VIBRATION AND NOISE ATTENUATION ON A STRAIGHT CHAIN CONVEYOR SYSTEM

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May 2019

This dissertation is submitted to
Universiti Sains Malaysia
As partial fulfilment of the requirement to graduate with honours degree in
BACHELOR OF ENGINEERING (MECHANICAL ENGINEERING)

School of Mechanical Engineering
Engineering Campus
Universiti Sains Malaysia

DECLARATION

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

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ACKNOWLEDGEMENT

I would like to express my gratitude to all who gave me the possibility to complete this project. Without their patience and guidance, I would not able to carry out the investigation of my final year project. Thereby, I would like to take this opportunity to thank them for their unceasing help and support.

First and foremost, I would like to thank my project supervisor, Prof. Zaidi Mohd. Ripin, who spent his precious time to discuss and guide me throughout my whole final year project. He gave me plenty of suggestions, ideas and guidance for my project. I am greatly indebted to him for all the effort and faith he had on me.

Secondly, I want to thank the technicians especially Mr. Wan Mohd Amri Wan Mamat Ali who always works with me for the experimental set up and some fabrications. Besides that, I would like to express my appreciation to Dr. Chan Ping Yi, who provided me suggestions and guidance regarding to my research. Next, I would like to thank Mr. Nabil for his help in the setup of the devices. Million thanks to them for helping me to conduct the experiment and assisting me with the instrumentations.

Not to be forgotten, my gratitude to my fellow friends who also tried their best to complete their own project. I want to thank them for all their help, support, motivation and valuable hints.

Lastly, I would like to thank my family for their love and continuous support during my intricate times in accomplishing this project.

VOON ZHEN LIANG

May 2019

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LIST OF ABBREVIATIONS

Symbol	Description
Disp.	Displacement
FFT	Fast Fourier Transform
SP	Sound Pressure
SPL	Sound Pressure Level
SPLD	Sound Pressure Level Difference
SPPR	Sound Pressure Percentage Reduction
PR	Percentage Reduction
z-Acc.	Vertical Acceleration

ABSTRAK

Kebisingan mesin penghantar ditunjukkan sebagai komposit dari pelbagai mekanisme penjanaan bunyi bising, seperti hentaman antara rantai dan roller idler, hentaman pada sisi dinding, hentaman pada plat flex, pengikisan geseran dan getaran struktur. Dalam projek ini, sistem penghantar siri FlexMove FL 150 telah dipelajari. Sistem penghantar ini menghadapi masalah kebisingan berkala manakala tahap tekanan bunyi tertinggi berlaku pada sambungan kanan serta balok sokongan dengan 81 - 83 dBA. Penyelidikan ini bertujuan untuk menentukan punca sumber bunyi dominan semasa operasi dan seterusnya, mencadangkan satu cara penyelesaian untuk mengurangkan tahap bunyi. Tiga pendekatan untuk mengekspresikan sinaran bunyi dilakukan - pengukuran pelepasan menegak dan sisi tanpa penghidupan mesin dan pengukuran gabungan antara anjakan laser, pecutan dan mikrofon tahap tekanan bunyi. *Fast Fourier Transform (FFT)* dilakukan atas beberapa parameter yang diukur untuk mengenal pasti kekerapan berkaitan. Keputusan eksperimen menunjukkan bahawa bunyi dominan yang berlaku di bahagian bawah sistem penghantar ini adalah disebabkan terutamanya oleh anjakan menegak unit rantaian, yang mengakibatkan ketulan unit-unit rantai atas rel panduan. Untuk mengurangkan daya impak, lapisan lembut dipasang pada rel panduan untuk menyerap hentaman. Hal ini mengurangkan kadar hentaman. Selepas penampalan Polimer A ke beberapa segmen rel panduan, tahap tekanan bunyi dan pecutan struktur telah berjaya dikurangkan sebanyak 4.4 dB dan 86.27 % masing-masing.

ABSTRACT

The conveyor noise is shown to be a composite of noise generating mechanisms, such as impacts between the chain and the idler roller, impacts on the sidewalls, impacts on the flex plate, frictional scraping and structure-borne vibration. In this project, the FlexMove FL 150 series conveyor system was studied. This conveyor system encountered periodical noise issues where the highest SPL was recorded at the right joint and the leg support with 81 – 83 dBA. This research aims to ascertain the source of the dominant noise during operation and subsequently, propose a solution to attenuate the sound pressure level. Three approaches for expressing the noise radiation were conducted – offline vertical and lateral clearance measurement and combined measurements of laser displacement, acceleration and microphone sound pressure level. Fast Fourier Transform (FFT) was performed on some measured parameters to identify the relevant frequency. Experimental results revealed that the dominant noise in the lower portion of the conveyor was mainly caused by the vertical displacement of the travelling chain units, resulting in the slapping of the chain units onto the slide rail. To reduce the impact force, a soft layer was installed on the slide rail to absorb the impact. This reduces the peak impact. After the adoption of Polymer A onto several segments of the slide rails, the sound pressure level and the structural acceleration had been successfully reduced by 4.4 dB and 86.27% respectively.

CHAPTER 1 INTRODUCTION

1.1 Overview

In this chapter, the definition and types of industrial conveyors, types of chain conveyors, factors that introduce noise to the conveyor system and the effects of noise towards health are introduced in the following subtopics. This project is endorsed by FlexMove System Sdn. Bhd. to be used as final year project. The short foundation of this undertaking is clarified, and the project layout is exhibited.

1.2 Project Background

In the material handling and packaging industries, a conveyor is a significant mechanical handling equipment involving a quick and efficient conveyance from one location to the other for different loads, especially for bulky materials. Conveyor systems are used extensively due to their high durability and reliability in material's conveyance and warehousing. There are various types of conveyor systems based on different principles of operations namely: gravity, belt, screw, bucket, vibrating, pneumatic/hydraulic, chain, spiral, grain conveyor systems etc. (Gupta et al., 2015). Among all, chain-type is one of the commonly used conveyor systems in the manufacturing industries.

Chain conveyors are powered by a continuous chain consisting of links with special top sections that create an articulated by essentially continuous, flat surface for handling objects placed on it. Chain conveyors are prominent in the ease of installation, flexible layouts and material flow designs and requires minimal maintenance. Hence, they are extensively used in transporting heavy loads particularly in assembly and painting operations such as automotive and appliance manufacturing plants (Chain Conveyors, 2019).

There are five types of chain: cast iron chain, cast steel chain, forged chain, steel chain and plastic chain (Otoshi, 1997). Plastic conveyor chains are used in the automotive, electronics, furniture, food, beverage and packaging industries and in the production and conveying of cosmetic, pharmaceutical and chemical products (Mitschke, 2010). Meanwhile, metal conveyor chains are widely used in mining operations and has caused dominant noise during operations (Peter et al., 2009). Plastic conveyor chains are up to 40% lighter weight than their steel counterparts that make plastic chain conveyors can be driven by smaller and more energy-efficient motors. This significantly reduces the energy consumption and the operating costs. Furthermore, using plastic chain also cuts noise emission levels by up to 80% compared to metal chains (Mitschke, 2010).

However, the noise originated from the plastic chain conveyors are still not neglectable. Noise has influential effects on human health regardless of the level or the exposure period (Onder et al., 2012). Recent scientific studies have shown a strong relationship between environmental noise and an elevated risk for high blood (Haralabidis et al., 2008). Emotionally, long-term exposure to excessive noise in the industries correlates with a higher risk of depression, anxiety and insomnia (Haralabidis et al., 2008). Therefore, noise attenuation is important for the development of chain conveyor design.

There are many possible reasons for the noise occurrence such as impacts and scraping at various conveyor structure, impact between the roller idler and the chain, impacts on the sidewalls at path points of discontinuity under horizontal pivot and impacts on the flex plate (Durr et al., 2003), structure-borne noise (Brown, 2004), collision of the chain rollers and the sprocket teeth ("Noise Reduction in Chains and Sprockets,"), collision between chain units and the slide rail during ascending or descending and noise from the motor during operation.

In this project, investigation was done on FlexMove FL 150 mm aluminium conveyor provided by FlexMove System Sdn. Bhd. in order to ascertain the source of the dominant noise generated from this plastic chain conveyor systems. The right side of the conveyor system comprises an idler roller which permits the relative rolling movement of the chain units to the rotating torque of gear motor. The total number of chain units on the conveyor system is 153 units that makes up a closed loop running continuously in clockwise direction.

The previous obtained results were used as the reference. Based on the precedent results, it showed that noise dominates at a region near the right leg supports of the FlexMove FL conveyor system at a sound pressure level ranging from 75.5 to 81.5 dB(A). Within the noise level ranges from 65 to 95 dBA, long term of exposure may origin psychological disorders in people (Onder et al., 2012).

1.3 Problem Statements

When the FlexMove FL 150 series conveyor is operating, an apparent noise of multiple impacts and a high pitch noise from the geared motor could be noticed. The noise radiated is repetitive and annoying. From last year's senior thesis, it showed that the highest sound pressure level was recorded at the right joint and leg support area (81 – 83 dBA).

1.4 Objectives

The following objectives are aimed in this scope of study, parallel with the main concern in problem statements.

1. To ascertain the cause of the dominant noise generated from the bottom part of FlexMove FL series straight conveyor.
2. To develop an effective approach to attenuate the noise level.

1.5 Scope of Work

The scope of this research is limited to the investigation of noise due to secondary motions (vertical and lateral movements) in the dominant noise regions located along the bottom portion of the conveyor system. It is not the intention of this paper to address conveyor noise issues due to angular fluctuation of the driving chain, motor, friction and wear of components. The experimental results obtained in the previous study was utilised as the reference of the noise and vibration condition on the FlexMove FL series conveyor system as shown in (Figure A9 to Figure A11). Noise attenuation method was applied to mitigate the noise emitted due to secondary motions without altering the vibration level of the conveyor stringer. Note that all tests were conducted with the chain conveyor system in an unloaded condition.

CHAPTER 2 LITERATURE REVIEW

2.1 Overview

Conveyor system is the primary means of moving materials in the manufacturing process especially for high throughput industries. Chain conveyor finds wide application in the automotive, electronics and also food industries due to its flexibility and the ability to change direction within tight space (Patil, 2015). One of the biggest drawbacks of the chain conveyor is the high level of noise generated during operation which may affect the well beings, comfort and even hearing risks to the workers. In this chapter, past research related to noise and vibration study on conveyor system are reviewed. The review is segmented into the following sub-areas: related secondary motion analysis, vibration analysis and noise analysis and mitigation methods. This chapter also introduces and reviews the methods and focused measuring parameters that were used by previous researchers in determining the noise level.

2.2 Related Secondary Motion Analysis

Tan et al. (2014) discussed the technique to determine instantaneous piston skirt friction during piston slap. They believed that the cylinder block vibration and the radiation of undesirable noise were caused by the impact of the piston to the cylinder liner. Besides, one of the main sources of noise is due to the mechanical impact of the piston on the cylinder wall (Gerges et al., 2000). Investigation of the piston secondary movement utilising a laser displacement sensor demonstrated a solid relationship between the friction and piston rotational movement at high engine rate (Tan et al., 2011).

Veikos et al. (1992a) exhibited that the causes of the longitudinal and transversal vibrations on the chain strains were the combined action of the polygonal effect and the roller impact, in which the flexibility of the links plays a significant role. A discrete model composed of the polygonal action of the chain strands were also suggested. Furthermore, the driving span of the chain was modelled by lumped masses associated by linear springs and considers the coupling between the longitudinal and transversal vibrations and the moving boundary conditions. Nevertheless, the main concern was not situated at the effect of chain guides in the drive and the angular speed fluctuation of the driving sprocket.

Fritz et al. (1995) presented a methodology where the roller-sprocket and the guide chain contacts are treated as unilateral constraints. In this study an integrated model describing the complex dynamics of the roller chain drive including chain guides and moving sprockets is proposed.

Dolatabadi et al. (2014) investigated on the impact-induced oscillations and noise in lubricated conjunctions on the piston-liner. In his study, tribodynamics theories were implemented into the computation of the impact energy at the lubricated piston-liner conjunctions. Analysis was done on the desired locations on the engine block surface through the sound pressure level (SPL), which was pre-converted from the vibration power data. Besides, he utilised the non-linear characteristic vibration absorbers to control the piston's secondary motion, which in turn diminishing the severity of impact dynamics. In majority piston impact-noise investigation, simple spring-damper elements were implemented at the conjunctions for simplicity of analysis (Cho et al., 2002; Ohta et al., 2011). Establishment of the top and bottom eccentricities of the skirts were used as the approach to reveal the number of potential impacts.

2.3 Vibration Analysis

In the past study of conveyor has been concentrated on the belt conveyor, particularly in the vibration and also the noise level associated with the operation of such conveyor. The belt tension and dynamic response has also been the object of study of conveyor vibration emphasizing the longitudinal vibration of the belt (Li et el., 2018) . This is mostly related to the starting, braking and loading of the system (Qin et al., 2008) and also the transverse vibration related to the resonance (Zhang et al., 2014). Vibration monitoring, simulation and the analysis of conveyor driving unit of a belt-typed coal transporter has shown that vibration amplitude is caused by the luffing angle and the conveying rate especially in horizontal direction (Laksana Guntur et al., 2017). A more detailed analysis of the non-linear transversal vibration of the conveyor belt has shown the stabilization of the belt due to the weak nonlinearity of the system (Suweken et al., 2003). In most cases vibration measurements are carried out to monitor the condition of the machines associated with conveyor (Damnjanović et al., 2017; Liu et al., 2018; Ojha et al., 2014).

Salokyová et al. (2016) researched on the influence of the mechanical vibrations on the production machine to its rate of change of technical state, concerning the amount of vibration in the bearing house of a rotating lathe machine. In his research, Fourier transformation was used to transform the time stamp vibration's acceleration gathered during machine operation to the frequency spectrum in the range of 3.0 to 10.0 kHz. He also mentioned that small vibration can be neglected as it did not bring negative impact to the system. Vibration can be classified into two types which are free vibration and forced vibration (Ma´dl, 1990). Force fluctuation can be occurred even the vibration is invisible by naked eyes, and it is mainly caused by the unbalanced attributes of the processed material and irregularities of the processed surface as cited in Ma´dl (1990).

2.4 Noise Analysis and Mitigation Methods

Li et el. (2014) presented the theoretical and numerical investigation on impact noise emitted by the collision of two cylinders. The similar investigation was also performed by Nishimura et al. (1963), Koss et al. (1973) and Mehraby (2011) with the collision of two spheres. Li developed a modified theoretical prediction model of collision cylinders based on the Palmgren's cylinder contact empirical model and acoustics theory to accurately estimate the impact noise emitted by the colliding cylinders. This is followed by a numerical simulation method combining the finite element method (FEM) and the transient boundary element method (TBEM) to verify the altered theoretical model.

Research on the continuous coal mining machine (chain conveyor) has been conducted unceasingly by many experts from National Institute of Occupational Safety and Health (NIOSH). In 2001, chain conveyor was pinpointed as the main cause of noise in the coal miner. In 2002, NIOSH conducted a noise control method by developing a chain conveyor with laminated flights (Kovalchik et al., 2002). In 2003, NIOSH proposed a noise mitigation method with the coating of heavy duty and highly durable urethane to the metal to metal and metal to coal contact (Durr et al., 2003). In 2005, the research was continued with the design and utilization of the dual sprocket chain to further attenuate the noise radiation (Peter et al., 2009).

In as early as 1986, literature suggested that retrofitting of the chain conveyor with a urethane coating or sleeve on the chain flights and the idler roller, constrained layer damping of the decks, and the use of an altered take up plate were several effective noise attenuation approaches (Burks et al., 1986; Pettitt et al., 1986). Owing to its elastomeric nature and combined properties, the shearing of the coating at the flight ends was reduced while extending the usability of the conveyor chain (Durr et al., 2003).

Durr et al. (2003) presented that, in order to develop engineering noise controls to attenuate noise generated, it is crucial to first uncover the location with dominant sound sources. Reverberation room was used for determining sound power levels of the operating coal miner before and after applying urethane coating at the idler roller. This results in an average sound power difference of 3.6 dBA through the urethane layering. Noise field measurement was also performed by positioning two tripods with microphones in an established coordinate system where the origin was in the middle of the machine. Sound level measurement was performed in this case with two-meter increments. A spreadsheet of results were imported to MATLAB to generate noise profile for each octave band, including overall A-weighted sound levels to demonstrate the individual sound levels and radiation patterns in octave and 1/3 octave bands (Kovalchik et al., 2002).

Furthermore, researches on the belt conveyor had been done by (Horstmeir, 1981) and Brown (2004). Based on Horstmeir (1981), the most dominant noise origins from the dynamic interaction at the belt or idler roll interface and structure-borne noise. Horstmeir reasoned that alteration of the idler roll surface just as damping treatments were conceivable noise mitigation methodologies. Other studies claimed that application of damping treatments towards idle roller have insignificant effect in reducing noise. They were suggesting that idler rollers with low total indicator run-out values (TIR) produced quiet conveyors. TIR is a measure of the gross “out of roundness” of the roller. As the roller surface is rotated past the head of a contact dial gauge, resting on that surface, the TIR is measured as difference between the maximum contact gauge deflection versus the minimum contact gauge detection.

Brown (2004) presented the results of a comprehensive analysis and testing of conveyors and components involved in belt conveyors. Brown stated that the conveyor noise is a composite noise and the most dominant noise is from the interaction between the belt and the roller. He proposed an idler roll surface profile measurement parameter - Maximum Instantaneous Slope (MIS) to interpret the operating condition and noise radiation potential of existing equipment. Average A-weighted sound intensity measurements were utilised using the scanning method in inclination to sound pressure for reducing the capturing of the adjacent and reflected noise. The experimental results from this study depicted there was a strong relationship between the velocity of idler roll surface profile parameter and the conveyor noise emissions for the range of idler types examined.

Besides, strong correlation exists between the idler roller surface parameter and the noise emission. Measurement of noise form belt-roller interaction had shown the noise level to be in the range of 50-60 dBA with different peak acceleration frequencies depending on the direction (Klimenda et al., 2016). There had been attempts to reduce the noise level with the use of nanostructured metal mesh-polyurethane composite material (Liu et al., 2012). The new chain was shown to reduce the noise level by 10-16 dB depending on the speed, however this is limited to the chain noise and not the conveyor noise itself. The structural vibration is an important part in the overall noise attenuation approach, and this is usually the approach adopted. Finite element analysis of the support frame can indicate important mode such as the torsional mode and the swing mode and from there vibration isolation can be included to reduce the transmitted vibration and subsequently the ensuing noise (Li et al., 2012). This can even be in the form of two-stage isolation to minimize the vibration transmissibility of the conveyor structure (Fiebig et al., 2017).

2.5 Summary

From the review, it is clear that there is limited publication on the noise and vibration analysis of plastic chain conveyor system. This is due to the complexity of the system as each solution is turnkey to suit the demand of the industry and is not readily analysed because of the non-linearity of the system (chain looseness, lateral and vertical impact of the guides can introduce very severe non-linearity into the system). The work presented here is based on measurement of vibration and noise of a straight chain conveyor system and the action taken to attenuate the noise level

CHAPTER 3 METHODOLOGY

3.1 Overview

Figure 3.1 shows the overall flowchart of the implementation plan to achieve the objectives of this study. The methodology is carried out in five stages.

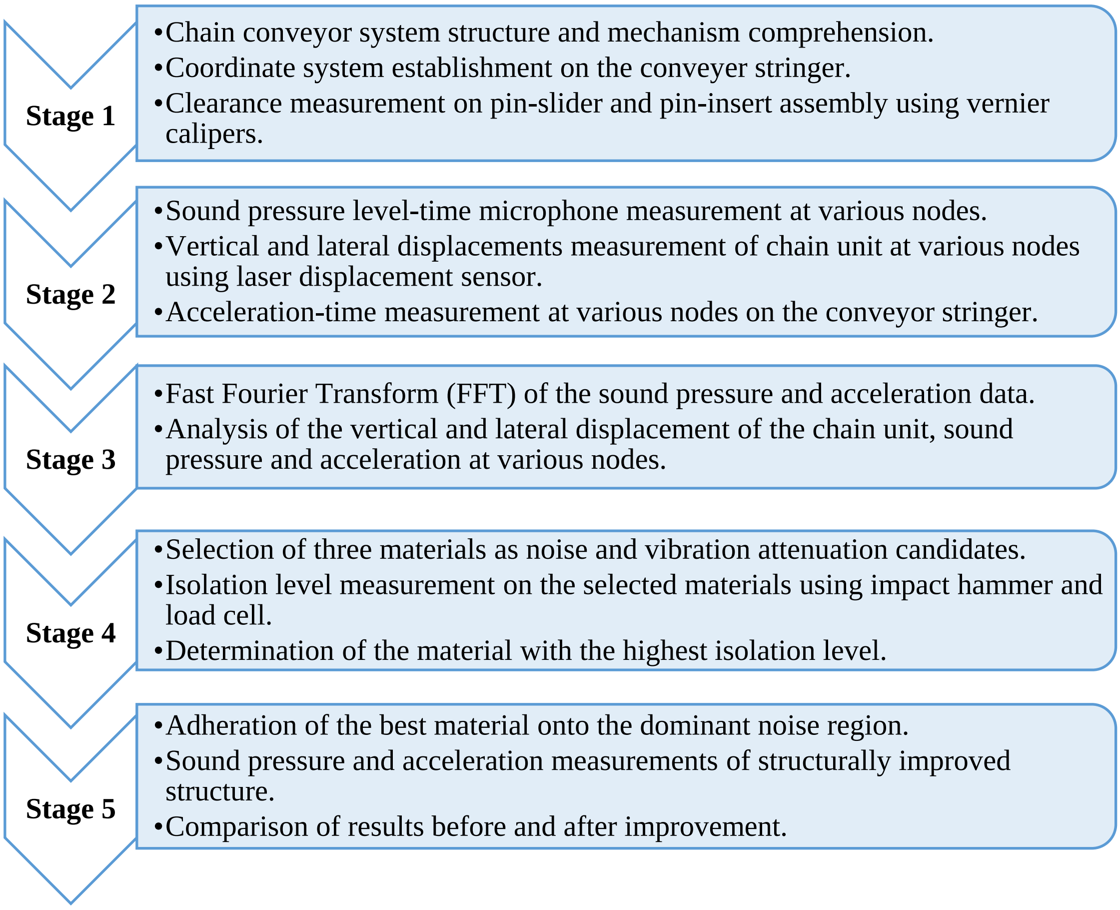


Figure 3.1: Flowchart of implementation plan

In the first stage, the project was initiated by comprehending the conformation of the structure of the conveyor system and the contacting mechanism of the chain units. Next, a coordinate system was established on the conveyor stringer for measurement purpose. The diameter of the pin and slider, pin and insert were measured using Vernier callipers to verify the existence of clearance in the assembly of pin into the slider and the insert.

In the second stage, BSWA MA211 ½” microphone was utilised to measure the sound pressure level (SPL) to determine the location with dominant noise in the bottom portion of the conveyor. The result from the SPL measurement was compared to the previous finding. Before the measurement of vertical and lateral displacements of the chain units, the vertical and lateral boundary limits were measured in offline condition by using Keyence LK-G152 laser displacement sensor. The similar sensor was then used to measure vertical and lateral displacement of the chain units at the dominant noise spots during machine operation. At the same time, sound pressure and vibrational acceleration measurements were executed by using BSWA microphone and DYTRAN 3055B2T accelerometer respectively to obtain data with the identical time stamp.

In the third stage, analysis work was done onto the obtained experimental results including vertical and lateral displacements, sound pressure level and acceleration to evaluate the relationship among the parameters. Fast Fourier Transform (FFT) function in MATLAB was carried out to convert the time-domain sound pressure and acceleration data into frequency spectrum. All results were tabulated and discussed in Results and Discussion.

In the fourth stage, three materials were chosen and the impact force relative to time of each material was determined by measuring the applied impact load using Kistler Quartz Impulse Force Hammer (Type 9724A5000) and the transmitted force measured using Kistler 4-Component Dynamometer (Type 9272) to investigate the effectiveness of the materials to reduce the impact.

In the final stage, the material that can reduce the most impact force was selected to be adhered to the noise spots to attenuate the noise radiated. The overall noise level and acceleration measurements were repeated at the six nodes. Eventually, comparisons of FFT graphs of sound pressure level and acceleration before and after alteration were evaluated to determine the effectiveness of the chosen material in mitigating noise and vibration.

3.2 Structure and Mechanism Establishment of the Conveyor System

3.2.1 Structure Comprehension of Conveyor System

The conveyor used in this analysis is a general-purpose chain conveyor with white acetal chain of individual length of 63 mm and 150 mm width (Figure A2) and a span length of five metres of the connected chains as shown in Figure 3.2. The conveyor is driven by a 0.55 kW gear AC motor (Figure A4) at 1350 rpm through a polyamide-glass fibre composite drive sprocket (Figure A1). The chain conveyor is supported on an extruded 150 x 75 mm aluminium beam (Figure A2) and has longitudinal guides to prevent the chain from excessive lateral motion. Based on the observation, there is a total number of 153 chain units to form a loop which rotates about a gear motor and a roller idler. The roller idler only provides support for the sliders to slide tangentially from the bottom part to the top part. Moreover, two chain units are linked together by a pivot pin passing through the slider hole of the first chain and the insert (wedge and friction) of the second chain as indicated in Appendices A (Figure A3).

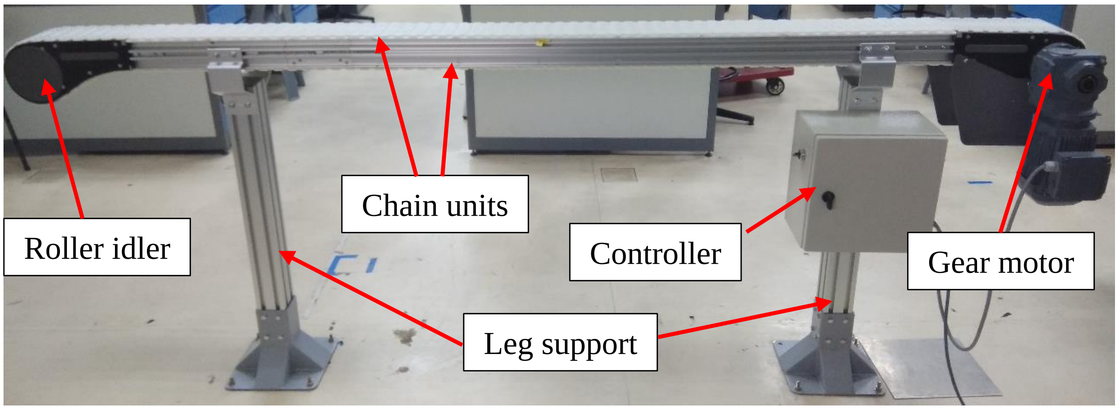


Figure 3.2: Chain type of conveyor system

3.2.2 Contacting Mechanism of Conveyor System

Based on Figure 3.3, the upper part's chain units travel along the slide rail with two contacting points located below the flat plate as indicated by the green triangles. Besides, the travelling chain units in the lower portion of the conveyor system comprises two contacting points located at two sliders as indicated by the red triangles.

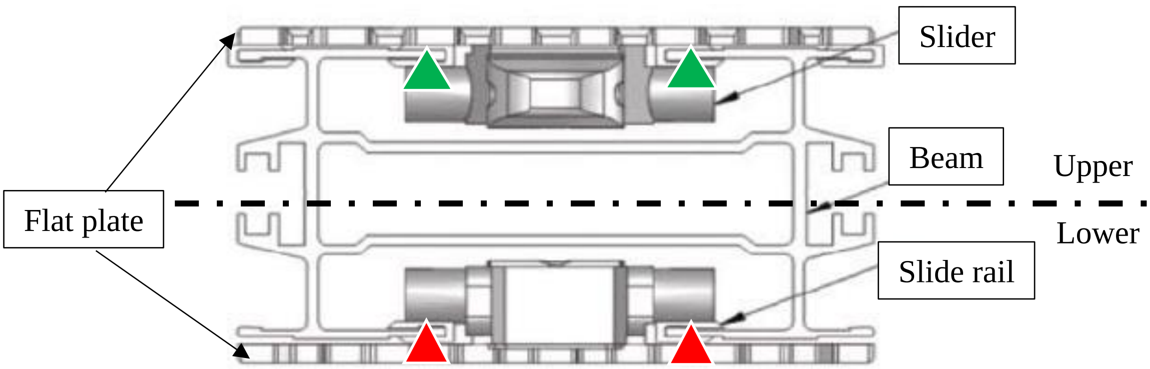


Figure 3.3: Upper and lower contacting mechanism of the chain unit on the rail

3.3 Coordinate System Establishment

The length of the conveyor horizontal beam between the motor and the idler roller end is 2 m. The beam was divided into 11 nodes (L20, L22, L24, …, L40) located 200 mm apart from each adjacent node as shown in the Figure 3.4. The established nodes were referred as the points of interest in which all the measurements were taken place at the nodes. The aim of the establishment is to standardize the measurement distance and to distant the influence of one node two the other, especially during the measurement of the sound pressure level.

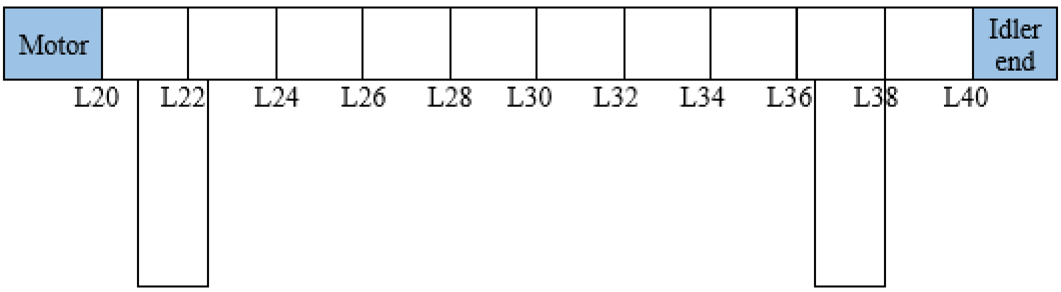


Figure 3.4: Coordinate system establishment

3.4 Pin-Slider and Pin-Insert Clearance Measurement

Figure 3.5 shows the image of the bottom view of the drive unit and the pin. The aim of the clearance measurement was to determine the existence of the clearance between the stainless-steel pivot pin and insert (wedge and friction), pivot pin and the slider, which can origin longitudinal impact during traveling. The inner diameters of the sliders and the insert and the outer diameter of the pin were measured using Vernier callipers and the results were tabulated in Result and Discussion.

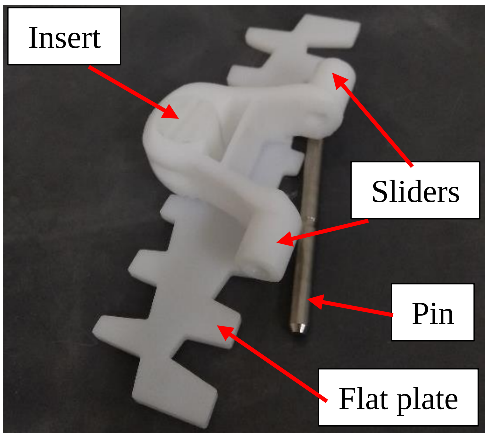


Figure 3.5: Chain unit and pin

3.5 Combined Displacement, Sound Pressure and Acceleration Measurements

Three experiments were conducted simultaneously to perform noise analysis which are laser displacement, sound pressure and acceleration measurements. The aim of combination of these three experiments is to determine the relationship between the vertical and lateral displacement of the chain unit, acceleration of the conveyor structure and the noise radiated at similar time stamp at various locations when the conveyor system is operating. The overall experimental setup was illustrated in Figure 3.6. More detailed explanation for each individual setup will be discussed in the following sub-section.

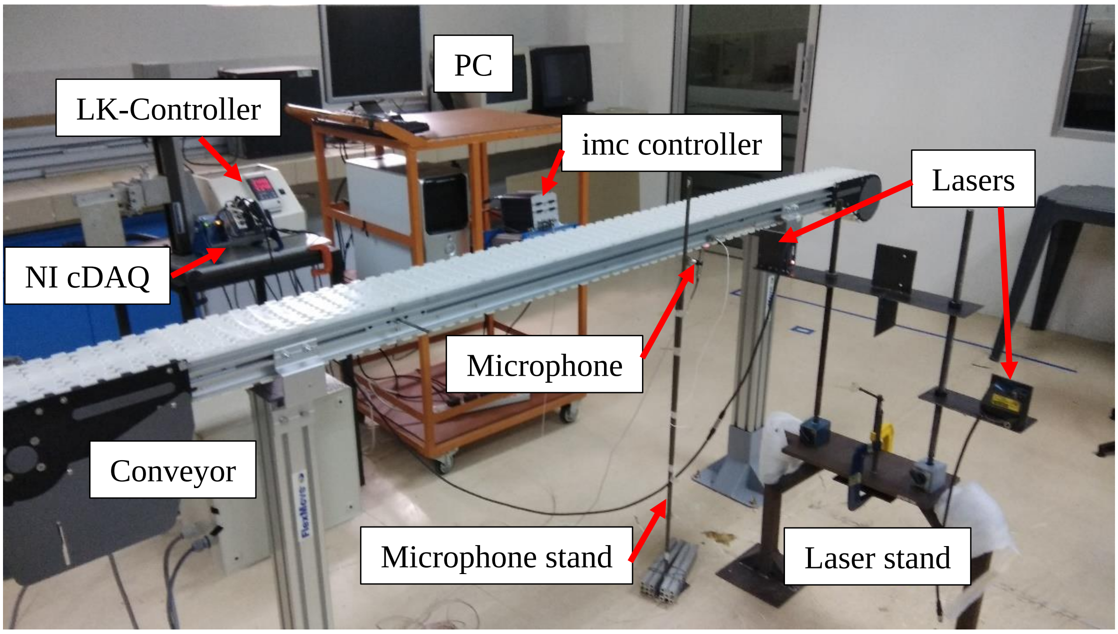


Figure 3.6: Experimental setup for the combined test

3.5.1 Sound Pressure Measurement

This experiment was conducted to determine the sound pressure and sound pressure level at established nodes along the lower portion of the conveyor system during machine operation. BSWA MA211 1/2" pre-polarized microphone which is readily available in Vibration Lab was utilised. The BSWA MA211 microphone is high performance pre-amplifier with low inherent noise, high input impedance and flat frequency response which ensures high quality acoustic measurements. A stand was fabricated to lift the microphone to the desired height and a microphone holder was used to hold the microphone in place during measurement as shown in Figure 3.7. Besides, IMC CS-8008 controller (Figure 3.8) was used as the data acquisition device for the microphone sound pressure and sound pressure level readings. The results were displayed in a graphical form in the imc DEVICES software installed in PC.

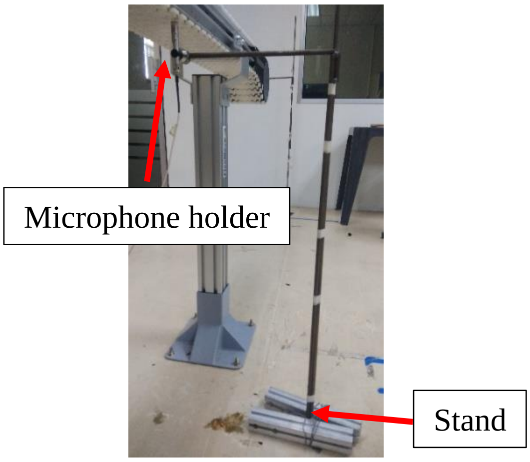


Figure 3.7: Microphone stand

Figure 3.8: imc CS-8008

During data acquisition, the microphone was offset 5 mm from the chain units at each specific node. Before measurement, the conveyor was allowed to run few seconds until it reaches steady operating speed. The sound pressure readings were recorded simultaneously with displacement and acceleration measurements for 30 seconds and the data from both graphs (units in mPa and dB) were exported to FAMOS for saving purpose. Moreover, the data will be imported into MATLAB to perform Fast Fourier Transform (FFT). The coding was inserted in Appendix as Figure A13. These steps were repeated after structural alteration.

Besides that, the background noise and motor noise were also measured to determine the frequency range of both noises via FFT plot. Therefore, the sound wave which oscillates at the same frequency and amplitude as background and motor noise can be eliminated during data analysis.

3.5.2 Laser Displacement Measurement

This experiment was conducted to determine the boundary limits and secondary movement of the chain unit at different locations along the lower portion conveyor system by using the mid-range small spot Keyence LK-G152 laser displacement sensor available in Vibration Lab of Mechanical Engineering School. This is due to its non-contact measurement mechanism, high sampling frequency (20 µm), high linearity (±0.05% F.S.), high repeatability (0.5 µm) and long measuring range of 150±40 mm. Besides, it is suitable for both diffused and specular reflecting surfaces, which made it appropriate for the measurement of the plastic surface chain. The secondary motion measurement was divided into two parts: vertical displacement measurement and lateral displacement measurement.

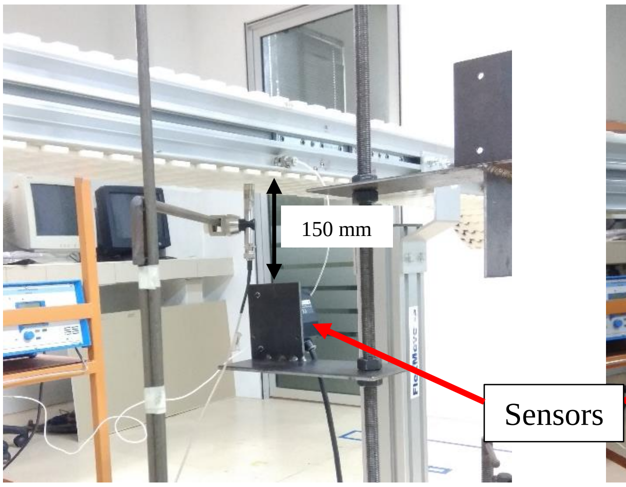


Figure 3.9: Setup for lower vertical displacement measurement

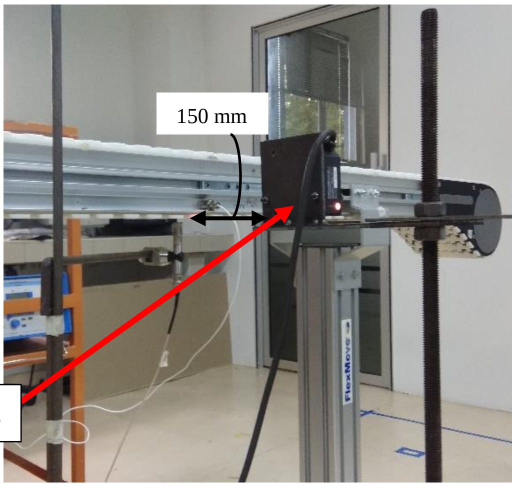


Figure 3.10: Setup for the lateral displacement measurement

During the experiment, the laser head was located 150 mm away from the measurement point and the position of the laser head could be adjusted by using a customized stand composed of lead screws, nuts, G-clamps and a sensor holder platform (Figure 3.11).

Figure 3.11: Customized laser displacement sensor stand

The readings from the laser will be displayed on the LK-GD500 display panel coupled with a LK-G3001(P) separate controller, which was used for control purpose. The output obtained from the controller was directed to NI cDAQ 9138 device to convert the series of displacement data into time-domain voltage data. The voltage-based displacement data was transferred via a USB cable to a developed GUI by using NI LabView installed in personal laptop as shown in Figure 3.13.

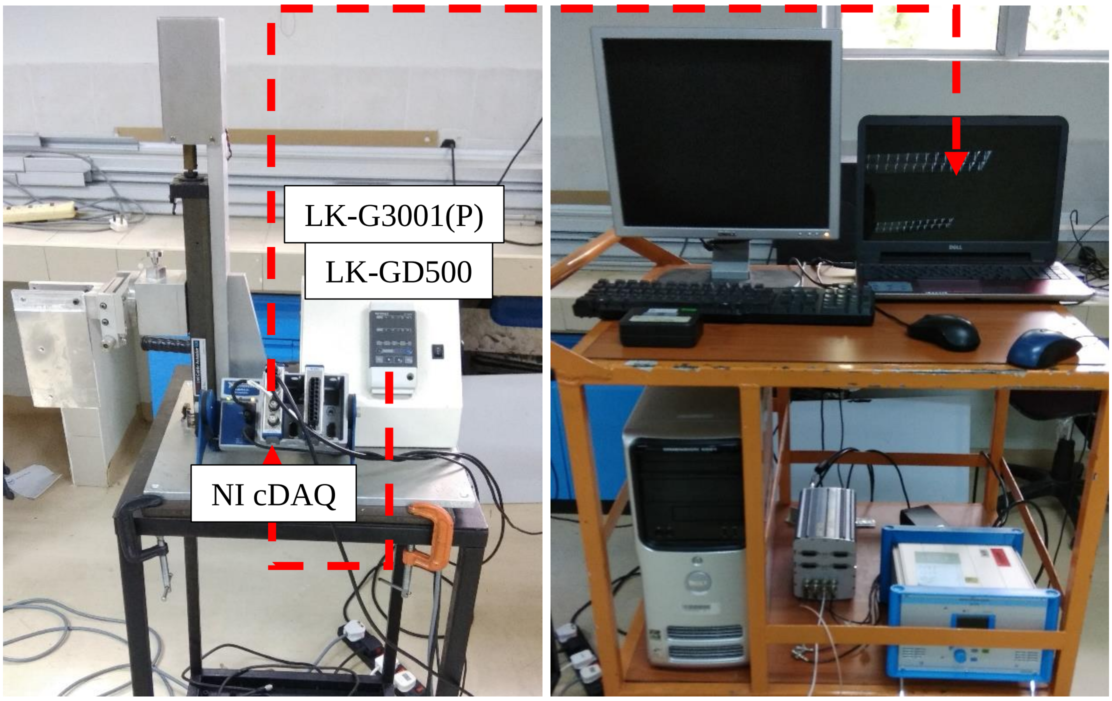


Figure 3.12: Flow of the data acquisition process

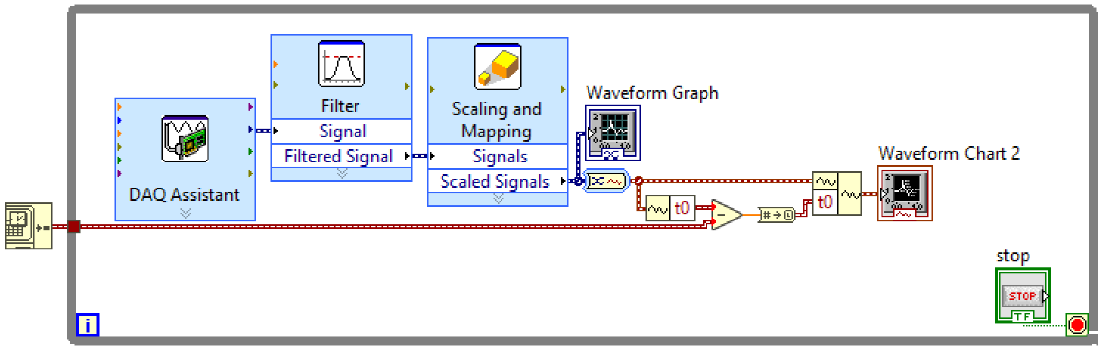


Figure 3.13: GUI for conversion of voltage to displacement

Based on Figure 3.13, cDAQ Assistant acts as the configuration block for receiving the voltage signal from the NI CompactDAQ where the sampling frequency and the number of signals retrieved can be adjusted in this block. Digital current was set in DAQ assistant as to yield bipolar (positive and negative) voltage values. Filter block is used to lowpass filter the white noise in the data obtained. In this case, 100Hz Butterworth filter was used. Meanwhile, since the voltage output is not equivalent to the displacement readings. Hence, Scaling and Mapping block was used to map the voltage value to the displacement value with suitable gradient (m) and intercept (c) values. The gradient and intercept were set as 2 and 0 respectively: 10 mm displacement will output 5 V whereas -10 mm displacement will output -5 V to the laptop via USB.

The boundary limits are the constraints that restrict the chain unit from excessive secondary motion. In this experiment, boundary limits were divided into four types which were upper limit and lower limit for vertical displacement whereas left limit and right limit for lateral displacement. The boundary limits were exhibited in Figure 3.14 and Figure 3.15 for better understanding. The space contained within the boundary limits is called clearance, which is the allowable space that permits motion. In this research, the vertical and lateral clearances were measured by using the laser displacement sensor as the parameters to set the boundary limits.

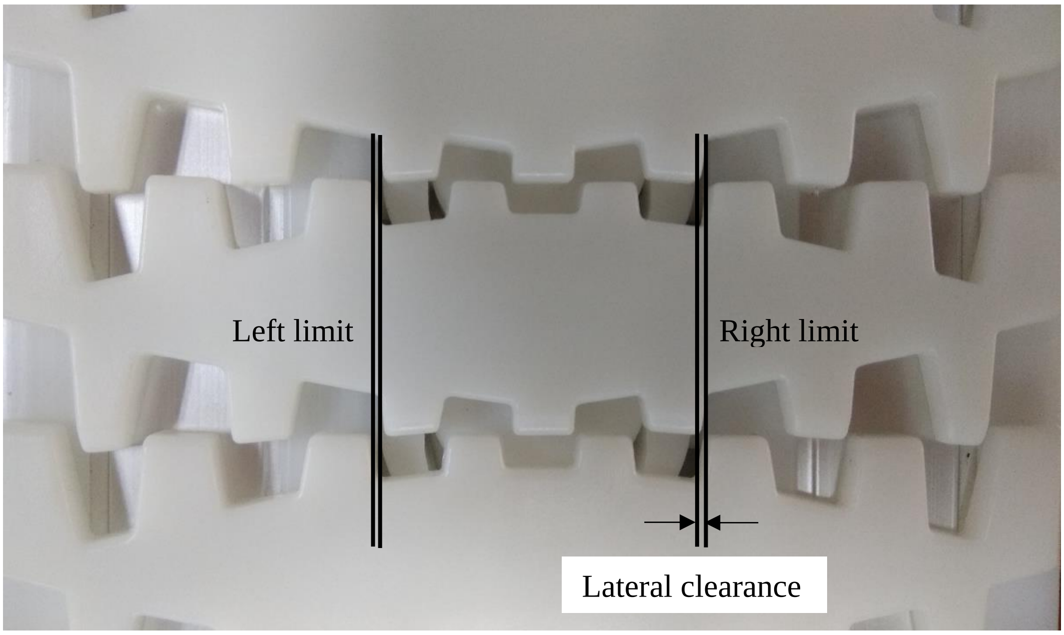


Figure 3.14: Indication of lateral clearance

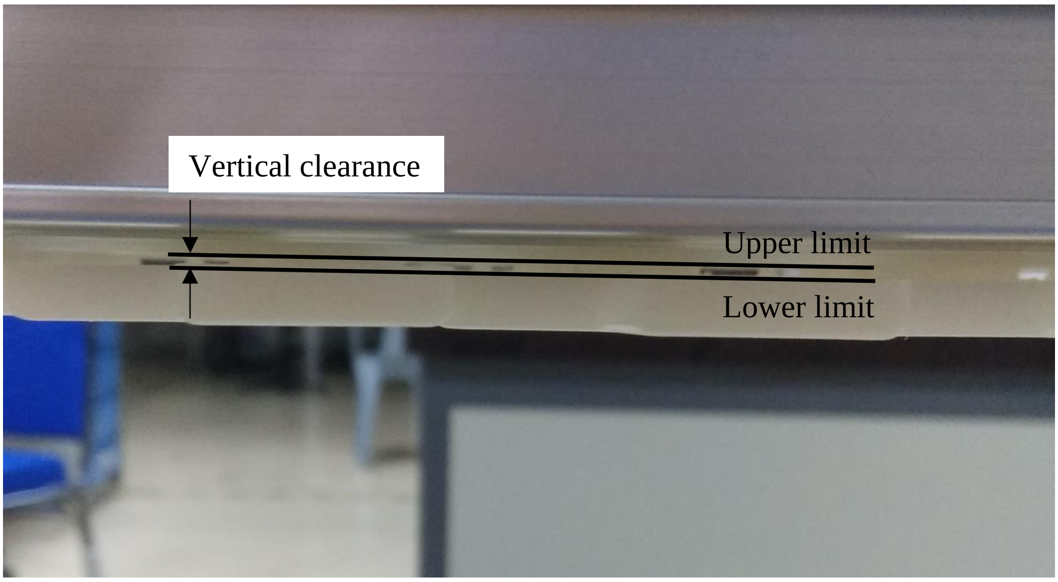


Figure 3.15: Indication of vertical clearance

3.5.3 Acceleration Measurement on Conveyor Stringer

For the acceleration measurement of the conveyor structure, DYTRAN 3055B2T accelerometer was used to measure the vertical acceleration at the established nodes. IMC CS-8008 controller was also used as the data acquisition device for accelerometer reading and the results were displayed in a graphical form in the imc DEVICES software installed in PC.

Figure 3.16: DYTRAN accelerometer mounted at one of the nodes on conveyor structure

During the experiment, the accelerometer was mounted at the node of interest on the conveyor stringer by using wax. The acceleration readings were recorded simultaneously with displacement and acceleration measurements for 30 seconds and the data from the output graph was exported to FAMOS for saving purpose. The data was saved into an excel file. Moreover, the data will be imported into MATLAB to perform Fast Fourier Transform (FFT). These steps were repeated after structural alteration. Besides that, the sensible background vibration was also measured to determine the frequency of background vibration via FFT plot. From here, the interference from the background vibration with certain frequency range can be filtered for more valid results.

3.6 Impact Test using Various Materials

Figure 3.17 shows the setup of the impact test. This test aims to determine the impact force reduction of various specimens. The setup of this experiment included a Kistler Quartz Impulse Force Hammer (Type 9724A5000) and a Kistler 4-Component Dynamometer (Type 9272). The dynamometer was connected to the multichannel charge amplifier (Type 5070) to output an electrical charge which varies in direct proportion with the load acting on the sensor (4-Component Dynamometer for Cutting Force Measurement in Drilling Type 9272, 2005). Moreover, the impact hammer was used to deliver a measurable force impulse to excite the chain connecting module under test (Quartz Impulse Force Hammer for Medium Force Range, Type 9724A…., 2009). The electrical signal from the impact hammer and the charge amplifier will afterwards be transmitted to the imc cs-8008 data acquisition device for real-time analysis in graphical form.

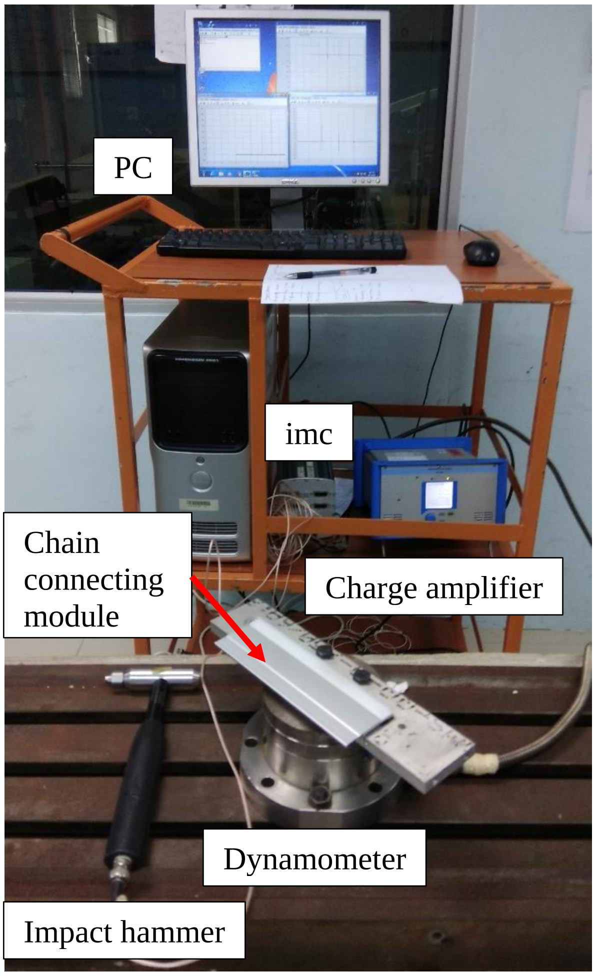
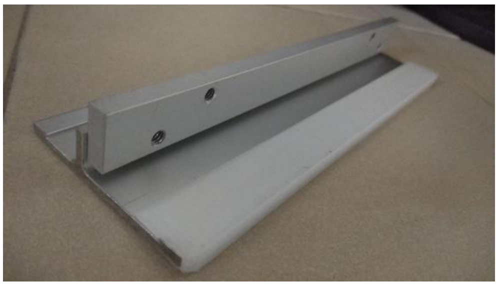
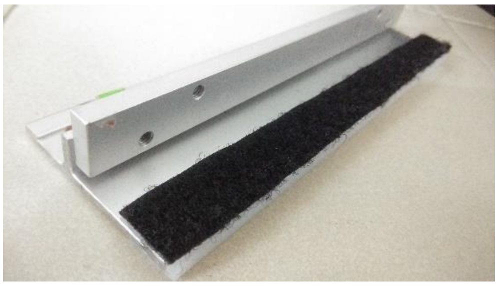


Figure 3.17: Setup of the impact test

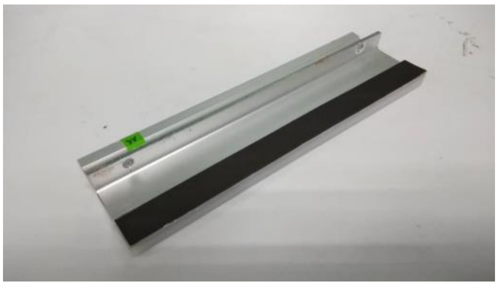
Three materials were selected to undergo the impact test such as 0.8 mm Polymer A, 0.9 mm wood and 1 mm elastomer. Before initiating the test, these materials were adhered to the slide rail on the chain connecting module as shown in Figure 3.18.



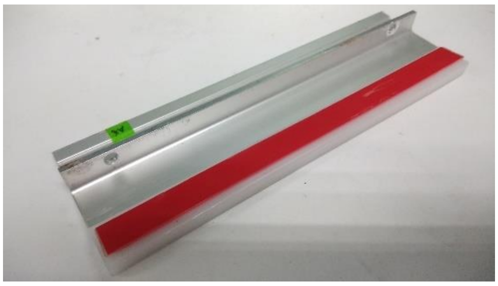
(a) Coating-free



(c) Slide rail with polymer



(b) Slide rail with wood



(d) Slide rail with elastomer

Figure 3.18: Slide rail under different surface conditions

Thereafter, the connecting module without coating was inverted and positioned onto a fabricated holder using aluminium bar to make the specimen remained static upon impact. The aluminium bar with the module was then mounted on the dynamometer by using two M8 bolts as shown in Figure 3.19

Figure 3.19: Mounting of the module on the dynamometer

Once the setup was done, the *Meas* button on the amplifier was pressed to start recording. The impact hammer was then used to knock on the slide rail (Figure 3.20) for several times to obtain a constant impulse. Two real-time forces graphs will be displayed on the PC's imc Device software including the impact forces applied on the hammer head and the transmitted forces to the dynamometer. The graphs were exported to FAMOS for the extraction of values at each period. The extracted data was then transferred to an excel file for post analysis. The same procedure was repeated for the connecting module with Polymer A, wood and elastomer coatings. Graph of impact force versus time for all specimens were plotted in the same graph. From here, the response of the force relative to time for each specimen could be contrasted.

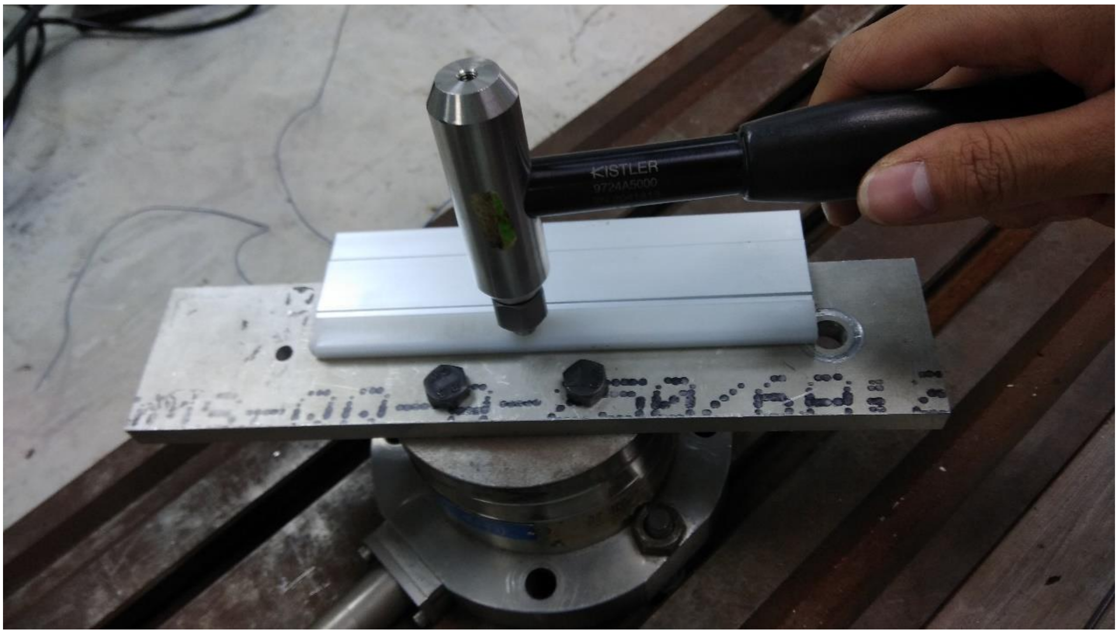


Figure 3.20: Impact hammer knocking on the slide rail

3.7 Structural Alteration

The material with the most effective response in mitigating the impact force was chosen to adhere to the slide rails where the dominant noise locates, as indicated by the red boxes below.

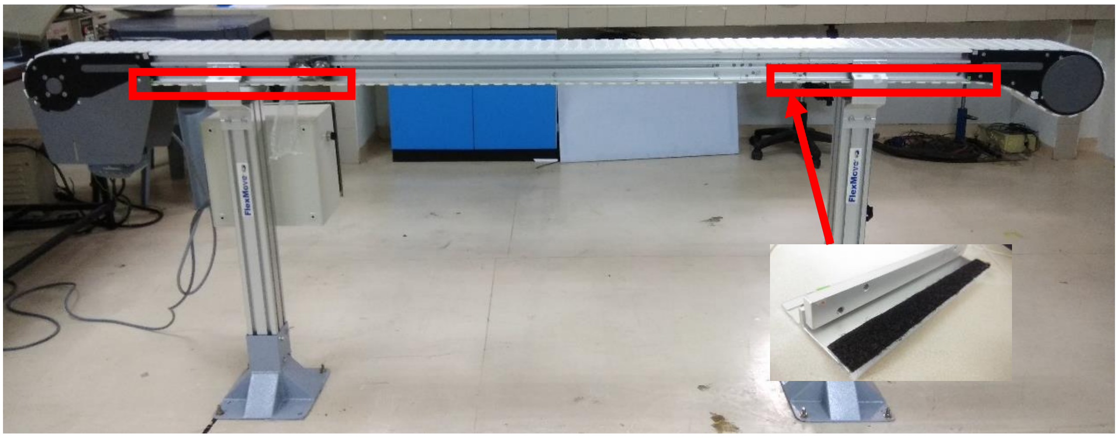


Figure 3.21: Region of structural alteration

Hereafter, the sound pressure and the acceleration measurements were repeated to determine the effectiveness of the material in attenuating noise and vibration. The evaluation of effectiveness was done by calculating the percentage reduction in the peak with the identical frequency of each individual FFT plots before and after alteration made.