DETERMINATION OF FREQUENCY DEPENDENT GLOBAL DYNAMIC PROPERTIES OF ENGINE MOUNT

BY

OOI LU EAN

Thesis submitted in fulfillment of the

requirements for the degree of

Doctor of Philosophy

SEPTEMBER 2014

ACKNOWLEDGEMENT

The successful completion of this thesis is made possible with the guidance and assistance from my supervisor, friends and family. Their encouragement and continuous support help me to overcome all kind of challenges and difficulties throughout this study.

First of all, I would like to express my sincere gratitude to my supervisor, Professor Dr. Zaidi bin Mohd. Ripin for his patience, understanding and continuous encouragement during my study period. His kind sharing of the novel ideas and the experiences always inspire me in the handling of different challenges. His valuable guidance has not only driven me to complete the research but also motivated me and make me more confident and to be positive in my life.

Special thanks to Mr. Wan Mohd Amri Wan Mamat Ali and Mr. Baharom Awang for helping me in all kinds of experimental work and fabrication of the testing setup.

During my study, I have a large group of friends and colleagues who always keep me company and sharing ideas with me when I was facing difficulty, helping me in transportation, technical aspects and all kinds of learning challenges. For all your help, I am deeply indebted.

Last but not least, I would like to indicate my deepest gratitude to my family and Mr. Loo Twan Sing for their love and support of my PhD study. All the challenges during PhD study were overcome with their warmest encouragement.

OOI LU EAN

MAY 2014

TABLE OF CONTENTS

ACKNOWLEDGEMENT ii

TABLE OF CONTENTS iii

LIST OF FIGURES vii

LIST OF TABLE xi

LIST OF SYMBOLS xii

LIST OF ABBREVIATIONS xvii

ABSTRAK xviii

ABSTRACT xix

CHAPTER ONE INTRODUCTION

1.0 Overview 1

1.1 Brief introduction 1

1.2 Problem statement and motivation of the work 5

1.3 Objectives 6

1.4 Contributions 6

1.5 Thesis scope and outline 7

CHAPTER TWO LITERATURE REVIEW

2.0 Overview 10

2.1 Overview of different types of engine mount 10

2.2 Engine mount properties 16

2.2.1 Frequency-dependent properties and frequency-independent properties 16

2.2.2 Dynamic compression stiffness and dynamic shear stiffness 19

2.3 Current measurement technique 23

2.4 Dynamic characterization of the engine mounting system 25

2.5 Design of the engine mounting system and optimization 29

2.6 Discussions 33

CHAPTER THREE METHODOLOGY

3.0 Overview 36

3.1 Workflow summary 36

3.2 Descriptions of engine rubber mount 39

3.3 Concept of global dynamic properties 41

3.4 Experimental measurement of the engine mount properties 44

3.4.1 Calculation of frequency-dependent stiffness and loss factor in compression and shear mode 49

3.4.2 Piecewise polynomial curve fitting 54

3.5 Validation of the global dynamic properties 56

3.5.1 Validation with the experimental measurement results 57

3.5.2 Application of the developed method on different type of engine rubber mounts 58

3.6 Optimization 58

3.6.1 Application of the developed optimization model to a small utility two-stroke engine 59

3.7 Statistical analysis 62

CHAPTER FOUR THEORETICAL DEVELOPMENT

4.0 Overview 64

4.1 Development of the transformation technique to determine global dynamic properties 64

4.2 Derivation of the force transmissibility with the consideration of frequency-dependent stiffness and loss factor 70

4.2.1 Minimization of the mean force transmissibility for a range of frequencies 73

CHAPTER FIVE RESULTS AND DISCUSSION

5.0 Overview 76

5.1 Dynamic compression stiffness and dynamic shear stiffness 76

5.1.1 Frequency-dependent stiffness and loss factor of engine mount 80

5.1.2 Results for piecewise polynomial curve fitting 84

5.2 Global dynamic properties of engine mount 87

5.2.1 Validation of the transformation technique with the experimental measurement results 94

5.3 Results for the application of the transformation technique on different type of engine mount 97

5.4 Optimization 101

5.4.1 Mean force transmissibility of the engine mounting system 102

5.4.2 The small utility two-stroke single-cylinder engine 111

CHAPTER SIX CONCLUSION

6.0 Conclusion 113

6.1 Recommendation for future work 114

REFERENCES 116

LIST OF PUBLICATIONS 125

LIST OF FIGURES

		Page
Figure 2.1	(a) A typical rubber-to-metal bonded mount (b) different types of rubber mount available in the market	12
Figure 2.2	Voigt model for rubber mounts (Swanson and Miller, 1993)	13
Figure 2.3	Simple hydraulic mount equipped with (a) orifice and (b) decoupler (Ushijima et al., 1988)	14
Figure 2.4	Comparison of the receptance function model with frequency-dependent and frequency-independent stiffness results with experimental measurement results (Lin et al., 2005)	17
Figure 2.5	Different testing specimen for shear testing of rubber mount (a) double shear sandwich structure (Roger, 2006) (b) simple sandwich structure (Roger, 2006) (c) improved structure for high frequency testing of rubber to metal mount (Cambiaghi et al., 2006) (d) design for testing of pressure dependent properties in shear and compression mode (Alkhader et al., 2012)	21
Figure 2.6	Example of adaptors used in measuring performance of rubber mounts	24
Figure 2.7	Four points engine mounting system in the global coordinate system (Park and Singh, 2008)	28

Figure 2.8	Six DOFs mode of the vehicle engine (Geck and Patton, 1984)	31
Figure 3.1	The workflow summary for the methodology of the study	38
Figure 3.2	The engine mount used in the study (a) engine mount design A, (b) engine mount design B, and (c) engine mount design C	40
Figure 3.3	Combination effect for engine mounts under compression force and shear force (dotted line indicates the deformation of mount under compression force and shear force)	42
Figure 3.4	The concept of global coordinate system and local coordinate system	44
Figure 3.5	The application of force and response direction of (a) dynamic compression stiffness (b) dynamic shear stiffness	45
Figure 3.6	The instrumentation for the dynamic characterization of engine mounts	46
Figure 3.7	The experimental setup for (a) excitation in Z-direction and (b) excitation in X-direction	47
Figure 3.8	Voigt model for the modeling of engine rubber mount	49
Figure 3.9	Determination of frequency region limits from the response amplitude curve	56
Figure 3.10	The installation conditions and the adaptors used for (a) 30° (b) 45° and (c) 60°	57

Figure 3.11	A small two-stroke single-cylinder utility engine used in this study	59
Figure 3.12	Dynamic forces in the single cylinder engine model	61
Figure 4.1	The schematic diagram of the engine mounting system in the global coordinate system	65
Figure 5.1	The measured (a) impact compression force and (b) impact shear force (c) acceleration response in compression direction and (d) acceleration response in shear direction	77
Figure 5.2	The measurement results for the (a) receptance function and (b) dynamic stiffness of the engine mount in X-, Y- and Z-directions	79
Figure 5.3	The measured and curve fitted data for the frequency-dependent stiffness in the X-, Y- and Z-directions	81
Figure 5.4	The measured and curve fitted data for the loss factor in the X-, Y- and Z-directions	83
Figure 5.5	The predicted global dynamic stiffness of engine mount A at different orientation angles: (a) about X-axis (b) about Y-axis and (c) about Z-axis	88
Figure 5.6	The global dynamic rotational stiffness of engine mount A at different orientation angles: (a) about X-axis (b) about Y-axis and (c) about Z-axis	92

Figure 5.7	The natural frequency of the system at different orientations of the mounts	93
Figure 5.8	Validation of the global dynamic properties determined using the transformation technique and the experimental measurement results for engine mount A	95
Figure 5.9	Validation of the application of transformation technique on engine mount B and C using experimental measurement results	98
Figure 5.10	Frequency and orientation angle dependent global dynamic stiffness of engine mount A, B and C	101
Figure 5.11	The design parameters for the engine mounting system	104
Figure 5.12	Comparison of the force transmissibility in the (a) X-, (b) Y- and (c) Z-directions for the original and optimized design (The shaded area is the effective zone of isolation (transmissibility < 1))	106
Figure 5.13	Force transmissibility in (a) X-, (b) Y- and (c) Z-directions using frequency-independent properties and frequency-dependent properties	109
Figure 5.14	The experimental validation of the optimum parameters on the real modified system (Z-direction)	110
Figure 5.15	Spectra of shaking force (a) in the Z-direction, (b) in the X-direction at 4800 rpm and 6000 rpm.	111

LIST OF TABLE

Table 3.1	Specifications of the engine in the case study.	60
Table 5.1	The summary for the curve fitting equation for $k(\omega)$ and $\eta(\omega)$ in X-, Y- and Z-directions	85
Table 5.2	Comparison of the modal loss factor using polynomial curve fitting method and the quality factor method	85
Table 5.3	Comparison of parameters for the original and optimized systems.	104
Table 5.4	Comparison of the transmitted force at 6000 rpm and 4800 rpm in the vertical direction.	112

LIST OF SYMBOLS

Symbols	Descriptions
C	damping coefficient
D	piston diameter (bore)
F_T	force transmitted to the foundation of the system
$F(t)$	external applied force
$H(j\omega)$	receptance function
$H_{x/y}(j\omega)$	receptance function in shear mode
$H_z(j\omega)$	receptance function in compression mode
I	moment of inertia
i	numbering of mount
$\mathrm{Im}\{H(j\omega)\}$	imaginary part of receptance function
$\mathrm{Im}\{H_{x/y}(j\omega)\}$	imaginary part of receptance function in shear mode
$\mathrm{Im}\{H_z(j\omega)\}$	imaginary part of receptance function in compression mode
j	imaginary notation of complex number
K	frequency-independent stiffness
$K_D(\omega)$	dynamic driving point stiffness

$K_T(\omega)$	dynamic transfer stiffness
$k_{11x/y}(\omega)$	dynamic driving point shear stiffness
$k_{11z}(\omega)$	dynamic driving point compression stiffness
k_a	compression stiffness
k_b	shear stiffness
$k(\omega)$	frequency-dependent stiffness
$k_{x/y}(\omega)$	frequency-dependent stiffness in shear mode
$k_z(\omega)$	frequency-dependent stiffness in compression mode
$k'(\omega)$	individual local complex stiffness
l	length of connecting road
m	preload mass
m_A	equivalent mass at crank pin
m_B	equivalent mass in wrist pin
$\eta(\omega)$	frequency-dependent loss factor
$\eta_{x/y}(\omega)$	frequency-dependent loss factor in shear mode
$\eta_z(\omega)$	frequency-dependent loss factor in compression mode
r	ratio of excitation frequency to natural frequency

r_a	crank radius
r_c	engine compression ratio
$\mathrm{Re}\{H(j\omega)\}$	real part of receptance function
$\mathrm{Re}\{H_{x/y}(j\omega)\}$	real part of receptance function in shear mode
$\mathrm{Re}\{H_z(j\omega)\}$	real part of receptance function in compression mode
T	oscillating torque
T_{mean}	mean force transmissibility
$\tan_{x/y}\theta$	phase lag between dynamic displacement and excitation force in shear mode
$\tan_z\theta$	phase lag between dynamic displacement and excitation force in compression mode
u_x	translational motion in X-direction
u_y	translational motion in Y-direction
u_z	translational motion in Z-direction
ω	frequency
ω_n	natural frequency
$x(t)$	displacement response of the system
$\ddot{x}(t)$	acceleration response of the system

XYZ	axes label in global coordinate system
$(XYZ)_g$	global coordinate system
$(XYZ)_{mi}$	local mount coordinate system
$[A]$	Euler transformation matrix
$[F]$	excitation force matrix
$[K(\omega)]$	global complex stiffness matrix
$[K_D(\omega)_{Global}]$	total dynamic driving point stiffness matrix in the global coordinate system
$[K_T(\omega)_{Global}]$	total dynamic transfer stiffness matrix in the global coordinate system
$[k_i(\omega)]$	global complex stiffness matrix for each individual mount in global coordinate system
$[k'_i(\omega)]$	local complex stiffness matrix for each individual mount in local coordinate system
$[k_{Di}'(\omega)]$	local dynamic driving point stiffness matrix
$[k_{Ti}'(\omega)]$	local dynamic transfer stiffness matrix
$[M]$	rigid mass matrix
$[r_i]$	position vector for each individual mount

$[x]$	translational and rotational displacement matrix at the center of gravity
$[\ddot{x}]$	translational and rotational acceleration matrix at the center of gravity
α	rotational motion along X-axis
β	rotational motion along Y-axis
γ	rotational motion along Z-axis
θ_x	orientation angle of individual mount along X-axis
θ_y	orientation angle of individual mount along Y-axis
θ_z	orientation angle of the individual mount along Z-axis

LIST OF ABBREVIATIONS

Notations	Descriptions
CG	center of gravity
CV(RMSE)	coefficient of variation of RMSE
DMA	dynamic mechanical analyzer
DOFs	degree of freedoms
EM	engine mount
FFT	fast fourier transform
FRF	frequency response function
HAVs	hand-arm vibration syndrome
ISO	international standard organization
RMSE	root mean square error
SDOF	single degree of freedom
UTM	universal testing machine

PENENTUAN SIFAT DINAMIK BERSANDAR FREKUENSI GLOBAL PELAPIK GETAH ENJIN

ABSTRAK

Sifat dinamik pelapik getah enjin mempengaruhi prestasi untuk pelapik getah enjin. Sifat kekakuan dan faktor kehilangan bersandarkan frekuensi menerangkan sifat pelapik enjin dengan lebih tepat berbanding model yang menggunakan sifat redaman tak bersandar frekuensi. Sebelum ini, teknik impak telah digunakan di dalam pengukuran kekakuan dinamik titik memacu dalam keadaan paksi tegak di mana sifat kekakuan ricih diabaikan. Kekakuan ricih dinamik penting untuk menentukan sifat dinamik global pelapik getah enjin. Di dalam kajian ini, teknik impak dikembangkan untuk pengukuran kekakuan dinamik mampatan dan kekakuan dinamik rich. Kesan gabungan antara kekakuan dinamik mampatan dan kekakuan dinamik ricih mengubah kekakuan dinamik global pelapik enjin semasa pelapik enjin dipasangkan pada keadaan condong. Satu model matematik telah dibangunkan dengan menggunakan teknik transformasi Euler yang merangkumi kekakuan dinamik mampatan dan kekakuan dinamik ricih pelapik enjin ini bagi menentukan sifat dinamik global pelapik enjin. Penggunaan cara pengukuran ini bersama dengan teknik transformasi telah disahkan dengan menggunakan tiga pelapik enjin yang berlainan. Akhir sekali, pengoptimuman telah dilakukan untuk membuktikan kesan sifat dinamik global yang merangkumi sifat bersandarkan frekuensi enjin dalam keadaan mampatan dan ricihan. Lokasi dan sudut condong yang optimum telah diperolehi bagi mengurangkan min kebolehpindahan daya untuk sistem pelapik enjin. Hasilan ini menunjukkan pengoptimuman yang berdasarkan sifat redaman tak bersandar frekuensi memberikan nilai puncak untuk kebolehpindahan daya yang lebih rendah.

DETERMINATION OF FREQUENCY DEPENDENT GLOBAL DYNAMIC PROPERTIES OF ENGINE MOUNT

ABSTRACT

The dynamic properties of engine mount influence the performance of the engine mounting system. The frequency-dependent properties represented the more accurate properties of the engine mount as opposed to the frequency-independent properties model. Impact technique in the past has been used to measure the dynamic driving point stiffness in the axial position which omitted the shear properties. In this research, the impact technique is extended to measure the global dynamic properties including the dynamic shear stiffness of the engine mount.. When the engine mount is oriented in certain angles, the combination effect of dynamic compression stiffness and dynamic shear stiffness altered the global dynamic properties of the engine mount. A transformation technique is developed to include both of the dynamic stiffnesses and used them to determine the global dynamic properties of engine mount. Three different types of engine mounts are used to validate the application of the developed technique. Finally, optimization was carried out to demonstrate the effect of the global dynamic properties based on the frequency-dependent properties of engine mount. The optimum locations and orientation angles of each individual mount are identified to minimize the mean force transmissibility of the engine mounting system. The reduction of the force transmissibility showed that optimization based on the frequency-independent properties underestimated the peak transmissibility.

CHAPTER ONE

INTRODUCTION

1.0 Overview

This chapter introduces the brief idea for the background of the study. The problem and motivation behind the study is discussed. The objectives and motivation of the study are listed and the chapter ended with the description of the thesis outlines.

1.1 Brief introduction

Rubber mounts may have as many as three important functions. First, they act as the attachment points for a part or a system to the chassis. Secondly, they function as isolation preventing noise and vibration from the engine or road conditions from being transferred to the driver and passengers. Rubber mounts can also be the adjustment point to keep the components properly aligned. The engine which is properly located with the suitable engine mount unit in the chassis operates smoothly in actual operation. They are basically inserted in between body or engine with the frame. In most of the application, engine mounts are designed to allow certain amount of rotation and translation to dampen the engine vibration. The failure of the engine mount can be caused by deterioration, cracking and drying out where the resiliency of the rubber disappeared. Broken part or fall out of the mounts will result in harsh, unpleasant ride and poor interior noise quality. They may also cause misalignment of critical control linkages for the throttle, clutch or transmission.

A rubber-to-metal engine mount is a common isolator used for reducing the engine vibration transmitted to the chassis and widely used by automotive manufacturers owing to its compact design. Although the recent development in engine mount is focusing on active or semi-active mount, rubber-to-metal engine mount still owes it advantages to its simple and compact design making it more suitable for small machine or operating tools (Yu et al., 2001). The frequency-dependent nature of the rubber due to its viscoelastic property make the rubber mounts very suitable in attenuation of engine vibration (Crandall, 1970). The design of the engine mounting system and the dynamic properties of the rubber influence the performance of the engine mount.

The global dynamic properties of the engine mount are the combined output of the dynamic compression stiffness and dynamic shear stiffness and also the orientation angles and the location of the engine mounts. The compression and shear properties are dependent of the stiffness and damping of each engine mount. Most of the application of engine mount are under the combination mode of compression and shear (Shaska et al., 2007, Ibrahim, 2008). In the review of the passive vibration isolators, Ibrahim (2008) highlighted the need to include the shear properties in the analysis of isolator to provide better modelling results of the isolators. The frequency and amplitude behaviour of the rubber should also be integrated into the model of isolators. Tarrago et al. (2007) mentioned the important of knowing the global properties of rubber where they characterized on the filled rubber bushing and developed the mathematical model using fractional derivative model to represent the axial and shear stiffness of rubber. These highlighted the importance of the characterization of global dynamic properties. However, none of the work on the engine rubber mount has included the measured dynamic compression stiffness and dynamic shear stiffness in the determination of global dynamic properties of engine mount. The inclusion of the measured dynamic properties will increase the applicability of the technique in representing the performance of engine mount.

In this study, the impact technique is applied in the direction of the compression and subsequently in the direction of the shear to capture both the principal compression stiffness and principal shear stiffness using a common measurement setup. The impact technique was used by Lin et al (2005) in the measurement of the dynamic driving point stiffness of engine mount is the compression direction. It is extended by Ooi and Ripin (2011) to measure the dynamic transfer stiffness of the engine mount in the compression direction. The impact technique is further developed in this study to measure the dynamic driving point stiffness of the engine mount in the shear direction. The measurement setup is proposed in this study with the impact technique to reduce the need of the complicated experimental jigs in the current testing device. The development of the impact technique in the measurement of shear properties improved the study of the engine mounting system as an isolation system in multi-directions (Kim and Singh, 2003). Impact technique is proposed in the measurement of dynamic compression and shear stiffness since it can overcome the frequency limitation and sensitivity of the actuator (Lin et al., 2005).

The proposed measurement setup with the implementation of impact technique overcomes the need of special jigs. The frequency-dependent stiffness and loss factor in both the compression and shear directions are derived from the experimental data. Both the frequency-dependent stiffness and loss factor are included in complex stiffness matrix and defined the global three dimensional properties of the engine mounting system. The Euler transformation matrix is used in the coordinate transformation to combine the dynamic compression stiffness and dynamic shear stiffness of each individual mount to obtain the global dynamic properties of an engine mount at any orientation angles. The prediction of the global dynamic properties of engine mounts at different orientations is compared with the experimental measurements for the purpose of validation. The technique proposed here will help designers of engine mounting systems to reduce the cost and time for the experimental measurement and characterization of engine mounts.

The force transmissibility of the engine mounting system is minimized through the search of optimal mount locations and orientation angles of the individual engine rubber mounts characterized by the frequency-dependent stiffness and loss factor. The existing optimization model on the engine mounting system only considers the frequency-independent stiffness and damping of the rubber behaviour (Tao et al., 2000; Dimitrovova and Rodrigues, 2010; Tey et al., 2013). The dynamic optimization model is developed for the engine mounting system to include the measured frequency-dependent stiffness and loss factor and minimize the force transmissibility of the engine mounting system over a range of frequencies instead of several frequency points. The force transmissibility and the response of the engine mounting system are calculated for both frequency-independent and frequency-dependent properties. A comparison of the optimization results using frequency-independent properties and frequency-dependent properties is carried out. The rubber mount element is modelled as a tri-axial frequency-dependent stiffness and loss factor. The location and orientation angles of each engine mount are defined as the design variables in the engine mount system. The measured properties of the frequency-dependent stiffness and loss factor are used throughout the study.

1.2 Problem statement and motivation of the work

Most of the actual engine mounting system consists of few units of engine mounts which are installed in the inclination conditions. The deformations of engine mount when it is installed in the vertical position are pure compression and pure shear respectively and cannot used to represent the dynamic characteristic of engine mount in the inclined condition (Ibrahim, 2008). The dynamic stiffnesses which change with the orientation angles contributed to the global dynamic properties of the engine mount. However, the current measurement technique requires many adaptors and constrained by the size and shape of the engine mount. There is a need to develop a technique to combine the dynamic compression stiffness and the dynamic shear stiffness when the engine mounts are measured under pure deformation conditions and used them to determine the global dynamic properties of the engine mounting system when the engine mounts are installed at different orientation angles. Since the current optimization model on the engine mounting system only considers the frequency-independent properties of the rubber (Tao et al., 2000; Dimitrovova and Rodrigues, 2010; Tey et al., 2013), it is a need to develop an optimization model which can include the frequency-dependent properties of engine mount to search for the optimum orientation angles and the location in order to minimize the force transmissibility of the engine mounting system.

1.3 Objectives

There are four main objectives to be achieved through this study:

- To extend the impact technique from the measurement of the dynamic compression stiffness to the measurement of dynamic shear stiffness of the engine mount with the frequency-dependent stiffness and loss factor are determined for both the compression and shear modes

- To develop a transformation technique which can combine the dynamic compression stiffness and dynamic shear stiffness under pure deformation conditions

- To characterize the global dynamic properties of engine mount under different orientation conditions by using the measured dynamic compression stiffness and dynamic shear stiffness of the engine mount

- To search the optimum orientation angles and locations of the engine mount using the frequency-dependent stiffness and frequency-dependent loss factor of engine mount

1.4 Contributions

The present research analyzes the dynamic properties of engine mounting system in three-dimensions. Previous dynamic analyses of engine mount were done using hydraulic actuators and shakers as the source of excitation. However, there is no study on the global dynamic properties of engine mount. The results from tests done in a particular orientation cannot be easily translated to performance on characteristics in the others axis. As such the use of the test results are limited to the axis used in the test. The main contribution of this study is the development of the transformation technique which includes the dynamic compression stiffness and dynamic shear stiffness of engine mount and predicts the global dynamic properties of engine mount. This allows for the characteristics of the mount at any orientation. The second contribution of the study is the optimization model which considers the orientation-dependent and frequency-dependent stiffness and frequency-dependent loss factor of engine mount. This enabled the determination of the optimized engine mounting orientation and location.

1.5 Thesis scope and outline

The scope of the thesis focuses on the development of the transformation technique and the optimization model which include the orientation-dependent and frequency-dependent stiffness and frequency-dependent loss factor of engine mount. First, the transformation technique is developed based on the measured dynamic compression stiffness and dynamic shear stiffness using impact technique. Second, the optimization model with the frequency-dependent properties is done based on the search of global minimization of the force transmissibility. The developed technique is focused on the rubber mount only. The frequency range of the measurement is defined as 0-200 Hz. The three-point mounting system is considering in the development of the technique. Only small amplitude excitation is considered in the measurement to exclude the effect of amplitude-dependent properties.

This thesis is presented in six chapters which included the brief introduction, literature study of the research, development of the theoretical concept and the model, methodology, results and discussion and finally the conclusions. The first chapter explains the brief introduction on the research on engine mount. The problem which motivates the research and also the objectives of the study are included. The contributions of this study are also highlighted.

In the second chapter, literature study of the dynamic analysis of the engine mount and also the characterization of different engine mount properties are discussed. The current measurement methods used in the dynamic analysis of the engine mount are reviewed. The importance of the development in global dynamic properties is highlighted in the literature survey. The previous optimization work done on the engine mounting system is also included.

In the third chapter, the theoretical concept for the dynamic compression stiffness and dynamic shear stiffness and their relationship with the global dynamic properties are presented. The details of experimental measurement of the dynamic compression stiffness and dynamic shear stiffness using impact technique are explained. The derivation of the frequency-dependent stiffness and loss factor from each direction and the polynomial curve fitting method are included. The explanation also covers the optimization to determine the optimal orientation angles and location of mount to achieve minimum force transmissibility.

In chapter four, the outcome of the transformation technique with its details of development to determine the global dynamic properties are explained. The development of the optimization model to include the frequency-dependent properties is explained.

In chapter five, the outcome from the development work is presented including the modeling results and the experimental results. The optimum parameters to achieve minimum force transmissibility of the engine mounting system are also discussed. The chapter ends with the validation of the developed technique and also its application on different types of engine mount.

Finally, the thesis ends with the conclusions and recommendation for future work.

CHAPTER TWO

LITERATURE REVIEW

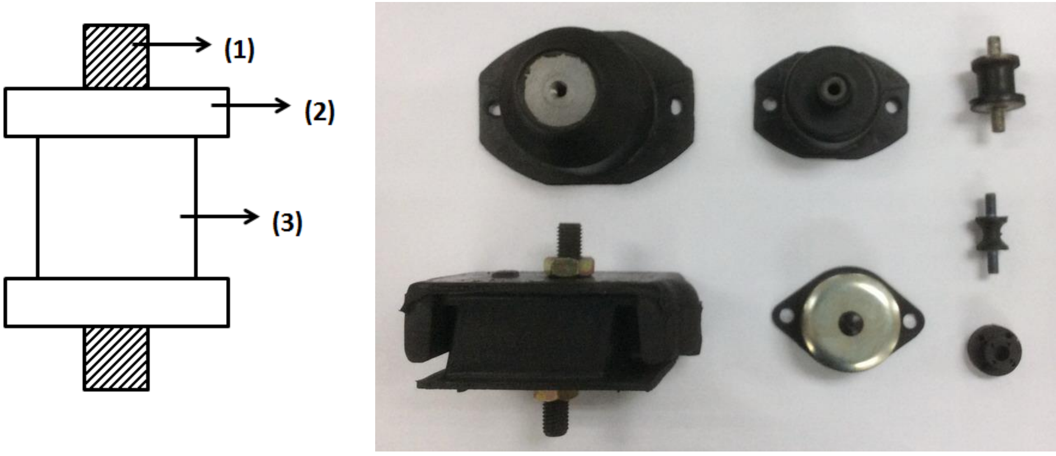
2.0 Overview

The previous work on the engine mount research is presented in this chapter. The development in the measurement of dynamic properties of engine mount and respective measurement techniques are reviewed and summarized together with the importance of the global dynamic properties of the engine mount. The work is extended to the design and the optimization of engine mounting system. The review highlights the need to include the frequency-dependent properties in the dynamic optimization model of the engine mount.

2.1 Overview of different types of engine mount

An engine mounting system commonly consists of several engine mounts to isolate the unwanted vibration of the engine from being transmitted to the chassis and the passengers in the vehicle. The mounting system helps to reduce fatigue failure and attenuate sound transmission from the engine. The high-frequency vibration of the engine generated sound which annoys the passengers or the machine users. The unbalanced engine force from the engine and other dynamic excitations of the engine can also results in unpleasant ride for passengers in a vehicle. Engine mounts have also been used in the grass trimmer and hand-held vibrating machine to isolate the vibration force from the machine users. Research has showed that the prolonged operation of the vibration tools or machine will caused the hand-arm vibration syndrome (HAVs).

Engine mounts are generally divided into few groups including rubber-to-metal mounts, passive hydraulic mounts, semi-active mounts, and active mounts (Yu et al., 2001). Although a lot of work have been carried out on the active mounts because of their better performance, the relatively simple rubber-to-metal mounts are advantageous in terms of their cost effective, compact design, and maintenance free nature. Rubber mounts are widely used as cost-effective engine isolation device and even as handle isolators to reduce acceleration transmissibility in hand-arm vibration (Ko et al., 2011). There are many types of rubber mounts in different sizes and shape to fit different applications. The simple rubber-to-metal bonded mount is common type of rubber mount used as vibration isolation. A typical rubber-to-metal bonded mount has three parts as shown in Figure 2.1(a). Part (1) labeled in Figure 2.1 is the screw which hold the mount together with the object to be mounted. There is also different type of engine mount with the screw hole at the bottom or center as shown in Figure 2.1(b). The hollow in the center of mount allows the changing of mount dimension during deformation without changing the stiffness of mounts (Rivin, 2003). The part (2) is the metal which acts as mechanical stop to limit the flexibility of the rubber elements. The part (3) is the rubber element which provides stiffness properties to the mount as an isolator.



(a) **(b)**

Figure 2.1 (a) A typical rubber-to-metal bonded mount (b) different types of rubber mount available in the market

The design of rubber mount as a vibration isolation device is based on the stiffness and damping properties and these can be formulated to provide better isolation performance (Schmitt and Leingang, 1976). The conventional rubber mount is usually represented by Voigt model shown in Figure 2.2 consisting of spring element which represented a frequency-independent stiffness (K) property and a damper element which represented a damping coefficient (C) property (Swanson and Miller, 1993). The spring and damper elements are arranged in parallel in the standard Voigt model. However, Voigt model is only suitable to represent the frequency-independent property in a single direction (Kim and Singh, 2003). Maxwell model is another model for rubber mount where the spring and damper elements are arranged in series. Maxwell model has always been used in the modeling of the fluid inertial property of the rubber, which influences the viscoelastic property of the rubber mount. The modeling of the rubber mount is further developing in the combination of the Maxwell and Voigt element to represent the better model of rubber mount (Jones, 2001). Zhang and Richard (2006) applied the Maxwell-Voigt model in the modal analysis of rubber mounting system. The study compared three different rubber models using different time coefficient. The results from the study revealed that the use of the Maxwell element in Maxwell-Voigt model produces higher frequency response function compared to the standard Voigt model.

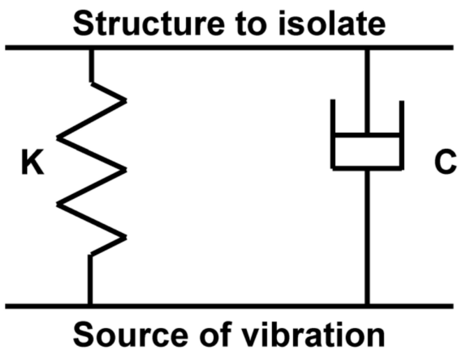
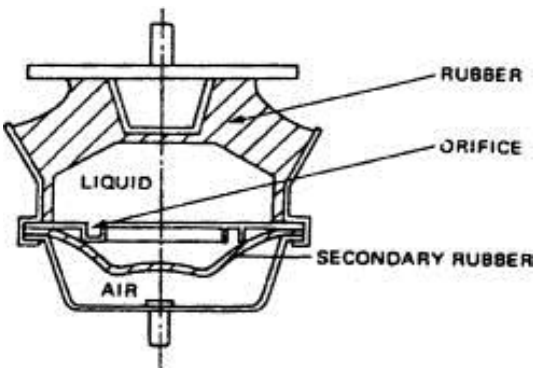
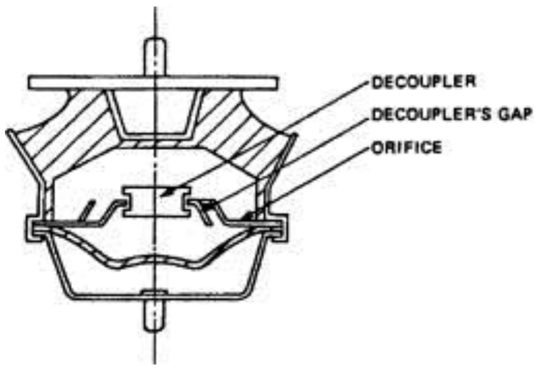


Figure 2.2 Voigt model for rubber mounts (Swanson and Miller, 1993)

The development in the vehicle industry in producing smaller and lighter vehicle requires more tunable devices to fulfill high static and low dynamic properties requirement in the design of the engine mounting system. Various types of passive hydraulic mounts are developed to improve the vibration isolation effect for higher and lower frequency range. The passive hydraulic mounts are similar in the conceptual design but the details of the actual structure used are different (Yu et al., 2001). The common structure designs which can achieve tunable function are orifice, inertia track and inertia track with decoupler. Two examples of simple hydraulic mounts are shown in Figure 2.3. Figure 2.3(a) shows the hydraulic engine mount with orifice and Figure 2.3(b) shows the hydraulic engine mount with decoupler. The hydraulic mount with orifice tunes the distance and number of orifice in the mount to allow the flow of the liquid in the mount and change the damping properties of the mount. The existence of the decoupler in the hydraulic mount limits the low amplitude movement of the mount and enhances the performance of the hydraulic mount with only orifice design. The modeling of passive hydraulic mount has been focusing on the performance on linear aspect and this limits the performance of the mount and did not reflect the actual mount properties (Golnaraghi and Jazar, 2001). Recent study of passive hydraulic mount is aimed to include the nonlinear properties in the design and modeling of the hydraulic mount. Shangguan (2009) reviewed and categorized the development in the hydraulic mount into three generation based on the design improvement and performance.





(a) **(b)**

Figure 2.3 Simple hydraulic mount equipped with (a) orifice and (b) decoupler (Ushijima et al., 1988)

The engine mount for a vehicle system required more tunable parameters to achieve better dynamic characteristic. The passive hydraulic mounts are suitable for certain application but still not fulfilling all the requirement of the vehicle design.

The performance of the hydraulic mount is only good when the input disturbance is sinusoidal. However, the actual road conditions or vibration from the high specification engines are always superimposed by different types of inputs (Ushijima et al., 1988). The hydraulic mounts are always pre-tuned for different targeted vehicle. The inaccuracy in manufacturing process or variations found under different conditions may change the actual performance of the mount. Therefore, semi-active and active mounts are developed to solve the problem of the passive mount (Yu et al., 2001). The semi-active and active mount created control force as feedback to attenuate the disturbance force. The closed loop system is used to provide the force to counteract the target vibration sources. Many different types of controllers and actuators are used in the semi-active and active system. Piezo-actuator is recently developed in the active mount system (Choi et al., 2008).

The above revision pointed that the development of engine mount is more advance and many more components are involved to provide better attenuation effect. The dynamic response of the system can be tuned using the feedback dynamic response. However, most of the general machinery mount still use the rubber mount unit as the fundamental design and also to properly support the engine in case of the system failure (Yu et al., 2001). Different types of mounts are required for different application. High cost and maintenance complexity prevented the semi-active and active mount from being applied in general and simple machine even though the semi-active and active mount can help to enhance performance (Kim and Kim, 1997; Richard and Singh, 2001). Thus, the research on the dynamic characterization on rubber mount properties and also the measurement technique is still important because there is still room for improvement due to its nonlinear and viscoelastic characteristic.

2.2 Engine mount properties

Rubber is commonly used in engine mount. Rubber has both elastic and viscous properties which are very useful in controlling the vibration level. Conventional description of engine mount properties has always been based on the frequency-independent properties, the modal mass, frequency-independent stiffness and the damping coefficients. However, recent development in the analyses of the engine mount showed that the engine mount behaved with frequency-dependent properties rather than frequency-independent behavior (Lin et al., 2005; Ooi and Ripin, 2011; Austrell and Olsson, 2012). The development in the modeling and measurement technique of engine mount properties which include the frequency-dependent properties has become important. Proper description of the rubber behavior of the engine mount will provide better characterization of isolator behavior during simulation or design for engine mount.

2.2.1 Frequency-dependent properties and frequency-independent properties

The frequency-dependent stiffness and damping are important dynamic properties of an engine mount which influence the vibration response of the engine (Park and Singh, 2010). The frequency-dependent characteristics have been shown to increase the accuracy when predicting the engine dynamic response (Lin et al., 2005; Ooi and Ripin, 2011). Lin et al. (2005) reconstructed the frequency response function of an engine mounting system by using the frequency-dependent stiffness and frequency-independent stiffness respectively. The reconstructions results are validated with the experimental measurement results and the observation by Lin et al. (2005) are shown in Figure 2.4. The result of comparison as shown in Figure 2.4 indicated that the frequency response function based on the frequency-independent properties over predicted the responses. On the other hand, the frequency response function using the frequency-dependent stiffness is closely related to the measurement results.

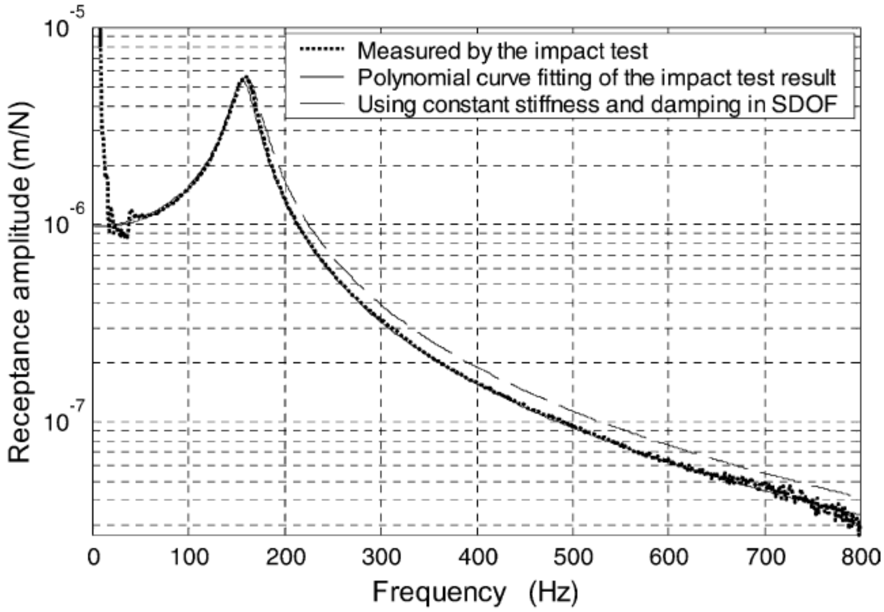


Figure 2.4 Comparison of the receptance function model with frequency-dependent and frequency-independent stiffness results with experimental measurement results (Lin et al., 2005)

Past analyses of engine mounting systems, including optimization studies and transmissibility or motion decoupling studies of engine mount properties, have been carried out assuming frequency-independent stiffness and damping properties (Tao et al., 2000; Koizumi et al., 2003; Sudhir and Annop, 2009). Tao et al. (2000) searched the optimum stiffness and orientation angles which are needed for the marine engine mount to minimize the engine force transmitted to the floor. The frequency-independent stiffness parameter is selected as the design parameter of the optimization. Jeong and Singh (2000) employed the frequency-independent stiffness and damping coefficient in their proposal of the analytical model which is used to decouple torque roll axis effect in the automotive engine mount. Park and Singh (2008) improved the method of decoupling torque roll axis effect by introducing the non-proportional damping in their model. The introduction of non-proportional damping in the analytical model showed better effect in the decoupling of torque roll axis. However, the stiffness and damping coefficient in the model is still frequency-independent. Cho and Kwak (2012) developed an analytical model to describe the dynamic characteristics of the railway vehicle suspension system. Similarly, the frequency-independent longitudinal and lateral stiffnesses were used in the development of the analytical model. By assuming frequency-independent properties in the vibration analysis helps to simplify the solution of the analytical problem but this does not reflect the actual behavior of the engine mount.

Recent development in the characterization of engine mount for both the analytical model and measurement technique have been focusing on the inclusion of frequency-dependent properties of the rubber. He and Singh (2005) proposed a procedure for estimating the frequency-dependent properties of the engine mount using the limited measurement data. The objective of the work was to improve the parameter estimation for the engine mount modeling. The authors highlighted the need of including frequency-dependent properties in the analysis of engine mounting system. Park and Singh (2010) filled the gap of research of Jeong and Singh (2000) and Park and Singh (2008) by examining the effect of including the frequency-dependent properties in the torque roll axis decoupling of the engine mount. The finding from Park and Singh (2010) on the eigen solution for the decoupling method with the frequency-dependent properties shows closer values with measurement results compared to the eigen solution with the frequency-independent properties. Austrell and Olsson (2012) revised the finite element model of the engine mount in the characterization of both the frequency and amplitude dependent properties for elastomers. The frequency-dependent properties for rubber specimen are measured and compared with the finite element model. The results show that the finite element model overestimated the stiffness values and underestimated the damping coefficient values.

2.2.2 Dynamic compression stiffness and dynamic shear stiffness

The compression stiffness is the stiffness properties under compression deformation while the shear stiffness is the stiffness properties under shear deformation (Gen, 2001). The dynamic compression stiffness and dynamic shear stiffness are important properties for the engine mount because they influence the ability of the engine mount to isolate the vibration level from sources. However, the actual installation conditions of the engine mounts in automobiles are not limited to the vertical direction but can be orientated at various angles, as specified by the manufacturer. In such cases, the dynamic performance of the engine mounts will differ from the results of the tests if they were conducted in the vertical direction. When the engine mount is tested when it is installed in the vertical position, the stress on the principal direction is considered as principal stress and the strain is considered as principal strain (Shaw and Machnight 2005). These stress and strain behaviors are similar to the principal compression stiffness and principal shear stiffness of engine mount. The principal directions of the compression stiffness and shear stiffness in the engine active plane are changed when the individual engine mount is oriented in the inclination position. Thus, the inclusion of the principal compression stiffness and the principal shear stiffness in the dynamic analysis of the engine mount model are necessary in order to correctly predict the dynamic performance of the engine mounting system.

The experimental setup for measuring the dynamic shear stiffness is different from the dynamic compression stiffness of an engine mount. In most cases, the dynamic characterization is done by using a small piece of rubber and put it under the dynamic mechanical analyzer (DMA) or universal testing machine (UTM) instead of using the actual engine mount. A special design adaptor or rubber specimen is needed for the rubber testing to study its multi-directional properties (Mars and Fatemi, 2004). However, the dynamic properties of real rubber are strongly influenced by the specimen geometry (Castellucci et al., 2008). The testing which is performed on a simple rubber specimen might not fully represent the actual performance of a rubber mount when they compared the different testing specimens which are used for the dynamic characterization of rubber. The results show that the specimen shape with simple shear deformation is the best geometry in the dynamic characterization of rubber.

In the conventional dynamic measurement using UTM, it is difficult to install the engine mount in its shear direction. It is common to use two rubber blocks as testing specimen in the measurement of shear stiffness. A double shear sandwich structure which consists of two pieces of rubber (Gent, 2001) is typically used in the dynamic analysis of rubber. However, experimental measurements of the shear stiffness using a double shear sandwich structure have shown inconsistent values, even for similar rubber samples (Misa and Kamaruddin, 2010). The compliance of test rig also significantly affects the accuracy of shear modulus measurement. Another way to measure the shear stiffness of the rubber mount is by orientating the mount in the shear direction so that it is parallel to the excitation line of force, as demonstrated by Xu et al. (2009). Recently, Alkhader et al. (2012) presented a new design in the testing of shear and compression properties of rubber. Their experiment was more focused on the pressure dependency of the shear compression properties of rubber instead of the frequency-dependent properties. For all the above-mentioned cases, different setups are necessary, which are costly because they require additional equipment. The testing device used by Alkhader et al. (2012) is shown in Figure 2.5 (d). The device is installed in the vertical and horizontal direction to measure the shear and compression properties. Figure 2.5(a) and (b) show double sandwich and simple sandwich structures which are commonly used in measurement of shear properties.

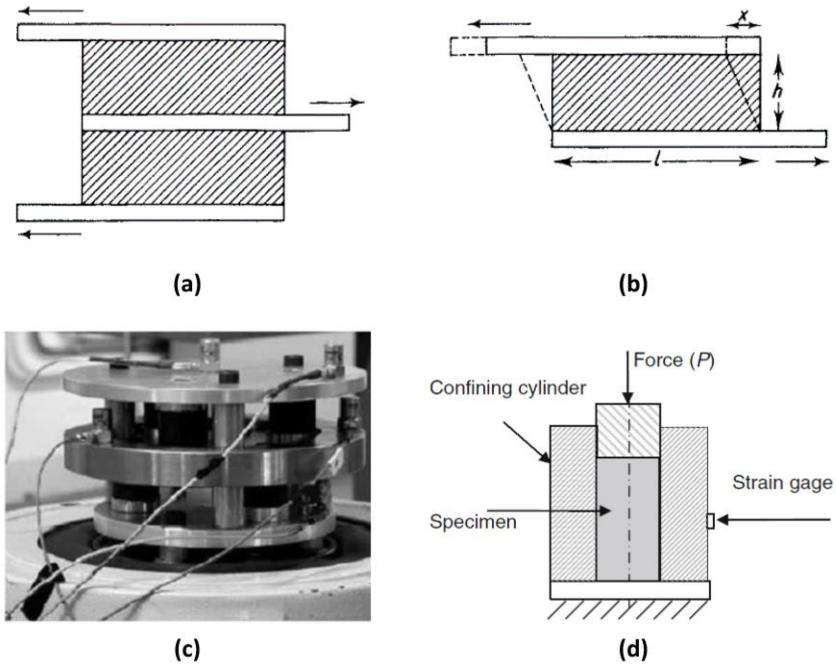


Figure 2.5 Different testing specimen for shear testing of rubber mount (a) double shear sandwich structure (Roger, 2006) (b) simple sandwich structure (Roger, 2006) (c) improved structure for high frequency testing of rubber to metal mount (Cambiaghi et al., 2006) (d) design for testing of pressure dependent properties in shear and compression mode (Alkhader et al., 2012)

The dynamic stiffness in the compression direction is always a concern in the dynamic analysis of the isolation system. Thompson (1998) developed an indirect method to measure the dynamic properties of the engine mount. The indirect method proposed by the author allowed the estimation of the dynamic properties of engine mount at higher frequency range and also in multi-direction. Dynamic characterization of rubber isolator in multi-direction was experimentally investigated by Richards and Singh (2001) using the electrodynamic shaker. Frequency-independent stiffness values were measured at different excitation force levels to examine the nonlinear effect on the stiffness in all directions. Rao et al. (2004) did the dynamic characterization for the isolator of the automotive exhaust hanger. The shaker and the standard hydraulic test actuators are used to excite the system. However, all the excitations involved in above cases are restricted to the vertical direction only. Other motions like lateral bending might happen in the actual cases. The dynamic characterization which includes other excitation, i.e shear and compression mode, must be considered in the dynamic analysis of engine mount to reflect the true characteristic of the isolator in the actual conditions. Cambiaghi et al. (2006) developed the methodology for determination of dynamic tensile modulus and dynamic shear modulus of the rubber to metal elastomer component by using the electrodynamic shaker. The designed structure used to measure the shear properties is shown in Figure 2.5 (c). In the measurement of shear properties, the elastomer is rotated so that it is perpendicular to the excitation direction. The measurement is not done for the elastomer which is installed similar with the tensile modulus measurement. The approach successfully captured the dynamic properties of engine mount at up to 500Hz.

2.3 Current measurement technique

Hydraulic actuators and the electrodynamics shaker are devices commonly used as excitation source in the dynamic analysis of engine mount. Hydraulic actuators are commonly used for applying dynamic load in the measurement of the dynamic properties of isolators (Heino and Peter, 2004; Rao et al., 2004; He and Singh., 2005). Electromagnetic shaker can also be used to provide vibration excitation in the dynamic stiffness measurement of engine mounts (Nader and Saunders, 2002; Mundo et al., 2006; Jie and Christopher, 2007). Jie and Christopher (2007) did the dynamic characterization on three different mounts using shaker. The damping coefficients are calculated using the half-power bandwidth method. The measured properties are used to identify the parameters for the three different models of engine mount. More recently, the electromagnetic shaker was used by Pananandaet al. (2013) to validate base excitation response of single degree of freedom (SDOF) system caused by the cubic damping. Discrete frequency with constant amplitude provided by the shaker and the dynamic response from the system is measured. Most hydraulic systems cannot function well at very high frequency for the vibration analysis due to the frame of the machine suffers from resonances at higher frequency range. The ability of electromagnetic shaker to provide excitation up to a higher frequency range (up to 5000 Hz) overcomes the frequency limitation of hydraulic actuators (Nader and Saunders, 2002).

Dynamic testing using uniaxial UTM is limited to the axial direction of the engine mount and, in general, limited to frequencies below 300 Hz (Nader and Saunders, 2002). In order to perform the dynamic testing in any direction other than the axial direction, a mechanical jig or adaptors are required to orientate the engine mount due to the physical setup of the UTM, as shown in Figure 2.6. This figure shows an example of the dynamic measurement of an engine mount oriented at 45°, where adaptors are used to orientate the engine mount at the particular angle. Different sets of adaptors are needed for the measurements at different orientation angles. This increases the difficulty of the measurement work. Similarly, the dynamic testing for the shear properties using UTM also require special sandwich specimen and rotated parallel with the excitation axis of the UTM.

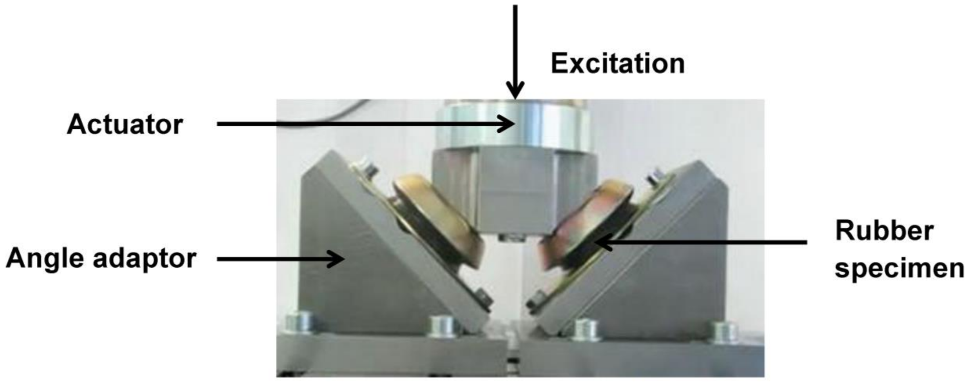


Figure 2.6 Example of adaptors used in measuring performance of rubber mounts

Another excitation technique involves the use of impact force to provide a relatively wide band of excitation frequencies. Lin et al. (2005) demonstrated the measurement of the frequency-dependent stiffness of a rubber mount by using the impact technique, where an impact hammer was used to apply an impact force on a preloaded engine mounting system. Ooi and Ripin (2011) extended the application of impact technique to the measurement of dynamic transfer stiffness. With this technique, the response from the system is captured, and the receptance function is calculated from the measured response. The dynamic driving point stiffness and the dynamic transfer stiffness are measured simultaneously using a single experimental setup. The frequency-dependent stiffness and loss factor are calculated from the