Modelling the Sound of a Golf Ball Impacting a Titanium Plate

DELAYE, S, STREETER, P, MORALES, E, WOOD, P, SENIOR, Terry <http://orcid.org/0000-0002-3049-5724>, HART, J <http://orcid.org/0000-0002-6142-4515> and ALLEN, T

Published version

DELAYE, S, STREETER, P, MORALES, E, WOOD, P, SENIOR, Terry, HART, J and ALLEN, T (2016). Modelling the Sound of a Golf Ball Impacting a Titanium Plate. Procedia engineering, 147, 354-359.

Copyright and re-use policy

Available online at www.sciencedirect.com

ScienceDirect

Procedia Engineering 147 (2016) 354 – 359

Procedia Engineering

www.elsevier.com/locate/procedia

11th conference of the International Sports Engineering Association, ISEA 2016

Modelling the sound of a golf ball impacting a titanium plate

Simon Delaye[a], Patrick Streeter[a], Eric Morales[b], Paul Wood[b], Terry Senior[a], John Hart[a] & Tom Allen[a,c]*

[a]*Centre for Sports Engineering Research (CSER), Sheffield Hallam University, Collegiate Campus, Sheffield, S10 2BP, UK*
[b]*Ping Golf, 2201 W Desert Cove, Phoenix, AZ 85029, USA*
[c]*School of Engineering, Manchester Metopolitan University, Manchester, M1 5GD, UK*

Abstract

A model was developed to predict the sound of a ball impacting a USGA CoR plate, as a first step towards simulating the acoustics of a ball/driver impact. A ball was dropped from 2.5 m onto a free-free plate with the impact sound recorded with a microphone. The experiment was replicated in Ansys/LS-DYNA, with both the exact Boundary Element Method and the Rayleigh method applied to predict the sound. The Rayleigh method predicted lower acoustic pressure than the Boundary Element Method, and was less accurate at predicting relative amplitudes of the frequency spectrum. The models under-predicted decay time, although, increasing mesh density improved agreement with the experiment. Further work should look to improve agreement between model and experiment for decay time, while investigating the effect of impact speed for a range of plate thicknesses.

Peer-review under responsibility of the organizing committee of ISEA 2016

Keywords: Sound; feel; performance; finite element; sporting equipment; perception

1. Introduction

The ‘feel’ of sports equipment can influence the user’s perception of its quality [1, 2] and any physical or psychological discomfort can decrease performance [1]. While the feel of a golf stroke with a driver is a combination of vibrations felt at the hand and the impact sound [3, 4], it is dominated by the acoustic response of the club [5]. Sound is a subjective characteristic, but golfers tend to differentiate drivers in terms of perceived loudness and sharpness [2]. Computer simulation represents an alternative to physical testing, with Finite Element (FE) based structural analysis used for club design [6, 7]. As highlighted by Roberts et al. [4], if impact sound could be predicted using modelling techniques, it would be possible to manipulate the acoustics of a club earlier in the design process.

Several simulation methods are available for solving vibroacoustic problems. The FE method requires discretisation of the entire acoustic domain, while the Boundary Element Method (BEM) only requires the surface of the vibrating structure to be meshed. LS-DYNA (Livermore Software Technology Corporation) couples an FE solver with a BEM solver [8, 9]. The structural response of the object is computed in the time domain, and then transformed into the frequency domain via a Fast Fourier Transform (FFT) as a boundary condition for the BEM solver. The Rayleigh method is a similar alternative to the BEM, in which each element of the vibrating surface is assimilated to a plane surface mounted on a rigid baffle and vibrates independently, it is more efficient but less reliable at predicting acoustic pressure, particularly for curved surfaces [10, 11].

LS-Dyna has been used to model the sound of a golf ball impacting a circular titanium plate with a reduced central section (CoR plate, United States Golf Association [12]). Mase et al. [13] used the BEM to predict the plate frequencies excited by impact, while Allen et al. [14] (2014) applied the Rayleigh method to plates of varying thickness. Mase et al. [13] also simulated a ball/driver impact using the BEM and Rayleigh method, although some frequencies did not match the experiment as the geometry in the model was simplified. It is important to ensure the capabilities of LS-DYNA at predicting the acoustic response of a ball impacting a

* Corresponding author. Tel.:+44 (0)161 2476265
E-mail address: t.allen@mmu.ac.uk

1877-7058
Peer-review under responsibility of the organizing committee of ISEA 2016
doi:10.1016/j.proeng.2016.06.309

simple plate, before developing a ball/club impact model. This investigation, therefore, explored the time and frequency dimensions for ball/plate impact models.

2. Methods

A ball (Titleist ProV1x) was dropped 2.5 m onto the CoR plate placed face up on anechoic foam, to give an impact speed of 7 ms^{-1}. The drop test overcame issues of obtaining consistent impact velocity and location, as encountered when using a high-speed projectile device [14]. The plate was impacted three times at both the centre and 20 mm off-centre, with the sound recorded at 44,100 Hz with a microphone (Behringer 140 ECM 8000) located 2.5 m away. Matlab (Version R2014a) was used to subtract background noise and convert the signal to the frequency domain via an FFT at a resolution of 1 Hz. The drop test was replicated numerically and solved using both the exact BEM and approximate Rayleigh method in Ansys/LS-Dyna.

The ball and plate were modelled in ANSYS 15.0, following the methods of Allen et al. [14], and the simulations were run using the LS-DYNA solver (version R7.0.0). The ball diameter was 42.7 mm, with 22,736 8-node constant stress brick elements. The same elements were used for the plate, with an aspect ratio of four. Modal simulations were used to identify plate frequency modes, which showed low dependency to mesh density when there were at least four elements through the reduced section. Two meshes were selected for the acoustic simulations with, i) 4 elements through the reduced section and 5,040 in total (coarse) and ii) 8 elements through the reduced section and 36,192 in total (fine). The material properties assigned to the plate were modified slightly from those used by Allen et al. [14], to improve agreement between model and experiment (E = 117 GPa, ν = 0.34, ρ = 4,388 $kg.m^{-3}$). Density was reduced by 112 $kg.m^{-3}$ to match the mass of the plate in the model with that of the actual plate and Young's modulus was increased by 1 GPa to improve agreement between simulation and experiment for modal frequencies.

For the acoustic simulations, the ball was set to impact the free-free plate at 7 ms^{-1} at both centre and off-centre locations. A massless acoustic node was set 2.5 m away, corresponding to the microphone position. Acceleration was written from the structural response every 5 μs, and converted to the frequency domain via an FFT with a raised cosine window, to serve as the boundary condition for the BEM solver. The simulation time was set to 0.15 s for the coarse plate according to a time domain preliminary study and to 0.25 s for the fine plate following the work of Volkoff-Shoemaker [15]. The acoustic medium was air at room temperature with a density of 1.21 kgm^{-3}, a speed of sound of 340 ms^{-1} and a reference pressure of 20 μPa.

LS-DYNA computes the acoustic pressure in the frequency domain as a complex variable, containing magnitude and phase information for each frequency. Output frequencies were set to range from 20 to 20,000 Hz, corresponding to the human ear audible range [16], at a resolution of 5 Hz. The microphone used in the experiment was not calibrated for acoustic pressure so the relative amplitudes of excited modes were investigated, which represents the sound timbre [16]. The time domain was also explored, which informs us about the sound behaviour over time [17]. The simulations provided the magnitude of the complex acoustic pressure in the time domain, which does not represent sound as recorded by the microphone as the phase delay is missing.

To obtain the actual time domain representation of the sound, Matlab was used to apply an Inverse Fast Fourier Transform (IFFT) to the complex pressure computed by the solver in the frequency domain. The real and imaginary parts of the complex acoustic pressure were provided. The frequency range was set to 0 to 9,000,000 Hz. After applying the IFFT the resulting sampling frequency was 54,613 Hz which is in accordance with the Nyquist theorem [16] to prevent aliasing when frequencies up to 20,000 Hz need to be discerned. The time domain was investigated for a centre impact with the computationally efficient coarse Rayleigh model.

3. Results

Table 1 compares frequencies from experimental impacts and a modal simulation of the plate, up to the 6th mode. The model tended to slightly under-predict experimental frequencies. The 1st, 3rd and 6th mode were not excited for a centre impact, and the 3rd mode was not excited for an off-centre impact, as the impact location corresponded to nodes for these modes. A few experimental frequencies did not match any mode of vibration from the modal simulation and were attributed to noise. One of these modes had a frequency of 3,570 to 3,735 Hz, which was likely to correspond to the ball as a frequency of 3,415 Hz was identified in a modal simulation of the ball. In addition, Hocknell et al. [18] attributed frequencies recorded from ball/club impact up to 3,500 Hz to vibrations of the ball.

Table 1. Frequencies obtained from experimental impacts and model. Model corresponds to modal simulation for fine mesh.

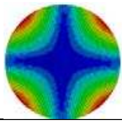
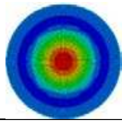
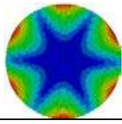
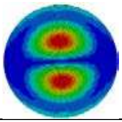
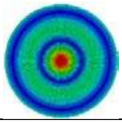
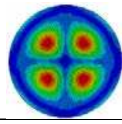
Mode	1st	2nd	3rd	4th	5th	6th
Exp. centre frequency (Hz)	-	4319 - 4323	-	7948 - 7961	12138 - 12160	-
Exp. off-centre frequency (Hz)	2854 - 2856	4316 - 4320	-	7959 - 7971	12100 - 12163	13063 - 13104
Model frequency (Hz)	2868	4244	7770	7847	12030	12886
Model mode shape						

Fig. 1a shows results in the frequency domain for the experiment and fine BEM simulation for centre impacts. The model predicted the frequency of the 2nd and 5th mode reasonably well, but did not identify the 4th mode. The model identified the 2nd mode as the dominant mode, but the amplitude of the 5th mode was 48% of the 2nd mode in comparison to only 2% for the experiment. For the same mesh the BEM and Rayleigh method predicted the same frequencies, with the coarse mesh slightly under-predicting the fine mesh (Fig. 1b). The amplitude of the 5th mode was 26% of the 2nd mode for the coarse Rayleigh model and 20% for the coarse BEM model, while the fine Rayleigh model incorrectly predicted similar acoustic pressure for both modes. The Rayleigh method predicted much lower acoustic pressure than the BEM.

As none of the models excited the 4th frequency mode the experimental impacts were assumed to have been slightly off-centre, based on the corresponding mode shape in Table 1. This was confirmed by running a 5 mm off-centre impact simulation with the coarse BEM model, which excited the 4th mode with low amplitude. Based on the results presented for centre impacts, only the faster solving models with the coarser mesh were used for off-centre impacts.

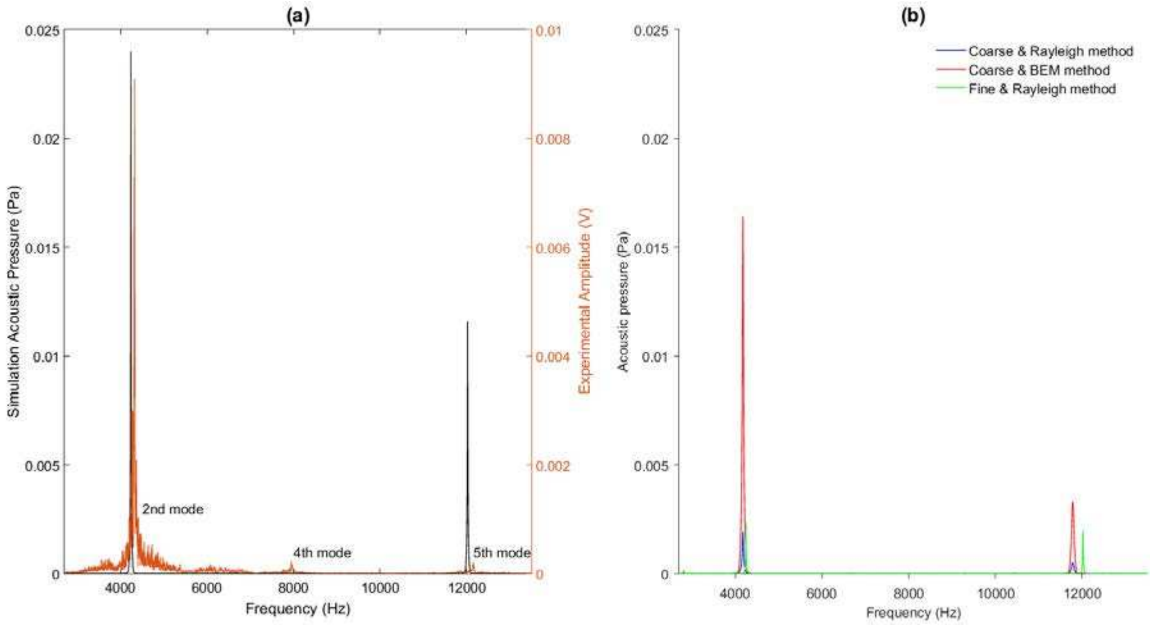


Fig. 1. Centre impact results in the frequency domain a) fine BEM simulation and mean experiment and b) simulation results.

Fig. 2a shows results in the frequency domain for the experiment and coarse BEM simulation for off-centre impacts. The model identified the 2nd mode as the dominant mode, but the 6th mode was not excited. The amplitude of the 1st mode was 29% of the 2nd mode for the model in comparison to only 9% for the experiment. Fig. 2b shows the Rayleigh model predicted lower amplitude than the BEM, in agreement with the results for the centre impact. In contrast to the experiment, the Rayleigh method predicted similar acoustic pressure for the first two modes.

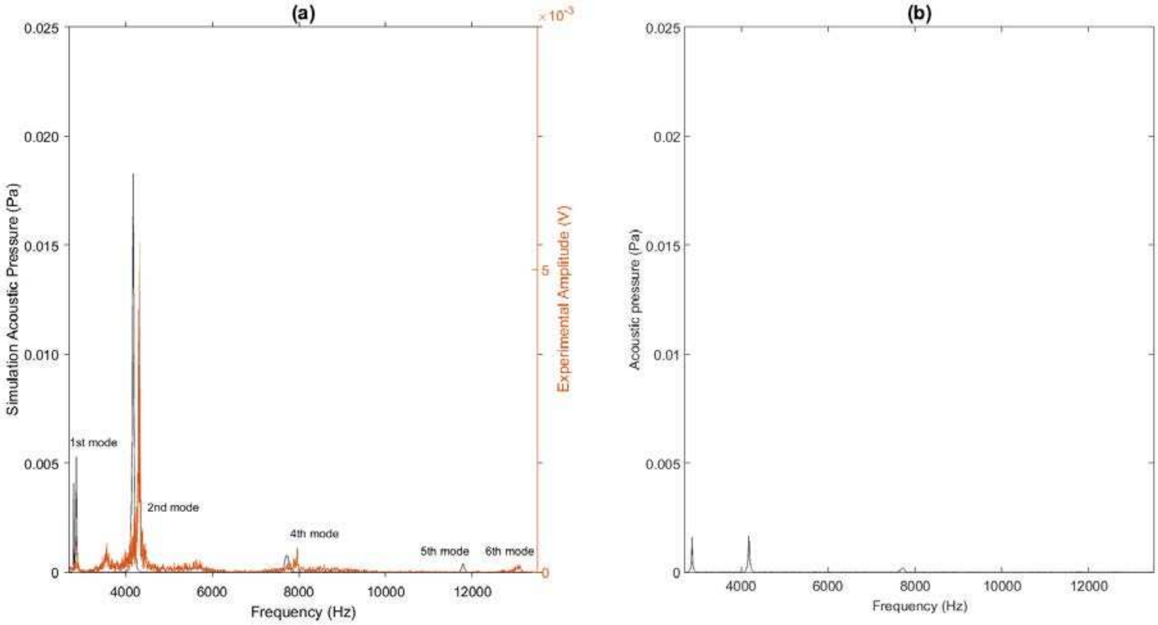


Fig. 2. Off-centre impact results in the frequency domain a) Experiment and coarse BEM, b) coarse Rayleigh.

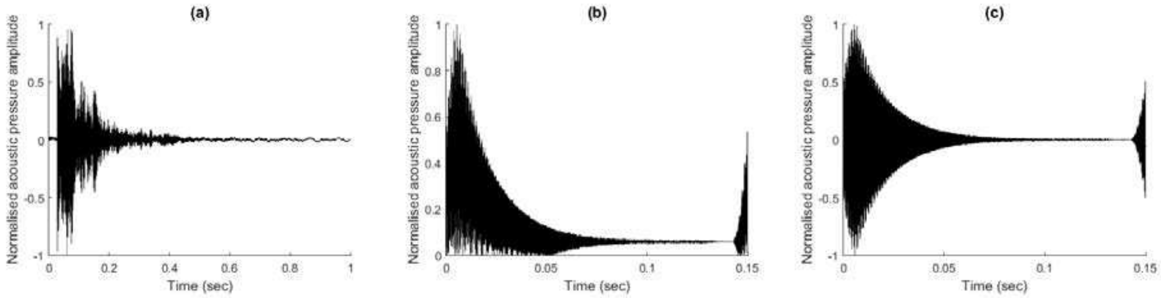


Fig. 3. Centre impact results in time domain a) Experiment b) Predicted magnitude of complex acoustic pressure with coarse Rayleigh method c) True acoustic response computed from coarse Rayleigh method.

Fig. 3 compares experimental and coarse Rayleigh simulation waveforms for a centre impact. The simulation waveforms exhibited a sound pressure increase towards the end which was probably due to a boundary error. The shape of the true temporal acoustic pressure (Fig. 3c) was very different from the magnitude of the complex accoustic pressure (Fig. 3b), which was not symmetrical about the abscissa axis. The global shape of the envelope was similar for both simulations as the impact sound decayed in ~0.1 s, although this was shorter than ~0.5 s for the experiment.

As the decay time of the impact sound was similar for computed true time domain and provided magnitude of the complex temporal acoustic pressure, the latter was used to check for the impact sound duration predicted by the different models. Decay time depended on the mesh density of the plate independently of whether the BEM or Rayleigh method was used. The fine model produced an impact sound which decayed in ~0.25 s, in better agreement with the experiment. Furthermore, the acoustic pressure was not fully damped when the simulation completed, so the termination time should be increased to capture the entire impact sound.

4. Discussion

The BEM and Rayleigh method were able to predict the frequencies excited by a ball/plate impact. For the same mesh density, the BEM and Rayleigh method provided the same frequencies. A plate with 36,192 elements more closely matched experimental frequencies than one with 5,040 elements, and further work should investigate the influence of mesh density in more detail. The BEM was better at predicting the relative acoustic pressure of the frequency spectrum. The Rayleigh method is recommended if only excited frequencies are of interest as these simulations solved at least three times faster.

The Rayleigh method predicted lower acoustic pressure than the BEM, as reported by Mase et al. [13] for a ball/driver impact. To a lesser extent, for a given computation method models with the coarser mesh predicted lower acoustic pressure that those with the finer mesh. Models with a coarser mesh were better predictors of relative amplitudes of the frequency spectrum for centre impacts. This research did not determine if the simulations were able to predict true acoustic pressure in the frequency domain. Further work should, therefore, look to improve the experimental methodology with a view to validate the acoustic pressure predicted by the model.

Investigation in the time domain showed the finer mesh to provide a better prediction of decay time independently of the computation method. The magnitude of the complex acoustic pressure in time domain, however, was used to reach this conclusion, which did not exactly represent the true time domain acoustic response. A method was also developed to obtain the exact time domain sound pressure from an acoustic simulation. Further work should investigate the effect of mesh density on decay time for the true acoustic response.

Further study should investigate the effect of impact speed on predicted relative amplitudes and decay times. This would inform us about the loudness of the impact sound for a given speed which is an important parameter involved in the acoustic perception of a golfer [4]. Further work should also apply the techniques present here to the range of plate thicknesses used by Allen et. al. [14] before developing and validating a model of a ball/driver impact. It would also be interesting to determine if participants can distinguish between recorded audio clips and those generated by models.

5. Conclusion

This research developed models to predict the sound of ball/plate impacts. The Rayleigh method proved sufficient to predict excited frequency modes. The BEM provided more accurate predictions of the relative acoustic pressure of the frequency spectrum. The models under-predicted decay time, although increasing mesh density improved agreement with experiment. Further work should look to improve agreement between model and experiment for decay time, and investigate a range of impact speeds for plates of varying thickness.

Acknowledgements

The authors would like to thank Dr George Dixon and Dr Yun Huang for their guidance. They would also like to thank Dr Heather Driscoll, Dr Amy Beeston, David Koncan and Jim Gough for feedback on the manuscript.

References

[1] Mitchell S, Roberts J, Jones R, Rothberg S, Harwood C. Human perceptions of sports equipment under playing conditions. J Sports Sci. 2001; 19(7):485-497.
[2] Roberts JR, Jones R, Rothberg SJ, Mansfield NJ, Meyer C. Influence of sound and vibration from sports impacts on players' perceptions of equipment quality. Proceedings of the Institution of Mechanical Engineers, Part L: Journal of Materials: Design and Applications. 2006;220(4):215-227.
[3] Roberts JR, Jones R, Mansfield NJ, Rothberg SJ. Evaluation of vibrotactile sensations in the 'feel' of a golf shot. J Sound Vibrat. 2005; 285(1):303-319.
[4] Roberts JR, Jones R, Mansfield NJ, Rothberg SJ. Evaluation of impact sound on the 'feel' of a golf shot. J Sound Vibrat. 2005; 287(4):651-666.
[5] Hocknell A, Jones R, Rothberg SJ. Engineering 'feel' in the design of golf clubs. The Engineering of Sport: Proceedings of the 1st International Conference on the Engineering of Sport.1996. 333-337.
[6] Petersen W, McPhee J. Shape optimization of golf clubface using finite element impact models. Sports Engineering. 2009; 12(2):77-85.
[7] Tanaka K, Teranishi Y, Ujihashi S. Finite element modelling and simulations for golf impact. Proceedings of the Institution of Mechanical Engineers, Part P: Journal of Sports Engineering and Technology. 2013; 2012; 227(1):20-30
[8] Huang Y, Souli M, Liu R. BEM methods for acoustic and vibroacoustic problems in LS-DYNA, Proceedings of 8th European LS-DYNA users conference, Strasbourg, 2011
[9] Huang Y., Souli M., Simulation of acoustic and vibro-acoustic problems in LS-DYNA® using boundary element method, Proceedings of the 10th International LS-DYNA Users' Conference, 2008.
[10] Smith D, Bernhard R. Verification of numerical acoustic radiation predictions. Proceedings of the 1989 noise and vibration conference, vol1, Society of automotive engineers, 1989.
[11] Herrin D, Martinus F, Wu T, Seybert A. A new look at the high frequency boundary element and rayleigh integral approximations. Society of Automotive Engineers, Inc. 2003.
[12] United States Golf Association, Procedure for measuring the velocity ratio of a clubhead for conformance to appendix II, (5a.). Available at: http://www.ewp.rpi.edu/hartford/users/papers/engr/ernesto/hillb2/MEP/Other/Articles/SLEProcRev3_1-02_created_2-10.pdf. Accessed 22 June 2015.
[13] Mase T, Sharpe R, Volkoff-Shoemaker N, Moreira S. Modeling the sound of a golf club impact. Proceedings of the Institution of Mechanical Engineers, Part P: Journal of Sports Engineering and Technology. 2012; 226(2):107-113.
[14] Allen T, Gough J, Koncan D, James D, Morales E, Wood P. Modelling the acoustics of a golf ball impacting a titanium plate. Procedia Engineering. 2014; 72:587-592.
[15] Volkoff-Shoemaker N. Finite element analysis of a golf ball and driver head impact understanding the feasibility of an acoustical optimization (unpublished). California Polytechnic State University, MSc Thesis, 2010.
[16] Smith SW, Audio processing, Chapter 22, in The scientist and engineer's guide to digital signal processing, 2nd ed.,San Diego, CA, California Technical Publishing, 1999.
[17] Howard D, Angus J. Introduction to sound, Chapter 1, in Acoustics and Psychoacoustics, 4th ed., Oxford, Elsevier Ltd., 2009.

[18] Hocknell, A., Jones, R., & Rothberg, S. J. Engineering ‘feel’in the design of golf clubs. *The Engineering of Sport*, 1996, 333-337.