

DYNAMIC ANALYSIS OF AN AIR HANDLING UNIT

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ABSTRACT

In this paper, vibration methods were conducted for an air-handling unit (AHU) assembly that experiences a vibration-related problem. The methods that were utilised are the Finite Element Analysis (FEA), the Experimental Modal Analysis (EMA) and the Operating Deflection Shapes (ODS). Mathematical and analytical modelling were also done before the experimental methods where they are used as reference for estimation of the results. Results between the theoretical and experimental methodologies were compared for analysis. The corresponding mode shapes were used as reference for finding the relevant experimental mode shapes of the AHU. The ODS results show the belt pulley system that connects the motor and the fan of the AHU experienced a strong axial vibration as supported by the initial vibration testing results. It was hypothesised that the AHU might be suffering from belt misalignment problem that introduces vibration to the whole AHU structure resulting in high vibration displacements.

Keywords : *Vibration analysis, Finite element analysis, Operational deflection shape, Experimental modal analysis, Air handling unit.*

1. INTRODUCTION

In plants and factories, there are many types of machinery that are in operation. Every machinery has their own purpose and thus works differently. The common thing for mechanical machineries are that they have moving parts which can introduce dynamic vibration as well as induced fatigue over high load of operation cycles. In the industry, the original equipment manufacturers (OEM) and the equipment users have employed vibration measurement and analysis methods to observe any vibration-related abnormality on their machineries. Vibration testing will be conducted during the transient (during start up of the machine) and the steady state conditions of the machine. Most of the problems are common misalignment, imbalance and bearing damages where those problems can be resolved directly. Some also implements a continuous vibration monitoring on their machineries especially if the machineries are very sensitive to excessive vibrations (i.e. running at very high speeds) and demand high expenditures on repair costs for the damages. The downside to this method is its costly maintenance and therefore should not be applied to more robust equipments. Another method is only conducting vibration testing to the machinery when experiencing a faulty problem. After conducting the necessary analysis, the problem will be solved appropriately usually using a heuristic (trial-and-error) method. This step is where finite element analysis (FEA) can be implemented to predict the outcome of any solutions which can save time and costs. Lots of research has been done on the finite element analysis and the modal analysis of a structure or a system. This either involves the theoretical aspects as well as the experimental part of solving a vibration problem in structures. Numerous textbook on vibration explain these theories and the

relevant experimental procedures. A number of works have been dedicated to solve vibration problems in pumps which involve rotating machineries. Marscher [1] has published an article on how to utilise modal testing to solve vibration in pumps. Onari and Boyadjis [2] have carried out a lot of FEA and EMA in order to solve various practical machinery problems. In their journal publication, they presented four case studies that involve pumps and motor turbines which experienced high vibration problems. The problems were then solved using a combination of FEA and EMA. They mentioned that the importance of vibration data gathered at the attachments such as flanges and interface areas to identify any separation between parts of the structures/machines. They concluded their research by stating the significance of the ODS as a powerful troubleshooting tool to understand a practical vibration problem. Combining the ODS, EMA and the FEA will result in more rapid problem diagnosis and proposed solutions. Some relevant works on fans were also done by Wang and Pan [3]. They conducted an EMA on a centrifugal fan with an operating range of 100-2400 rev/min, which is a part of a harvester threshing machine. They obtained the fan resonance frequency in the region of 125 Hz. The natural frequencies of motors were also investigated as done by Wang and Lai [4]. They concluded their research by stating that the structural details of the induction motor should be modelled as closed to the actual dimensions and features to obtain a better approximation of the natural frequencies and mode shapes. They found that the experimental natural frequencies of the 2.2kW 50 Hz three pole induction motor are determined by the individual parts of the motor. The fundamental frequency caused by the rotor is around 295 Hz and the fundamental frequency of the stator and casing is around 390 Hz. However another research conducted by Tonoki et. al. [5] established the fundamental frequency of an induction 2.2 kW general purpose motor as 1250 Hz.

2. CASE STUDY

The AHU is currently in operation in circulating air-conditioned air in a large office building. The mechanical assembly of the AHU comprises of a fan, a motor positioned on beams on top of a beam frame structure. The frame is positioned on top of four vibration spring isolators at its corners. The motor is the driving component of the AHU which in turn drives the fan via a belting or pulley system. The dimensions of the components were measured manually since they are not readily available in the manual provided. The vibration spring isolators specifications are also unavailable and therefore an approximation based on other colour coded factory manufactured springs were referred for future calculations. Figure 1 shows the AHU components.

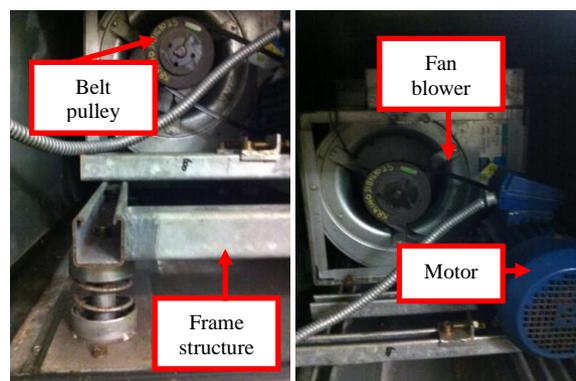


Figure 1 : View of the inner components of the AHU.

A. Initial Vibration Assessment

The preliminary testing conducted were done in accordance to the British Standards BS7854-6 (1996) titled "*Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts: Part 6 Reciprocating machines with power ratings above 100kW*" where the overall r.m.s. velocity measurements were obtained at seven selected locations at the components of the AHU assembly. The vibration severity is based on the highest overall r.m.s. vibration velocity amplitudes at any given axis of the particular component. They are then graded based on the severity chart available in BS7854-6 (1996) which also takes into account the machinery class (categorised by its power rating) and classified into 'zones'. Both the motor and fan are classified as Zone D which is defined in the standards as 'Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine' as shown in Table 1.

Table 1 : Vibration severity assessment of the motor and fan of the AHU.

Machine	Overall R.M.S. Velocity (mm/s)	Axis	Vibration Grade	Vibration Severity
Motor	9.46	Axial	11	Zone D
Fan	12.67	Axial	11	Zone D

METHODOLOGY

There are numerous methodology that will be described further on which are used to analyse the AHU analytically, using FEA method and also experimental methods such as ODS and EMA.

B. Analytical Analysis

There are three main analytical models used for comparison with FEA and experimental results. Two of them are fundamental natural frequency estimators which are the Dunkerley's equation and the Rayleigh's energy method. The other model is the Euler-Bernoulli beam theory under transverse vibration to estimate the mode shape profile of the AHU frame structure.

The fundamental frequency and the mode shape can be approximated by using the Dunkerley's equation with the assumptions that the masses are of concentrated mass where in this case the mass of blower and mass of motor, and the boundary conditions of the beam are clamped at both ends. The approximation is a lower bound approximation to the exact solution.

Equation 2.9 shows the Dunkerley's equation where ω_n represents the fundamental frequency of the whole system and $\omega_1, \omega_2, \omega_3$ and so on represents the natural frequency of each individual mass without the effect of other masses.

$$\frac{1}{\omega_n^2} = \frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2} + \dots + \frac{1}{\omega_{nn}^2} \quad (1)$$

There are three natural frequencies for consideration which are the frame's natural frequency and the natural frequencies due to the position of mass of the fan and the motor. The stiffness of the beam due to the masses was calculated using the stiffness of the beam equation obtained from Table 6-1 [6].

Rayleigh's method can be used to estimate the fundamental natural frequency of continuous structures. As shown by Rao [7], this is done by equating the maximum kinetic and potential energies and Rayleigh's quotient.

Weaver et. al. [8] used the Rayleigh technique and derived an expression calculating the fundamental natural frequency and the corresponding mode shape of a beam under transverse vibration with a concentrated mass with fixed-fixed boundary conditions as shown in Figure 2.

Equation (2) shows the natural frequency and Equation (3) shows the corresponding mode shape equations.

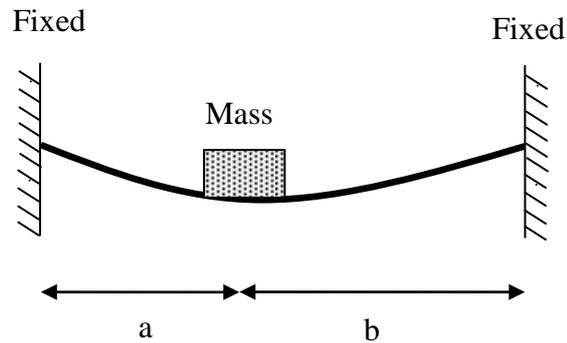


Figure 2 A beam with fixed-fixed boundary conditions with an external mass undergoing lateral vibration.

$$f = \frac{4}{\pi} \sqrt{\left(\frac{3EI}{L^3 [M + (\alpha + \beta)M_b]} \right)}$$

where

$$\alpha = \frac{a}{a+b} \left[\frac{(3a+b)^2}{28b^2} + \frac{9(a+b)^2}{20b^2} - \frac{(a+b)(3a+b)}{4b^2} \right]$$

$$\beta = \frac{b}{a+b} \left[\frac{(3b+a)^2}{28a^2} + \frac{9(a+b)^2}{20a^2} - \frac{(a+b)(3b+a)}{4a^2} \right]$$

(2)

$$\begin{aligned} \text{Mode Shape, } \tilde{y}_i \left(\frac{x}{L} \right) &= \left(\frac{x}{L} \right)^2 \left(\frac{3ax}{L^2} + \frac{bx}{L^2} - \frac{3a}{L} \right) \\ &; 0 \leq x \leq a \\ &= \left(1 - \frac{x}{L} \right)^2 \left(\frac{3b+a}{L} \left(1 - \frac{x}{L} \right) - \frac{3b}{L} \right) \\ &; a < x \leq b \end{aligned}$$

(3)

Below are the assumptions done for the Rayleigh's Technique analysis for the AHU case study:

- The assumptions of a Euler-Bernoulli beam are applied.
- The material of the beam is of the same material to galvanised steel.
- The area moment of inertia is calculated by taking into account the U-shaped cross section of the beam using Blevins's equations [6].
- An external mass assumed to be of the combination of the fan and the motor is located at 0.5 m from the first clamped end.

The theory that can be used to describe a beam structure which undergoes a transverse or lateral vibration is by assuming the Euler-Bernoulli Beam theory [7]. The general assumptions are stated by Blevins [6] as follows:

- The beams are uniform along the span and are composed of linear, homogenous, isotropic elastic material.

- The beams are considered as slender beams.
- There is no other mass acting on the beam.
- No axial loads are acting on the beam.
- The shear centre of the beam cross section is on the same point with the centre of mass of the beam.

$$c^2 \frac{\partial^4 w}{\partial x^4}(x, t) + \frac{\partial^2 w}{\partial t^2}(x, t) = 0 \tag{4}$$

where

$$c = \sqrt{\frac{EI}{\rho A}}$$

The solutions for free vibration wave equations as shown in Equation (4) gives the eigenvalue and eigenvector solutions which represent the natural frequencies and mode shapes respectively. For a fixed-fixed beam, the natural frequencies and mode shapes are given by Equation (5) and (6).

$$f = \frac{\beta^2}{2\pi} \sqrt{\frac{EI}{\rho A}} \tag{5}$$

$$\text{Mode Shape, } \tilde{y}_i\left(\frac{x}{L}\right) = \cosh \frac{\lambda_i x}{L} - \cos \frac{\lambda_i x}{L} - \sigma_i \left(\sinh \frac{\lambda_i x}{L} - \sin \frac{\lambda_i x}{L} \right) \tag{6}$$

Below are the assumptions done for the Euler-Bernoulli Beam theory analysis for the AHU case study:

- The assumptions of Euler-Bernoulli beam are applied.
- The material of the beam is of the same material to galvanised steel.
- The area moment of inertia is calculated by taking into account the U-shaped cross section of the beam using equations by [6].

C. Finite Element Analysis (FEA) Method

The FEA is based on the finite element method (FEM). This method was first developed in the 1940s to investigate structural mechanics [9-11]. The limitations of the FEM were liberated by the emergence of digital computers in the 1950s to allow computational of solving stiffness equations in the matrix form [12-14].

The CAD modelling was done using the SolidWorks 2011 software. The AHU dimensions were measured manually and transferred to the CAD model. Nevertheless the model is not a perfect model. Assumptions were made to the model as listed below:

- The particular details of the motor were assumed to be a solid. This means the armatures, the rotating shaft, the shell thickness of the motor and other components of the motor were ignored.
- The insides dimensions and the rotating shaft and vane blades of the fan were not modelled. The fan was modelled as a solid.
- The belt pulley was not modelled although it is one of the connections between the motor and the fan. The mass of the motor and the blower in the CAD model is different from the mass specified in the technical specification manual of the AHU.

The flow chart of the process for FEA is as shown in Figure 3.

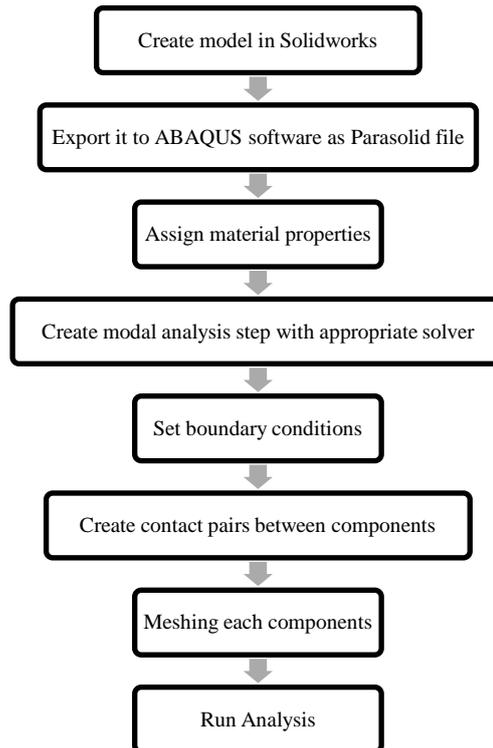


Figure 3 Diagram of the data acquisition setup for EMA method.

The model from SolidWorks 2011 was imported to the ABAQUS CAE software for FE analysis. ABAQUS was able to retain the information from the imported file such as the geometry and acknowledges the individual parts of the assembly. The material properties, boundary conditions and meshing properties were set. The final step for the pre-processing which is submitting the job for analysis in the Job module were done to obtain the natural frequencies and the corresponding mode shapes for the AHU model.

D. Experimental Modal Analysis (EMA)

According to Fu and He [15], modal analysis can be defined as the process of determining the inherent dynamic characteristics of a system in the form of natural frequencies and mode shapes. This analysis can then be used to obtain the mathematical model for the system's dynamic behaviour. The analysis are both theoretical and experimental in nature.

The data acquisition setup for EMA method is as shown in Figure 4. A roving tri-axial accelerometer was used to measure the AHU vibrations while the AHU is in free conditions (i.e. shut down) with a force impact hammer used as the excitation force. The data acquisition software DeweFRF was used to capture frequency response functions (FRFs) signals while the analysis was done using the Me'scopeVES v5.

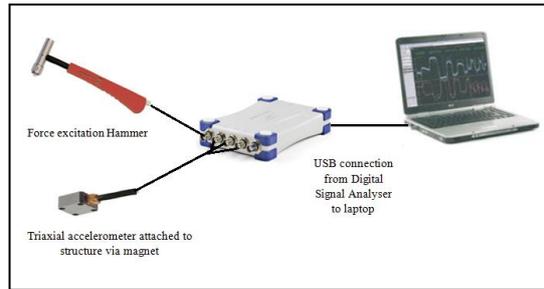


Figure 4 Diagram of the data acquisition setup for EMA method.

A minimum of five 'good' impacts from the hammer were needed for the averaging of the FRFs for one location. Characteristics of a 'good' impact test are as follows:

- There are no double hits or bad hits. This can be observed in the time and spectrum domain of the impact excitation force. A bad hit will introduce ripples in the force spectrum and double hits can be observed in the time domain as well.
- Excellent coherence. The coherence after each hit can be observed in the GUI. A value of one (unity) is ideal to ensure excellent relationship between input and output.
- Smooth FRFs obtained after the five impact tests.

The EMA then proceeded to the next stage which was the assignment of the measurement FRFs equations to the relevant points in the structure model. The option of creating interpolating equations was also used in order to create a smoother animation of the modes for the AHU.

The next step of analysis is determining the mode natural frequencies and its relevant mode shapes of the AHU. This required a good analysis of the Bode plot which is the magnitude and phase analysis of the FRFs. Also, the coherence of the FRFs were observed to ensure the reliability of the data measured for analysis. Four basic requirements in the Bode plot and coherence analysis was needed to identify the AHU's natural frequencies which are:

- There is a significant peak in the FRFs magnitude plot
- There is a phase shift occurring at the assumed resonance peaks
- Ideally the resonance peaks will have a 180° phase shift
- The coherence is ideally is a value of 1 (unity) at the peaks

The author's judgement in deciding the appropriate natural frequencies that reflect the resonance frequencies of the associated whole frame structure to the beam theory was required. Natural frequencies of individual components were ignored. Once the natural frequencies are determined, quick fitting method was used to compare and quick fitting all the FRFs to obtain a more precise and accurate natural frequency together with its damping properties. Then the residues (mode shapes) at the frequencies can also be calculated and shown visually in the structure model after the necessary measurement equations were created.

E. Operation Deflection Shapes (ODS) Method

This is a dynamic testing method that finds the deformation shape of a structure/machinery that is operating at its operating frequency (steady-state conditions). According to Boyadjis and Onari [2], the ODS method is performed on the entire unit structure and done at its worst case dynamic conditions within range of its operating frequencies.

For the AHU case study, the data acquisition setup for ODS analysis is as shown in Figure 5. A roving triaxial accelerometer was used to measure the AHU vibrations while in operating conditions with a uniaxial accelerometer used as reference. The data acquisition software DeweFRF was used to capture the vibration time signals while the analysis was done using the Me'scopeVES v5.

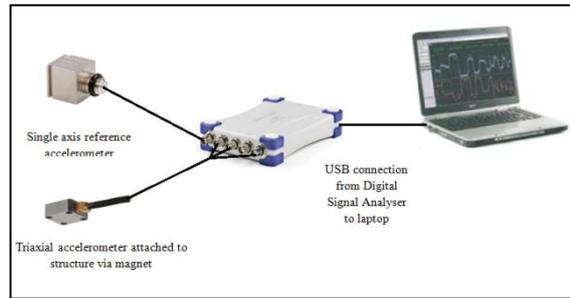


Figure 5 Diagram of the data acquisition setup for ODS analysis.

The measurement locations for the reference and roving accelerometers were established. It was advised to position the reference accelerometer at the most significant deflected part of the structure to obtain a good phase comparison with other measurement locations as were done by Boyadjis and Onari [2]. Therefore for this reason the non-driving-end (NDE) of the motor was chosen as the position for the reference accelerometer. A total number of 23 roving accelerometer locations were done.

The time domain signal files will then be imported into the software via a Data Block file for the project and each channel will be assigned with its specific Degree-of-Freedom (DOF) information and linked to their respective measurement locations. For example, the DOF properties of a x-axis channel with its respective reference accelerometer measured at the z-axis plane at location or point 1 will be named '1X : 1Z'.

The ODS Frequency Response Function (FRF) are needed for a frequency based ODS analysis. Me'scopeVES enables users to calculate ODS FRFs from various inputs. One of such inputs is the time domain signal. Therefore, the time domain signal data block file is then transformed into an ODS FRF data block file using the software features.

The new ODS FRF data block file will be then be used to match with the measurement locations in AHU model of the structure file. Since the structure file has all the relevant ODS FRF information at all its points, then the animation of the AHU's operating shapes can be done using the 'Draw | Animate Shapes' function available in the structure file options. The behaviour of the AHU was investigated based on the ODS FRF resonance peaks. ODS FRFs graphs and the relevant animations were recorded and stored for analysis. The vibration auto spectrums were also calculated from the time domain signals obtained from the tri-axial accelerometer measurements.

3. RESULTS AND DISCUSSION

The results from the various methods were compared and used to analysed the vibration-related problem of the AHU.

A. Comparison of Theoretical and Experimental Results

The most significant component of the AHU structure for comparison purposes with the analytical beam theory in this study was identified to be the frame structure that supports the motor and fan blower components. Figure 6 and Figure 7 shows the comparisons of the mode shape profiles of the AHU frame structure.

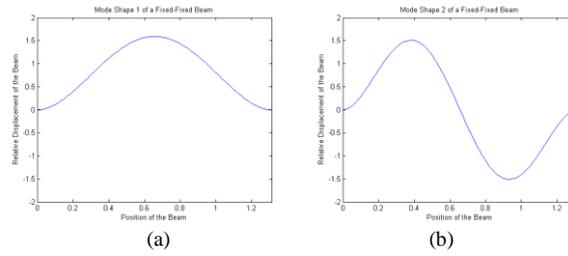


Figure 6 The resulting mode shapes of the Euler-Bernoulli beam
 a) first mode shape at 206.2 Hz
 b) second mode shape at 568.4 Hz for fixed-fixed boundary conditions.

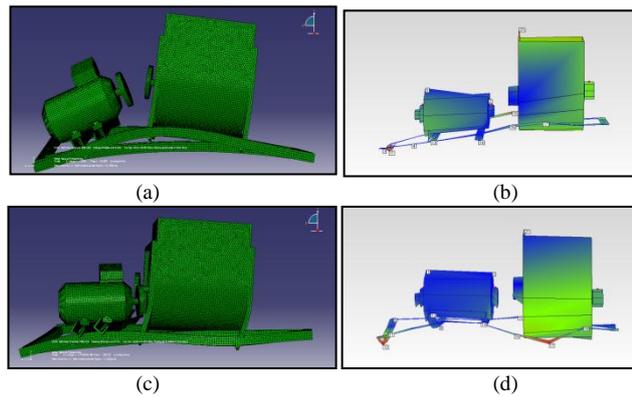


Figure 7 Similarities between associated mode shapes profiles with the beam theory obtained from finite element analysis and experimental modal analysis

- a) first associated mode shape FEA
- b) first associated mode shape EMA
- c) second associated mode shape FEA
- d) second associated mode shape EMA.

Comparisons of the theoretical results for the Dunkerley's, Euler-Bernoulli Beam theory, the Rayleigh's methods and FEA methods with the experimental results were made as shown in Table 2.

Table 2 Comparison of theoretical results with experimental values

Methodology	First mode shape	Difference to experiment value (%)	Second mode shape	Difference to experiment value (%)
Experimental Modal Analysis	50.6	-	228.0	-
Dunkerley's Equation	43.7	9.7	-	-
Euler Bernoulli's Beam	206.2	-307.5	568.4	-149.3
Rayleigh's Method	41.9	17.2	-	-
Finite Element Method	43.7	13.7	175.0	23.2

The Dunkerley's method gives the best estimation of the fundamental frequency for a fixed-fixed beam with the masses on the beam. A difference percentage of 9.7% between the fundamental frequency (45.7 Hz) and the associated experimental natural frequency of the frame structure (50.6 Hz) was obtained. Therefore, it can be said that it could be a good estimator for future problems that involves structures made up predominantly by beams. An analyst can use this method as a basis for knowing the fundamental frequency of the problem and use it as a reference before experimentation. Nevertheless, the weakness of the Dunkerley's method is that it only calculates the fundamental frequency and also does not give out any mode shapes for comparisons.

The Euler-Bernoulli Beam theory gives high difference percentages compared to the experimental natural frequencies which are -307.5% for the first associated natural frequency (206.2 Hz) and -149.3% for the second associated natural frequency (568.4 Hz) of the AHU. The negative percentages means that the theoretical values are higher than the experimental values. This is not surprising since that the beam was assumed not supporting the mass of the motor and the fan which is not the case for the real structure. The added mass will further reduce the natural frequencies of the system especially since the mass combination of the motor and fan is far heavier than the whole frame structure.

The difference percentages are caused by the oversimplification and assumptions made for each analytical method that makes them not resembling the true case study of the AHU. However, it can be concluded that the Dunkerley's and Rayleigh's methods can be used to estimate the fundamental frequency of a lightly damped beam structure. The corresponding mode shapes can be estimated by checking with the Euler-Bernoulli and Rayleigh's technique mode shapes as the references for experimental analysis.

More comprehensive analytical methods can be applied to this project. Further works can be done to improved the analytical study for the AHU problem based on transverse vibrations on beams with concentrated masses in two and three dimensional planes and solutions obtained using computational numerical methods. The author acknowledges the limitations of the analytical solutions that they will never provide the exact solutions to real case studies due to the complexities and non linearity available in real life problems as well as the various assumptions made in the analytical models. Even so, better analytical models and solutions may provide better base reference for analyst to approach a vibration problem before conducting the experiments.

For the FEA results, the percentage differences are below 20% for the first natural frequency and slightly above 20% for the second natural frequency. FEA modifications were done to investigate the varying of Young's Modulus of the frame structure material, the spring's stiffness as well as the mass of the fan. It was found that increasing the Young's Modulus of the material improved the results. Doubling the spring's stiffness has no effect on the natural frequencies. Higher mass values for the fan reduces the FEA natural frequencies and this increases the percentage difference. Nevertheless, the percentage difference is predictable since that the FEA model is bounded by limitations, assumptions and thus errors can be introduce into the results.

The limitations and errors that arise from the comparisons might be caused by the oversimplification of the modelling process for the FE analysis. The model was not sketched precisely as the AHU structure due to lack of drawings information. The motor and fan blower were modelled as a lump mass with a user defined material to be calibrated to the actual mass of these components. The cross section shape of the frame structure was also simplified and the bolts and connectors were modelled not according to specification. The connectors were also assumed to be bonded to the frame structure. The boundary conditions between the components

might also vary. The spring stiffness was estimated based on product catalogues. This uncertainties add up to make the percentage difference of the results.

It is important to note that the FEA methods is actually best utilised at the design stage of building a structure where the intrinsic material properties and the specifications can be determined precisely and accurate. FEA in approaching a real vibration problem is quite difficult unless there are sufficient information regarding all the materials and connections of intricate parts of the machines and structures. Even FEA modifications can be a complicated job due to the limitations that has been previously mentioned. On the other hand, FEA can be a great tool to assist analyst to further predict any structural dynamics modifications if the need arises.

B. Dynamic Analysis of AHU

The forcing excitation frequencies were identified from the vibration auto spectrums calculated from the ODS analysis and tabulated as shown in Table 3. The location 1 which is the driving end (DE) of the motor was chosen as the vibration spectrum analysis due to its proximity to the belt pulley which drives the fan blower. The peaks representing the driver and driven frequency can be observed in the vibration spectrums including their harmonics. There are also many other peaks that cannot be described due to lack of information regarding the motor and blower specifications. It can be assumed that some of this harmonics represents the bearings passing frequencies inside the motor and also other internal moving parts. The vane pass frequency was calculated to be around 1114 Hz. Nonetheless, there are no evident peaks in the vibration auto spectrum at measurement location 1.

Table 3 Observed forcing excitation frequencies in the axial and vertical vibration auto spectrum at location 1.

Force excitation description	Observed frequency (Hz)
Driver frequency 1X (motor)	21.5
Driven frequency 1X (fan)	26.0
Harmonics of driver frequency (2X, 3X ..)	43, 64, 85.5
Harmonics of driven frequency (2X, 3X ..)	53.5, 77.5, 104
Driver frequency 1X (motor)	21.5

Judging from the ODS shapes in the Figure 8 and Figure 9 show the motor is moving axially away from the blower (where the belting connection is located) at the driver frequency, 21.5 Hz. At the 26 Hz driven frequency, the fan blower panels are moving flexibly like a thin plate and the motor experiences axial and horizontal vibrations.

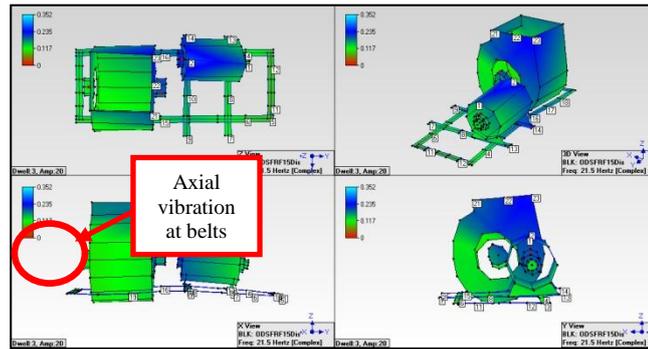


Figure 8 Four views (top, 3D, front and side clockwise) of the operating deflection shapes motion of the AHU structure at the driver frequency 21.5 Hz.

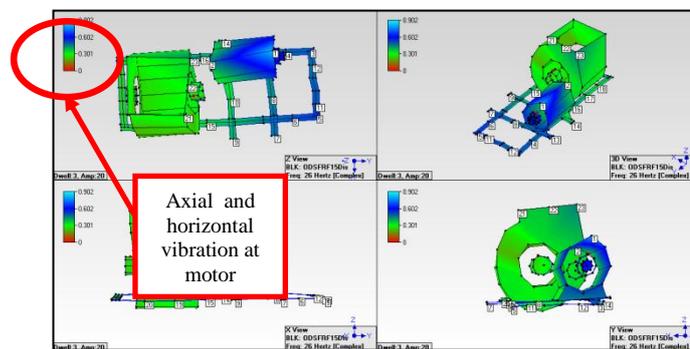


Figure 9 Four views (top, 3D, front and side clockwise) of the operating deflection shapes motion of the AHU structure at the driver frequency 26 Hz.

There were also extra information regarding the AHU servicing history. The AHU broke down in August 2012 which was after the initial vibration testing measurements were conducted and the problem was the belting failure. The belt had failed maybe after prolong expanding and twisting motions during the AHU operating conditions. Figure 10 shows the condition of the broken belt.



Figure 10 Picture of the broken belt.

It is inferred that the belt may suffer a fatigue type failure caused by misalignment of the belting system by [16]. The author's hypothesis is that there is possibly an offset misalignment that shows an increase of the axial vibrations especially at the driver or driven frequency. Based on

the high axial overall vibrations measured at the fan blower and DE of the motor during the initial vibration testing (as referred to Table I) before the belt failure, it can be assumed that this is the most probable diagnose of the vibration problem experienced by the AHU. The misalignment will cause the motor and fan moving axially thus the vibration propagates throughout the whole AHU structure. It is assumed that it is not a resonance effect especially for the frame structure because the operating conditions are well below the first natural frequency of the AHU supporting frame structure which are around 48% (driven fan blower frequency) and 58% (driver motor frequency).

For further works, there is no requirement for any structural modifications for the AHU. Instead, proper installation and alignment of the belt pulley system will ensure a longer lasting machine life. Regular servicing may also be needed to clear the dusts that accumulate in the fan blower because higher mass will reduce the natural frequency of the system. Although the loss of the belt system is relatively cheap, however the belt misalignment can induce vibrations to the whole structure which may cut short its working life conditions of the motor and the blower. For expensive structures such as turbines in oil and gas industries, simple problems such as misalignments, looseness and others will prove to be catastrophic in financial terms.

Strictly speaking this is still a hypothesis and must be proven by doing further works in checking whether the motor and fan pulley is truly misaligned. If the pulleys are not misaligned, therefore the hypothesis will be proved incorrect and another theory can be hypothesised that the operating conditions might be exciting other natural frequencies that may belong to the motor and fan or any other components of the AHU structure. This in turn provides the motion described by the ODS results where the resonance affects axial vibrations at the belting pulley system and causes it to experience fatigue failure. Other further works that can be done is to simulate the rotational force of the belting system in the FEA model and tune it to the ODS results to obtain a better simulation behaviour of the AHU under different conditions. This method was consistently used by [2] for approaching vibration problems in pumps. Therefore the utilisation of the analytical, FEA and experimental methods which are the modal and operating deflection shapes analysis are important as tools to solve vibration problems.

4.0 CONCLUSIONS

Theoretical analysis was done and results were compared with experimental values. No proposal of design modifications using FEA were done due to the time constraint and the author's conclusion that the problem of the AHU structure is hypothetically caused by the misalignment of the belt pulley system.

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REFERENCES

1. W. D. Marscher, "How to Use Impact Testing to Solve Pump Vibration Problem," in Proceedings EPRI Power Plant Pumps Symposium, Florida, 1991.
2. M. M. Onari and P. A. Boyadjis, "Solving Structural Vibration Problems Using Operating Deflection Shape and Finite Element Analysis," in Proceedings of the Twenty-Fifth International Pump Users Symposium, 2009.
3. L. Wang and Z. Pan, "Experimental Modal Analysis of Fan Vibration Frequency," Applied Mechanics and Materials, pp. pp. 138-139, 2012.
4. C. Wang and J. C. S. Lai, "Vibration Analysis of an Induction Motor," Journal of Sound and Vibration, pp. pp. 733-756, 1999.
5. F. Ishibashi, M. Matsushita, S. Noda and K. Tonoki, "Change of Mechanical Natural Frequencies of Induction Motor," IEEE Transactions on Industry Applications, pp. pp. 922-927, 2010.
6. R. D. Blevins, Formulas for Natural Frequency and Mode Shape, New York: Van Nostrand Reinhold Company, 1979.
7. S. Rao, Mechanical Vibrations: Fifth Edition, New Jersey: Prentice Hall, 2011.
8. S. Timoshenko, D. H. Young and J. W. Weaver, Vibration Problems in Engineering (4th ed.), New York: John Wiley, 1974.
9. A. Hrennikoff, "Solution of Problems in Elasticity by the Frame Work Method," Journal of Applied Mechanics, vol. Vol. 8, no. No. 4, pp. pp. 169-175, December 1941.
10. D. McHenry, "A Lattice Analogy for the Solution of Plane Stress Problems," Journal of Institution of Civil Engineers, vol. Vol. 21, pp. pp. 59-82, 194.
11. N. M. Newmark, "Numerical Methods of Analysis in Bars, Plates, and Elastic Bodies," in Numerical Methods in Analysis in Engineering, Macmillan, 1949.
12. S. Levy, "Structural Analysis and Influence Coefficients for Delta Wings," Journal of Aeronautical Sciences, vol. Vol. 20, no. 7, pp. pp. 449-454, 1953.
13. J.H. Argyris, "Energy Theorems and Structural Analysis," in Aircraft Engineering, 1955.
14. J. H. Argyris and S. Kelsey, Energy Theorems and Structural Analysis, London: Butterworth, 1960.
15. J. He and Z.-F. Fu, Modal Analysis, Oxford: Butterworth-Heinemann, 2001.
16. C. Scheffer and P. Girdhar, Practical Machinery Vibration Analysis & Predictive Maintenance, MA: Elsevier, 2004.