NOISE AND VIBRATION STUDY OF COMMERCIAL CONVEYOR SYSTEM

By

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DECLARATION

This work has not previously been accepted in substance for any degree and is not being concurrently submitted in candidate for any degree.

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NOMENCLATURE

DESCRIPTIONS	SYMBOL	UNIT
Mass	m	kg
Stiffness	k	N/m
Density	ρ	kg/m^3
Moment of inertia	I	m^4
Young modulus	E	Pa
Cross-sectional area	A	m^2
Length	l	m
Force	F	N
Angular natural frequency	ω	rad/s
Natural frequency	f	Hz
Damping coefficient	c	kg/s
Damping ratio	ζ	-
Mass ratio	m'	-
Displacement amplitude	x	m
Excitation force	p	N
Phase shift	φ	degree
Excitation of angular frequency	Ω	rad/s
Characteristic function (deflection n^{th} mode)	$\emptyset_n$	-
Deflection of beam	δ	m

ABSTRAK

Penghantar rantaian ialah mesin yang digunakan dalam pelbagai jenis industri kerana ia menyediakan pengangkutan mudah untuk barang dalam industri. Penghantar siri FlexMove FL 150 yang disediakan oleh FlexMove System Sdn. Bhd. ialah sebuah sistem penghantaran jenis rantaian. Penghantar rantaian ini menghadapi masalah getaran dan bunyi. Apabila sistem penghantaran ini sedang beroperasi, getaran dan bunyi yang dihasilkan oleh struktur mewujudkan suasana kerja yang tidak sihat untuk pekerja. Matlamat projek ini adalah untuk mengurangkan tahap getaran dan bunyi struktur secara keseluruhannya pada julat frekuensi operasi penghantar. Dalam projek ini, penyerap getaran dinamik tertala (DVA) telah direka dan digunakan pada struktur penghantar untuk mengurangkan tahap getaran dan bunyi. DVA direka sebagai julur rasuk dan bersama dengan bahan ikat-kenyal sebagai peredam. Pengukuran bunyi *Microflown Scan & Paint* dilakukan untuk menentukan paras tekanan bunyi yang tinggi dari struktur. Untuk pengukuran isyarat getaran dari struktur, eksperimen anjakan *LK-G Laser Head*, Analisis Modal dan Analisis Spektral telah dijalankan. Analisis Modal adalah platform untuk melaksanakan pengubahsuaian dinamik struktur melalui simulasi. Setelah DVA dipasang pada struktur penghantar, pengurangan tahap getaran dan bunyi ditaksir dengan mengulangi pengukuran yang telah dilakukan. Analisis modal penghantar yang diperbaiki membuktikan bahawa amplitud getaran dikurangkan sebanyak 9.59%. Pengukuran akustik menunjukkan bahawa penghantar yang diperbaiki menunjukkan pengurangan tahap tekanan bunyi maksimum sebanyak 2.5dB(A).

ABSTRACT

Chain conveyors are one of the machines used in various type of industries as they provide a convenient mean of transport for goods within the industry. The chain conveyor used in this study is FlexMove FL 150 series chain conveyor which is provided FlexMove System Sdn. Bhd. This particular chain conveyor encounters vibration and noise problem. When the conveyor is under operation, the vibration and noise issue found in the conveyor system contributes to an unhealthy working environment for operators. The aim of this project is to attenuate the vibration and noise level of the whole structure at the operating frequency range. In this project, tuned damped dynamic vibration absorber (DVA) was deployed on the conveyor structure to attenuate the vibration and noise level. The damped DVA was designed as a cantilever beam and together with viscoelastic material as damper. Microflown Scan & Paint sound measurement was conducted to determine sound pressure levels from the structure. For vibration signal acquisition from the structure, LK-G Laser Head displacement experiment, Modal Analysis and Spectral Analysis were conducted. Modal analysis was the platform to execute structural dynamic modification through simulation. The tuned damped DVA was attached to the conveyor structure and the attenuation of vibration and noise level is assessed by repeating the measurements. Modal analysis of the improved conveyor proves that the vibration amplitude is reduced by 9.59% with the attachment of tuned damped DVA. The acoustic measurements show that the improved conveyor endures maximum sound pressure level reduction of 2.5dB(A).

CHAPTER ONE

INTRODUCTION

1.0 Overview

In this chapter, the following sub topics related introduction are discussed. This project is approved by FlexMove System Sdn. Bhd. to be utilized as final year project. The brief background of this project is explained, and the project outline is presented.

1.1 Project Background

A conveyor system is a mechanical handling equipment that transports materials from one location to another. Conveyors are very helpful in applications involving the transportation of heavy or bulky materials. Conveyor systems allow quick and efficient transportation for a wide variety of materials and hence, often deployed in material handling and packaging industries. These systems can also be found in common consumer applications such as supermarkets and airports where conveyors are used to transport goods. Conveyor system can be categorized into two major types; chain conveyor system and belt conveyor system.

Chain conveyors utilize a powered continuous chain arrangement and the chain configuration can be different from each other. The chain arrangement depends on the customization and required function of the transport line. The chain arrangement is driven by a motor, and the materials suspended on the chain units are conveyed. Chain conveyors are generally easy to install and have very minimum maintenance. Many industry sectors utilize chain conveyor technology in their production lines. The automotive industry commonly uses chain conveyor systems to transport car parts through paint plants.

Chain conveyor systems are preferred over belt conveyor systems due to the flexibility of chain conveyor provides when the manufacturing line turns more than 30 degrees and hold the goods in place while the chain ascends or descends. However, there are few problems with chain conveyor where it leads to noise and vibration problem. There are a lot of possible reasons for noise and vibration problems in chain conveyor system such as friction between the chain units and the conveyor beam, friction between chain units and the guide rails, collision between chain units during a turn, sudden ascend or descend of conveyor and noise from motor during operation [1].

The study of noise and vibration is crucial for chain conveyor design development [2]. Noise and vibration control of conveyor structures is vital to avoid severe consequences. Unsuccessful vibration control will cause fatigue in the structure, ultimately resulting in its failure and excessive noise can cause hearing impairment, discomfort, stress and unhealthy working environment to the machine operators. There is growing awareness of the problem related to vibration and noise control in conveyors and manufacturers are requested to reduce the vibration and noise produced by all mechanisms involved in conveyor system.

In order to determine root causes of noise and vibration in chain conveyor systems out of many possibilities, FlexMove FL series straight conveyor is targeted in this project. FlexMove FL series straight conveyor is a chain conveyor system provided by FlexMove System Sdn. Bhd. for research purposes. The structure of this conveyor is made of aluminium and the chain units are made of plastic. In this study, measurements are carried to identify the sound and vibration localization source. From the measurements, a damped DVA is applied to the conveyor structure to attenuate the noise and vibration level.

1.2 Problem Statement

There are few problems with chain conveyor where it leads to noise and vibration problem. There are a lot of possible causes for the rise of vibration and noise problems. In order to identify the sources of high level of vibration and noise, FlexMove FL series straight conveyor was targeted, and different types of noise and vibration experiment were carried out on the FlexMove FL series straight conveyor. From the measurement results, few noise, and vibration spots were identified.

1.3 Objectives

The following objectives are targeted in this scope of study, parallel with the main concern stated in problem statement.

- To identify the peak vibration and noise spots on FlexMove FL series straight conveyor.
- To develop damped dynamic vibration absorber (DVA) for structural modification of FlexMove FL series straight conveyor.
- To evaluate the performance of structurally improved conveyor in terms of vibration and noise level.

1.4 Scope of Project

The vibration and noise level of the conveyor system was determined in a previous study where the experimental results provide fundamental data which can be used to choose the appropriate vibration and noise control method. The measurements done earlier in the previous study are repeated to verify the results and analysis. The vibration response is further analyzed using the modal analysis of the constructed geometric model of beam in LMS Test Lab. From the analysis, the structural dynamic modification (SDM) will be carried out using modification prediction of LMS Test Lab and the simulation results from SDM will be used to design a tuned mass absorber to reduce the vibration. Once the tuned mass absorber is fabricated, it will be attached to the conveyor beam at specific node analyzed earlier from modal analysis. The measurements and experiments conducted earlier will be repeated to evaluate the reduction in vibration and noise level. Experimental work will be carried out in the laboratory to prove the concept and final stage of the project will be the implementation of the prototype on the conveyor.

CHAPTER TWO

LITERATURE REVIEW

2.0 Overview

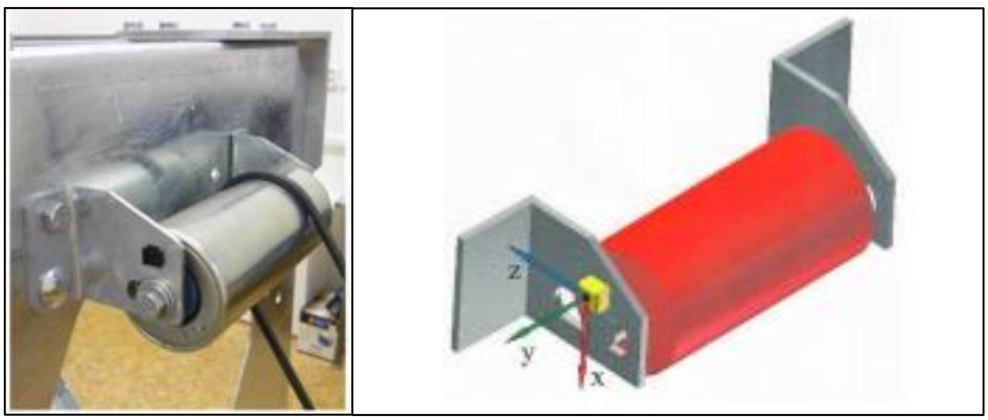
In this chapter, the following contents are discussed for literature review.

- Noise and vibration study on conveyor structure
- Noise and vibration control on machines
- Application of Dynamic Vibration Absorber (DVA)

2.1 Noise and vibration study on conveyor structure

In the context of noise and vibration identification of conveyors, Brown, S.C. [3] presented the results of a comprehensive analysis and testing of conveyors and components involved in conveyors. Conveyor noise is a composite of noise generating mechanisms and the most dominant noise is from the dynamic interaction at the (belt/idler) roll interface. The idler roll surface profile is proven to be an important factor for excitation of vibration and noise radiation for most conveyors. An idler roll surface profile measurement parameter was developed as Maximum Instantaneous Slope, (MIS) - which can be utilized to analyze the operating condition and noise generation potential of existing equipment. The experimental results from this study indicated that a strong correlation exists between the velocity of idler roll surface profile parameter and the conveyor noise emissions for the particular range of idler types tested. This study can be referred as one of the platform discussing the importance of noise and vibration control of large, outdoor belt conveyor systems. The solution discussed in this paper helps to evaluate the operating condition and the solution is more towards an inspection method to identify the noise generating conditions.

Klimenda, F., et al. [4] discussed the noise and vibration measurement of rollers for belt conveyor in this study. Vibration measurements were carried on conveyor rollers and this study emphasized on the procedure and standard methodology to measure and identify vibration in conveyor rollers. Rollers are transport with classic rubber strip, which were rolled into a tube. This modification contributed to reduction of dust in the vicinity of the conveyor . However, this modification lead to problems with the observance of hygienic limits for noise, especially near built-up areas. Due to this limitation, it is necessary to emphasize on the reduction of noise during the design of the conveyor, particularly to eliminate noise and vibration of the individual components. Hence, measurements were conducted on different type of rollers from different manufacturers (Figure 2.1). The solution presented in this study focused more on the roller design and the improvement did not cover the conveyor structure as a one whole structure to reduce the overall vibration and noise level of operating conveyor.



(a) (b)

Figure 2.1: (a) Conveyor roller design (b) Location of 3-axis accelerometer on the holder of roller. [4]

Guntur, H. L., & Krisnahadi, Y. [5] presented the results of vibration monitoring, simulation and the analysis of conveyor driving unit of a coal transporter. Coal transporter is an important equipment for coal handling and energy supply in steam power plant. Conveyor driving unit (CDU) is the unit in a coal transporter which drives the conveyor and transports coal from the coal yard to the burner. Transported load, luffing angle and conveying rate affect the vibration of CDU. Hence, vibration-based condition monitoring was conducted to maintain the reliability of the CDU. In this paper, vibration monitoring result and analysis of influence of the luffing angle and conveying rate (transported load) to vibration were emphasized. The CDU was mathematically modelled, and the vibration of CDU was simulated. From the simulation results, it is manifested that larger luffing angle and conveying rate increased the vibration amplitude, specifically at horizontal (x) direction. The baseline of this study comprises more on inspection of the parameters affecting the vibration amplitude and did not include the vibration control technique.

A new type of double pitch silent chain (Figure 2.2) was proposed by Liu, X., et al. [6] in order to overcome current problem of high noise level of double-pitch roller chain for conveyors and to yield higher efficiency and accuracy. This was achieved by replacing some plates with nano-structured metal mesh - polyurethane composite material. The noise characteristics of the new type double-pitch silent chain for conveyor were tested and analyzed. Based on the experimental results, this particular chain effectively increased the noise reduction effect, eased chordal action and reduced the impact between chain plates and the sprocket tooth. This paper highlighted that the design could improve the conveying conditions distinctly and therefore, the design of double-pitch silent chain for conveyors has a value of application. The solution proposed in this study have been tested out in real time application and the feasibility of the solution is proven.

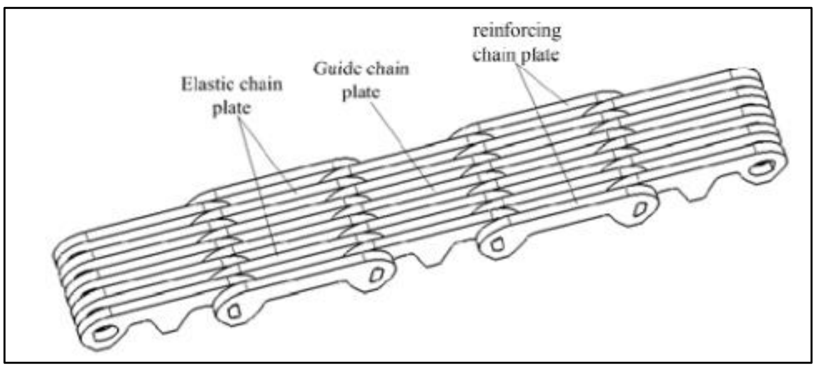


Figure 2.2: Schematic of the new type of double-pitch silent chain for conveyors. [6]

A study on noise reduction for COMAS vibrating conveyor is presented by Li, S., et al. [7] COMAS vibrating conveyor is one of the important material handling equipment in the cigarette silk production line. ANSYS software was used to analyze the modal of the frame and from the analysis results, it was discovered that the structure has torsion vibration and swing. These vibrations are proven to be contributing to adverse effects and noise. A vibration isolation system was designed and installed on the vibrating conveyor. The effectiveness and efficacy of the isolation component was proved with static and dynamic analysis of the vibration isolation system through finite element method. Numerical analysis of this study indicated that the load bearing of vibration isolator was relatively uniform, and the reaction forces were less than the limit load of the vibration isolator. With the adoption of vibration isolator, the energy transfer of vibration and the noise level were successfully reduced. Based on this study, a vibration isolator was proven to be one of the effective vibration and noise control method for conveyor structures.

On the current design improvement of conveyor design, Fiebig, W. and J. Wróbel [8] focused on effectiveness of two stage vibration isolation on a vibratory conveyor. Vibratory conveyors are commonly used in foundries for separating the casting from the mold. The shake-out conveyor has been supported directly on foundations or metal structures. The high amplitudes of vibrations on the foundation which are being transmitted to the building structure has been observed and analyzed. Active vibration isolation systems generate sophisticated solutions to eliminate vibration problems and hence, to reduce the vibration transmission from the conveyor to the foundation, two-stage vibration isolation (Figure 2.3) has been deployed on the conveyor. The mathematical model of two stage vibration isolation is demonstrated in this paper. Based on the experimental results and numerical analysis, a significant reduction of vibrations at the building structure has been accomplished. The design improvement discussed in this study emphasized more on the isolation of vibration energy from reaching the ground or support area. The solution proposed did not focus on the reduction of vibration on the machine structure itself.

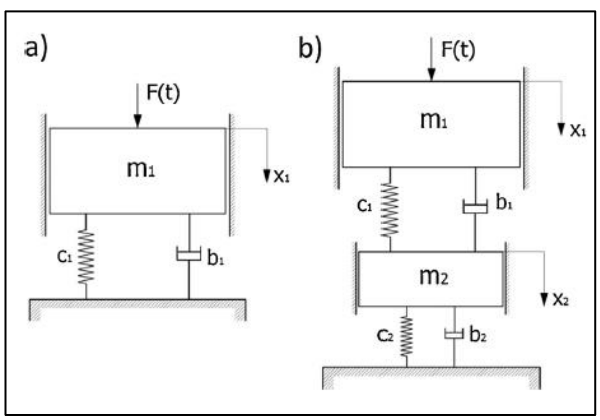


Figure 2.3: Schematic of the (a) single-stage (SSVI) and (b) double-stage (DSVI) conveyor's vibration isolation system. [8]

2.2 Noise and vibration control on machines

Noise and vibration control methods on other machinery or structures such as rotary machines, motor and beams which consists of mechanisms and structures almost similar to conveyor system can be adapted into conveyor systems to improve the noise and vibration control. In the scope of other machinery and structures vibration control, Gripp, J.A.B. and D.A. Rade [9] presented a strategy for the attenuation of noise and vibration in mechanical structures using piezoelectric shunt damping. Piezoelectric shunt damping consists of connecting piezoelectric transducers integrated in a structure to electric or electronic circuits. This alternative has a big potential for usage in small or mid-scale machines and structures. A systematic literature review of different piezoelectric shunt damping strategies designed for the attenuation of vibration and noise in mechanical systems is presented in this study. The basic principles underlying the electromechanical behavior, design procedures and numerical modeling of piezoelectric shunt damping devices is discussed as well. The scope covered by this study is more towards the viability of deploying piezoelectric shunt damping device to a harmonically excited mechanical structure.

Werner, U. [10] developed a design where actuators are deployed between the motor feet and soft foundation based on multibody model for vibration attenuation. The aim of this study was to analyze the active vibration reduction of soft mounted machines via actuators. The actuator forces were inserted directly in the vibration model without the aid of a feedback control system. The aim of this study is to reduce the forced vibrations, which are caused by typical excitations of electrical motors (eccentricity of rotor mass, bent rotor deflection and magnetic eccentricity). A simplified multibody model was derived and proposed. Based on this model, the mathematical coherences were derived, and a numerical example was generated, where different vibration reduction concepts were analyzed, and the necessary actuator forces were calculated. Based on the simplified multibody vibration model, actuators have the potential in suppressing vibrations from electrical machines. It is proposed that the multibody model has to be implemented into a feedback control system in future for better performance.

Switched-reluctance motors (SRM) exhibit major acoustic drawbacks that prevent their usage in electric vehicles despite the widely-acknowledged robustness and low manufacturing costs. SRM stator is completely enclosed with a viscoelastic resin. Millithaler, P., et al. [11] proposed a tuning methodology for reducing the noise emitted by SRM in operation by utilizing the advantage of the high damping capacity possessed by a viscoelastic material in particular temperature and frequency ranges. This paper represents the application of computing representative electromagnetic excitations and structural response of the stator including equivalent radiated sound pressure levels following the introduction of tuning process aspects. An optimized viscoelastic material was discovered, and the peak sound pressure levels were reduced up to 10 dB, compared to the initial value. Based on this paper, optimum tuning methodology can contribute to efficient sound pressure level reduction.

An approach for vibration mitigation based on an adaptive tuned vibration absorber group for vehicle powertrain system was introduced by Gao, P., et al. [12] Based on a dynamic model of a vehicle powertrain system, natural vibration analyses and sensitivity analyses of the eigenvalues were conducted to determine the crucial values for each natural vibration of a powertrain system. The results were then used to optimize the installation position of each adaptive tuned vibration absorber. Based on the optimal frequency ratio, the optimum parameters of the auxiliary vibration absorber and the optimal damping ratio of the passive vibration absorber were calculated. Then, the optimal tuning scheme for the adaptive tuned vibration absorber group was proposed, and related numerical simulations were carried out. Based on the numerical simulations, the optimal tuning scheme for the adaptive tuned vibration absorber group significantly reduced the variable frequency vibrations of a powertrain system.

Noise and vibration control methods on machines and structures such as rotary machines were gathered and reviewed. The approach discussed in every method was reviewed and the advantages of every method was considered. Based on the discussion, vibration absorber method is preferred and in the next section, the applications of dynamic vibration absorber are reviewed.

2.3 Application of Dynamic Vibration Absorber (DVA)

F. S. and F. Pellicano [13] presented a study to analyze the efficiency of dynamic vibration absorbers (DVA) in suppressing the vibrations of a simply supported beam subjected to a sequence of constantly moving loads (Figure 2.4). The supported beam subjected to constantly moving loads can be referred to conveyor beam which supports the constantly moving chain units when the conveyor is operating. In this study, few types of DVA were considered: linear, cubic, higher odd-order monomials and piecewise linear stiffness; linear, cubic and linear-quadratic viscous damping. The reason of considering few types of DVA was to investigate for any improvements possessed by nonlinear DVAs with respect to the classical linear devices.

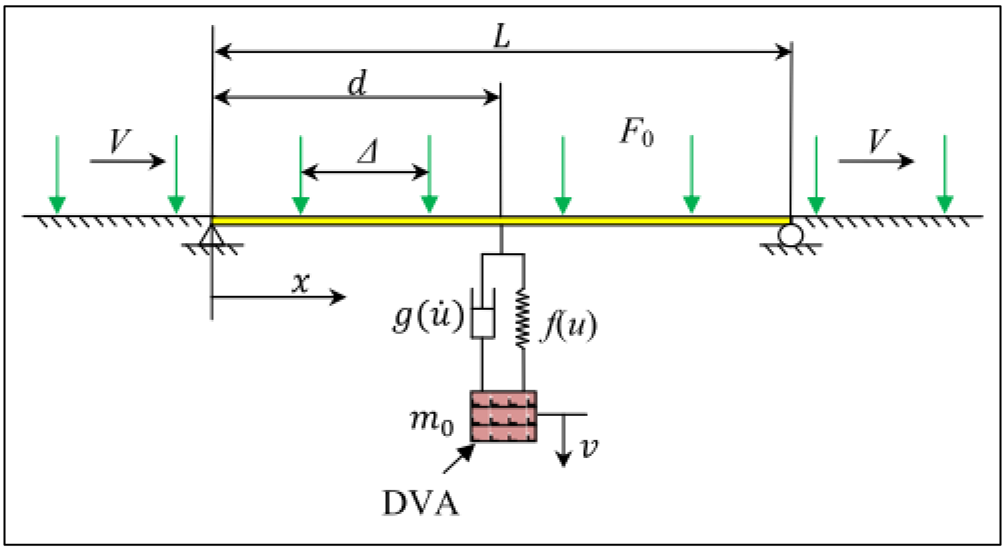


Figure 2.4: The beam model subjected to a sequence of moving load. [13]

A study on suppressing a rectangular cantilever plate from excessive resonance amplitude was presented by Arpaci, A. and M. Savci [14]. The suppressing effect of excessive resonance amplitude was achieved by optimum tuning of the damping parameters. The proposed method was deployed by utilizing an auxiliary distributed system for suppressing the excessive vibration. Structural damping of the cantilever plate was incorporated into the main and auxiliary systems by considering them as a complex elastic modulus. A cantilever beam was designed as a dynamic absorber and attached to the rectangular cantilever plate which was driven by a harmonically varying force (Figure 2.5). The proposed method proved that the adoption of damping into the auxiliary system can effectively attenuate the amplitudes generated at the resonant frequencies.

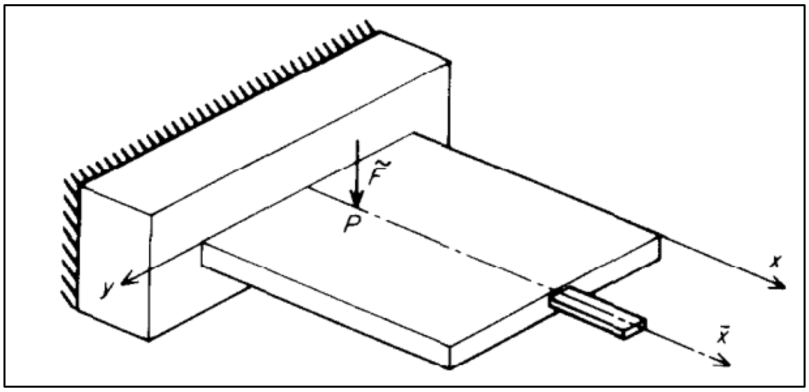


Figure 2.5: Rectangular cantilever plate with a cantilever beam attached to it as a dynamic absorber. [14]

Another study related to DVA which was presented by Kefu Liu, Gianmarc Coppola [15], focuses on the optimum design of the damped dynamic vibration absorber (DVA) for damped primary systems in this study. In this study, the DVA damper was connected between the absorber mass and the ground. This method can be deployed by attaching a secondary mass to conveyor leg structure and connecting the secondary mass to ground. DVA can shift the natural frequency of a structure out of phase and damping the inertia force energy acting on the structure but the DVA is only effective within a narrow band of frequency.

The introduction of a dynamic vibration absorber (DVA) to a vibrating structure could be a sophisticated solution for vibration attenuation if the absorber is accurately designed and installed to the structure but on the other hand, the vibration attenuation performance of this type of DVA is limited by the ratio between the absorber mass and the mass of the primary structure. Hua, Y., et al. [16] proposed a beam-based DVA and optimized it for reduction of the resonant vibration of a general structure. The mass ratio, the flexural rigidity and length of the beam affect the vibration attenuation performance of the proposed beam DVA. The proposed beam DVA exhibits flexibility in vibration control design compared to the traditional sprung mass DVA due to more design parameters. Based on this study, under same mass constraint, the proposed beam DVA can provide better performance than the traditional DVA by designing the beam DVA with appropriate parameters of mass ratio, flexural rigidity and length of beam. Based on this study, it is justified that appropriate parameters of DVA is compulsory to achieve optimum vibration level reduction in the conveyor structure.

Esmailzadeh, E. and N. Jalili [17] proposed an optimal dynamic vibration absorbers (DVA) for a structurally damped beam system subjected to an arbitrary distributed harmonic force excitation (Figure 2.6). The effects of rotatory inertia and shear deformation was discussed based on the Timoshenko beam theory. This method provided flexibility of choosing the number of absorbers. This method eased the process of choosing the number of absorbers. The optimum stiffness and damping coefficients were calculated for each absorber with a selected mass. Direct Updated Method was used as the optimization procedure for the DVA to obtain the optimum parameters such as mass, stiffness and damping value. Few factors of achieving optimum condition of suppressing vibration absorber are the location where the absorbers are attached, the position of the applied force and beam characteristics such as boundary conditions, cross sectional geometry, and structural damping. These factors can be highlighted in the procedure of designing the DVA for the conveyor structure.

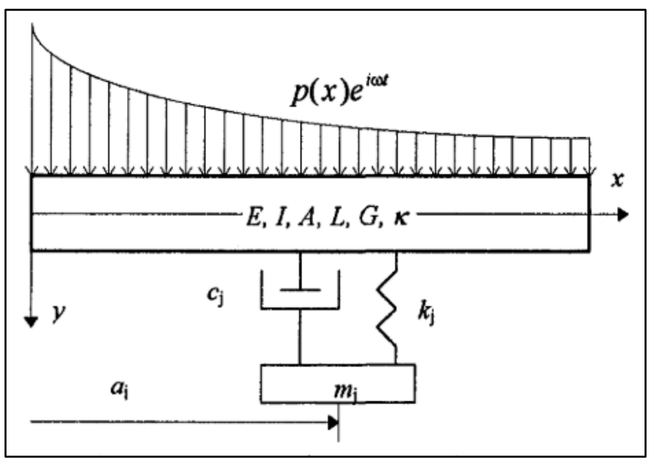


Figure 2.6: Structurally damped beam system subjected to an arbitrary distributed harmonic force excitation attached to an absorber. [17]

Huang, X., et al. [18] introduced a dynamic vibration absorber (DVA) with negative stiffness to suppress the longitudinal vibration transmission along a marine shafting system. The developed stiffness models of the DVA were made of a rubber pad and a Belleville spring. By setting up a similar model of a propeller, an analytical model of the shafting system was derived. This analytical model was derived together with and without the proposed DVA. Vibration control in the low frequency range was deployed onto this analytical model. Optimal parameters of the DVA with negative stiffness were obtained by conducting a stability analysis. The influence of the parameters of the DVA and negative stiffness on the vibration suppression was determined through a parametric study. Based on the analysis, the developed DVA with negative stiffness manifests enhanced vibration transmission suppression performance for broader absorber frequency range around resonance.

A new design of tuned mass damper was proposed by Lee, C.Y. [19] to reduce the structural vibration of a machine platform subjected to excitation force. The design is based on tunable fluid mass driven by micropump. The absorber mass was manipulated by pumping of fluid between the liquid chambers of the vibration absorber. The natural frequency of the absorber could be tuned without altering the stiffness unlike the traditional tuned mass damper. In this study, the vibration reduction effect from the absorber was retarded due to effect of damping increment originated from the liquid sloshing inside liquid chamber. Hence, a horizontal separation panel was introduced inside the liquid chambers and the liquid sloshing was successfully reduced. This effectively reduced the damping ratio of absorber and contributed to higher control efficiency.

Vibration absorbers methods available were gathered and reviewed. Most of the absorbers were ultimately designed to eliminate the vibration of a harmonically excited system by introducing a secondary system. A damped DVA is selected in this project to attenuate the vibration and noise level at the operating frequency range of the structure.

CHAPTER THREE

METHODOLOGY

3.0 Overview

Based on the objectives, the vibration problem was investigated before proceeding to the technique related to overcome the problem. In this study, methodology covered two different area; identifying the vibration and noise profile and then, solving the problem. First of all, the conveyor structure was modelled into a simpler structure to ease the process of analysis and a geometry were constructed for measurement and simulation procedure. Noise level were determined with Microflown Scan & Paint sound measurement. Vibration level were obtained with spectral analysis, modal analysis and vibration displacement measurement. The experiments were followed by structural dynamic modification (SDM) through simulation. Once, the simulation results of SDM are validated, a tuned damped dynamic vibration absorber was fabricated and deployed on the conveyor structure. The measurements carried out earlier were repeated again to evaluate the noise and vibration attenuation level.

3.1 Flowchart of methodology

Figure 3.1 shows the overall flowchart of the completed methodology to achieve the objective of this study.

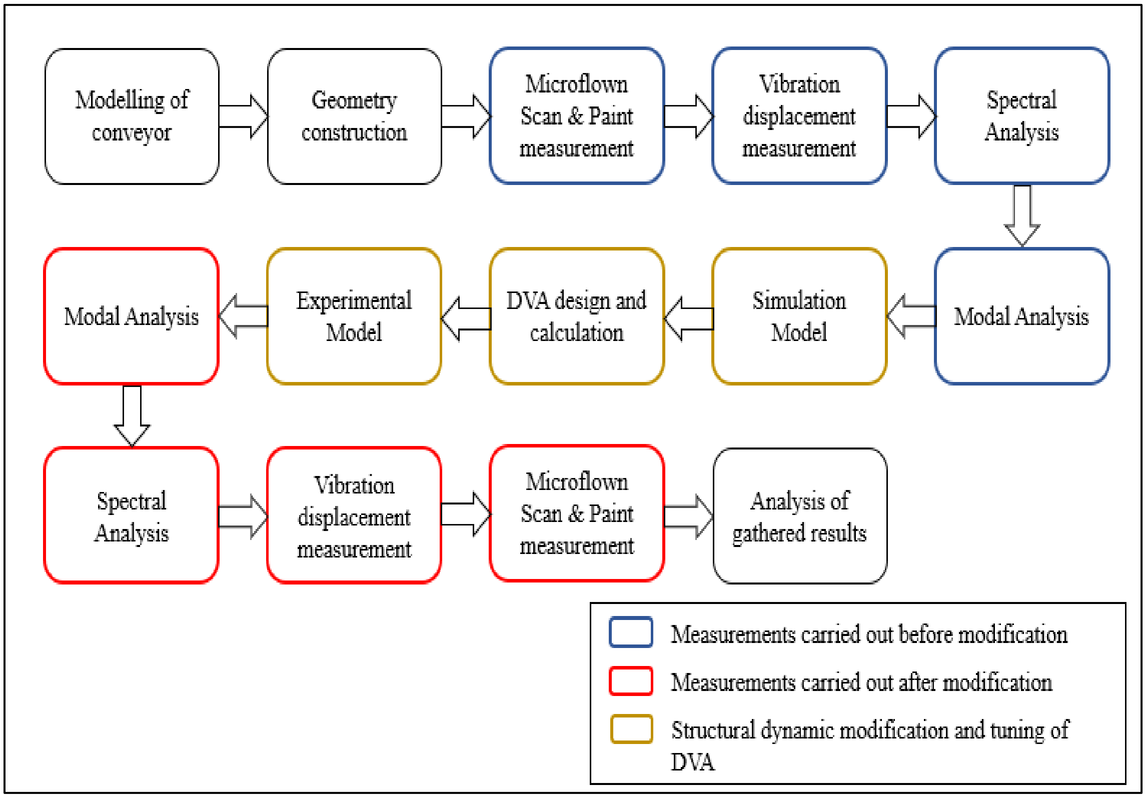


Figure 3.1: Flowchart of methodology and measurements conducted

First of all, the conveyor is modelled into a simplified structure to ease the analysis later. Then, based on the simplified structure, a geometric representation of the structure is created. In this study, there are two parts of measurement; before and after modification measurements. The first measurement conducted was Microflown Scan & Paint measurement for acoustic measurement and followed with vibration displacement measurement, spectral analysis and modal analysis for vibration signal acquisition. These four sets of measurements account for the before modification measurements and results. Once the modal analysis was completed and studied, structural dynamic modification (SDM) which consists of three different processes was carried out. The three different processes were simulation modelling, DVA designing and experimental modelling. Once SDM was completed, the damped DVA was attached to the structure and after modification measurements were conducted. The before and after measurements results were gathered and tabulated for discussion.

3.2 Modeling of the conveyor structure

The conveyor beam is a complicated structure with guide rails and connecting chains inside the structure. The purpose of the conveyor beam is to hold the running chains and components inside the operating chains in position. Hence, the structure was modelled into a simpler structure which has similarities to the conveyor beam in terms of material properties, geometric properties and functions in order to ease the analysis of the conveyor structure. The structure was modelled as an aluminium beam as the conveyor beam itself is made of aluminium [20].

Table 3.1: Material properties of an aluminium beam

Material properties	**Value**
Young modulus, E	69GPa
Poison ratio, v	0.35
Density, p	2800kg/m^3

Beam is a structure element that can withstand load especially by resist bending. In engineering field, beam has several types of boundary condition such as free-free, fixed-free, fixed-pinned, fixed-fixed and pinned-pinned. The conveyor structure is fixed to two legs support and hence, the beam model adapted to the conveyor modelling is fixed-fixed beam or known as simply-supported beam.

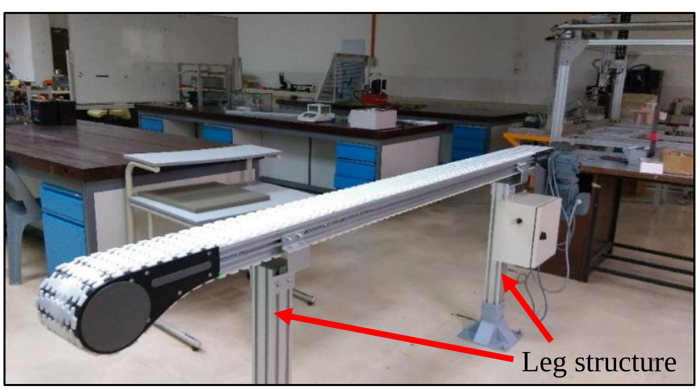


Figure 3.2: Conveyor structure with two leg support

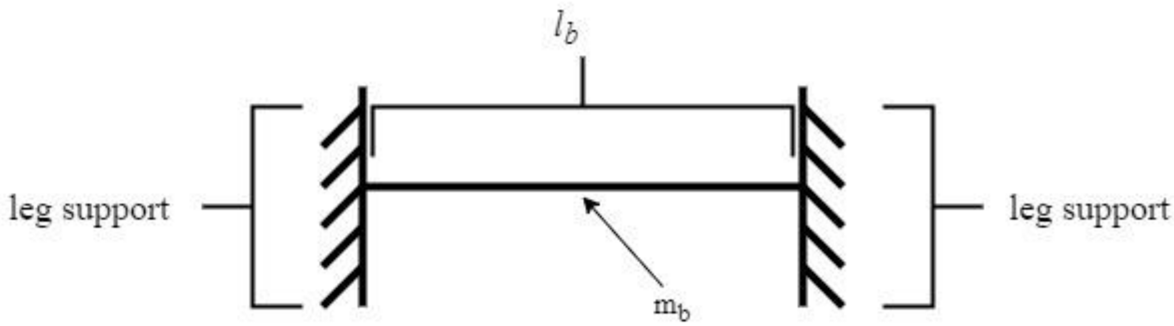


Figure 3.3: Schematic diagram of simply supported beam (conveyor model) with two supports.

To calculate the natural frequency of a simply supported beam with the mass of beam and length of beam, the following governing equations are required;

The equations of motion of beam are derived according to the Euler's differential equation:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho A\frac{\partial^2 y}{\partial t^2} = 0 \tag{3.1}$$

The deflection curve is approximated in the form of equation

$$y(x,t) = \emptyset_n(x)e^{i\omega_n t} \tag{3.2}$$

Equation (3.1) is substituted into Equation (3.2) and results in the following equation:

$$\frac{\partial^4 \emptyset_n(x)}{\partial x^4} - {\beta_n}^4\emptyset_n(x) = 0 \tag{3.3}$$

The following equations for determining the natural frequency is determined by differentiation of Equation (3.3) :

$$\omega_b = {\beta_n}^2\sqrt{\frac{EI}{\rho A}} \tag{3.4}$$

$$\omega_b = (\beta_n l)^2\sqrt{\frac{EI}{\rho A {l_b}^4}} \tag{3.5}$$

Where; $\beta_n l \approx (2n+1)\frac{\pi}{2}$ (constant for simply supported beam)

$m_b = \rho A l_b$ (mass with length of the beam l_b)

Second moment of inertia of beam, $I = \frac{1}{4}\pi r^4$

Young modulus of aluminium, E=69GPa

3.3 Geometry of conveyor beam

The structure of the conveyor consists of a horizontal beam made of aluminium supporting the guide rails, chain units and connecting plates. The horizontal beam is supported by two leg structures made of aluminium too. The geometry constructed was based on the horizontal beam only and this geometry was referred throughout this project.

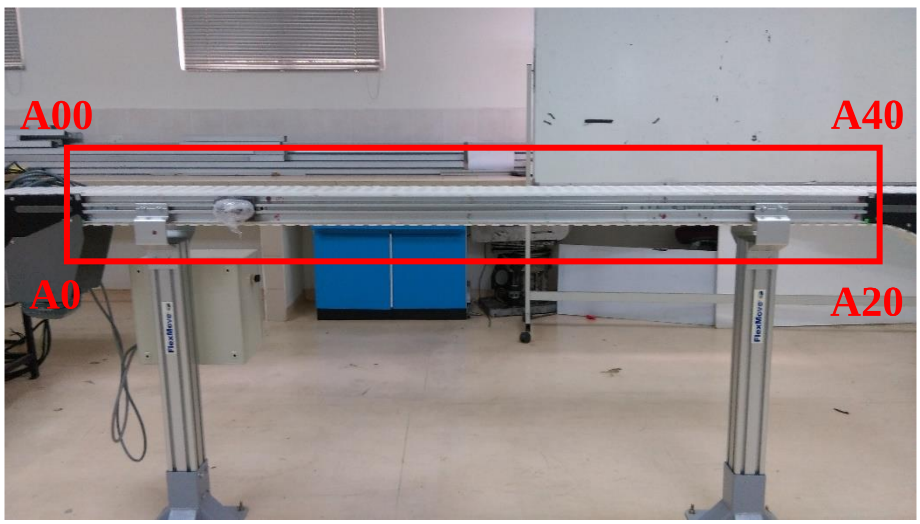


Figure 3.4: Horizontal aluminium beam of the conveyor

The length of the horizontal beam is 200cm and was divided into 11 points with 20cm interval in between. There are upper and lower points as the surface area of the beam is quite large and hence, there is 22 points on each side. The beam was divided into two sides; side A and side B. Side B is where the motor is located and a total of 44 points geometry was constructed. Side A coordinates starts with A0 and ends with A40 and the similar denotation applied on side B. Figure 3.5 and 3.6 illustrate the geometry constructed on the beam. This geometry was later referred in all the measurements including simulation model as well. The same geometry was constructed in LMS testlab for modal analysis.

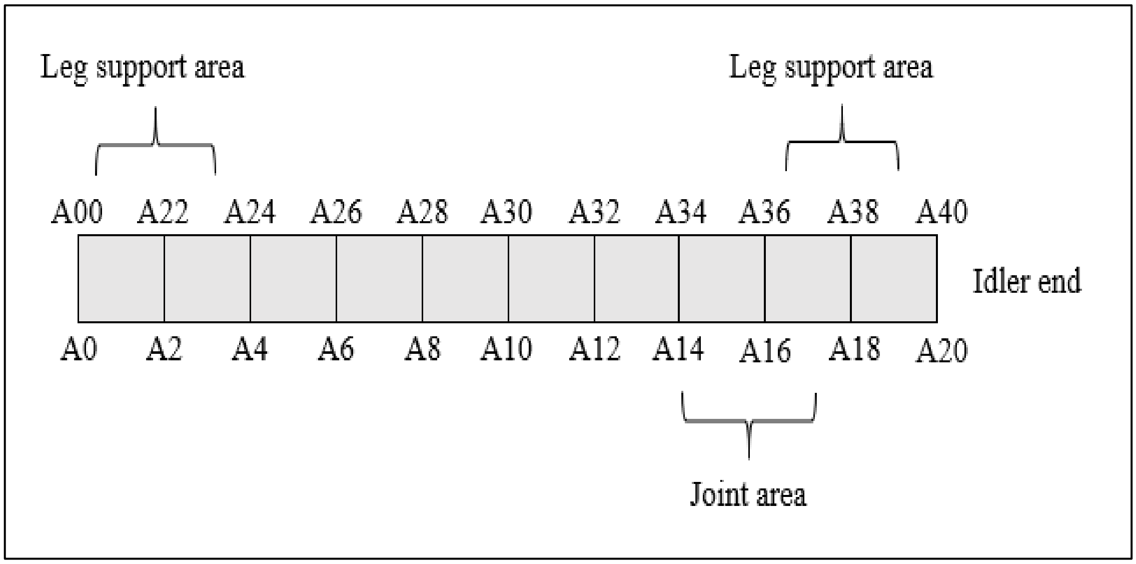


Figure 3.5: Side A of 22 points.

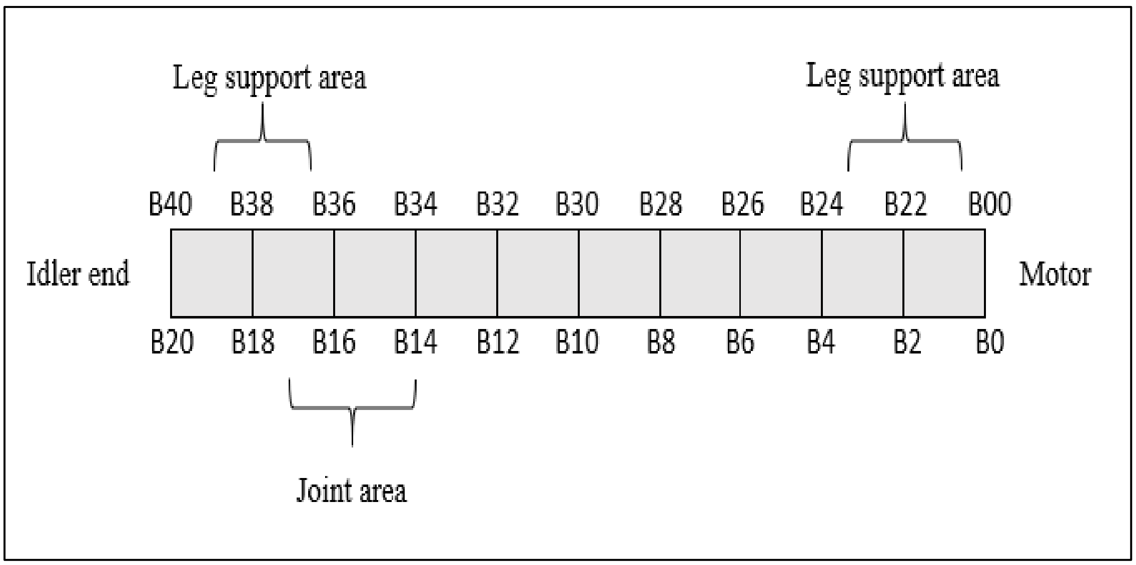


Figure 3.6: Side B of 22 points.

3.4 Microflown Scan & Paint sound measurement

Microflown Scan & Paint sound measurement offers a fast solution for processing near field scanned measurements for sound source localization [21]. Microflown Scan & Paint system is a complete setup for acoustic measurement and it is portable which allows users to utilize the setup in any kind of measurement environment. Figure 3.7 shows the actual measurement setup and the equipment involved are Microflown Scan & paint software, PU regular probe, video camera and Scout 422 Scoutend Data Acquisition hardware. The measurement was done by moving the PU probe over the object in a specified path while the measurement being recorded by a video camera.

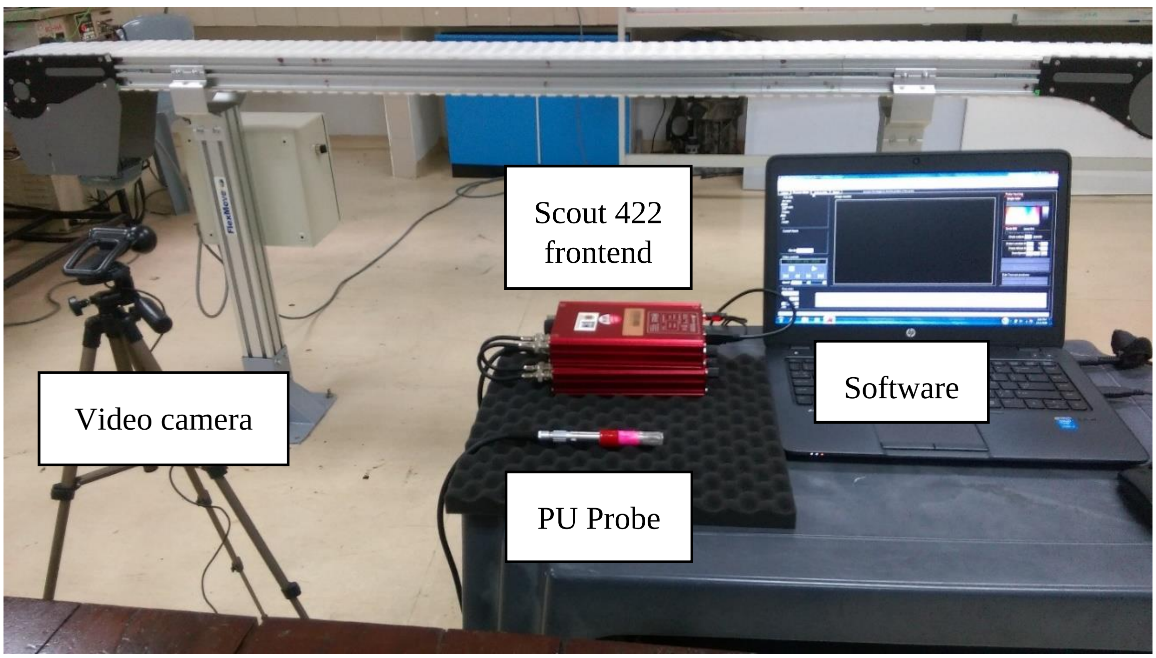


Figure 3.7: Sound measurement setup using Scout 422 frontend (Data Acquisition), video camera and software.

The conveyor was divided into three main categories; side A, side B and motor side. Side A and side B were divided into three parts as the conveyor beam is too long and the camera couldn't cover the total length. The surface of the conveyor system which is perpendicular to the camera lens was scanned with PU probe and the whole measurement process was recorded. The recorded audio and video data were synchronized atomically by the software and the measurements were ready for further processing. For further processing, the probe was marked to help the software to identify and mark the probe location. From each frame of the video, the position of the probe was detected, and auto-tracking feature was enabled to develop the full path of the probe. The resulting path was processed automatically by the software and the resulting readings were laid upon the image of the video which in turn provides the colour map of noise level on the image itself. This helped to visualize the noise level and originating source of the noise. The colour map generated was analyzed for sound source localization [22]. High sound pressure level area was highlighted and referred again after the structure was modified. This experiment was repeated with modified structure and the improved conveyor sound pressure level was compared.

3.5 Vibration Displacement measurement

This experiment was conducted to determine the vibration in terms of displacement at different points along the conveyor beam [23]. The LK-G series laser head was used to measure the displacement per time frame at different points. The laser head was located 10cm from the measurement point and the displacement of the laser was adjusted using the displacement adjustor which was controlled by all-in-one controller LK-G30001V. The data recorded by the laser head was sent to the laptop via the USB cable.

The provided software, LK Navigator which was incorporated together with the LK-G series laser head acts as the data acquisition platform and collects all the data to be stored in the internal memory of the LK-G. Other than displaying real time measurements, LK Navigator software functions features enlarging, reducing and overlapping the reading of measurements using the cursor and other functions for data analysis. For instance, the data collected by laser head were gathered and plotted into a displacement curve against time. Few points on the constructed geometry were marked, and the measurement was taken accordingly at all the points.

During the experiment, few precautions were taken to improve the accuracy of the experimental results. The average of three repeated measurements was taken. Besides, the laser was aligned outside the chain units because the chain units' displacement was not considered as shown in Figure 3.8. For this experiment, the primary target was the displacement of the conveyor beam at specific points and hence, it was vital to ensure the laser was aligned outside the chain units to prevent the chain units from interfering the results.

Figure 3.8: Laser alignment outside the chain units

From this experiment, a displacement curve against real time was obtained. This displacement curve provided information on maximum and minimum displacement. Displacement difference was calculated from the maximum and minimum value. The displacement difference values were collected at different points and this experiment was repeated again after the DVA was installed at the conveyor system. The displacement difference value before and after modification were compared to validate the feasibility of the tuned mass absorber.

3.6 Spectral Analysis

Spectral analysis is where structural vibration can be measured with electronic sensors that convert vibration response into electrical signal. The signal is generally divided into frequency and time domain. Figure 3.9 shows the actual measurement setup on the conveyor structure and the equipments involved are accelerometers, LMS SCADAS mobile data acquisition hardware and LMS testlab software. Accelerometers were used to detect vibration response and transmit the electronic signal to LMS SCADAS mobile data acquisition hardware and into the LMS testlab software. The software is able to convert time signals into frequency domain by Fourier Fast Transform (FFT) automatically in real time.

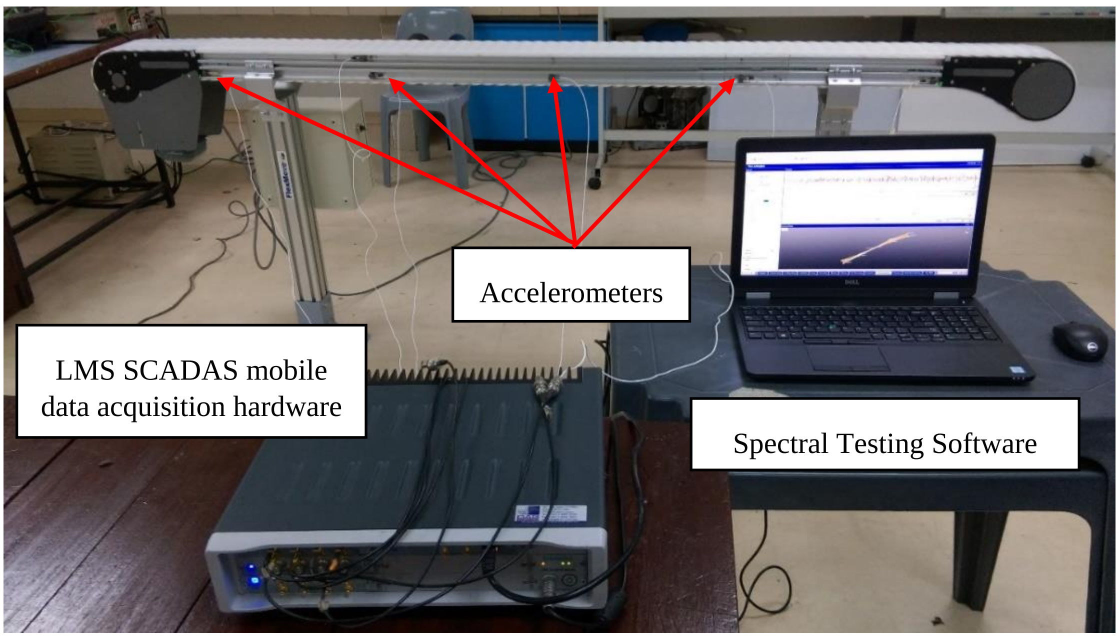


Figure 3.9: Actual measurement setup of spectral analysis

The accelerometers used were Dytran model 3055B, General Purpose Accelerometer as shown in Figure 3.10. One of the accelerometers was denoted as the reference point and other accelerometers were placed on different points. The selected measurement points on the structure were converted to a 3-Dimensional geometry by using LMS software. Accelerometers were connected to LMS SCADAS mobile data acquisition hardware and the hardware was connected to the laptop with the installed LMS testlab software (Spectral Testing) via LAN cable. The measurement was done with the first step of calibrating the accelerometers using calibration exciter Type 4294 Bruel & Kjaer with appropriate parameters related. Then, the vibration signal was collected from point to point based on the constructed 3-D geometry from the software while the conveyor was running at normal condition.

As a result, a geometric model of the conveyor structure moving in a pattern proportional to the recorded data was visualized and animated. The vibration response (frequency and time domain) of the conveyor structure running at 20Hz in the y-direction was taken and investigated with spectral testing of the LMS software. The acceleration response in frequency and time domain were studied further to identify the peak acceleration response and to identify the natural frequency of the structure. The acceleration response of the structure was measured again through spectral analysis after the structure was modified with damped DVA. The acceleration response of the improved conveyor was compared with the acceleration response of the original conveyor to validate the effectiveness of the damped DVA by distinguishing the peak response.

Figure 3.10: Dytran model 3055B, General Purpose Accelerometer