Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)
ISSN 1673-565X (Print); ISSN 1862-1775 (Online)
www.jzus.zju.edu.cn; www.springerlink.com
E-mail: jzus@zju.edu.cn

Automated impact device with non-synchronous impacts: a practical solution for modal testing during operation*

Zhi Chao ONG†[1], Hong Cheet LIM[1], Anders BRANDT[2]
[1]Department of Mechanical Engineering, Faculty of Engineering, University of Malaya, Kuala Lumpur 50603, Malaysia
[2]Department of Technology and Innovation, University of Southern Denmark, Campusvej 55, DK-5230 Odense M, Denmark
†E-mail: alexongzc@um.edu.my

Received Aug. 15, 2017; Revision accepted Sept. 14, 2017; Crosschecked May 9, 2018

Abstract: Previous study has shown that synchronization of phases between impacts and the cyclic load component should be avoided to improve the effectiveness of operational modal testing, i.e. impact-synchronous modal analysis in obtaining a cleaner frequency response function (FRF) estimation with fewer number of averages. However, avoiding the phase synchronization effect is rarely achievable with the current manual impact hammer because of the lack of control of the impact timing. We investigate how to improve FRF estimation in the presence of harmonic disturbances, such as those present in operating rotating machines. An auto impact device is therefore introduced to replace the manual impact hammer. This device ensures that impact intervals can be applied at non-synchronous instances with respect to the harmonic disturbance. We demonstrate that this new device is a viable option for operational modal testing. It allows significant improvement in FRF estimation and shows good correlation of modal extraction data with benchmark experimental modal analysis results.

Key words: Auto impact device; Frequency response function (FRF); Impact-synchronous time averaging; Manual impact hammer; Phase synchronization
https://doi.org/10.1631/jzus.A1700431 **CLC number:** TH16

1 Introduction

Common problems related to vibrations occur due to inherent imbalance in engines for prime movers, in blade and disk vibrations on turbines, and in reciprocating machines, etc. Vibration problems are often more serious when the frequency of the excessive vibration coincides with the natural frequency of the structure. In such a case, the response of the structure is amplified, causing excessive deflections which in some cases can cause immediate failures. Thus, it is valuable to know how an operating system responds to a harmonic excitation. In general, the response of a structure to harmonic vibration coinciding with a natural frequency depends on three factors: (1) the amount of damping, (2) the excitation frequency, and (3) the relationship between the mode shape coefficients in the excitation and response points (William and Marie, 1998).

Modal analysis is an important and established tool in various engineering fields and can be used to address such vibration issues. Engineers use the modal parameters obtained from modal analysis, i.e. natural frequencies, damping ratios, and mode shapes (Avitabile, 2001; Wang et al., 2010; Loh et al., 2011; Fayyadh and Razak, 2013), to design structures or machines based on the desired characteristics so achieving high efficiency during operation, as well as for applications such as structural modification,

* Project supported by the Fundamental Research Grant Scheme (No. FP010-2014A), Postgraduate Research Grant (No. PG011-2015A), and Advanced Shock and Vibration Research (ASVR) Group of University of Malaya, Malaya
ORCID: Zhi Chao ONG, https://orcid.org/0000-0002-1686-3551

sensitivity analysis, structural health monitoring, and structural damage detection (Thomson, 1983; Ewins, 1984; Fransen et al., 2011; Garcia-Perez et al., 2013; Cakir and Uysal, 2015; Dziedziech et al., 2015; Xu et al., 2016). The two most widely used modal analysis techniques are experimental modal analysis (EMA) and operational modal analysis (OMA). EMA describes the dynamic characteristics of the system based on measured input and output data. The analysis can be carried out either in the time or frequency domain, depending on user convenience (Cunha and Caetano, 2006). However, there is a significant constraint on using this technique as the systems or machines are not allowed to operate. In the oil and gas industry, for example, production downtime can equate to hundreds of thousands of dollars' loss per day, and thus it is not feasible to shut down the operating machines to carry out EMA.

In practice, when the structure is excited by external or internal dynamic forces, e.g. a wind excited building or bridge, engine vibration excited cars or machines, OMA is preferred over EMA. OMA, also known as output only modal analysis or ambient modal analysis, is a suitable technique when a machine or system cannot be shut down for EMA purposes (Mohanty and Rixen, 2004; Hashim et al., 2013; Rahman et al., 2014). OMA does not require the input excitation to be measured, but only the output responses. Thus, the total time and cost required for the modal analysis test are greatly reduced. A limitation with OMA for the purpose of achieving the sensitivity of a structure to harmonic loading is that it does not result in scaled mode shapes, and thus cannot answer what a machine's sensitivity is to a particular (harmonic) force.

Impact-synchronous time averaging (ISTA) has been introduced to address the issue of frequency response function (FRF) estimation in the presence of harmonics and random noise (Mohanty and Rixen, 2004; Fayyadh and Razak, 2013; Rahman et al., 2014; Ong et al., 2016). ISTA is based on the averaging of several impacts in the time domain, where after the fast Fourier transform (FFT) is applied the FRF estimation is obtained as the ratio of the spectra. It was shown that the suppression of the harmonics was achieved, although part of the harmonics remained in the FRF estimation, producing modal parameter estimations that were not entirely reliable (Rahman et al., 2014).

In Ong et al. (2017), it was commented that the main reason for the limitation in the ISTA performance was the lack of control on the phase angle distribution of the disturbance at the impact instances; i.e. the harmonic could potentially have the same phase angle in different time blocks. Simulation and experimental studies showed that the effectiveness of impact-synchronous modal analysis (ISMA) can be enhanced when the phases of the impacts are not synchronized with the phase of the periodic response of the cyclic load. However, avoiding the phase synchronization effect is rarely achievable with the current manual impact hammer due to the lack of control on impact timing. In the present study, an auto impact device allowing controlled impact events is introduced. This device can apply impacts at controlled time intervals which are always asynchronous with respect to the responses from the cyclic load components. With a minimum number of impacts, all the responses contributed by unknown sources of the force contained in the acceleration response are filtered out when the phase of the periodic responses is not consistent with the impact signature for every impact applied. Thus, synchronization of phase between impacts and disturbances is avoided through the use of the auto impact device to enhance the estimation of FRF. Modal analysis parameter extraction is then performed on data from measurements during operation on a test structure, in order to validate the effectiveness of removing the harmonics by the ISTA method.

2 Mathematical background

2.1 Effect of phase synchronization in ISTA

Vibration response is acquired and processed in blocks (e.g. 4096 samples). When each individual block of a sinusoidal signal, i.e. $y(t)=A\sin(\omega t+\beta)=a\cos(\omega t)+b\sin(\omega t)$, is recorded in blocks of time series at a different phase, β, mathematically it will lead to different values of a and b even though the amplitude A does not change corresponding to that individual block of time series. Performing block averaging on the recorded sinusoidal signal tends to diminish these non-synchronous components, i.e. a and b, and can subsequently reduce A to zero. Considering the cyclic load component as the sinusoidal signal when performing modal testing during operation, performing ISTA will eventually filter out the harmonic disturbances.

2.2 Control of non-synchronous constant impact interval using auto impact device

A mathematical model has been developed in previous study to control the auto impact device in performing modal testing with a controlled impact interval (Ong and Lee, 2015). The auto impact device has yet to be tested in operational modal testing. Based on the mathematical model which consists of a digital square wave signal, the 'on' and 'off' states of the auto impact device are controlled by the crest and trough of the signal, respectively. The parameters that are involved in the control of the auto impact device are frequency (f) or period (T) of the square wave, sampling rate (SR), block size (BS), duty cycle (DC), time of response block (t_{block}), number of cycles in a time block (n), length of time for active pulse (t_{pulse}), time difference (Δt), the number of blocks for active pulse (N), solenoid 'ON' time (t_{on}), impact interval (T_{impact}), and impact frequency (f_{impact}). The parameters to be manipulated to get different impact profiles are BS, SR, DC, and f (or T) of the square wave.

The impact interval is calculated as

$$T_{\text{impact}} = \frac{T}{\Delta t} \cdot t_{\text{block}}. \tag{1}$$

The impact frequency (Hz) is determined by the inverse of Eq. (1) as

$$f_{\text{impact}} = \frac{1}{T_{\text{impact}}}. \tag{2}$$

3 Measurement procedures and instrumentations

3.1 FRF measurement using manual impact hammer

Fig. 1 shows the experimental setup for ISTA using manual impact hammer excitation. The test rig was made of a motor coupled to a rotor shaft system. The manual impact hammer was set to impact at point 1 while a tri-axial accelerometer roved from point 1 to 20 measuring the responses of the structure in the x, y, and z directions. Twenty averages were taken at each point and the operating speed of the test rig was set at 20 Hz. A data acquisition device (DAQ) consisting of National Instrument NI-USB-9234 modules controlled by the DASYLab software was used. The ME'scope software was used to draw the 3D structural model of the test rig in coordinate points where every point was connected by straight lines (Fig. 2), and for modal parameter extraction. Table 1 shows the descriptions of the instrumentations used in this study.

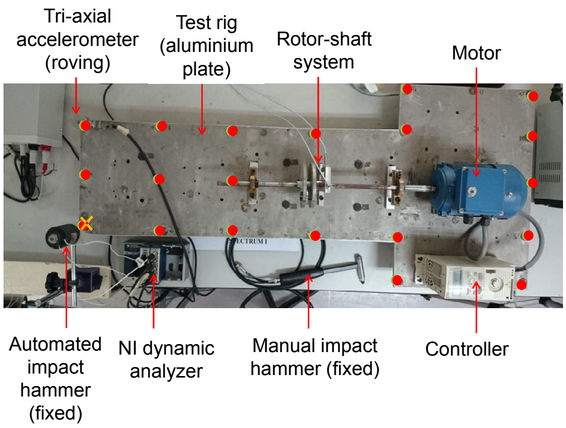

Fig. 1 Measurement locations of motor driven test rig

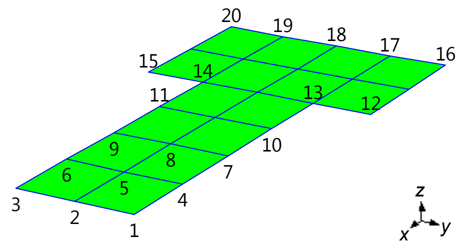

Fig. 2 Structural model of the fault simulation rig

3.2 FRF measurement using auto impact device

The auto impact device replaces the manual impact hammer while sharing the same experimental procedures. This idea was developed to ensure that the impacts are more equal in the force level and position, as the manual impacts may suffer from drawbacks on both these factors. The optimum distance between the test rig and the impact hammer tip was determined by energizing the auto impact device until the moment the impact hammer tip has just contacted the surface of the test rig.

Table 1 List of instrumentations

Instrument	Detail
UM simulation rig	Used as a test rig to perform ISTA
PCB impact hammer (Model 086C03)	Sensitivity: 2.16 mV/N Tip type: medium tip with vinyl cover Hammer mass: 0.16 kg Frequency range: 8 kHz Amplitude range: ±2200 N peak Impact period: random
Automated impact device and impact forcing sensor (Model 208C04)	Clamped with retort stand and connected to channel 1 of National Instrument Dynamic Analyzers Sensitivity: 1.162 mV/N Tip type: medium tip with vinyl cover a. Consistent impact interval setting: Auto impact sampling rate: 50 000 samples/s; Auto impact block size: 1024 samples; Frequency: 97.78 Hz; Duty cycle: 0.0050; Impact period: 8.069 12 s b. Random impact setting: Auto impact sampling rate: 1000 samples/s; Auto impact block size: 1 sample; Multiple of cyclic load frequency: 0.2 Hz; Random number: 0–1; Impact period: random
IMI tri-axial accelerometer (Model 604B31)	Sensitivity: 100 mV/g Frequency range: 0.5–5000 Hz Amplitude range: ±50 g peak
NI USB dynamic signal acquisition module (Model NI-USB-9234)	Number of channels: 4 ACD resolution: 24 bits Minimum data rate: 1650 samples/s Maximum data rate: 51 200 samples/s
DASYLab v10.0	Sampling rate: 2048 samples/s Block size: 4096 samples Channel 1: manual impact hammer/automated impact device Channel 2: accelerometer (*X*-axis) Channel 3: accelerometer (*Y*-axis) Channel 4: accelerometer (*Z*-axis) Application of exponential window in time response Adjustment was made in pre-setting mode
ME'scope v4.0	To process FRF obtained through DASYLab, curve fitting is done using ortho-polynomial method to extract damped natural frequency, modal damping, and residue mode shape

3.2.1 Random impact excitation

The idea of generating random impacts is that it should ensure impacts are imparted at random time instances. To imitate this, the frequency of the control signal was set at 0.2 Hz (multiple of cyclic load frequency, i.e. 20 Hz) with the addition of a pseudo random number (PRN) of a cyclic load period:

$$T_{\text{random impact}} = \frac{1}{f_{\text{multiple}}} + \left(\text{PRN} \times \frac{1}{f_{\text{cyclic load}}}\right). \tag{3}$$

3.2.2 Non-synchronous impact excitation with constant impact interval

As mentioned in (Ong and Lee, 2015), the impact contact time and impact interval of the impact device are determined by f, BS, DC, and SR of the control signal. Note that the BS and SR used should produce a quick time response for the impact device within the data acquisition time of 2 s. The general setup is shown in Fig. 3. A control signal in the form of a pulse was generated to trigger the impact device to excite the structure. As the operating speed of the motor was set at 20 Hz, synchronization of impacts with cyclic load component could be avoided as the impact frequency of 0.1239 Hz determined is a non-integer multiple of the cyclic load frequency (Fig. 4). It effectively creates a consistent but non-synchronous impact interval and ensures that the impacts are not synchronized with the cyclic load component. Time averaging of 20 blocks would diminish the cyclic load component and the desired response originating from impulse on the structure remains unchanged over time. The ideal combination of the parameters to be set into the DAQ was experimentally determined and the impact interval was calculated as tabulated in Table 2. Note that 20 averages were used for each experiment. This could differentiate the effectiveness of using the manual impact hammer, random impacts, and non-synchronous impacts with constant impact interval by the auto impact device in FRF measurement and modal parameter extraction.

4 Results and discussion

4.1 Comparison of FRF estimation

In this subsection, we compare the FRF estimation using three different measurement strategies: (1) manual impact hammer, (2) auto impact device with random impacts, and (3) auto impact device with non-synchronous impacts.

Figs. 5–7 depict the experimentally determined FRFs through modal testing during operation using the manual impact hammer, random impacts, and non-synchronous impacts by auto impact device, respectively. A better FRF is the result of the output response of a structure divided by the input excitation only. By comparing these estimated FRFs, the highest peak is observed at 20 Hz using manual impact hammer, followed by auto impact device with random impacts. The cyclic load component at 20 Hz is dominant and covers up the adjacent modes and consequently seriously affects the FRF estimation. Meanwhile, using the auto impact device with non-synchronous impacts and constant impact interval yields a lower disturbance component at 20 Hz, and thus the adjacent modes appear and are significantly enhanced.

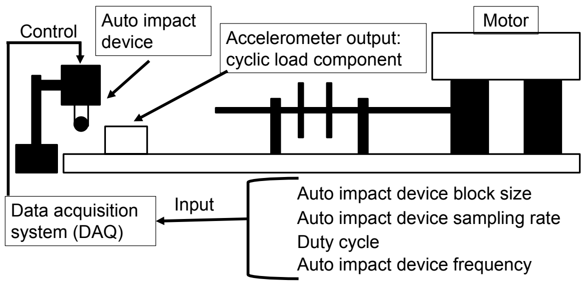

Fig. 3 General instrumentation setup for an auto impact device with non-synchronous impacts with constant impact interval

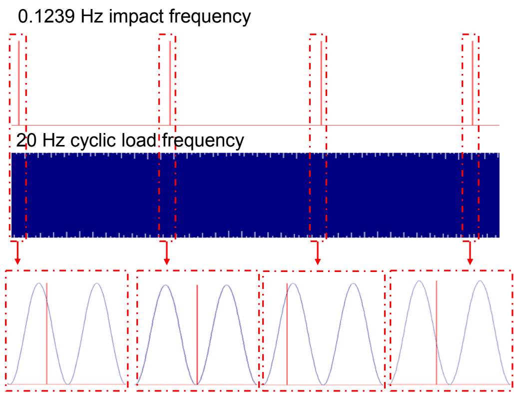

Fig. 4 Simulation of non-synchronous impacts applied corresponding to the periodic response of the cyclic load

Table 2 Summary of input parameters and output response for the auto impact device with non-synchronous impacts

Item	Detail
Input signal to DAQ	Sampling rate (SR): 50 000 samples/s Block size (BS): 1024 samples Duty cycle (DC): 0.5% Frequency (f): 97.78 Hz
Output response of the auto impact device	Impact interval: 0.1239 Hz

The highest cyclic load component that is observed in the FRF estimation using the manual impact hammer is possibly due to any of three reasons:

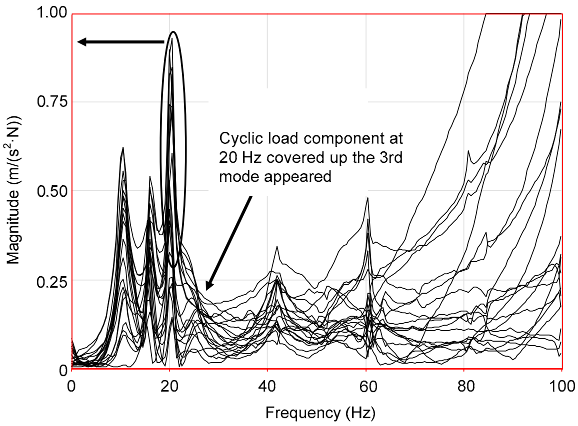

Fig. 5 Estimation of FRFs using manual impact hammer

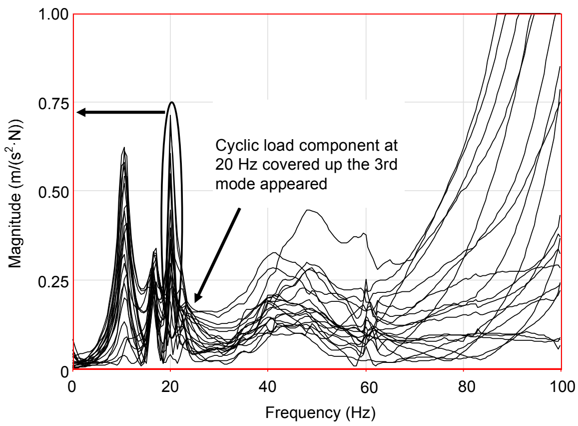

Fig. 6 Estimation of FRFs using auto impact device with random impacts

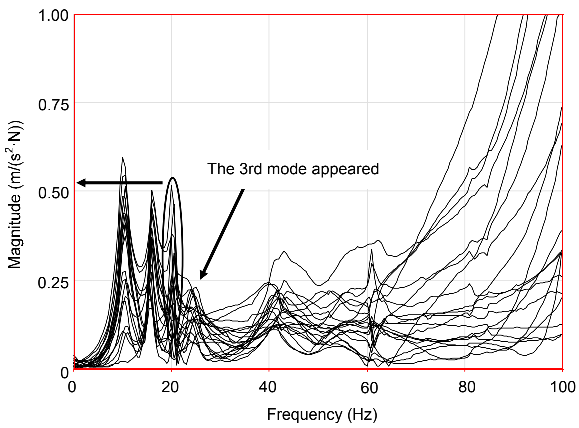

Fig. 7 Estimation of FRFs using auto impact device with non-synchronous impacts

(1) inconsistency in force level of input excitation, (2) inconsistency in excitation location between the impacts, and (3) inefficient removal of the harmonic components when the impact instants are random.

Previous study has shown that a cyclic load component can significantly affect the quality of the estimated FRF such that the modal parameter extraction stage becomes difficult. When the excitation is manually conducted by the user, the amount of excitation force for each impact may vary. When the input excitation force for a particular impact is very small compared to the response from the cyclic load component, the natural modes of the test structure are hardly excited. Consequently, the signatures triggered by impacts are dominated by the response from the cyclic load. Another reason is that the user may perform the excitation at locations which slightly deviate from the predefined location. In addition, eliminating the harmonic response using random trigger instants requires large number of averages.

Replacing the manual impact hammer by an auto impact device using the random timing of the impacts tends to solve some of the limitations of using an impact hammer manually. The force level of input excitation is more consistent as the input force is well controlled. This is important to ensure that each impact has a force level higher than the cyclic force in order to excite all the natural modes of the test structure. Moreover, the auto impact device is clamped firmly by the retort stand, and thus it is able to consistently impart the impacts at the predefined location. It is worth mentioning that the random impact approach by an auto impact device will be useful and able to enhance the FRF estimation provided that all the random impacts are well distributed over the period of the cyclic load. However, the reduction of the cyclic load component at 20 Hz is not very efficient. Further investigation on the time response signal of all measurement points shows that at the phase position where each impact starts, the phases of the periodic response of cyclic load with respect to 20 impacts applied at each measurement point tend to be concentrated or closely spaced within certain areas and are not equally distributed (Fig. 8). This shows the inefficiency of using random timing.

An enhancement on FRF estimation is obtained using the auto impact device with impacts with a fixed time interval, non-synchronous with the cyclic load. A significant decrease of the cyclic load component at 20 Hz, of 45% compared to using a manual impact hammer, is observed. The time response signals of all measurement points are again examined at the phase position where every impact or acceleration response starts. The phase positions of periodic response from cyclic load with respect to impact for all 20 averages are equally distributed as shown in Fig. 9. Thus, a better FRF estimation is generated using the auto impact device with non-synchronous impacts.

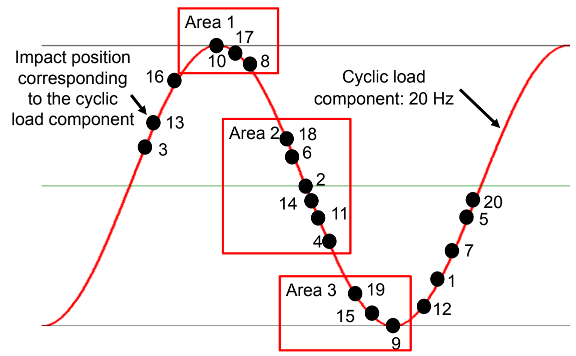

Fig. 8 Phase positions of random impacts corresponding to respective periodic response of cyclic load

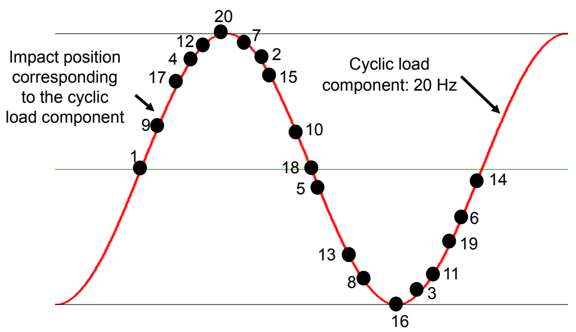

Fig. 9 Phase positions of non-synchronous impacts corresponding to respective periodic response of cyclic load

4.2 Comparison of modal extraction data

Next, we used the estimated FRFs described in Section 4.1 to obtain the modal parameters. EMA results were used as a benchmark to compare and validate the effectiveness of using different excitation strategies, and results are tabulated in Table 3. As can be seen, the first two natural modes are excited by both manual impact hammer and the auto impact device with random impacts. For the case using manual impact hammer the first two modes are estimated at 10.5 Hz and 15.9 Hz respectively, whereas using the auto impact device with random impacts results in estimations of 10.5 Hz and 16.5 Hz respectively. The third natural mode is covered up by the harmonic disturbance at 20 Hz. On the other hand, the auto impact device with non-synchronous impacts achieved good elimination of the harmonic at 20 Hz, resulting in estimations of the first three modes, i.e. 10.4 Hz, 15.8 Hz, and 24.4 Hz. Also, a higher percentage of difference in damping ratio is observed for excitation using the manual impact hammer and the auto impact device with random impacts for the first two natural modes (Table 4). The percentage of difference between the benchmark and auto impact device with non-synchronous impacts in modal damping estimation for the third natural mode is 12.54%. The fact that the harmonic at 20 Hz is not totally eliminated could cause the error in damping ratio estimation, although the errors are small, indicating a good suppression of the harmonic.

Modal assurance criterion (MAC) values between the benchmark EMA data and ISTA data using manual impact hammer and auto impact device are summarized in Table 3. The first and second natural modes are far from the cyclic load frequency at 20 Hz where it has little effect on modal extraction and this, in turn, yields a stable and high MAC value when only 20 averages are taken in all three experiments. The third natural mode could be estimated when the auto impact device with non-synchronous impacts was used. The correlation of the mode shape with the benchmark EMA is high with an MAC value of 0.902 whereas using the manual impact hammer and auto impact device with random impacts could not effectively reduce the dominance of harmonic disturbances at 20 Hz, and thus the third natural mode could not be identified using data from these two cases.

5 Conclusions

The ISTA technique is able to filter out non-synchronous harmonic disturbances in FRF estimation. The technique, however, has some basic limitations such as lack of control of the impact with respect to the phase angle of the disturbances when using an impact hammer manually. This limits the effectiveness and practicality of this novel technique. In this paper, an auto impact device with non-synchronous impacts has been introduced to apply non-synchronous

Table 3 Summary of natural frequencies and mode shapes comparison between modal parameter extraction based on FRFs from a benchmark (BM) measurement without the harmonic and using the ISTA with manual impact hammer (A), auto impact device with random impacts (B), and auto impact device with non-synchronous impacts (C)

Mode	Natural frequency (Hz)				Percentage of difference (%)			MAC		
	BM	A	B	C	BM vs. A	BM vs. B	BM vs. C	BM vs. A	BM vs. B	BM vs. C
1	10.4	10.5	10.5	10.4	1.00	1.00	0.00	0.922	0.899	0.925
2	15.9	15.9	16.5	15.8	0.00	3.77	0.60	0.892	0.890	0.893
3	24.0	N/A	N/A	24.4	N/A	N/A	1.67	N/A	N/A	0.902

N/A: not available

Table 4 Summary of damping ratios from modal parameter extraction based on FRFs from a benchmark (BM) measurement without the harmonic and using the ISTA with manual impact hammer (A), auto impact device with random impacts (B), and auto impact device with non-synchronous impacts (C)

Mode	Damping ratio				Percentage of difference (%)		
	BM	A	B	C	BM vs. A	BM vs. B	BM vs. C
1	0.0832	0.0949	0.0935	0.0933	14.06	12.38	12.14
2	0.0448	0.0436	0.0499	0.0440	2.68	11.38	1.79
3	0.0566	N/A	N/A	0.0495	N/A	N/A	12.54

impacts in determining dynamic characteristics of an operating machine. The enhancement of the effectiveness of ISTA has been demonstrated through comparisons of FRF estimation obtained by a manual impact hammer, by the auto impact device with random impacts, and by the auto impact device with non-synchronous impacts. Results showed that an enhancement in FRF estimation was obtained using the auto impact device with non-synchronous impacts, making it possible to estimate a third mode which, with the other two methods, is hidden by the harmonic disturbance. The cyclic load component was decreased and adjacent modes were found to be enhanced significantly. Enhanced FRF estimation was also found to lead to more accurate modal parameters. Results showed that using the auto impact device with non-synchronous impacts, the first three natural modes were successfully determined and all three modes achieved good correlation with benchmark EMA results with relatively low percentage of difference in natural frequency, of less than 1.67%, 1.79%–12.54% in damping ratio, and MAC values between 0.893 and 0.925. Therefore, an auto impact device which applies non-synchronous impacts with respect to the phase of the harmonic disturbance is a viable option instead of a manual impact hammer to enhance the FRF estimation and modal extraction data.

References

Avitabile P, 2001. Experimental modal analysis—a simple non-mathematical presentation. *Sound and Vibration*, 35(1):20-31.

Cakir F, Uysal H, 2015. Experimental modal analysis of brick masonry arches strengthened prepreg composites. *Journal of Cultural Heritage*, 16(3):284-292. https://doi.org/10.1016/j.culher.2014.06.003

Cunha A, Caetano E, 2006. Experimental modal analysis of civil engineering structures. *Sound and Vibration*, 40(6): 12-20.

Dziedziech K, Staszewski WJ, Uhl T, 2015. Wavelet-based modal analysis for time-variant systems. *Mechanical Systems and Signal Processing*, 50-51:323-337. https://doi.org/10.1016/j.ymssp.2014.05.003

Ewins DJ, 1984. Modal Testing: Theory and Practice. Research Studies Press, Letchworth, UK.

Fayyadh MM, Razak HA, 2013. Damage identification and assessment in RC structures using vibration data: a review. *Journal of Civil Engineering and Management*, 19(3): 375-386. https://doi.org/10.3846/13923730.2012.744773

Fransen S, Rixen D, Henriksen T, et al., 2011. On the operational modal analysis of solid rocket motors. *In*: Proulx T (Ed.), Structural Dynamics, Volume 3. Springer, New York, USA, p.453-463. https://doi.org/10.1007/978-1-4419-9834-7_43

Garcia-Perez A, Amezquita-Sanchez JP, Dominguez-Gonzalez A, et al., 2013. Fused empirical mode decomposition and wavelets for locating combined damage in a truss-type structure through vibration analysis. *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)*, 14(9):615-630.

https://doi.org/10.1631/jzus.A1300030

Hashim H, Ibrahim Z, Razak HA, 2013. Dynamic characteristics and model updating of damaged slab from ambient vibration measurements. *Measurement*, 46(4):1371-1378. https://doi.org/10.1016/j.measurement.2012.11.043

Loh SK, Faris WF, Hamdi M, et al., 2011. Vibrational characteristics of piping system in air conditioning outdoor unit. *Science China—Technological Sciences*, 54(5): 1154-1168. https://doi.org/10.1007/s11431-011-4360-x

Mohanty P, Rixen DJ, 2004. Operational modal analysis in the presence of harmonic excitation. *Journal of Sound and Vibration*, 270(1-2):93-109. https://doi.org/10.1016/S0022-460x(03)00485-1

Ong ZC, Lee CC, 2015. Investigation of impact profile and isolation effect in automated impact device design and control for operational modal analysis. *Journal of Dynamic Systems Measurement and Control*, 137(9):094504. https://doi.org/10.1115/1.4030526

Ong ZC, Lim HC, Khoo SY, et al., 2016. An experimental investigation on the effects of exponential window and impact force level on harmonic reduction in impact-synchronous modal analysis. *Journal of Mechanical Science and Technology*, 30(8):3523-3532. https://doi.org/10.1007/s12206-016-0712-6

Ong ZC, Lim HC, Khoo SY, et al., 2017. Assessment of the phase synchronization effect in modal testing during operation. *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)*, 18(2):92-105. https://doi.org/10.1631/jzus.A1600003

Rahman AGA, Ismail Z, Noroozi S, et al., 2014. Enhancement of impact-synchronous modal analysis with number of averages. *Journal of Vibration and Control*, 20(11):1645-1655. https://doi.org/10.1177/1077546312475147

Thomson WT, 1983. Theory of Vibration with Applications. Allen & Unwin, London, UK.

Wang H, Zou KG, Li AQ, et al., 2010. Parameter effects on the dynamic characteristics of a super-long-span triple-tower suspension bridge. *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)*, 11(5): 305-316. https://doi.org/10.1631/jzus.A0900496

William TT, Marie DD, 1998. Theory of Vibration with Applications. Prentice Hall, New Jersey, USA.

Xu YL, Zhang XH, Zhu SY, et al., 2016. Multi-type sensor placement and response reconstruction for structural health monitoring of long-span suspension bridges. *Science Bulletin*, 61(4):313-329. https://doi.org/10.1007/s11434-016-1000-7

中文概要

题　目： 实现异步冲击的自动冲击装置：一种针对操作过程中模态测试的实用方案

目　的： 目前手动冲击锤设备缺乏对冲击时间的控制，容易引起冲击相位和周期荷载的周期性响应的相位同步问题。本文旨在通过使用异步自动冲击激励的自动冲击装置代替同步冲击模态分析中传统手动冲击锤的方法来解决上述问题。

创新点： 1. 引入具有可调冲击参数的自动冲击装置；2. 该装置可通过控制施加冲击的时间步来确保冲击和来自循环载荷组件的响应异步；3. 当周期性响应的相位与装置所施加的冲击信号不一致时，加速响应中未知力源的影响会被降到最小。

方　法： 1. 分别使用数字方波信号的波峰和波谷来控制自动冲击装置的“开”和“关”状态；2. 通过调控样本大小（1024 个）、采样率（50 000 个/秒）、占空比（0.5%）和冲击频率（97.78 Hz）（或周期）等参数得到不同的冲击图形。

结　论： 1. 使用可实现异步冲击的自动冲击装置可以估算第 3 阶自然模态；2. 前 3 种自然模态可以被成功确定并与基准实验模态分析结果具有良好的相关性，表现为低于 1.67%的自然频率差异，1.79%~12.54%的阻尼比差异以及介于 0.893 和 0.925 之间的模态置信度。3. 针对谐波干扰对相位的影响，相比于使用手动冲击锤来增强频率响应函数估计和模态提取数据，使用可实现异步冲击的自动冲击装置是一种更可行的选择。

关键词： 自动冲击装置；频率响应函数；冲击同步时间均值；手动冲击锤；相位同步