field tests conducted on a 54″ wide professional lawn mower by the authors confirmed that the main source of noise comes from the cutting deck, which produces an oscillatory aerodynamic pressure field inside the cutting chamber. The sound pressure measurements in Fig. 5 indicate that this oscillatory aerodynamic pressure is the only significant contributor to the very high noise level in the frequency range of 20 and about 1000 Hz. Unfortunately, the blade has ten vibration modes in this frequency range; several natural frequencies are identical or very close to the aerodynamic excitation frequencies.

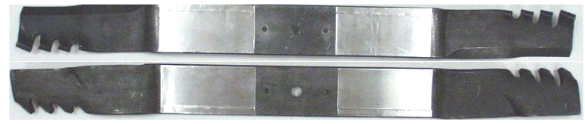

Fig. 13: Blades with viscoelastic foil applied to the top and bottom surfaces

To reduce noise, the blade should not have any natural frequencies at all below 1000 Hz, or have natural frequencies away from the excitation frequencies. The first approach is very difficult or impossible to achieve. The second approach is not easy to accomplish either because the blade is a very sophisticated structure having multiple functionalities-grass-cutting, grass clippings compaction, grass clippings lifting and air sucking and blowing.

In this study, the authors attempted to use a passive vibration isolation approach by applying a layer of viscoelastic foil on the top and bottom surfaces of the blade at locations away from the blade tips. This practice does not affect the performance of the blade in any conceivable way. The objectives are to test whether the viscoelastic foil can be used as an effective vibration energy absorber to achieve the goal of noise reduction. In a commercial application, the viscoelastic foil may not be a good solution due to its low resistance to wear and erosion.

As a test, four 9 cm by 6 cm strips of the 3M™ 2552 damping foil were placed onto the cutting blade as shown in Fig. 7. The stiffening effect of the four strips of damping foil is small for the steel-made blade. However, application of the damping foil does introduce damping to the system. It is anticipated that the damping foil should reduce the overall vibration of the blade and reduce the overall noise level. The measurements of sound pressure for blades with the damping foil are shown Fig. 13. A comparison with the previous measurements shown in Fig. 5 indicates that the damping foil is very effective. The application of the damping foil eliminated the sound pressure spikes at 440 and 528 Hz and reduced the sound pressure level at 176 Hz. However, it should be noted that, because of the stiffening effect, application of the damping foil slightly increased the sound pressure at 264 and 352 Hz due to change in natural frequencies. The sound pressure at 88 Hz remained unchanged (Fig. 14). From Table 1, sound pressure level of 101.2 dBA at 400 Hz centre frequency band, caused by the 440 Hz sound produced from the cutting deck, is now significantly reduced as a result of the application of damping foil. The second highest sound pressure level of 98.3 dBA at 528 Hz will be significantly reduced as well.

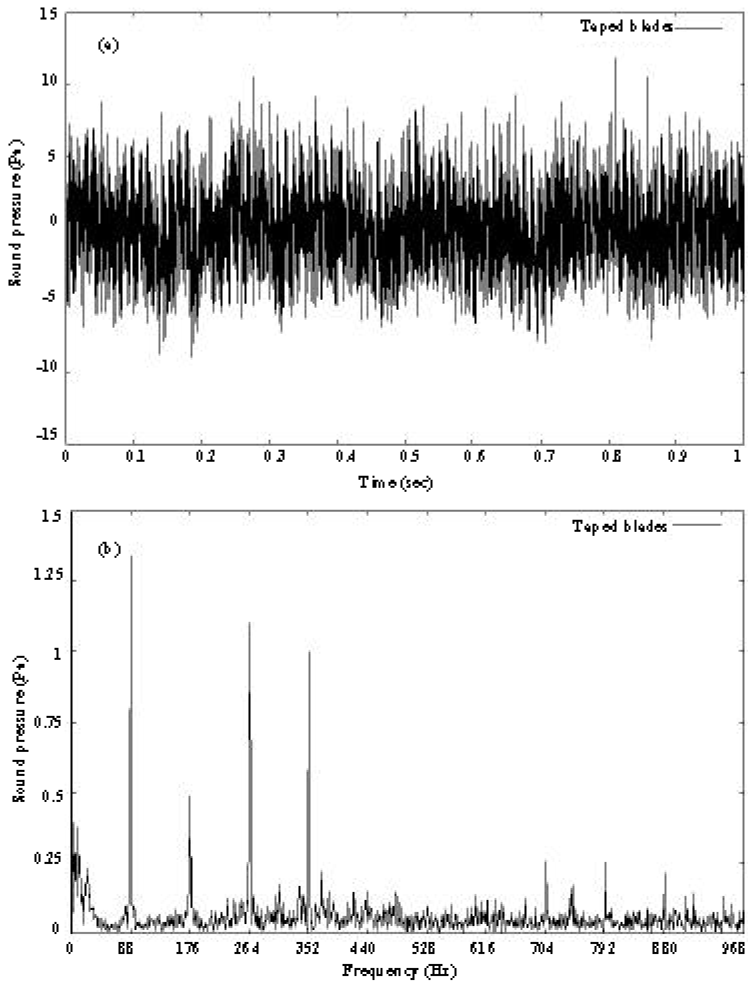

Fig. 14: Measured pressure of sound generated by the taped blades: (a) time domain, (b) frequency domain

CONCLUSION

This study presents an experimental study of sound and vibration generated by a pair of counter-rotating grass-cutting blades for professional use. Experimental data show that the oscillatory dynamic pressure field is periodic and contains only even orders of harmonics in relation to the blade rotational speed. For the tested blades, the highest order of meaningful harmonics is about 24. The highest sound pressure level on an A-weighted scale occurs at the 10th order frequency.

To explain the extraordinarily high sound pressure levels at several frequencies, a three-dimensional finite element model for the cutting blades was developed to determine the dynamic natural frequencies including the effect of spinning. It was found that, during operations, the blades experienced several modes of resonant vibrations below the 24th order frequency. The resonant vibrations of the cutting blades contributed to the very high noise levels at 10 and 12th order frequencies.

Use of viscoelastic damping foil is very effective in reducing blade vibration and the sound pressure levels at the 10 and 12th order frequencies. This measure helps reduce the overall noise produced from the cutting deck.

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