Study of vibration transmission in hard disk drive for ultra high track density

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Study of Vibration Transmission in Hard Disk Drive

For Ultra High Track Density

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School of Mechanical & Production Engineering

A thesis submitted to the Nanyang Technological University
in fulfilment of the requirement for the degree of
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Abstract

The continuing increase of recording density of magnetic hard disk drives (HDDs) requires a higher positioning accuracy within a narrower track width, making vibration control one more critical issue in HDD industry. A good knowledge of the vibration transmission characteristics within HDD may help the vibration reduction a lot. The objective of this study is to study the vibratory energy transmission within complex structures such as HDD using structural intensity.

Firstly, the experimental determination of structural intensity in vibrating structures is studied and a novel experimental data processing scheme, B-spline surface fitting (BSSF) is proposed. The performance of the B-spline approximation in structural intensity calculation is investigated from a viewpoint of wave number domain (k -space) filtering. Relationship between parameters of the B-spline approximation and the k -space filter is established. The BSSF method as well as the experimental determination procedure is validated through both simulated and measured velocity data. With the experimental method established, the structural intensity within a near HDD base casing is investigated experimentally. Preliminary results show that it is feasible to visualise the vibratory energy within the base casing; however, further study is needed to eliminate the false energy source in the structural intensity pattern.

Approaches of diverting the structural intensity in the base casing is studied using 3D solid elements. Simulation results show that proper addition of stiffener can change the orientations of energy flow in the base casing significantly. Therefore, it opens a door to divert the vibratory energy flow in the base casing effectively.

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Nomenclature

Symbol	Description
B	bending stiffness coefficient
E	Young's modulus
f	frequency
$F()$	Fourier transform
h	thickness
$i_k(t)$	instantaneous structural intensity
I_k	structural intensity
J_k	reactive intensity
k	wave number
k_B	bending wave number
M_i	bending moment
M_{ij}	torsion moment
N_i	force
t	time
v	velocity
$\hat{v}_l$	laser measured velocity
w	normal displacement
W	energy flow, work
α_i	nodal velocity
$\beta^n(x)$	n -th order B-spline basis function
λ	wavelength
ω	angular frequency
ρ	mass density
σ_{ij}	stress
$\tilde{\Pi}_k$	complex structural intensity
ν	Possion's ratio

Chapter 1

Introduction

1.1 Background

Hard Disk Drives (HDDs) play a dominant role in data storage technology. Over the last few decades, magnetic recording technology has been going through a series of dramatic transformations and innovations. The progress in the disk drive industry is generally measured in the dimensions of the reduction of disk size and increase of data storage density. Higher storage capacity, faster data access and lower production cost are always the impetus of the HDD technology. It is projected that the capacity in individual 2.5 inch disk drive would break through 400 GBytes in 2005 (IBM, 2001).

A fundamental way in the HDD industry to increase the storage capacity is to increase the areal density, which is the product of linear density, measured in the number of bits per inch (BPI) and track density, measured in track per inch (TPI). High areal density requires high track density, which is limited by the head positioning accuracy over the disk. Factors governing the head positioning accuracy include the noise of the servo signal, the synchronous and non-synchronous vibration with the rotation of the disk, and the transient mechanical vibration right after the high-speed movement of the head.

Traditionally, there has been intense focus on reducing the effect of actuator mechanical resonant modes on the head-positioning servo to achieve higher track density. The head actuator assembly (HAA) has been optimised and various passive dampers have been designed to improve the vibration characteristics of the HDD actuator. In these studies,

the vibrations concerned are mainly generated from windage, seek induced residual vibration, etc.

As areal recording density of HDD grows at an average growth rate of 60-100% over the years, track densities of 500k TPI and higher will become necessary in the near future. It means the track mis-registration (TMR) must be limited within 5 nano-meters or below. For such high head positioning accuracy, sources of vibration that have previously been within tolerance limit now become the focus of further improvement effort. One of such concerns is the vibration of the base casing.

There are several factors that may contribute to the vibration of the base casing. Firstly, the external shock and vibration may cause the base casing to vibrate. There have been some studies on novel servo control designs to cancel the external disturbances. Another source of the vibration comes from the spindle motor rotation, which is synchronous with the spindle revolution. Moreover, when the actuator seeks, the base casing receives a reactive force resulting from the actuator driving force, which causes transient dynamics of the base casing. Because of the action of the pivot bearing, the rotational vibration of the base casing can result in relative angular displacement between the actuator and the disk. In addition, the vibration may be transmitted from the base casing to the actuator via the pivot. All of these factors pose a great challenge to the head positioning accuracy for pushing the track density higher.

1.2 Objectives

This project seeks to study the issue of vibratory energy transmission in complex structures both experimentally and numerically. The main objective is to establish a

methodology to study how the vibratory energy is transmitted within the HDD, which will be applicable to the vibration control problem for ultra high track density.

1.3 Scope

The scope of this study is limited to investigate the vibration transmission in HDD base casing using structural intensity technique. It includes experimental determination method, numerical simulation and applications of structural intensity to vibration control in HDD.

1.4 Outline of the thesis

Given the objective of this project, structural intensity, which is adopted to visualise the vibratory energy in complex structures, is investigated. Due to the limitations of the current data processing schemes, a new method, B-spline surface fitting is developed to extract structural intensity from measured data. With the measurement methodology established, preliminary experiment is carried out on a near HDD base casing. In addition, ways to divert the power flow in complex structures is also investigated numerically. The rest of the thesis is organised as follows,

Chapter 2 first reviews the major energy flow analysis methods and the differences between these methods. The measurement and numerical simulation of the structural intensity is addressed in detail.

Chapter 3 addresses the theoretical development of structural intensity technique. After introducing several different data-processing schemes, the new scheme, BSSF is

proposed. Analytical cases are also given to exemplify the basic concepts of structural intensity.

Chapter 4 validates the structural intensity measurement method discussed in chapter 3. Both numerical simulation and experiment are carried out. In addition, the performance of BSSF in structural intensity calculation is investigated from a viewpoint of k -space filtering. Relationship between parameters of B-spline approximation and k -space filter is established.

Chapter 5 presents some preliminary experimental and numerical results on HDD to examine the potential applications of structural intensity in HDD.

Chapter 6 concludes the thesis and proposes topics for future work.

Chapter 2

Literature Review

In this chapter, three major energy flow analysis methods are first reviewed. The structural intensity measurement method is then addressed in detail and limitations of current measurement techniques are briefly discussed. With the application of supercomputer, numerical simulation is becoming more and more popular in the study of structural intensity. Hence, computation of structural intensity using finite element method is also reviewed. Finally, the applications of structural intensity, including its application in HDD, are discussed.

2.1 Overview of energy analysis methods

In the vibration analysis of complex structures, it is often needed to know how vibration energy in one part of the structure will be transmitted to other parts of the structure. For example, an automotive engineer may need to determine how the engine vibration will affect the passenger compartment. In order to fulfil this task, the concept of power flow was proposed and several methods to determine the power flow in structures were developed. In the literature, power is defined as the time-averaged product of the phased generalised force and velocity component in the direction of the force. Since the direction of the generalised forces is known, the direction of the power is also determined. Therefore, the power can be plotted in a flow type pattern.

In order to determine the spatial distribution of vibration energy, many works have been reported. There are several ways of utilising energy quantities in connection with a

vibration analysis, e.g., statistical energy analysis (SEA), energy finite element method (EFEM) and structural intensity technique.

2.1.1 Statistical Energy Analysis (SEA)

SEA is an approximate power flow analysis method in which the complex structure is modelled as a collection of interconnected subsystems. The vibration level within each subsystem is characterised by the average energy per mode, and energy flow across the junctions between subsystems is characterised by a set of coupling loss factors (Lyon and Dejong, 1995). Since the SEA model includes power balance information, it can be used to trace the flow of energy through the system and to identify important transmission paths from the source locations to the response locations. For example, given an excitation applied at the supporting beams in the floor of a high-speed train, one may calculate the power flow to the floor plates and to the passenger compartment (Lalor, 1996).

SEA has been used to predict power flow in many types of engineering structures, such as automobiles, aircrafts and ships. Its particular value from the conceptual point of view is that it establishes a link between the stored energy and the energy flow. However, as its results are expressed only by time-averaged and spatially integrated energy, SEA cannot provide a spatial mapping of the power in the structure and detailed spatial variations in the power are unable to be obtained. Furthermore, SEA is based on the assumption that there are sufficient vibration modes in the frequency band of interest, which render it only a useful tool in high-frequency range.

2.1.2 Energy Finite Element Method (EFEM)

In order to calculate the spatial distribution of vibration energy, Nefske and Sung (1989) suggested the so-called energy finite element method (EFEM) or power flow finite element method (PFFEM) by using the vibration conduction derived for one-dimensional vibrating structures. It is an SEA equivalent method based on local energy variables in analogy with heat conduction. Bouthier and Bernhard (1992) extended the PFFEM to two-dimensional vibration analysis composed of plane wave components. Power flow models of reinforced beam-plate coupled structures and coupled thin plates were derived to examine the propagation of energy waves (Seo, et al., 2003; Park, et al, 2001,2003).

However, Langley (1995) showed that the two-dimensional vibration conduction equation would be valid only for the structure having a relatively small damping factor or highly reverberant wave field. Carcaterra and Sestieri (1995) also showed that the vibrational-thermal analogy may be out of reach where a general case is concerned.

2.1.3 Structural Intensity

Noiseux (1970) first introduced the concept of structural intensity to extend the vector acoustics approach to energy flow in structure-borne sound fields. His goal was to develop a method that would un-mask the energy path and provide a spatial map of the power flow in structures. Later, Pavic (1976) proposed a method for measuring the power flow due to flexural waves in beam and plate structures by using multiple transducers and digital processing techniques. Cross-spectral density methods were presented by Verheij (1980) to measure structural power flow in beams and pipes. Since then, many measurement methods and numerical prediction approaches have been proposed.

Structural intensity indicates the magnitude and direction of structure-borne sound in vibrating systems. With the structural intensity vector fields, the vibratory energy transmission path and positions of energy sources and energy sinks can be easily determined. Therefore, dissipative elements, mechanical modifications and active vibration control can be adopted to control or divert the energy flow paths within the structure and the amount of mechanical energy injected into the structure.

In addition to SEA, EFEM and structural intensity, other power flow methods have also been proposed and studied, for example, ray tracing method (Chae and Ih, 2001) and power mode approach (Ji. et. al., 2003a, 2003b). One major difference between structural intensity and the rest of the power flow methods mentioned above is that structural intensity describes accurately the spatial distribution of vibration energy, however, the others are statistical methods and can only provide approximate results. In order to study how the vibratory energy is transmitted within the HDD, a spatial distribution of vibration energy is preferred. Therefore, structural intensity is adopted to investigate the vibration transmission in this study.

2.2 Measurement of structural intensity

The determination of structural intensity in thin-walled structures requires the information of surface velocity and stress, as well as the associated phase relationship between them. According to the ways to measure the velocity and/or stress, the structural intensity measurement can be grouped into two main categories: contact method and non-contact methods. Contact method usually adopts transducers such as accelerometers and strain gauges, while for non-contact method, laser, interferometry or near-field acoustic holography could be used.

The initial works on structural intensity measurement are mainly based on contact method. Noiseux (1970) used two accelerometers and one strain gauge to determine the flexural waves. Pavic (1976) also used the accelerometer to measure the velocity but in his study he adopted finite difference technique instead of the strain gauge to obtain the generalised forces. A complete formulation using this method requires a minimum of 4 sensors, 13 measurement points and 34 different measurements between the pairs of sensors in order to determine the amplitude and phase between the sensors. Many contact measurements use this principle. Verhiej (1980) adapted the finite-difference approach and used a frequency-domain, cross-spectral density formulation for the calculation of structural intensity from experimental measurements, which provided a more convenient formulation for practical measurement purposes.

The drawback of contact methods is obvious as the sensors may impose additional loading on the test structures, which will cause a change in the structural intensity map. In addition, if the number of measurement points is large, it will be very time-consuming to carry out the measurement.

The first non-contact structural intensity measurement method, called Structural Intensity from the Measurement of Acoustic Pressure (SIMAP), is based on the near-field acoustic holography (Williams, et al., 1985). The main drawback of this approach is the requirement for the test set-up. It is impractical to transport a holographic set-up so that in-field testing can be performed. However, it proposes a novel idea of utilising the spatial Fourier transform (SFT) and k -space low-pass filtering, rather than finite difference techniques, to calculate the required spatial derivatives of the velocity.

With the application of laser techniques in the measurements of vibration, a new non-contact method is proposed based on Laser Doppler Vibrometer (LDV) or scanning LDV. In the early stage, LDV is used to replace the accelerometers to measure the surface velocity (McDevitt and Koopmann, 1990; Hayek et al., 1990; Wagstaff, et al., 1990; Lee, et. al., 1993; Benes, et al., 1995; Morikawa et al., 1995). Only a few points on the surface are scanned. This method removes the effect of additional loading imposed by accelerometers. However, as the measurement principle is the same as the one used in contact method, the near-field effect and inter-phase error still exist. Scanning LDV uses electronically controlled mirrors to position the laser beam on the structure and are capable of high spatial density measurement of the vibrating surface. Recently, special interest has been paid to use scanning LDV to study structural intensity, its divergence, as well as to visualise the structural intensity within the structure (Spalding and Mann, 1995; Zhang and Mann, 1996(a), 1996(b); Blotter, et al., 2002; Nejade and Singh, 2002; Pascal, et al., 2002).

Apart from scanning LDV, interferometry technique has also been used to realise the high spatial density measurement of the vibrating surface and then to obtain the structural intensity distribution (Petzing and Tyrer, 1999; Chambard, et al., 2002). The use of scanning LDV and interferometry to measure structural intensity is similar, except that in-plane vibration can be determined more easily via interferometry than scanning LDV.

2.3 Various experimental data processing schemes

Whether contact or non-contact method is used, usually, only the vibration velocity or displacement information is recorded. So, some post processing must be done in order to extract the structural intensity from the experimental data. Based on the theoretical

expressions for the structural intensity, the data processing methods must be capable of calculating the spatial partial derivatives up to the third order or higher. Finite Difference approximation and Spatial Fourier Transform (SFT) are two frequently used methods to carry out such a job.

Finite Difference approximation is first used by Noiseux (1970). In his study, a measurement technique with four sensors based on far field and free field hypotheses is used, which leads to a simplified finite difference approximation formula. Substantial errors near the sources and discontinuities can be introduced due to the far field and free field hypotheses. In order to overcome the stringency of these hypotheses, Pavic (1976) presents a complete finite difference formulation for structural intensity in plate and beam. For this formulation, 34 cross-spectral measurements between 13 points are required to compute the two components of the intensity vector in a plate. This scheme is not practical for experimental procedures if the number of measurement points is large. The divergence of the structural intensity in a planar plate, which can well be used to locate mechanical excitation, is expressed in terms of higher order partial derivatives. Hence, another drawback of finite difference approximation is that it cannot obtain the divergence of the structural intensity due to the limitation of measurement points. In addition, finite difference approximation has a limitation by inter-transducer phase mismatch at low frequency and finite difference spacing errors at high frequency. Carroll (1990) shows that for moderate to lightly damped structures, phase accuracy on the order of 0.1 degree is required to measure the power flow accurately. This high phase accuracy requirement is necessary to separate the energy carried in the standing and travelling waves.

As mentioned in section 2.2, Williams et al. (1985) propose a novel idea of utilising the spatial Fourier Transform and k -space low-pass filtering, rather than finite difference techniques. By using the SFT and processing in the wave number domain, it is possible to obtain various spatial derivatives needed to calculate the intensity and its divergence. Although Williams's work is based on acoustic holography, the proposed data processing method can be easily applied to laser measurement. Pascal, et al., (1993) showed the combination of the advantages of SFT and direct measurement by laser vibrometry to obtain the structural intensity field over a plate-like structure. Application of SFT has been addressed further by Zhang and Mann (1996(a); 1996(b)). Some issues regarding the utilisation of k -space low-pass filtering and spatial windowing using band-pass k -filtering were discussed. Morikawa and Ueha (1996) also study the problem of plate intensity measurements using SFT. Through a comparative study, they establish the superiority of SFT over the finite difference method.

However, it is well known that the use of the SFT is often accompanied by large distortions if the signal has discontinuities in the space periodicity. If the SFT is directly used, the results are very sensitive to the boundary conditions and one cannot obtain a good estimation of the spectrum in wave number domain. As a result, significant error may exist in the calculated structural intensity field. A method called Regressive Fourier Transform is proposed to solve this problem (Arruda, 1992). With a better estimation of the wave number spectrum, the structural intensity can be calculated with higher accuracy. However, the algorithm is sometimes hard to be automatically used because of the trial-and-error parameters. Pascal et al., (2002) propose a mirror processing technique to overcome the leakage effects caused by non-periodical boundary conditions. A noticeable improvement on the quality of the spectrum can be obtained by using this

technique. Nejade and Singh (2002) discuss the effect of reflections from the plate edges and that of the uncorrelated noise on the intensity. In their study, SFT approach is also adopted to estimate flexural intensity in finite plates with arbitrary boundary conditions. The concept of spectral filtering instead of the low-pass filtering is implemented.

Although SFT can obtain better structural intensity estimation than finite difference approximation, a rectangular measurement zone without discontinuity is preferred in order to implement the two-dimensional Fourier transform and the inverse transform. For measurements on irregular geometry with holes and other discontinuities, it is difficult to apply the SFT to calculate the structural intensity. Regressive Discrete Fourier Transform can minimise the errors due to non-periodicity and is also able to deal with non-equally spaced and non-rectangular domain data (Roberto and Arruda, 1992). However, the algorithm is sometimes hard to be automatically used because of trial-and-error parameters. Blotter, et al., (2002) presented a technique known as Experimental Spatial Power Flow, which can provide spatially continuous representation of the power flow based on measurements from a scanning LDV. The essence of this technique is to find the continuous 3-D complex-valued velocity representation of the true structure velocity. The structural intensity of a steel plate was measured and calculated using this method. Theoretically, this technique can be extended to the determination of power flow distribution on any thin-walled structures. In his study, a simply supported rectangular plate has been studied. Although the best FEM mesh should be determined through comparing the convergence of the computed power between different mesh schemes, the idea of surface fitting provides a possibility of visualising the structural intensity in real complex structures.

2.4 Numerical simulation

Although structural intensity was first proposed as an experimental concept, it also can be predicted using numerical methods such as FEM, which is now widely used in engineering analysis. Hambric (1990) first investigates the computation of the structural intensity using FEM. Formulations for calculating input and output powers, power flows, and mechanical intensities for beam and plate/shell element types have been derived. In his study, NASTRAN is used to calculate the required velocity, force, and stress results of an analysis, from which the power flow quantities are calculated by a post processor. Test models include a simple truss and a beam-stiffened cantilever plate. Both test cases show reasonable power flow fields over low to medium frequencies.

In another FEM formulation presented by Gavic and Pavic (1990, 1993), mode summation is accepted to predict the displacement and stresses. High accuracy requirements of structural response are satisfied by employing the “swept static solution”, which effectively takes into account the influence of higher modes. This approach offers possibility to model rather complicated thin-walled structures and to calculate the intensity distribution. The drawback of this method is a large number of modes have to be used to obtain a good approximation of the structural intensity distribution, but this would increase the computer storage and computing time substantially.

The above two FEM formulations utilise beam, plate or shell element. However, for situations where wave numbers are significant in all directions or the beams and plates are connected by thick joints, the thin wall or shell theory becomes invalid. Therefore, three-dimensional intensity techniques should be developed to quantify accurately the power flow fields. The first effort to use solid finite elements to compute structural

intensity is performed by Hambric and Szwerc (1999) in a T-beam model. The capability is validated using the INCE standard Lexan T-beam. It is found that accurate modelling of the total cross-sectional power carried by flexural waves requires at least two and preferably three or four solid elements through the thickness of the cross section carrying the waves. For longitudinal waves, a single element should be sufficient. It has also been found that the surface intensity fields may give misleading indication about the power transmission path when flexural and longitudinal waves coexist in a structure. Nevertheless, more rigorous validations are needed for a more expansive solid structure.

For numerical prediction of structural intensity in complex structures, one challenge is to model various connections and joints between different sub-assemblies correctly. Rook and Singh (1998) calculate the structural intensity of a compliant plate-beam structure connected by bearings. In order to model the moment coupling introduced by the ball bearing, a new formulation for bearing stiffness matrix has been developed. In order to avoid the discontinuities of the stress values at the nodes, intensities are evaluated at the integration points. In some recent study, special attentions are also paid to model the abrupt changes and bolts connections (Cieslik, 2003; Khun, et al., 2004).

In order to overcome the disadvantage of FEM when dealing with high frequency problems and large complex spatial structures due to the prohibitive computational cost, spectral element method (SEM) is presented to predict the structural intensity in beams (Ahmida and Arruda, 2001). A structure of two connected beams is studied. One disadvantage of SEM is the available element library is too small.

It should be pointed out that, although the structural intensity can be predicted numerically, experimental determination is preferable since the dynamic loading and boundary conditions cannot be accurately modelled. In this study, FEM is used to study several ideal cases to validate the experimental method and explore the effects of mechanical modification on the change of structural intensity distribution.

2.5 Applications of structural intensity technique

2.5.1 Applications in industry

Structural intensity has the same physical meaning as the acoustical intensity and the measurement techniques for determination of structural intensity in beams and plates have been studied for about 30 years. It is believed that the structural intensity technique has a wide range of potential applications, i.e., visualising and predicting vibratory energy flows in complex structures, spotting vibration sources and sinks, ranking various propagation paths and characterisation of damping treatments of decoupling systems, acting as a diagnostic indicator of machinery failure (Tourret, 1990; Banks and Hambric, 2000; Li and Lai, 2000). However, industrial applications of structural intensity have had a slower development than the corresponding acoustic intensity methods. One reason is the difficulty of measuring structural intensity in real situations and in the definition of the transmission paths in the case of relative complex structures (Bobrovnitskii, 1993).

Moreover, most experimental studies are carried out on homogeneous thin and thin-walled structures – beams, plates, shells and pipes (Gavric, et al., 1997; Audrain, et al., 2000; Freschi, et al., 2000). Small structural thickness implies the linear distribution of stresses, reducing the dimension of the problem. In industry, many structures, however,

are not thin-walled or are extremely inhomogeneous and complex. The existing experimental technique is not directly applicable and more powerful approaches are necessary. One example is to visualise the power flow inside an ultrasonic tool using a hybrid method (Koike, et al., 2000). Because the three-dimensional measurement of the entire vibration distribution of the ultrasonic tool is impossible, a FEM model together with the partial vibration data measured at the surface of the tool was used to calculate the power flow. Since such a problem is ill-posed, improvement is required to obtain precise power flow vector maps.

2.5.2 Applications in HDD

One major objective of this project is to establish a methodology to study how the vibration energy is transmitted within the HDD system, so that strategic measures can be taken to satisfy the vibration requirement for ultra high track density. The construction of a typical HDD is shown in Fig. 2.1. Data is stored as magnetic transitions on the thin media layer that coats the glass disks. The arm pivots about a fixed bearing, and it is driven by a voice coil motor that swings between two permanent magnets. The read/write heads are located at the tip of the arm above each disk surface. The whole HDD is mounted on a compliant chassis cabinet (the mounting system), which is not shown in this figure.

According to the above discussions, structural intensity indicates the magnitude and direction of the vibratory energy transmitted in the system. Once the structural intensity distribution inside the HDD is known, proper mechanical modifications or damping treatments can then be adopted to divert the vibratory energy. As a result, structural intensity may be a very powerful tool to solve the vibration transmission problem in

HDD for ultra high track density. However, to the author's knowledge, the application of structural intensity to HDD is so far rare. Hence, a systematic study should be carried out to establish a methodology to determine the structural intensity distribution in structures such as HDD base casing. Also, ways concerning how to divert or contain the structural intensity distribution should also be investigated.

Fig. 2.1 Certain mechanical components within a typical hard disk drive

Chapter 3

Structural intensity and its calculation

This chapter presents the concept of structural intensity, its expressions in thin-walled structures, and various data processing methods in its experimental determination. B-spline surface fitting method is then proposed and issues on its use are investigated. One analytical case is given to exemplify the basic concepts. In this project, FEM is used to study several ideal cases to validate the experimental method and explore the effects of mechanical modification on the change of structural intensity distribution. Hence, the procedure of numerical simulation using ANSYS® is also studied and validated by comparison between the simulated structural intensity with the analytical one.

3.1 Introduction

The concept of structural intensity is introduced to extend the vector acoustics approach to energy flow in structure-borne sound fields. Acoustic intensity technique has brought a revolution in the field of acoustics. Now the acoustic energy density and vector power flow can be directly measured in air by commercially available intensity meters, which greatly widen the possibilities in studying complex acoustic fields and increase the number of practically solvable problems. Source locating on mechanical equipment or machine sound power determination in severe environments now becomes feasible as well as in situ acoustical materials performance evaluation, quantification of various propagation paths and understanding of the complex sound radiation mechanisms.

Structural intensity is defined as the net energy flow per unit area in the given directions. Obviously, this definition of structural intensity has the same physical meaning as the acoustical intensity in a fluid medium. The instantaneous structural intensity can be defined by (Gavric and Pavic, 1993),

$$i_k(t) = -\sum_l \sigma_{kl}(t) v_l(t) \quad l = 1, 2, 3 \quad (3.1)$$

where $v_l(t)$ is velocity in l^{th} direction and $\sigma_{kl}(t)$ is the corresponding stress.

The time-averaged structural intensity of the k^{th} instantaneous intensity component, I_k , represents the net power flow through the structure,

$$\begin{aligned} I_k &= \langle i_k(t) \rangle \\ &= -\sum_l \langle \sigma_{kl}(t) v_l(t) \rangle_t, \quad k, l = 1, 2, 3. \end{aligned} \quad (3.2)$$

where $\langle \Lambda \rangle$ denotes time averaging.

For steady-state vibrations, structural intensity in k^{th} direction at frequency ω is defined as,

$$\tilde{\Pi}_k(\omega) = -\frac{1}{2} \sum_l \tilde{\sigma}_{kl}(\omega) \tilde{v}_l^*(\omega) = I_k(\omega) + jJ_k(\omega) \quad l = 1, 2, 3 \quad (3.3)$$

where $\tilde{v}_l^*(\omega)$ is the complex conjugate of the velocity and $\tilde{\sigma}_{kl}(\omega)$ is the complex stress.

The real part of the quantity $I_k(\omega)$ is the net energy flow, i.e. the structural intensity defined in Eq.(3.2), at frequency ω in the structure. The imaginary part $J_k(\omega)$ is called reactive intensity and its contribution to net energy flow is zero. The real part of Eq.(3.3), the structural intensity, is associated with the propagative vibration fields, while reactive intensity characterises the reverberance in the field (Honda and Irie, 1990; Alfredsson, 1993). Further study shows that the reactive intensity is proportional to the time averaged

Lagrangian energy, i.e. the difference of kinetic and potential energies, and the proportionality between the reactive intensity and the difference of kinetic and potential energies has already been established (Alfredsson, 1997; Pavic, 2003).

It is worth noting that the net intensity $I_k(\omega)$ integrated over all frequency components is equal to structural intensity defined by time-averaging in time-domain in Eq.(3.2),

$$I_k = \langle i_k(t) \rangle = \int_{\omega} I_k(\omega) d\omega \quad (3.4)$$

The relationship between the structural intensity I_k , the complex structural intensity $\tilde{\Pi}_k$, and the instantaneous structural intensity $i_k(t)$ can be established for a single frequency component (Gavric, et al., 1990). Suppose the complex stresses $\tilde{\sigma}$ and velocity $\tilde{v}$ have phase shifts ψ and ϕ respectively. Then the following relationship holds,

$$i_k(t) = I_k - |\tilde{\Pi}_k| \cos[2(\omega t + \varphi)] \quad (3.5)$$

where $|\tilde{\Pi}_k| = \sqrt{I_k^2 + J_k^2}$ is the absolute value of the complex structural intensity,

$\varphi = \frac{1}{2}(\psi + \phi)$ is the phase of complex intensity.

It is clear that the instantaneous intensity is equal to the rate of change of energy density in a given point of structure. Based on Eq.(3.5), the change can be divided into two parts: one constant in time, equal to the structural intensity; and the other one equal to the absolute value of complex intensity, fluctuating sinusoidally at a frequency twice as much as the excitation frequency.

3.2 Structural intensity in thin-walled structures

The expressions for the structural intensity, its divergence and excitation force distributions in thin-walled structures have been well documented in literatures (Noiseux, 1970; Pavic, 1976; Romano, et al., 1990; Gavric and Pavic, 1993, Zhang and Mann III, 1996b). Therefore, only a brief outline is presented in this section.

3.2.1 Formulations of structural intensity in thin-walled structures

In thin-walled structures (plates, beams, shells and their assemblies) the stress distribution through the thickness (shell and plates) or through the cross-section (beams) is assumed to be linear. Displacements of any point of a thin-walled structure can be expressed by translational and angular displacements of the middle surface. Therefore, the structural intensity can be completely defined by stress resultants and translational and angular displacements of the middle surface. For shells and plates, structural intensity may be conveniently defined as the net energy flow per unit length in given direction (structural intensity integrated over the thickness):

$$I_x = -\frac{\omega}{2} \text{Im} [\tilde{N}_x \tilde{u}_0^* + \tilde{N}_{xy} \tilde{v}_0^* + \tilde{Q}_x \tilde{w}_0^* + \tilde{M}_x \tilde{\theta}_y^* - \tilde{M}_{xy} \tilde{\theta}_x^*] \quad (3.6a)$$

$$I_y = -\frac{\omega}{2} \text{Im} [\tilde{N}_y \tilde{v}_0^* + \tilde{N}_{yx} \tilde{u}_0^* + \tilde{Q}_y \tilde{w}_0^* - \tilde{M}_y \tilde{\theta}_x^* + \tilde{M}_{xy} \tilde{\theta}_y^*] \quad (3.6b)$$

where I_x and I_y are the x and y components of the structural intensity; $\tilde{N}_x, \tilde{N}_y, \tilde{N}_{xy} = \tilde{N}_{yx}$ are the complex membrane forces; $\tilde{M}_x, \tilde{M}_y, \tilde{M}_{xy} = \tilde{M}_{yx}$ are the complex bending and torsion moments; $\tilde{Q}_x$ and $\tilde{Q}_y$ are the complex shear forces; $\tilde{u}_0^*, \tilde{v}_0^*$ and $\tilde{w}_0^*$ are the complex conjugate displacements in x, y and z directions; $\tilde{\theta}_x^*$ and $\tilde{\theta}_y^*$ are the complex conjugate rotations about x and y axes. Fig. 3.1 shows the positive orientations of internal forces

and displacements used to formulate the structural intensity in Eq.(3.6). Formulations for beams and shells can be found in reference (Gavric and Pavic, 1993).

In Eq.(3.6), both in-plane and out-of-plane waves are included in the expression of structural intensity. If only out-of-plane waves are considered, Eq.(3.6) then can be simplified as,

$$I_x = -\frac{\omega}{2} \text{Im} [\tilde{Q}_x \tilde{w}_0^* + \tilde{M}_x \tilde{\theta}_y^* - \tilde{M}_{xy} \tilde{\theta}_x^*] \quad (3.7a)$$

$$I_y = -\frac{\omega}{2} \text{Im} [\tilde{Q}_y \tilde{w}_0^* - \tilde{M}_y \tilde{\theta}_x^* + \tilde{M}_{xy} \tilde{\theta}_y^*] \quad (3.7b)$$

Using Kirchoff plate theory, the generalized forces in Eq.(3.7) can be expressed in terms of spatial derivatives of the normal displacement w_0 . Considering the relationship between vibration displacement and vibration velocity, the structural intensity defined in Eq. (3.7) (power flow per unit length) can be expressed equivalently in terms of the normal velocity $v(x, y)$,

$$I_x = \frac{B}{2\omega} \text{Im} \left\{ \frac{\partial}{\partial x} (\nabla^2 \tilde{v}) \tilde{v}^* - \left(\frac{\partial^2 \tilde{v}}{\partial x^2} + \nu \frac{\partial^2 \tilde{v}}{\partial y^2} \right) \frac{\partial \tilde{v}^*}{\partial x} - (1-\nu) \frac{\partial^2 \tilde{v}}{\partial x \partial y} \frac{\partial \tilde{v}^*}{\partial y} \right\} \quad (3.8a)$$

$$I_y = \frac{B}{2\omega} \text{Im} \left\{ \frac{\partial}{\partial y} (\nabla^2 \tilde{v}) \tilde{v}^* - \left(\frac{\partial^2 \tilde{v}}{\partial y^2} + \nu \frac{\partial^2 \tilde{v}}{\partial x^2} \right) \frac{\partial \tilde{v}^*}{\partial y} - (1-\nu) \frac{\partial^2 \tilde{v}}{\partial x \partial y} \frac{\partial \tilde{v}^*}{\partial x} \right\} \quad (3.8b)$$

where $B = Eh^3/12(1-\nu^2)$ is the bending stiffness coefficient; E is Young's modulus, h is the plate thickness and ν the Possion's ratio.

Therefore, the structural intensity in plates can be determined solely from knowledge of the out-of-plane movements, in the form of products of different space derivative of the vibration velocity in the normal direction.

The structural intensity defined in Eq.(3.8) may be written in terms of the three wave types that propagates in the plate, namely, bending, twisting, and shearing waves. For example, the structural intensity in the x direction can be written as,

$$I_x = I_{x,bending} + I_{x,twisting} + I_{x,shearing} \quad (3.9)$$

where,

$$I_{x,bending} = -\frac{B}{2\omega} \text{Im} \left\{ \left(\frac{\partial^2 \tilde{v}}{\partial x^2} + \nu \frac{\partial^2 \tilde{v}}{\partial y^2} \right) \left(\frac{\partial \tilde{v}^*}{\partial x} \right) \right\} \quad (3.10a)$$

$$I_{x,twisting} = -\frac{B}{2\omega} \text{Im} \left\{ (1-\nu) \frac{\partial^2 \tilde{v}}{\partial x \partial y} \frac{\partial \tilde{v}^*}{\partial y} \right\} \quad (3.10b)$$

$$I_{x,shearing} = \frac{B}{2\omega} \text{Im} \left\{ \frac{\partial}{\partial x} (\nabla^2 \tilde{v}) \tilde{v}^* \right\} \quad (3.10c)$$

It should be noted that Eq.(3.8) to Eq.(3.10) are based on the Kirchhoff plate theory. This plate model neglects the effect of rotary inertia and shear deformations. Mindlin (1951) suggested a new plate theory for the flexural motion of isotropic and elastic plates including the influence of rotary inertia and shear deformation. The structural intensity formulations can also be developed based on Mindlin plate theory (Romano, et al.,1990). Zhang and Mann III (1996a, 1996b) investigate the two formulations and conclude that at low frequencies, where such effects are negligible, both formulations produce the same results. The usual condition $\lambda = 6h$ can be used to define the upper limit of validity of these derivations. Here, λ is the wavelength of interest, and h is the thickness of the plate concerned.

3.2.2 Injected power

In order to compute the injected power, Eq.(3.8a) and Eq.(3.8b) are combined into one equation using the 3D nabla operator,

$$I = \frac{B}{2\omega} \text{Im} \left\{ \nabla(\nabla^2 \tilde{v}) \tilde{v}^* - \nabla^2 \tilde{v} \nabla \tilde{v}^* - \frac{1-\nu}{2} \nabla \times \nabla \times (\tilde{v} \nabla \tilde{v}^*) \right\} \quad (3.11)$$

Assume an energy flow $W(x, y)$ crosses the surface of the plate. This flow may be produced by mechanical excitation (source) and local or distributed damping on the structure (sinks: dissipating elements, or, if applicable, acoustic radiation). Then, it can be demonstrated that the divergence of structural intensity in the plate is equal to the surface density of injected (or dissipated) power (Pascal, et al., 2002). That is,

$$\nabla \cdot I(x, y) = W(x, y) \quad (3.12)$$

Substituting Eq. (3.11) into Eq. (3.12), yields,

$$\nabla \cdot I = \frac{B}{2\omega} \text{Im} \left\{ \left(\nabla^4 \tilde{v} \right) \tilde{v}^* \right\} \quad (3.13)$$

Thus, the divergence of structural intensity can be determined from the normal vibration velocity in a plate.

3.2.3 Structural intensity in thin plate: an analytical example

So far, the concept of structural intensity and its measurement techniques have been introduced. In this section, a case of structural intensity due to point excitation and an attached damper, which appears in literature (Gavric and Pavic, 1993), is given to demonstrate the physical meaning of structural intensity and its superiority over modal

analysis in the study of vibratory energy transmission. In order to illustrate the relationship between structural intensity and modal shapes, the mode summation method is used to compute the structural intensity analytically.

The test object is a simply supported plate with an attached viscous damper, excited by a sinusoidal force. The plate is $3m$ long and $1.7m$ wide with a thickness of $1cm$. The material properties used for the plate are as follows: the Young's modulus is $210GPa$, the Poisson's ratio is 0.3 and the mass density is $7800kg/m^3$. The plate is assumed to be dissipationless, i.e. without structural damping. The excitation with an amplitude of $1000N$ is applied at $x=0.6m$ and $y=0.4m$ as shown in Fig. 3.2. A viscous damper with the damping rate of $100Ns/m$ is attached to the plate at $x=2.2m$ and $y=1.2m$. The structural intensity is calculated for excitation frequency of $50Hz$. The first 10 modes calculated by FEM are given in Table 3.1. It is evident that the excitation frequency, $f=50\text{ Hz}$, is between the (2,2) mode and (4,1) mode. Fig. 3.3 shows the corresponding four mode shapes adjacent to the excitation frequency, i.e. (1,2), (2,2), (4,1) and (2,3) modes.

For a simply supported plate, the complex response of the plate can be computed analytically using the mode summation method. After the complex displacement field and velocity field are obtained, the structural intensity field is evaluated by using Eq.(3.8).

Fig. 3.4 shows the spatial distribution of structural intensity computed with 20×20 modes. It can be seen that the position of excitation force (energy source) and damper (energy sink) are clearly indicated in the total structural intensity plot (Fig. 3.4a). Furthermore, as expected, the structural intensity vectors clearly indicate the energy transmission path from the source to the sink. In order to indicate the energy transmission

path more clearly, the structural intensity is represented with uniform vectors in Fig. 3.4b. This figure indicates that energy transmission path within the plate is consistent with the definition of structural intensity described in section 3.1. Different components of the total structural intensity are also shown in Fig. 3.4 (c)-(e) for bending, shear and twisting. For the purpose of comparison, the velocity distribution is shown in Fig. 3.5. Unlike the plot of structural intensity, the information contained in the velocity distribution is at best ambiguous, and neither the source/sink nor the energy transmission path can be identified from it. Likewise, the source/sink and energy transmission path cannot be identified from the mode shapes shown in Fig. 3.3 either. That is to say, the classic modal analysis cannot provide the energy transmission information as structural intensity does.

Fig. 3.6 shows the reactive part of the computed intensity. As its contribution to the net energy transfer is zero, the energy transmission path cannot be identified from this figure. At the same time, the positions of source/sink cannot be identified either. However, it does not mean that the study of reactive intensity has little significance. For example, according to Spalding and Mann III's experiment (1995), if small constrained layer damping patches are applied to a region of low shearing reactive structural intensity magnitude, a local velocity change can be achieved; on the other hand, if the shearing reactive intensity magnitude is high, then a global velocity change obtained.

Gavric and Pavic (1993) have studied the convergence performance of mode summation when structural intensity is concerned. Spatial distributions of intensity are computed with different number of modes. The net power flow per unit width over the closed curve containing the source/sink is also obtained through integration. Because the plate is assumed to be dissipationless, the whole injected power should be transferred to the sink

and then dissipated. Thus, difference between the energy injected into the system and energy propagating towards the damper can be used to evaluate the convergence of the mode summation. Based on the above computation, they conclude that much more modes have to be used to obtain a good approximation of the structural intensity distribution than to compute the structural response. This conclusion is helpful for choosing proper parameters when carrying out numerical simulation or experimental data processing to extract the structural intensity.

The integrated power flows expressed as percentage of input power are listed in Table 3.2. The structural intensity distribution with 6×6 modes is also given in Fig. 3.7. The positions of source and sink can not be located clearly with less modes used, as compared to that shown in Fig. 3.4(a), although the number of modes is sufficient to compute structural response.

In conclusion, the analytical case shows structural intensity has significant advantages over RMS velocity measurement and modal analysis approach if vibration transmission is of interest. With the aid of structural intensity, the energy sources/sinks, as well as energy transmission path can be easily determined. The knowledge of these factors will broaden the understanding of the dynamic properties of the structures and play a useful part in the noise and vibration reduction treatment.

3.3 Experimental determination of structural intensity

As mentioned in Chapter 2, the structural intensity measurement method can be grouped into two main categories, i.e. contact and non-contact. Contact method usually adopts contacting transducers such as accelerometers and strain gauges, while for non-contact

method, laser, interferometry or near-field acoustic holography could be used. Due to the advantages of non-contact method discussed previously, and the difficulty to attach enough transducers to the hard disk drive concerned, scanning LDV, one no-contact method, is used to carry out the vibration measurement in this work.

Scanning LDV uses electronically controlled mirrors to position the laser beam on the structure and is capable of high spatial density measurement of the velocity on the vibrating surface. According to Eq.(3.8), the structural intensity in plates can be determined solely from knowledge of the transverse movements, in the form of products of different space derivative of the vibration velocity in the normal direction. Hence, Scanning LDV can prepare all the data needed to determine the structural intensity in the test object.

With the velocity data measured, certain post processing must be done to calculate the spatial partial derivatives up to the third order in Eq. (3.8). To do so, a novel data processing method, B-spline surface fitting (BSSF), is developed to extract the structural intensity from the measured velocity data. In this section, Finite Difference approximation and SFT, which are usually used, are first introduced for the relationship between the proposed new method and the current ones. A comprehensive study of the use of BSSF in structural intensity is given in section 3.4.

3.3.1 Finite difference approximation

Finite difference (FD) technique is often used to determine the space derivatives in Eq.(3.8). As the structural intensity vectors are expressed by partial derivatives of velocity up to third order, standard finite difference method may cause large errors. By

applying the far field and free field hypotheses (Noiseux, 1970), the structural intensity defined by Eq. (3.8) can be reduced to a simplified formula,

$$I \approx \frac{B}{2\omega} \text{Im}\{2k_B^2 v \nabla v^*\} = \sqrt{BM_s} \text{Im}\{v \nabla v^*\} \quad (3.14)$$

where $k_B = (\omega)^{1/2} (M_s / B)^{1/4}$ is the bending wave number in the plane and M_s is the mass per unit area of the plate.

Because this approximation uses the far field and free field hypotheses, substantial errors may be caused near the sources and discontinuities. The formulation of structural intensity using standard finite difference is omitted in this dissertation. It can be found in reference (Pavic, 1976).

3.3.2 Spatial Fourier Transform (SFT)

The limitations caused by finite difference approximation for determination of structural intensity and its divergence can be overcome by the use of wave number processing technique. The principle of this method is to calculate the spatial derivatives by the SFT. For instance, when making use of the Cartesian coordinate system, the basic formula used to calculate a spatial derivative is as follows:

$$\frac{\partial^m \partial^n}{\partial x^m \partial y^n} [w(x, y)] = F^{-1} \{ (jk_x)^m (jk_y)^n F[w(x, y)] \} \quad (3.15)$$

where k_x and k_y are the wave numbers in $\hat{k}_x$ and $\hat{k}_y$ directions respectively, F and F^{-1} represent the direct and inverse 2-D Fourier transforms, defined as,

$$W(k_x, k_y; \omega) = F\{w(x, y; \omega)\} = \frac{1}{4\pi^2} \iint w(x, y; \omega) e^{-j(k_x x + k_y y)} dx dy \quad (3.16a)$$

$$w(x, y; \omega) = \iint W(k_x, k_y; \omega) e^{j(k_x x + k_y y)} dk_x dk_y \quad (3.16b)$$

Assume $V(k_x, k_y)$ is the spatial Fourier transform of normal velocity field $\tilde{v}(x, y)$. Let $K = k_x \hat{k}_x + k_y \hat{k}_y$. By using Eq.(3.15), the structural intensity defined in Eq.(3.11) can then be expressed as,

$$I(x, y) = \frac{B}{2\omega} \text{Im} \left\{ F^{-1} \left\{ jK^3 V(k_x, k_y) \right\} \tilde{v}^* - F^{-1} \left\{ K^2 V(k_x, k_y) \right\} F^{-1} \left\{ jKV^*(k_x, k_y) \right\} \right. \\ \left. + \frac{1-\nu}{2} F^{-1} \left\{ K \times K \times F \left\{ \tilde{v} \nabla \tilde{v}^* \right\} \right\} \right\} \quad (3.17)$$

Similarly, the divergence of structural intensity is expressed as,

$$\nabla \times I(x, y) = -\frac{B}{2\omega} \text{Im} \left\{ F^{-1} \left\{ K^4 V(k_x, k_y) \right\} F^{-1} \left\{ V^*(k_x, k_y) \right\} \right\} \quad (3.18)$$

It is evident that any measurement noise at high k_x or k_y will be magnified greatly. So, it is necessary to filter the noise in k -space before applying the inverse Fourier transform to get different orders of partial derivatives. Usually, an oval k -filter is used. As a rule of thumb, the filter size is chosen so that majority of large amplitude data in the centre of the wave number domain is kept (Spalding and Mann III, 1995).

Another problem associated with SFT is that when applied to truncated signal, the finite aperture restriction leads to an error of wraparound on the wave number spectrum, especially when the vibrating velocity is not zero at the edges of the plate. A plate with free-free boundary conditions is one of such cases. In order to solve this problem, a technique called mirror method has been developed (Pascal, et al., 2002). The idea of the mirror method is to build a continuous and periodic signal from the original one to be processed by SFT. The original image is mirrored with respect to its end using a link function. Therefore, the resulting signal becomes continuous and periodic in the end. In

Pascal's formulation, the image signal is multiplied by a cosine function before connecting the original signal,

3.4 Application of BSSF in structural intensity measurement

In order to overcome the above-mentioned drawbacks of FD and SFT and expand structural intensity measurement to more general structures, a novel approach, BSSF (B-spline surface fitting), is developed to process the measured velocity data. The basic procedure of the use of B-spline surface fitting in structural intensity calculation is established. Meanwhile, the low-pass filtering property of B-spline approximation is addressed, which makes a foundation for a further study of the performance of the proposed data processing scheme, BSSF, in next chapter.

3.4.1 Mesh based least-square surface fitting

The surface fitting concept is well suited to estimating multiple-order derivatives. After the data acquisition in an experiment, there is often a need to represent and/or differentiate the experimental data that has been recorded only at discrete locations within the region. To accomplish this task, the so-called surface fitting technique can be employed. Suppose the dependent variable that needs to be represented continuously is the normal velocity v . Consider a two-dimensional field throughout some region R . The procedure of finite element (FE)-based least-square surface fitting can be briefly expressed as follows:

Firstly, a FE mesh is established over region R . The basis function is selected in such a way that the required continuity of the dependent variable is guaranteed. The unknowns

in the mesh, such as nodal displacements, may represent the velocity value at that point, or just act as the FE model parameters without having any physical meaning. The nodal displacements can be determined by minimising the squared residuals between the known solutions, that is the measured velocity, and the approximated solution obtained from the FE model. The velocity and its derivatives at any point in any element can then be computed.

In order to compute the structural intensity and the injected power, spatial derivatives up to third and fourth order are needed respectively. Therefore, in this study, bivariate tensor product quintic B-spline is employed as the basis function.

3.4.2 B-spline and its tensor product

B-spline curve is constructed by a linear combination of polynomial segments or basis functions defined in uniform parametric space from 0 to 1. The basis functions are derived such that continuity through each individual segment of the curve and continuity between the curves is guaranteed.

The order of the basis functions determines the order of continuity of the curve. In order to have C^m continuity at the knots, $m+2$ basis functions of order $m+1$ are required. Knots are the parametric values at which the B-spline segments are joined. Control points are weighting coefficients on the B-spline basis functions much like the nodal coordinates in a conventional finite element model. The $m+2$ control points are required to define a C^m segment of the curve. The control points are simply weighting coefficients and do not represent the measured values at the knots. The entire B-spline curve $s(x)$ is expressed

as a linear combination of the control points, c_i , and basis functions, $\beta_i(x)$, as shown by Eq. (3.19).

$$s(x) = \sum_{i=-\infty}^{\infty} c_i \beta_i(x) \quad (3.19)$$

For a quintic B-spline curve, at least 12 knots, 6 control points and 6 fifth-order basis functions are required to define a section of the curve. The relationships between the knots, control points and the defined section are shown in Fig. 3.8. The defined section of the curve is the sum of all 6 curves between knot 6 and knot 7.

For quintic B-spline, 36 coefficients are needed to define the quintic B-spline basis functions. Based on the definition and properties of B-splines (de Boor, C., 1978), the coefficients can be computed and shown as follows,

$$\begin{Bmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \\ \beta_5 \\ \beta_6 \end{Bmatrix} = \begin{bmatrix} 1/120 & 0 & 0 & 0 & 0 & 0 \\ -1/24 & 1/24 & 1/12 & 1/12 & 1/24 & 1/120 \\ 1/12 & -1/6 & -1/6 & 1/6 & 5/12 & 13/60 \\ -1/12 & 1/4 & 0 & -1/2 & 0 & 11/20 \\ 1/24 & -1/6 & 1/6 & 1/6 & -5/12 & 13/60 \\ -1/120 & 1/24 & -1/12 & 1/12 & -1/24 & 1/120 \end{bmatrix} \cdot \begin{Bmatrix} x^5 \\ x^4 \\ x^3 \\ x^2 \\ x \\ 1 \end{Bmatrix} \quad (3.20)$$

In order to represent surface $s(x, y)$, the quintic B-spline has to been extended in some way to functions of more than one variable. The so-called tensor product splines are certainly the easiest to handle. The quintic B-spline surface $s(x, y)$ using tensor product splines can be expressed as,

$$s_{ij}(x, y) = \sum_{k=1}^6 \sum_{l=1}^6 \alpha_{i+k}(x) \beta_{j+l}(y) \quad (3.21)$$

A certain drawback of tensor product splines is that the problem domains are restricted to rectangular. In order to overcome this problem, subdivision based on the concept of finite element can be implemented. Of course there are many other methods that can manipulate arbitrary domains. However, they are more complex and therefore computationally less attractive.

3.4.3 Procedure of BSSF

In order to obtain a continuous, complex-valued velocity representation of the true structure velocity, a finite element formulation proposed by Montgomery, et al., (1994) is employed. This method provides a surface fit of the velocity data acquired by the scanning LDV. Since only normal velocity is concerned, a simplified procedure to obtain the smooth representation v from the measured velocity field, $\hat{v}_L$, can be expressed as follows.

Let v is the continuous representation of the measured velocity $\hat{v}_L$, and,

$$v_i = v(x_i, y_i) = \sum_{j=1}^{\hat{m}} N_j(x_i, y_i) \alpha_j \quad (3.22)$$

where $N_j(x, y)$ is basis function defined using tensor production, α_j is nodal velocity, $\hat{m}$ is the number of control points used to define each element. For the element defined by quintic B-splines, $\hat{m}=36$.

The nodal velocity α_j can be solved by minimising the errors between the measured velocity and the approximated one. The squared residual error Π is defined as,

$$\Pi = \sum_i^{\hat{n}} (v_i - \hat{v}_{Li})^2 \quad (3.23)$$

Here, $\hat{n}$ is the total number of measured velocity data.

Substituting Eq.(3.22) into Eq.(3.23) and minimising gives,

$$\frac{\partial \Pi}{\partial \alpha_j} = 2 \sum_{i=1}^{\hat{n}} e_i N_j(x_i, y_i) = 0 \quad (3.24)$$

and,

$$e_i = \sum_{k=1}^{\hat{m}} N_k(x_i, y_i) \alpha_k - \hat{v}_{Li} \quad (3.25)$$

$$\sum_{i=1}^{\hat{n}} \begin{bmatrix} N_{1i} N_{1i} v_{Li} & N_{1i} N_{2i} v_{Li} & \Lambda & N_{1i} N_{\hat{m}i} v_{Li} \\ & N_{2i} N_{2i} v_{Li} & \Lambda & N_{2i} N_{\hat{m}i} v_{Li} \\ & & \text{O} & \\ \text{symmetric} & & & N_{\hat{m}i} N_{\hat{m}i} v_{Li} \end{bmatrix} \begin{Bmatrix} \alpha_1 \\ \alpha_2 \\ \text{M} \\ \alpha_{\hat{m}} \end{Bmatrix} = \sum_{i=1}^{\hat{n}} \begin{Bmatrix} N_{1i} \hat{v}_{Li} \\ N_{2i} \hat{v}_{Li} \\ \text{M} \\ N_{\hat{m}i} \hat{v}_{Li} \end{Bmatrix} \quad (3.26)$$

Eq.(3.26) has similar expression of the governing equation of structural finite-element analysis and it can be rewritten in a simple form,

$$[K]_E \{D\}_E = \{P\}_E \quad (3.27)$$

$[K]_E$ and $\{P\}_E$ are called the pseudo element stiffness matrix (ESM) and the pseudo load vector, respectively. The pseudo ESM and load can be assembled together based on the connectivity information of the finite element mesh. The nodal velocities α_i can then be obtained using a typical finite element procedure. Thus, the velocity at any point in any element can be computed from Eq.(3.22). As the basis functions are continuous up to fourth order, the derivatives needed to compute structural intensity can be easily obtained through differentiating Eq.(3.22).

3.4.4 Low-pass filtering property of B-spline approximation

The B-spline approximation procedure described in section 3.4.3, which relies on the use of standard least square approximation techniques, can be used to fit arbitrarily spaced measurements data. In doing so, the proper number and position of the knots and control points must be determined. However, there is no known efficient method for providing an optimal solution for the B-spline approximation problem assuming that both the knot locations and control points are priori unknown. Fortunately, this issue can be simplified in this study because the data to be approximated have some known properties. The mesh density corresponding to the knots location and control points can be determined by considering the low-pass filtering effects of B-spline approximation.

To illustrate the low-pass filtering effects, take the curve fitting with uniform spaced knots. Because the basis function $b_i(x)$ can be constructed by integer shifts of a single function $\beta(x)$, Eq (3.19) can be rewritten as,

$$s(x) = \sum_{i=-\infty}^{+\infty} c_i \beta(x-i) \quad (3.28)$$

The interpolation function, $s(x)$, can be expressed in terms of the discrete function values themselves:

$$s(x) = \sum_{k=-\infty}^{+\infty} s(k) \eta(x-i) \quad (3.29)$$

where $\eta(x)$ is the cardinal spline with the same order of $\beta(x)$. Function $\eta(x)$ represents the continuous impulse response of the polynomial spline interpolator. In the approximation theory literature, this function is also sometimes referred to as the fundamental spline.

Evaluating the function $s(x)$ in Eq.(3.29) at an integer value n , gives,

$$s(n) = \sum_{k=-\infty}^{+\infty} s(k)\eta(n-k) \quad (3.30)$$

which has the exact form of a discrete convolution. The cardinal spline $\eta(x)$, evaluated at integer values, is digitally convolved with the sampled values of the function $g(x)$. Therefore, $\eta(x)$ works as a digital filter. In addition, the frequency response of n^{th} order B-spline interpolator $\eta^n(x)$ can be computed (Aldroubi, et al, 1992). Figs.3.9 and 3.10 display $\eta^3(x)$, $\eta^7(x)$ and their continuous spectra respectively. With the order n increases, $\eta^n(x)$ converges to an ideal low-pass filter.

For the least square spline used in this study, Unser et al. (1993) demonstrate that when dealing with discrete signals, the approximation is conceptually similar to re-sampling a signal at a lower rate. So, the control points and knots can be chosen according to the band limit of the data to be processed and the low pass effect of the cardinal spline interpolator. Although only one-dimensional case is considered here, the results are directly applicable to two-dimensions through the use of tensor product splines.

It is important to realize the difference between B-splines and the corresponding cardinal splines. For illustration, Fig. 3.11 shows a $\beta^5(x)$ and its spectrum. More discussion can be found in the reference (Aldroubi, et al., 1992).

3.5 Numerical simulation of structural intensity

Different finite element analysis software packages can be used to calculate the field variables of the structural intensity model in Eq. (3.7). The commercial FEM code

ANSYS is employed for all the simulation in this study. The correctness of this method is justified in this section.

The steady state dynamic analysis is used to obtain the magnitude and phase angle of the response of a harmonically excited system. As only the out-of-plane wave is considered, the plate is modelled using eight-node iso-parametric shell elements. The necessary internal forces and moments are obtained through numerical integration. The full method for harmonic response solution is chosen. Unlike the mode summation method, the response is directly computed from the mass, damping and stiffness matrices of the system without any simplifications.

In order to demonstrate the correctness of the FEM simulation of structural intensity, the case in section 3.2.3 is re-examined using ANSYS. A mesh scheme much finer than the one necessary to compute the structural response is adopted to provide accurate stress outputs. The structural intensity field computed is shown in Fig. 3.12. It is quite clear that the numerical results in Fig.3.12 are in good agreement with the analytical results in Fig. 3.4(a). Hence, the numerical results can be assumed to represent the real structural intensity fields.

Table 3.1 The first 10 modes of a simply supported steel plate

Mode number	Frequency (Hz)
(1,1)	11.26
(2,1)	19.46
(3,1)	33.14
(1,2)	36.85
(2,2)	45.02
(4,1)	52.31
(2,3)	58.67
(5,1)	76.95
(4,2)	77.80
(2,5)	79.49

Table 3.2 Integrated energy flow expressed as percentage of input power
(Gavric and Pavic, 1993)

Modes used in computation	Input/output Power(%)	Intensity around the force (%)	Intensity around the damper (%)
2×2	53	4	5
4×4	99	26	19
6×6	100	42	53
8×8	100	76	65
15×15	100	118	120
20×20	100	113	113
30×30	100	101	103
50×50	100	98	98

Fig. 3.1 Plate element with positively defined internal forces and displacements

Fig. 3.2 Schematic of a simply supported steel plate with an attached viscous damper, excited by a point sinusoidal force (Dimensions in m)

(a) $m=1, n=2$ mode, at 36.85 Hz

(b) $m=2, n=2$ mode, at 45.02 Hz

(c) $m=4, n=1$ mode, at 52.38 Hz (d) $m=2, n=3$ mode, at 58.80 Hz

Fig. 3.3 The four adjacent mode shapes of the steel plate to the excitation frequency, (a) $m=1, n=2$, at 36.85 Hz; (b) $m=2, n=2$, at 45.02 Hz; (c) $m=4, n=1$, at 52.38 Hz; (d) $m=2, n=3$, at 58.80 Hz

(a) Total intensity with magnitude-based vector

(b) Total intensity with uniform vector

(c) Sub-intensity due to bending moment

(d) Sub-intensity due to shear force

(e) Sub-intensity due to twisting moment

Fig. 3.4 Analytical structural intensity and its components on a steel plate at 50 Hz. (a) Total intensity with magnitude-based vector; (b) total intensity with uniform vector; (c) sub-intensity due to bending moment; (d) sub-intensity due to shear force; (e) sub-intensity due to twisting moment

Fig. 3.5 Vibration velocity distribution on the steel plate

(a) Magnitude-based vector

(b) Uniform vector

Fig. 3.6 Reactive structural intensity on the steel plate at 50 Hz. (a) magnitude-based vector; (b) uniform vector

Fig. 3.7 Magnitude-based structural intensity on the steel plate computed with 6×6 modes

Fig. 3.8 A section of quintic B-spline curve (between knot 6 and 7), with 6 control points and 12 knots

Fig. 3.9 Effective third-order cardinal B-spline $\eta^3(x)$ (a) and its spectrum (b)

Fig. 3.10 Effective seventh-order cardinal B-spline $\eta^7(x)$ (a) and its spectrum (b)

Fig. 3.11 Fifth-order B-spline $\beta^5(x)$ (a) and its spectrum (b)

Fig. 3.12 Numerically predicted structural intensity on the steel plate with 4 edges simply supported, at 50 Hz

Chapter 4

Structural intensity in plates

The purpose of this chapter is to validate the structural intensity measurement method discussed in Chapter 3. The validation procedure is divided into two parts. In the first part, simulated velocity data are used to validate the data processing schemes, BSSF. The calculated structural intensity is compared with the numerical one by FEM to see whether the energy sources/sinks and energy transmission path can be identified correctly. In the second part, vibration measurement is carried out on a thin aluminium plate under three different boundary conditions and the structural intensity distribution results are presented and discussed. In addition, the performance of BSSF in structural intensity calculation is investigated from a viewpoint of k -space filtering. Relationship between parameters of B-spline approximation and k -space filter is established, which can help to determine the appropriate mesh size and improve the understanding of the proposed new data processing scheme.

4.1 Issues on the application of BSSF

In order to calculate the structural intensity accurately, the proposed data processing scheme ought to be able to remove the measurement noise properly. Otherwise, the measurement noise may be magnified greatly in the representation of the high order derivatives and results in a distorted structural intensity distribution pattern. Moreover, since the surface of the HDD base casing is irregular, the data processing scheme must be capable of dealing with signal irregularities. The first problem is investigated from a viewpoint of k -space filtering. Relationship between parameters of B-spline

approximation and k -space filtering is established, which can help to determine the appropriate mesh size to remove the noise while keeping the useful information. The other issue is discussed on the basis of the local support property of B-spline.

4.1.1 K-space filtering

The purpose of filtering is to remove the noise-contaminated components especially at high frequencies. According to the principle of mode summation, when a system is excited at a single frequency f , a number of its eigen-modes will be excited but with different degrees of strength. Although the overall response of the structure is a combination of all natural mode shapes, usually only those modes that make significant contributions to the motion need to be considered to determine the structural response. The importance of the contribution of each mode to the overall response depends on the proximity of the corresponding eigen-frequency to the frequency of the excitation. Hence, the properties of the filter can be approximately determined by keeping the significant mode shapes, which are associated with the excitation frequency and the structure's modal parameters.

Fig. 4.1 shows the wave-number spectrum of the surface velocity of an aluminium plate, excited at 100 Hz. The surface velocity data are simulated with ANSYS® and can be assumed to be noise-free. In order to demonstrate the noise contaminated wave number spectrum, 15% white noise is added to the original data. As the plate is excited at a single frequency, the strong wave number components, seen at each corner, correspond to the flexural wave number defined as,

$$k = \left(\frac{\rho h}{B} \right)^{1/4} (2\pi f)^{1/2} \quad (4.1)$$

and its adjacent modes. The other components represent the weak modes and uncorrelated noise. The purpose of k -filtering, therefore, is to remove the wave number components corresponding to the noise in the wave number spectrum. If the filter size is too large, the noise may not be removed, resulting in distortions in the structural intensity distribution pattern. If the filter size is too small, however, useful information may be removed along with the noise, and the structural intensity pattern may not indicate the real dynamic response of the structure.

In the case of SFT, an oval filter is usually used as the k -space filter. The filter can be described as follows,

$$F(k_x, k_y) = \begin{cases} 1 - \frac{e^{-\alpha}}{2}, & \text{for } r < c \\ \frac{e^{-\alpha}}{2}, & \text{for } r > c \end{cases} \quad (4.2a)$$

where,

$$\alpha = (1 - r/c)/s, \quad (4.2b)$$

$$r = \sqrt{k_x^2 + k_y^2}, \quad (4.2c)$$

$$c = \sqrt{c_x^2 + c_y^2}, \quad (4.2d)$$

$$c_x^2/a_x^2 + c_y^2/a_y^2 = 1, \quad (4.2f)$$

and s controls the filter slope. The bandwidth of the low pass filter can be adjusted in the k_x and k_y directions by changing the values of a_x and a_y . The determination of the filter parameters, i.e. a_x , a_y and s is a trial-and-error process. As a rule of thumb, the parameters are generally chosen as such that 90%-95% of the total energy is kept after filtering.

It has already shown that the BSSF is equivalent to a low-pass filtering procedure in section 3.4.4. Hence, a similar k -space filter may be implemented in the surface fitting. According to the study in section 3.4.4, the relationship between the parameters of B-spline approximation (mesh size) and the performance of k -space filter can be established as follows.

The filtering effect of B-spline approximation is determined by the FE-mesh density. If a coarse mesh is employed, then only components with relative low frequency are kept in the velocity approximation. On the other hand, if a fine mesh is used, noise may remain in the resulting velocity approximation. Hence, a proper mesh density becomes a key factor in the surface fitting. Since the velocity field is dominated by the components corresponding to the flexural wave number in the structure and its adjacent modes, the selected mesh density at least should kept these components while removing the uncorrelated noise. Suppose the cut-off frequency of the cardinal B-spline is ω_c , then a mesh size l will correspond to the components of (ω_c/l) in the wave number domain. Thus, if the dominant flexural wave number in the structure is k_s , the upper limit of the mesh size can be determined by letting $\omega_c/l > k_s$, which gives

$$l < \frac{\omega_c}{k_s} \quad (4.3)$$

Therefore, if a uniform mesh is adopted with a mesh size $l_x \times l_y$, the band limits of the k -space filtering are then $k_x < 2\pi f_c \cdot \left(1/l_x\right)$ and $k_y < 2\pi f_c \cdot \left(1/l_y\right)$, respectively. Fourier transform of the quintic cardinal spline $\eta^5(x)$ is given in Fig. 4.2. Apparently, $\eta^5(x)$ works as a low-pass filter. The upper limit of the cut off frequency $f_c(\omega_c = 2\pi f_c)$ is also indicated in this figure.

4.1.2 Signal singularity

One basic property of B-splines is local support (Dierckx, 1993), which means that the defined segments in Eq. (3.19) depend on just a few of the coefficients. These coefficients in turn depend on locally measured data. As a result, discontinuities due to holes, irregular scanning shapes, etc., can be manipulated with little effect on the overall performance of the reconstructed velocity field. Another important property is that both the knots in Eq.(3.19) and the measured points are not necessarily equally spaced, which means the velocity field can be reconstructed with different resolutions over the sub-regions of interest. One application of this property is to refine the structural intensity near sources/sinks, where sharp changes may exist in the velocity field.

For finite structures, the velocity on the boundaries of the measurement surface is not necessarily zero. As mentioned earlier, the resulting non-periodicity may cause great distortion in the wave number domain if SFT is directly performed. In order to minimise the distortion, windowing or mirror processing is often adopted. Since B-spline approximation does not assume that velocity is periodic at the boundaries, no special care is needed to treat the boundary conditions in this method. Figure 4.3 shows an original velocity data compared to the data obtained by reconstructing the same data polluted with 15% white noise. The data along the four edges are not zero. That is, the non-periodic boundaries exist. It can be seen from Fig. 4.3(c) that the data can be reconstructed accurately through B-spline approximation, without any other special measures to deal with the non-periodic boundaries.

4.2 Structural intensity with simulated data

An aluminium plate is taken as the test subject. The plate is of $0.7m$ long and $0.5m$ wide with a thickness of $3mm$. The Young's modulus is $70GPa$, the Poisson's ratio is 0.3 and the mass density is $2100kg/m^3$. A viscous damper with the damping rate of $2000Ns/m$ is attached to the plate at $x=0.55m$ and $y=0.15m$. The excitation force with an amplitude of $1N$ is applied at $x=0.1m$ and $y=0.35$.

Structural intensity distribution patterns computed by two different methods are compared to validate the ability of BSSF to visualise the vibratory energy transmission.

Method 1 consists of computing the structural intensity with ANSYS directly. The normal displacement $\tilde{w}_0$, shear forces $\tilde{Q}$, bending and torsional moments $\tilde{M}$, rotations $\tilde{\theta}$ and the normal velocity $\tilde{v}$ are first computed using ANSYS. Then the structural intensity in the test plate is computed according to Eq. (3.7). Based on the study in section 3.5, the structural intensity computed this way can be assumed to be accurate.

Method 2 consists of computing the structural intensity using BSSF. The simulated velocity data are taken as the experimental data. The proposed experimental data processing scheme is then used to process these discrete velocity data and compute the structural intensity based on Eq. (3.8). Since Eq. (3.7) and Eq. (3.8) are completely equivalent according to Kirchoff plate theory, the proposed data processing scheme, i.e. BSSF, will be validated by showing that the two methods provide similar results.

Three different cases are considered. The boundary conditions and the excitation frequencies for each case are listed in Table 4.1. In case 1 and case 3, the frequency 30 Hz and 100 Hz are close to the first and second resonance of the plate, respectively. In

case 2, the frequency 30 Hz is between the first and second resonance. The corresponding mode shapes of the plate within the frequency range of interest for case 1 and case 2 are shown in Figs. 4.4 and 4.5 respectively.

4.2.1 Structural intensity by FEM

In section 3.5, the steel plate is assumed to be simply supported along all four sides. In this section, because free boundaries conditions are considered, the stress gradient is much higher than that of the steel plate considered. The mesh density here is set to be much finer to obtain stress output accurate enough for the calculation of structural intensity. As in section 3.5, the 8-node structural shell element (Shell 93) is used to model the aluminium plate. The viscous damper is modelled using element (Combin 14).

The calculated structural intensity distribution patterns are shown in Fig. 4.6. According to previous discussion, these structural intensity vectors can be considered as accurate representation of the power flow in the plate.

Fig. 4.6(a) and (c) show the structural intensity distribution of the plate under the same boundary conditions. In both cases, the positions of the force and damper are located accurately, except that the energy transmission paths are slightly different. The difference can be seen more clearly with the uniform vector plot, as shown in Fig. 4.6 (d) and (f). It suggests that the energy transmission may be different at different frequency even for the same structure with the same boundary conditions. Figs.4.6 (b) and (e) show the structural intensity distribution of the plate at 30 Hz , with the 2 long edges free. Both Figs (d) and (e) show the structural intensity distribution of the plate at 30 Hz. The boundary conditions, however, are different in the two cases. Comparison between the

two figures indicates that the change of boundary conditions may result in great distortion in the energy transmission pattern.

4.2.2 Structural intensity extracted with BSSF

The surface velocity data are obtained by ANSYS at the same time as the stresses and displacements used in section 4.2.1 are calculated, ensuring that the structural intensity patterns calculated with the two methods are comparable. Fig. 4.7 shows the simulated velocity fields.

The simulated velocity data are taken as laser-measured velocity data and processed with BSSF to calculate the structural intensity. The mesh size of the B-splines approximation is determined according to the relationship established in Eq. (4.3).

The structural intensity patterns for the three different boundary conditions are calculated. In order to investigate the capability of BSSF to remove the measurement noise, white noise is added to the velocity data of the plate with four edges simply-supported at 100 Hz, as shown in Fig. 4.8. The calculated structural intensity patterns are shown in Fig. 4.9.

Figs. 4.9(a) and 4.9(c) show the structural intensity patterns of the plate with four edges simply supported at different frequencies. The directions of the structural intensity vectors indicate the positions of the excitation force and the viscous damper, as well as the vibratory energy flow path from the excitation force to the damper. The black dots show the locations of the forces/dampers. Comparing Fig. 4.9(a) and 4.9(c) with Fig. 4.6(a) and 4.6(c), it can be seen that the predicted force/damper positions and the energy flow path using BSSF are as accurate as the ones directly calculated with FEM.

Although the experimental data processing scheme results in the same source/sink positions and power flow path as the “accurate” one, some discrepancy occurs near the sources/sinks. Under ideal point force conditions, the largest intensity vectors should exist at the point of source/sink if the structural damping is neglected, as shown in Fig. 4.6. In Fig. 4.9, however, the largest structural intensity vectors appear slightly away from the positions of the source/sink. Since near a source/sink, the shear component of the structural intensity is dominant, a reasonable explanation is that the representation of velocity data does not have the resolving power to accurately model sharp changes in the shear force (Blotter, et al., 2002). From the viewpoint of k -space filtering, this phenomenon may be explained as some important high frequency components near the discontinuities (e.g., source/sink) are filtered with the globally optimal filter size. Hence, this problem can be solved through locally increasing the cut-off frequency of the low-pass filter. For BSSF, it can be achieved by local refinement of the mesh (non-uniform mesh). However, as the main objective of structural intensity analysis is to identify the locations of energy source/sink and energy transmission path, it is usually not necessary to do such a job unless accurate amplitude of the structural intensity near the source/sink is needed.

Fig. 4.9(b) shows the structural intensity pattern of the plate with the two long edges free. Great errors appear near the edges of the plate. As illustrated in the figure, some structural intensity vectors at the top and bottom edges point to the wrong direction unexpectedly, with exceptional large amplitude. It is not suggested that these are the correct values or representation of the structural intensity. One reason for this error is that less data is used to solve for the B-spline coefficients at the boundaries where the velocity gradient is larger. Although errors exist near the edges of the plate, the source/sink and

the energy transmission path still can be identified accurately from Fig. 4.9(b). That is to say, the proposed experimental data processing scheme is still effective.

Fig. 4.9(d) shows the structural intensity pattern with noise contaminated velocity data. In spite of the 15% white noise added, almost the same result is obtained as that of the noise-free velocity data (as shown in Fig. 4.9(c)). This example demonstrates the good anti-noise performance of the proposed BSSF data processing scheme.

For the purpose of comparison, the simulated velocity data are also processed using SFT. Similar results are obtained, as shown in Fig. 4.10. However, for non-periodic data, the use of SFT will become more complex than the proposed BSSF. For instance, if SFT is used, mirror processing or other measures must be taken to get relatively accurate results in case 2 where the two long edges of the plate are set free.

Concluding this section, we can say that the proposed BSSF method can effectively locate the locations of energy sources and sinks and visualise energy transmission path in a plate-like vibrating systems. The relationship established between the B-spline mesh size and the k -space filter can be used to determine the parameters of the B-spline mesh. With proper mesh size, the BSSF method is capable of removing the measurement noise properly and computing the structural intensity accurately from highly noise-contaminated experimental data.

4.3 Experimental example

This section shows an example of experimentally determining structural intensity in a

vibrating thin aluminium plate. The aim here is to examine the capability of the proposed method in calculating structural intensity in a real vibration system.

4.3.1 Experimental set-up

Scanning LDV is one of the fundamental components in the measurements of structural intensity. The scanning LDV system consists of two key components: an OFV-056 scanning head and a control/processing system. The OFV-056 scanning head contains XY deflection mirrors, live colour video camera and a LDV sensor. The deflection mirrors automatically steer the laser beam to the desired position on the target within a $40^\circ \times 40^\circ$ field of view, with a position resolution of 0.002° . The mirrors move very fast, so an area can be measured very quickly. The control/processing system includes an OFV-3001-S LDV controller, a junction box, as well as a data management system.

Fig. 4.11 shows the diagram of the experimental set-up. The test plate is secured to a rigid fixture (not shown in Fig. 4.11) with screws. The signal generator can produce two anti-phase harmonic signals, which are then input to the two micro shakers to excite the test plate with opposite phase. The real phase difference between the excitation forces provided by the micro shakers may be checked via the force transducers mounted between the shakers and the plate. Test shows that a constant phase difference, which is around 180 degree, can be obtained between the two shakers. Scanning LDV is adopted to measure the vibration velocity over the test plate during the experiment and the collected velocity data are input to the computer for further processing. A list of the equipment used (types and models) is given in Table 4.2.

In order to demonstrate the capability of BSSF to deal with complicated boundary conditions, the plate is mounted in such a way that two long edges are fixed to the rigid fixture with 6 screws and the other two short edges are set free. The two micro-shakers harmonically excite the plate at a constant frequency of 168 *Hz*. In order to maintain a constant vibratory energy flow between the two shakers, shaker 1 is tuned about 180 degree lagging shaker 2. The amplitude of shaker 1 is set approximately half of shaker 2. According to Williams *et al.* (1985) and Blotter *et al.* (2002), there may exist a significant vibration energy transmission path between the two shakers under such a condition. The schematic of the plate, the locations of the two shakers and the scanned region are illustrated in Fig. 4.12.

Fig. 4.13 shows a photograph of the assembly of the fixture, test plate and locations of the micro-shakers. Fig. 4.14 gives a photograph of the whole experimental setup, including the signal generator, scanning head and control/processing system.

4.3.2 Procedure

The experiment procedure can be divided into three basic steps: 1) acquisition of discrete velocity data; 2) reconstruction of a smooth and continuous velocity field; 3) extraction of the structural intensity from the reconstructed velocity data. The first step is carried out through the Scanning LDV. The second and third steps are realized with MATLAB based on the BSSF method established.

The test plate is 320 *mm* long and 230 *mm* wide, with a thickness of 2*mm*. A complex velocity field over a 72×50 grid is scanned. Figs.4.15 (a) and (b) show the real part and

imaginary part of the measured velocity field, respectively. As expected, the velocities along the four edges are not all zero due to the mounting conditions.

With the discrete velocity data measured, BSSF can then be used to reconstruct a continuous and smooth velocity field and compute the structural intensity subsequently. In order to reconstruct the velocity field properly, the B-spline mesh size must be determined in advance. According to Eq.(4.1), the dominant wave number component in the plate is $k_s = 17.4$. Therefore, a uniform 3×2 mesh is used. For such a mesh size, the cut-off wave numbers in k_x and k_y direction are 31.4 and 29.8, respectively, which are about 1.8 times as large as the structural wave number $k_s = 17.4$. Therefore, the dominant wave number components are kept in the reconstructed velocity field, while noise at higher wave number can be removed.

4.3.3 Results and discussion

Fig.4.16 shows the calculated structural intensity field over the plate. The two large black dots on the figure represent the relative size and locations of the two shakers. It can be seen from this figure that the two shakers are clearly localised from the structural intensity pattern. Moreover, the intensity vectors around shaker 1 are convergent while those around shaker 2 are divergent. It implies that shaker 1 really works as the energy source; shaker 2, however, works as an energy sink rather than a source during the excitation, and vibratory energy transmission does exist between the two shakers.

Some intensity vectors at the boundaries of Fig. 4.16 point to the wrong direction unexpectedly. It is not suggested that these are the correct values or representation of the

structural intensity. One reason for this error, as discussed in section 4.2, is that less data is used to solve for the B-spline coefficients at the boundaries. Hence, it is suggested that a larger area than that of interest be scanned at the stage of velocity measurement.

The main objective of this project is to establish a methodology to study how the vibratory energy is transmitted within complex structures such as the HDD. Usually, it is the locations of sources/sinks and the transmission path from the sources to sinks that are of interest. From this viewpoint, the proposed experimental procedure and the BSSF data processing scheme can fully meet the demand and act as a powerful tool to study the vibration transmission within the HDD.

Table 4.1 Descriptions of the plate in each simulation case

	Plate type	Boundary conditions	Excitation frequency (Hz)
Case 1	Aluminium	4 edges: simply supported	30
Case 2	Aluminium	2 short edges: simply supported 2 long edges: free	30
Case 3	Aluminium	4 edges: simply supported	100

Table 4.2 List of equipment used in the experiment

Name	Types & Models
Signal Generator	Sony Tektronix AWG2041
Force Transducer	B&K 8200
Charge Amplifier	B&K 2635
Micro Shaker	B&K 4810
Scanning LDV (including the controller, junction box and the data management system)	Polytec PSV300
Power Amplifier	B&K 2706

Fig. 4.1 Illustrative wave-number spectra of the surface velocity of an aluminium plate, (a) with added white noise; (b) without noise

Fig. 4.2 Fourier transform of the cardinal quintic spline $\eta^5(x)$

Fig. 4.3 An example of B-spline approximation. (a) original data; (b) data with 15% added noise; (c) data reconstructed with B-spline surface fitting

(a) (1,1) mode, at 49.63 Hz

(b) (2,1) mode, at 99.93 Hz

Fig. 4.4 The first two mode shapes of the aluminium plate with 4 edges simply supported

(a) (1,0) mode, at 16.29 Hz

(b) (1,1) mode, at 34.74 Hz

Fig. 4.5 The first two mode shapes of the aluminium plate with 2 short edges simply supported and the other two long edges free

(a) 4 edges simply supported, at 30 Hz, magnitude-based vector

(b) 2 short edges simply supported and 2 long edges free,
at 30 Hz, magnitude-based vector

(c) 4 edges simply supported, at 100 Hz, magnitude-based vector

(d) 4 edges simply supported, at 30 Hz, uniform vector

(e) 2 short edges simply supported and 2 long edges free, at 30 Hz, uniform vector

(f) 4 edges simply supported, at 100 Hz, uniform vector

Fig. 4.6 Numerically predicted structural intensity on an aluminum plate, (a) with 4 edges simply-supported, at 30 Hz, magnitude-based vector; (b) with 2 short edges simply-supported and 2 long edges free, at 30 Hz, magnitude-based vector; (c) with 4 edges simply-supported, at 100 Hz, magnitude-based vector; (d) with 4 edges simply-supported, at 30 Hz, uniform vector; (e) with 2 short edges simply-supported and 2 long edges free, at 30 Hz, uniform vector; (f) with 4 edges simply-supported, at 100 Hz, uniform vector

(a) with 4 edges simply supported, at 30 Hz

(b) with 2 short edges simply supported and 2 long edges free, at 30 Hz

(c) with 4 edges simply supported, at 100 Hz

Fig. 4.7 Numerically predicted vibration velocity distribution on an aluminum plate, (a) with 4 edges simply-supported, at 30 Hz; (b) with 2 short edges simply-supported and 2 long edges free, at 30 Hz; (c) with 4 edges simply-supported, at 100Hz

Fig. 4.8 Noisy vibration velocity distribution on an aluminum plate, with 4 edges simply-supported, at 100Hz

(a) with 4 edges simply supported, at 30 Hz

(b) with 2 short edges simply supported and 2 long edges free, at 30 Hz

(c) with 4 edges simply supported, at 100 Hz

(d) with 4 edges simply supported, at 100 Hz, white noise

Fig.4.9. Calculated structural intensity on an aluminum plate with BSSF, (a) with 4 edges simply-supported, at 30 Hz; (b) with 2 short edges simply-supported and 2 long edges free, at 30 Hz; (c) with 4 edges simply-supported, at 100Hz; (d)) with 4 edges simply-supported, at 100Hz, white noise added (all are plotted with magnitude-based vector)

(a) with 4 edges simply supported, at 30 Hz

(b) with 2 short edges simply supported and 2 long edges free, at 30 Hz

(c) with 4 edges simply supported, at 100 Hz

(d) with 4 edges simply supported, at 100 Hz, white noise added

Fig. 4.10 Calculated structural intensity on an aluminum plate with SFT, (a) with 4 edges simply-supported, at 30 Hz; (b) with 2 short edges simply-supported and 2 long edges free, at 30 Hz; (c) with 4 edges simply-supported, at 100Hz; (d)) with 4 edges simply-supported, at 100Hz, white noise added

Fig. 4.11 The experimental set-up

Fig. 4.12 Schematic of the test plate, scanned area and locations of the two shakers (dimension: mm)

Fig. 4.13 Photograph of the assembly of the fixture, test plate and the locations of micro-shakers (the two shakers are placed under the test plate and therefore invisible in this figure)

Fig. 4.14 Photograph of the whole experimental setup for structural intensity measurement on a thin aluminium plate

Fig. 4.15 Measured complex velocity on the test plate at 168 Hz (a) real part; (b) imaginary part

Fig. 4.16 Measured structural intensity on the test plate, at 168Hz, two black dots show the locations of the shakers.

Chapter 5

Applications in Hard Disk Drive

In the previous two chapters, we have studied the vibratory energy flow visualisation method using structural intensity technique. Both numerical simulation approach and experimental determination method are investigated. A new data processing scheme has been developed and validated with a specially designed experiment. The study so far has concentrated on plates. This chapter presents some preliminary experimental and numerical results on HDD base casing to examine the potential applications of structural intensity in HDD.

5.1 Structural intensity in HDD

The HDD is always subjected to dynamic loading caused by the eccentric force of the rotating motor. The vibration generated will be transferred to the suspension arm through the base casing and cause the reading/writing error/failure, which in turn limits the achievable track intensity. Hence, appropriate vibration control measures must be taken in order to push the track intensity higher. In this study, the structural intensity technique is used to investigate the vibratory energy flow in HDD. It is hoped that the knowledge of the vibratory energy flow can guide the mechanical modifications to reduce the unwanted vibration level.

Structural intensity distribution in the HDD base casing is first determined experimentally. Due to geometrical complexity of commercial available HDD base casing, e.g., structural holes and uneven surface, a near HDD base casing is manufactured

and employed as the test object in this experiment. The purpose of this experiment is to examine the feasibility of determining the structural intensity distribution in HDD base casing using the experimental method developed in previous chapters.

Meanwhile, the structural intensity in HDD base casing is also simulated numerically using solid finite elements. As already stated, with the structural intensity distribution visualised, appropriate means can be adopted to channel or divert the vibratory energy flow in HDD to reduce the vibration level at desired position in the structures.

5.1.1 Experimental set-up

Fig. 5.1 shows the photographs of part of the experimental set-up for measuring the vibration response of the near HDD base casing. The base casing is mounted to a specially designed fixture, just like the common way that the HDD is mounted in the computer. The size and material of the near HDD base casing is the same as the commercial one except that its bottom plane is almost flat so that its thickness can be assumed to be homogeneous across the bottom plane. A commercial spindle motor is mounted to the base casing in the same place as in the commercial one.

In order to obtain the relative phase information of the vibration velocity measured, a micro-shaker instead of the spindle motor is employed to excite the base casing. The micro-shaker is connected with the motor and excites the motor at 167.5 Hz vertically. The selected excitation frequency, 167.5 Hz, corresponds to the frequency of the spindle motor rotation rate 10000 rpm. As mentioned in Chapter 1, for a HDD with a rotary actuator system, the in-plane rotational vibration is the major motion type of the base casing that contributes to the TMR. However, the shaker's excitation mainly results in the

bending motion in the base casing. Hence, the vibration excited by the shaker cannot be assumed to represent the actual disturbance on the HDD actuator due to rotation of the spindle motor. From this point of view, the objective of the measurement carried out on the near HDD is only to demonstrate the feasibility to extract the structural intensity on such a small complex structure using BSSF. In order to explore the real power flow information on the HDD base casing, the spindle motor will be taken as the excitation instead of the micro shaker. However, as will be explained in next chapter, further study must be done to determine accurately the relative phases between the measured data.

Fig. 5.2 gives a schematic diagram of the whole experimental set-up. The equipment used is the same as that in Chapter 4 except that only one micro-shaker is employed in this experiment. The out-of-plane motion of the base casing is measured. The surface velocity data and the associated phase information are recorded over the bottom plane. The holes on the bottom plane are excluded from the scanning area.

5.1.2 Experimental results

The phase and amplitude of the velocity are measured and recorded. Fig. 5.3 shows the measured velocity amplitude on the base casing bottom plane. In order to calculate the structural intensity, the measured velocity data are processed using the data processing scheme, BSSF. The extracted structural intensity distribution over the bottom plane of the HDD base casing is shown in Fig. 5.4.

In this experiment, the micro shaker inputs energy to the HDD base casing and acts as the only energy source. For the given test structure, the base casing's internal (and small) damping as well as some sound radiation constitutes the main mechanism of the energy

dissipation. The structural intensity vectors, therefore, are expected to show the energy flow from the position of the micro-shaker to the rest of the base casing. In Fig. 5.4, although the structural intensity pattern shows the energy flows from the shaker to rest area, significant energy reflection from the edges can also be observed. One interpretation of this would be that the energy flow pattern is conditioned by the edge reflections due to the lack of a significant energy sink in the system. According to Neijade and Singh (2002), increasing the internal damping can reduce the effects of edge reflections to visualise the energy transmission path clearly.

Another difference between the expected structural intensity pattern and the extracted one is the existence of several false energy sources, i.e., source A, B and C. The three points correspond to the thick joints between the bottom plane and the sides of the near base casing. Because the bottom plane cannot be assumed to be a homogenous plate at the thick joints, the use of the structural intensity expressions which are based on homogeneous plate theory may leads to erratic results at these regions. Hence, further work should be done to deal with this problem in the future.

From the above discussion, we can say, although promising, extra efforts should be made to determine the structural intensity in HDD base casing experimentally.

5.2 FEM simulation results

In this section, the structural intensity within the HDD base casing is simulated using solid finite elements. Fig. 5.5 shows the FEM model of the whole base casing. A harmonic excitation force is applied through the beam. According to Hambric and Szwerc (1999), four elements are used across the thickness of the bottom plane to model

it in order to obtain correct structural intensity distribution. The calculated structural intensity is the sum of the power flow through the four elements.

Fig. 5.6 shows the calculated structural intensity distribution. The structural intensity pattern indicates that the vibratory energy flows from the position of the spindle motor to the rest area and dissipated by the internal damping. Hence, the energy flow pattern is in good agreement with the real situation. In next section, the solid element FEM model is used to study the effects of mechanical modification on the vibration energy flow.

5.3 Effects of structural modification on the vibration energy flow

Vibration energy flow, i.e. structural intensity, describes the transmission of vibration power by elastic waves through mechanical structures. Once the structural intensity distribution within the HDD is determined either experimentally or numerically, proper structural modifications or damping treatments can then be adopted to divert or contain the vibration energy flow, resulting in a reduced vibration level at certain locations in the structures.

Stiffened plate construction has been widely used in many engineering structures, such as aircraft and marine vehicles because of its high strength-to-weight ratio. The attached stiffeners can provide effective strength and resistance to vibration motions and alter the structural intensity distribution significantly (Xu, et al., 2004). At the same time, stiffeners can be easily realised in HDD base casing in the manufacturing. Therefore, the effects of stiffeners on the structural intensity distribution are addressed in this section.

Figure 5.7 shows a model of the HDD base casing with an added stiffener. The solid finite element procedure in section 5.1.3 is used to calculate the structural intensity distribution. With the same loading condition as that in section 5.1.3, the structural intensity distribution is calculated and shown in Fig. 5.8. The difference between Fig. 5.6 and Fig. 5.8 shows that there exists a significant energy flow along the stiffener attached and the stiffener changes the orientations of energy flow in the base casing at the regions of its effects. It implies that proper addition of stiffeners may be used to divert the vibration energy to regions where vibration has less significance.

Figs. 5.9 and 5.10 show the model of an HDD base casing with a rectangular hole of the size $6\text{mm}\times 3\text{mm}$ and the calculated structural intensity distribution. It can be seen that the mechanical hole has little effect on the energy flow pattern at the frequency concerned.

Figs. 11(a), (b) and (c) show the corresponding fundamental mode shapes for the original base casing, the stiffened base casing and the base casing with a mechanical hole respectively. It can be observed from the above examples that the introduction of stiffeners and mechanical holes does not remarkably affect the mode shape of the base casing or the coupled structure. On the other hand, the energy flow pattern is really affected by the stiffener to a large degree, as shown in Fig. 5.8. Since the vibration response is usually dependent on the modal shapes, the correlation between structural intensity and vibration modal shape must be studied in order to use structural intensity to guide the addition of stiffeners to base casing to reduce the vibration. Furthermore, the influence of stiffener's cross-sectional geometry properties on the changes of vibration energy flow transmitted from the source to the sink ought to be quantitatively investigated.

5.4 Summary

In this chapter, the structural intensity within a near HDD base casing excited by a shaker is studied. The results demonstrate that the vibration energy transmission path (indicated by the structural intensity pattern) is in agreement with the real situation on the whole. However, false energy sources and significant energy reflection from the edges are also observed in the extracted structural intensity pattern. It suggests, although promising, further study is needed to determine structural intensity in HDD base casing experimentally.

The structural intensity in the HDD base casing is also simulated using 3D solid elements. Simulation results show that addition of stiffener can change the orientations of energy flow in the base casing significantly. Therefore, it may be used to divert efficiently the energy flow in the base casing with the guide of structural intensity pattern.

Top view

Side view

Fig. 5.1 Photographs of the assembly of the fixture, the HDD base casing and the micro-shaker

Fig. 5.2 Schematic of the experimental set-up

Fig. 5.3 Measured velocity amplitude on the base casing bottom plane

Fig. 5.4 Extracted structural intensity on the near HDD base casing

Fig. 5.5 The 3D FEM model of the base casing

Fig. 5.6 Simulated structural intensity pattern in the HDD base casing

Fig. 5.7 HDD base casing with stiffener

Fig. 5.8 Simulated structural intensity pattern in the HDD base casing with stiffener (uniform vector field)

Fig. 5.9 The 3D HDD base casing, with a structural hole

Fig. 5.10 Simulated structural intensity pattern in the HDD base casing with mechanical hole (uniform vector field)

(a) the original HDD base casing, $f=1942\text{Hz}$

(b) the base casing with stiffener, $f=2067\text{ Hz}$

(c) the base casing with structural hole, $f=2139$ Hz

Fig. 5.11 Fundamental mode shapes of (a) the original HDD base casing, (b) the base casing with stiffener and (c) the base casing with structural hole

Chapter 6

Conclusions and recommendations

6.1 Conclusions

Track density of HDDs has been increasing rapidly in recent years with the demand for higher capacity data storage devices. As a result, the vibration transmitted from the base casing to the actuator arm, which has been negligible for a long time, now becomes more and more significant. Effective mechanical modifications or novel servo control design must be taken to solve this vibration challenge. On the other hand, if the vibratory energy transmission path from the spindle motor to the actuator arm is known, efficient measures can be taken to divert or curtail the vibratory energy transmitted to the actuator arm, and then reduce its vibration level effectively. In this project, structural intensity is used to investigate the vibration energy transmission in HDD.

The concept of structural intensity extends the vector acoustics approach to energy flow in structure-borne sound fields. In this thesis, the structural intensity in thin plates is first studied both analytically and numerically. Results show that the vibratory energy transmission path and positions of source and sinks can be determined with the structural intensity distribution pattern. Since structural intensity is directly related to the energy level, proper damping treatments, mechanical modifications or active vibration control then can be taken to dissipate or just alter the energy flow paths and divert the vibration energy to regions of less significance effectively. As a result, the vibration level at certain regions of the structure concerned is reduced.

In order to determine the structural intensity distribution experimentally, non-contact measurement approach based on Scanning LDV is studied. Emphases are put on the data processing methods to extract the structural intensity from the measured complex velocity data. Due to the capabilities of dealing signal discontinuities, a novel experimental data processing scheme, BSSF is proposed. The performance of the B-spline approximation in structural intensity calculation is investigated from a viewpoint of wave number domain (k -space) filtering. Relationship between parameters of the B-spline approximation and the k -space filter is then established. The BSSF method as well as the experimental determination procedure is validated both through simulated and measured velocity data.

Preliminary experiment has been carried out on a near HDD base casing. In the experiment, micro-shaker instead of the spindle motor is used to excite the test object. The results reflect the vibration transmission in the HDD base casing to some extent. Although promising, the current technique needs some improvements to determine structural intensity in HDD accurately. Moreover, effects of stiffener on structural intensity distribution in HDD base casing have been investigated with numerical simulation. It shows that adding stiffener to the base casing could be an effective way to divert the vibration energy.

6.2 Recommendations

In this work, the structural intensity in the near HDD base casing is investigated with a micro-shaker as the vibration source. Further study should be conducted on the real HDD base casing under the operating condition, that is to say, the vibration transmission due to the spindle motor rotation should be investigated. To do so, certain modifications on the

current measurement system must be taken to determine the phase information among different measurement points. Differential fiber vibrometer instead of the general scanning LDV may be used to do the measurement in order to obtain the relative phase information.

Effective solutions to divert or control the vibratory energy transmission path should be further studied. As shown in Chapter 5, adding stiffener may be an appropriate way to divert the energy transfer path. However, problems such as where to put the stiffener and how to decide the best cross section and other properties of the stiffener must be solved. Other approaches such as damping treatment may also be considered.

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