Modal Analysis of Rotary Piston Blower

By

Safinah Binti Ahmad

Dissertation submitted in partial fulfillment of
the requirements for the
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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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in partial fulfillment of the requirement for the
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Approved by,

____

(Ir. Hj. Idris Ibrahim)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
May 2012

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except in the reference and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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SAFINAH BINTI AHMAD

ABSTRACT

Dynamic characteristic of rotary piston blower is highly important for plant reference to operate the machine. The critical parameter of shaft is one of the sources of excitation which can affect the performance of blower due to resonance. However, the dynamic characteristic of rotary piston blower, K302 under various discharge flow condition has not been established by the operator. Thus, this project is carried out to investigate the dynamic characteristic of a rotary piston blower at different load condition using Modal Analysis approach.

Typically, researchers use a combination of analytical and experimental method, coupled with system identification system. However, only analytical method is used in this project. The modeling of a blower and its assembly are performed using Catia V5 while the simulation is done through Finite Element Analysis using Ansys Workbench software. The variable parameter used for the simulation is the blower operating speed/frequency and the result obtained is the system natural frequency and its related mode shape.

The resonance occurs at the drive end of the blower’s shaft at low operating frequency of 8.33Hz at mode shape 3 with a 756.21mm displacement. The result obtained could be very useful for plant operator to maintain the reliability of the machine. Any failure cause by vibration should be prevented.

ACKNOWLEDGEMENT

Bismillahirrahmanirrahim,

All praise to God the Almighty, The Most Gracious and The Most Merciful. Thanks to Allah, who with Him willing to give me opportunity to successfully complete my final year project with entitle 'Modal Analysis of Rotary Piston Blower'.

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ABBREVIATION AND NOMENCLATURE

ASME	American Society of Mechanical Engineers
HIS	Hydraulic Institute Standard
FEM	Finite Element Method
FEA	Finite Element Analysis
CAD	Computer-aided Design
IJEST	International Journal of Engineering Science and Technology
FFT	Fast Fourier Transform Analyzer

CHAPTER 1

INTRODUCTION

1.1 Background of Study

Most manufacturing plants use fans and blowers for industrial purposes which need an air flow. American Society of Mechanical Engineers (ASME) uses the specific ratio, which is the ratio of the discharge pressure over the suction pressure to define fans, blowers and compressors. Table 1.1 shows the difference between fans, blowers and compressors.

Table 1.1: Difference between fans, blowers and compressor [1]

Equipment	**Specific ratio ($P_{discharge}$ / $P_{suction}$)**	**Pressure rise (mmHg)**
Fans	Up to 1.11	1136
Blowers	1.11 to 1.20	1136-2066
Compressors	More than 1.20	-

Rotary piston Delta blower with model K302 manufactured by Aerzen is used by one of the Petronas petrochemical plant in Malaysia. K302 is a positive displacement blower under category lobe type of rotary blower. Figure 1.1 shows the tri-lobe roots blower of K302. The main function of blower is to move gas and air by the system pressure they must operate against. K302 blower with tri-lobe blower is popular in diverse industry because it offers superb sanitary quality, high efficiency, reliability, corrosion resistance and good clean-in place and sterilize-in place characteristic.

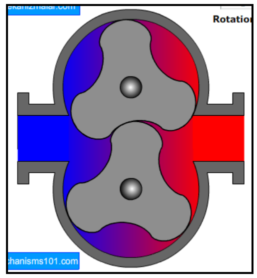


Figure 1.1: Tri-lobe roots blower [2]

Tri-lobe roots blower operates by pumping fluids with a pair of meshing lobes. However, the lobes do not make contact because the lobe contact is prevented by external timing gears located in the gearbox. Pump shaft support bearings are located in the gearbox, and since the bearing are out of the pumped, pressure is limited by bearing location and shaft deflection [3].

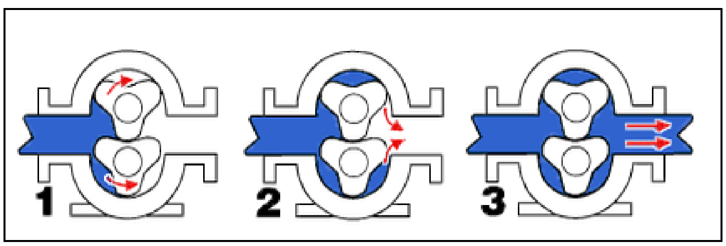


Figure 1.2: Working principle of blower [3]

Figure 1.2 shows the working principle of blower [3].

a) As the lobes come out of mesh, they create expanding volume on the inlet side of the pump. Liquid flows into the cavity and is trapped by the lobes as they rotate.
b) Liquid travels around the interior of the casing in the pockets between the lobes and the casing – it does not pass between the lobes.
c) Finally, the meshing of the lobes forces liquid through the outlet port under pressure.

Figure 1.3 shows the main components of blower unit which are emphasized in the modeling part of this project.

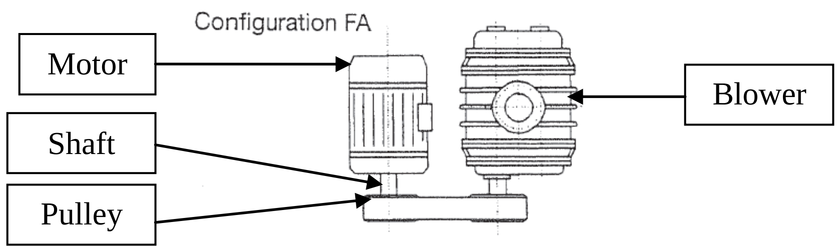


Figure 1.3: Main components of blower unit [4]

These constructional features have the following distinct characteristic:

a) The flow is largely dependent on the operating speed.
b) The input power is largely dependent on the total pressure across the machine.
c) The suction and discharge pressures are determined by the system conditions.
d) The temperature rise of the discharge air and machine is largely dependent on the differential pressure across it [3].

1.2 Problem Statement

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point [5]. Most prime movers have vibrational problems due to the inherent unbalance in the machines. In all these situations, the structure or machine component subject to vibration can fail because of material fatigue resulting from the cyclic variation of the induced stress [6]. The vibration will cause more rapid wear of machine parts such as bearing and gears and also creates excessive noise.

Vibration problems are most commonly associated with blower. From previous research paper “A Modal Approach for Vibration Analysis and Condition Monitoring of a Centrifugal Pump” by Ramana Podugu, J. Suresh Kumar, B. V. Ramana Murthy, N. Syam Kumar (2008) states that “The source of vibration can be categorized into three types such as mechanical causes, hydraulic causes and peripheral causes. Level of imbalance and the level of misalignment are the important reasons of mechanical and hydraulic causes. The peripheral causes of vibration include harmonic vibration from nearby equipment or drivers, operating the pump at critical speed". Any failure will show up as symptoms which include abnormal trend of vibration at certain natural frequency.

Vibration problem is affected by dynamic characteristics of blower system which much affect its performance and reliability. However, the dynamic characteristic of rotary piston blower with model K302 under various discharge flow condition has not been established by the operator.

On July 2011, it was observed that there was increasing in trend of vibration amplitude which resulted in failure of blower. Thus, a project is carried out to perform a modal analysis of the blower under various flow requirements from Finite Element Method (FEM) using Ansys Workbench software.

The importance of studying vibration is to predict the failure of machine before it happen. Earlier indication will give sign that the machine should undergo preventive maintenance before more cost is required for down of machine.

1.3 Aim and Objectives

The aim of project is

a) To investigate the mode shape of the rotary piston blower, K302 at different load condition.

The objectives of project are:

a) To study the rotary piston blower under various load condition using modal analysis technique.
b) To establish critical parameters of the blower dynamic characteristic such as resonance point, critical frequency, and deflection of shaft.

1.4 Scope of Study

Scopes of study for the project are:

a) Modeling using Catia V5 software and simulating through Ansys Workbench software.
b) Analyzing critical parameters from the simulation data.
c) Excluding of experimental analysis.

CHAPTER 2

LITERATURE REVIEW

2.1 Theoretical Background

Modal analysis is a study of the dynamic properties of structures under vibrational excitation which aim to determine the natural mode shapes and frequencies of structures. Modal analysis is the field of measuring and analyzing the dynamic response of structures and or fluids when excited by an input [7]. It is a common to use FEM to perform modal analysis because the object being analyzed can have arbitrary shape and the results of the calculations are acceptable. The FEM originated from the need for solving complex elasticity and structural analysis problems in engineering.

A variety of specializations under mechanical engineering discipline (such as aeronautical, biomechanical, and automotive industries) commonly use integrated FEM in designing and developing of the product. In practice, a Finite Element Analysis (FEA) usually consists of three principle steps [8]:

a) *Preprocessing:* The user constructs a model of the part to be analyzed in which the geometry is divided into a number of discrete sub regions, or element, connected at discrete points called nodes. These models consume time to prepare, and commercial codes vie with one another to have the most user-friendly graphical processor to assist in this rather tedious chore. Some of these preprocessors can overlay a mesh on a preexisting Computer-aided Design (CAD) file, so that FEA can be done conveniently as part of the computer drafting-and-design process.

b) *Analysis:* The dataset prepared by the preprocessor is used as input to the finite element code itself, which constructs and solves a system of linear or nonlinear algebraic equations.

c) *Post processing:* User would pore through reams of number generated by the code, listing displacement and stresses at discrete positions within the model.

In a structural simulation, FEM helps tremendously in producing stiffness and strength visualizations and also in minimizing weight, materials, and costs [9]. However, in spite of the great power of FEA, disadvantages of computer solutions are they do not necessarily reveal how the stresses are influenced by important problem variables (such as materials properties and geometrical features), and errors in input data can produce wildly incorrect result that may be overlooked by the analyst.

2.2 Structure Dynamic Properties of Finite Element Method

When the model is divided into finite element, the dynamic balance is [10, 11]:

$$\{F_i\} + \{F_c\} + \{F_s\} = \{F(t)\} \tag{1}$$

Where $\{F_i\}$ is inertia force vector quantity;

$\{F_c\}$ is damping force vector quantity;

$\{F_s\}$ is elastic force vector quantity;

$\{F(t)\}$ is dynamic load vector quantity.

For the system without damp or ignoring damp, the balance equation can be written by:

$$\{F_i\} + \{F_s\} = \{F(t)\} \tag{2}$$

The inertia force vector quantity can be substituted by quality matrix and node displacement vector. Then $\{F_i\}$ becomes:

$$\{F_i\} = M \frac{\partial^2}{\partial t^2} \alpha(t) \tag{3}$$

Where $\alpha(t)$ is a node displacement vector;

M is quality matrix.

The elastic force vector can be substituted by node displacement vector and rigidity vector. Thus, the $\{F_s\}$ can be written by:

$$\{F_s\} = K\, \alpha(t) \tag{4}$$

Where K is rigidity factor;

$\alpha(t)$ is node displacement.

When the system is natural vibration, $\{F(t)l\} = 0$, and the (2) can be transformed as the follow

$$M\, \ddot{a}(t) + K\, \alpha(t) \tag{5}$$

The characteristic equation based on quadratic parabola load equation is given by

$$(K - \omega^2 M)\, \varphi\, (t) = 0 \tag{6}$$

Where ω is intrinsic frequency.

Ansys Workbench software provides several arithmetic, such as subspace arithmetic, Interactive arithmetic, Lanczos arithmetic, Reduction arithmetic. In this paper, with the function of high operation speed and high precision, Lanczos arithmetic is used to calculate the natural frequency and fundamental main nodes.

The derivation steps of Lanczos arithmetic are given by the follows.

Given vector x_i $(i = 1, 2, 3, \ldots, r)$. Then, the equation (7) can be solved by:

$$K\overline{x_i} = Mx_{i-1} \tag{7}$$

Orthogonalization:

$$\overline{x_i} = x_i - \alpha_{i-1}x_{i-1} - \beta_{i-2}x_{i-2} \tag{8}$$

Where

$$\alpha_{i-1} = \overline{x_i}Mx_{i-1} \tag{9}$$

Regularization:

$$x_i = x_i / \beta_i \tag{10}$$

where $\beta_i = (\hat{x}_i M \hat{x}_i)^{1/2}$.

Generalized eigenvalue problem ($K\Phi_r = M\Phi\Omega_r$) translate into standard eigenvalue problem which is Lanczos vector towards to triangle matrix. Based on (5) to (11), $A=K^{-1}M$ can be solved by (12):

$$Ax_{i-1} = \beta_i x_i + \alpha_{i-1}x_{i-1} + \beta_{i-1} x_{i-2} \tag{11}$$

Where $(i = 1, 2, 3, \ldots, r)$, $x_0 = \{0\}$.

(11) may be written as matrix form

$$AX = XT \tag{12}$$

Where

$$T = \begin{bmatrix} \alpha_i & \beta_2 & & & \\ \beta_2 & \alpha_2 & \beta_3 & & \\ & \beta_3 & \alpha_3 & \beta_4 & \\ & & \cdots & \cdots & \\ & & & \cdots & \cdots \end{bmatrix} \tag{13}$$

$$X = \left[x_1 x_2 \cdots x_r \right] \tag{14}$$

The equation between original characteristic matrices and Lanczos vector is written as

$$\Phi_r = XZ \tag{15}$$

Where

$$\Phi_r = [\varphi_1 \ \varphi_2 \ldots \ \varphi_r] \tag{16}$$

Substitution (16) into original eigenvalues, use $XTMK\text{-}1$ premultiply and $\lambda = \Omega_r^{-1}$ post-multiply in equation two sides, then use (12) and $X^T MX = 1$ obtains the equation:

$$TZ = Z\lambda \tag{17}$$

The eigen solution of (17) is

$$\mathrm{Z} = [z_1, z_2 \ldots z_r], \ \Omega_r = \lambda^{-1} \tag{18}$$

Then $\Phi_r = XZ, \ \Omega_r = \lambda^{-1}$.

Accordingly, $W_i^2 = 1/\lambda_i^2 \ (i=1, 2, \ldots, r)$.

2.3 Literature Review

Performance and reliability of equipment is much affected by its dynamic characteristic. An article “A Modal Approach for Vibration Analysis and Condition Monitoring of a Centrifugal Pump” done by Ramana Podugu, J. Suresh Kumar, B. V. Ramana Murthy, N. Syam Kumar (2011) have studied natural frequency of centrifugal pump through mathematical model and FEA. The original centrifugal casing is modeled and simulated using Ansys software. The model of pump is meshed by using 20-node tetrahedral with solid 186 elements.

The first ten natural frequencies are calculated from FEA and listed in the Table 2.1.

Table 2.1: First ten natural frequencies [12]

Mode	Natural frequency (Hz)	(Operating speed) 1st speed (Hz)	Margin in %
1	63.25	62.5	1.20
2	83.89	62.5	34.23
3	106.83	62.5	70.93
4	129.83	62.5	107.73
5	156.72	62.5	150.75
6	166.32	62.5	166.11
7	182.50	62.5	192.00
8	192.70	62.5	208.32
9	225.54	62.5	260.86
10	245.73	62.5	293.17

This paper explains how reduction of separation margin between running speed and the natural frequency of machine is the main cause of high vibrations in machine during operation. Referring to the result calculated, it is observed that the first natural frequency is 63.25Hz which is very close to the operating speed with margin 1.20%. This less separation margin causes high vibration of machine. According to Hydraulic Institute Standard -9.6.4-2000 guideline (HIS), the first natural frequency should be 10% above or below the pump operating speed.

Throughout this paper, the dynamic characteristic of the centrifugal pump is improved by adding stiffeners to the pump pedestals under pump feet. Increasing the stiffness of the centrifugal pump assembly will increase the frequency of the centrifugal pump. Table 2.2 shows the first ten natural frequencies with modified pedestal.

Table 2.2: First ten natural frequencies with modified pedestal [12]

Mode No	Natural frequency (Hz)	(Operating speed) 1st speed (Hz)	Margin in %
1	74.311	62.5	18.89
2	91.026	62.5	45.64
3	138.48	62.5	121.56
4	151.50	62.5	142.40
5	163.01	62.5	160.81
6	166.70	62.5	166.72
7	206.90	62.5	231.04
8	250.40	62.5	300.64
9	269.06	62.5	330.49
10	276.61	62.5	342.57

The first margin between natural frequency and operational speed is 18.89% which is satisfy the HIS clause.

The research shows how the dynamic characteristic of centrifugal pump is improved by adding the stiffeners to the pump pedestals which increase the frequency of the pump. Application of vibration technique for monitoring the centrifugal pump is convenient and reliable in determining the failures in the early stage and can avoid unscheduled shutdowns and expensive repair costs [12].

Article written by P. R. Baviskar & V. B. Tungikar (2011), "Analysis of Crack in Shaft of Blower Using Finite Element Analysis and Experimental Technique" say that one of the failures of equipment might be due to the crack initiation and propagation in any of the moving part. The article proves that the presence of crack in the structure will change dynamic characteristic of shaft. Modal parameters that might be affected by crack are modal frequency, modal value, and mode shape associated with each modal frequency. It also changes the structural parameters like mass, damping matrix, stiffness matrix as well as flexibility matrix of the structure. Several methods have being done in order to compare the results and to reduce the error in the analysis. Theoretical analysis, modeling analysis on the structure is done using FEM in Ansys software meanwhile the experimental analysis is done using Fast Fourier Transform (FFT) Analyzer. Harmonic response analysis is used to determine the steady-state response of a linear structure to loads that vary sinusoidal with time.

This paper studies on crack initiation and propagation in shaft of blower. The natural frequency is monitored to access crack location and crack size in beam. It is verified that the design is against resonance, fatigue and other effects of forced vibrations. Result from this paper that the natural frequency decreases with increase in severity of crack. For the same severity, the frequency reduction is large for location of crack away from the support. The natural reduction in cracked beam is not due to removal of mass from beam, indeed the reduction in mass would increase natural frequency. The effect is due to removal of material which carries significant stresses when defect is a narrow crack [13].

Earlier research has being done by M. H. Sadeghi, S. Jafari and B. Nasseroleslami (2008) in the modal analysis of a turbo-pump shaft. This research studies on the mode shape of turbo-pump shaft using different analysis method. Modal properties of the turbo-pump shaft is studied using different excitation techniques, numerical solution using simplified geometric modeling combined with the FEM. High rotational speed of the shaft with unavoidable misalignment can be considered as one of the most important source of excitations, which may lead to critical conditions due to the resonance and result in failure of turbo-pump shaft.

The results from experimental method and FEM have being discussed. Table 2.3 shows the comparison a1mong natural frequencies and mode shape from Hammer test, Shaker test and FEM solution.

Table 2.3: Natural frequencies and mode shape from Hammer test, Shaker test and FEM solution [14]

Case	FEM (Hz)	Experiment (Shaker Test) (Hz)	Experiment (Hammer Test) (Hz)	Error (%)	Mode shape
1	49.52	51.5	48.5	0.96	1st Bending Mode
2	174.7	179	182	3.21	1st Torsion Mode
3	329.3	395	401	17.26	2nd Bending Mode
4	419.3	-	-		2nd Torsion Mode
5	499.0	520	Not excited	4.04	3rd Bending Mode
6	510.7	-	-	-	Complex undefined mode shape
7	632.0	622	Not excited	1.61	1st Longitudinal Mode
8	680.9	741	742	8.17	4th Bending mode shape
9	903.1	-	-	-	5th Bending Mode
10	966.2	-	-	-	Complex undefined mode shape
11	1023.1	-	-	-	6th Bending Mode
12	1051.7	-	-	-	7th Bending Mode
13	1085.2	-	-	-	Complex undefined mode shape

Error (%) shows the error between computational results and average Hammer and Shaker test result.

From this comparison, it reveals that some of mode shapes are not excited in experiment. This is mainly originated from common limitations of experimental tests. Unexcited mode shapes include second torsion mode, high frequency bending mode, and complex mode shape. These results are expected due to difficulty of excitation of these modes with the available setup [14]. However, the low percentage of error between the FEM and experimental result (except for the second bending mode) implies that attained modal parameters are desirable accuracy. The compatibility of the results validates the modeling and experimental processes.

2.4 Summary of Literature Review

The research papers which are referred in this project are summarized in this chapter. Most of the previous researches are based on the main component in the machine such as analysis on the centrifugal pump, turbo-pump shaft and crankshaft. Based on the previous researches that have been done, it is concluded that most of the source of vibration is caused by mechanical and hydraulic sources which is related to the level of imbalance and misalignment.

Title: A Modal Approach for Vibration Analysis and Condition Monitoring of a Centrifugal Pump [12].

Author: Ramana Podugu, J. Suresh Kumar, B. V. Ramana Murthy, N. Syam Kumar.

Objective: Study of dynamic characteristic of centrifugal pump based on variation in natural frequency.

Method:

- Mathematical model.
- FEA model using Ansys software.

Result:

- Reducing separation margin between running speed and the natural frequency of the machine is the main cause for high vibrations in machines during operation.
- Increasing of stiffness of the centrifugal pump assembly will increase the frequency of the centrifugal pump.

Title: Analysis of Crack in Shaft of Blower using Finite Element Analysis and Experimental Technique [13].

Author: P. R. Baviskar, V. B. Tungikar.

Objective: Study of crack location and crack depth on shaft based on observation of changes in natural frequency.

Method:

- Theoretical analysis by calculation.
- Experimental analysis.
- Model analysis by FEM, Ansys software.

Result:

- Comparing the result from different analysis method.
- Crack will reduce the natural frequency of shaft. Natural frequency decreases with increase in severity of crack.
- By using the proposed method, fault diagnosis in any rotating element with different boundary condition, geometrical shape and materials can be done.

Title: Modal Analysis of a Turbo-pump Shaft: an Innovative Suspending Method to Improve the Results [14].

Author: M. H. Sadeghi, S. Jafari, B. Nasseroleslami.

Objective: Study of modal properties of turbo-pump shaft to reduce noise-to-signal ratio which is resulting from classic suspension.

Method:

- Experimental modal analysis.
- Numerical solution combines with FEM analysis.

Result:

- Comparing the natural frequency and mode shape from experimental and FEM results.
- Low percentage of error between the FEM and experimental result implies that attained modal parameters are of desirable accuracy.

Title: Design and Modal analysis of Composite Drive Shaft for Automotive Application [15].

Author: Mohammad Reza Khoshravan, Amin Paykani, Aidin Akbarzadeh.

Objectives:

- Study of design method and vibrational analysis of composite propeller shafts.
- Two main sections: Design of composite shaft and design of couplings.
- Study of some parameters: Critical speed, static torque and adhesive joints

Method:

- Critical speed analysis by mathematical calculation.
- Model analysis of composite drive shafts using ANSYS

Result:

- Composite drive shaft made up high modulus carbon/epoxy multilayered composites has been designed.
- Replacing the composite material has resulted in considerable amount of weight reduction about 72% when compared to conventional steel shaft.
- Orientation of fibers has great influence on the dynamic characteristic of the composite shafts.

Title: Dynamic Problems concerning the Speed of Rotation Increase of a Turbine-blower Assembly [16].

Author: A. Castilho, G. Jacquet-Richardet, M. Lalanne.

Objective: Investigate the dynamic problems of a turbine-blower assembly when its speed of rotation is increased by roughly ten per cent to obtain better efficiency.

Method:

- Finite Element Method of the turbine-blower.
- Dynamic Response Analysis to an excitation by the base

Result:

- Turbine speed increase as far as:
  - a) the components transmitting power
  - b) certain blade profile are modified
  - c) the wired blades of low pressure being made on the rotors.
- Reducing of gear can be affected by slight modification due to speed increases.

CHAPTER 3

METHODOLOGY

3.1 Flow Chart

Figure 3.1 shows the flow chart of the project which involves three main parts; modeling, simulation and analysis. The expected output of this project is the critical point of model which includes resonance, critical frequency and shaft deflection.

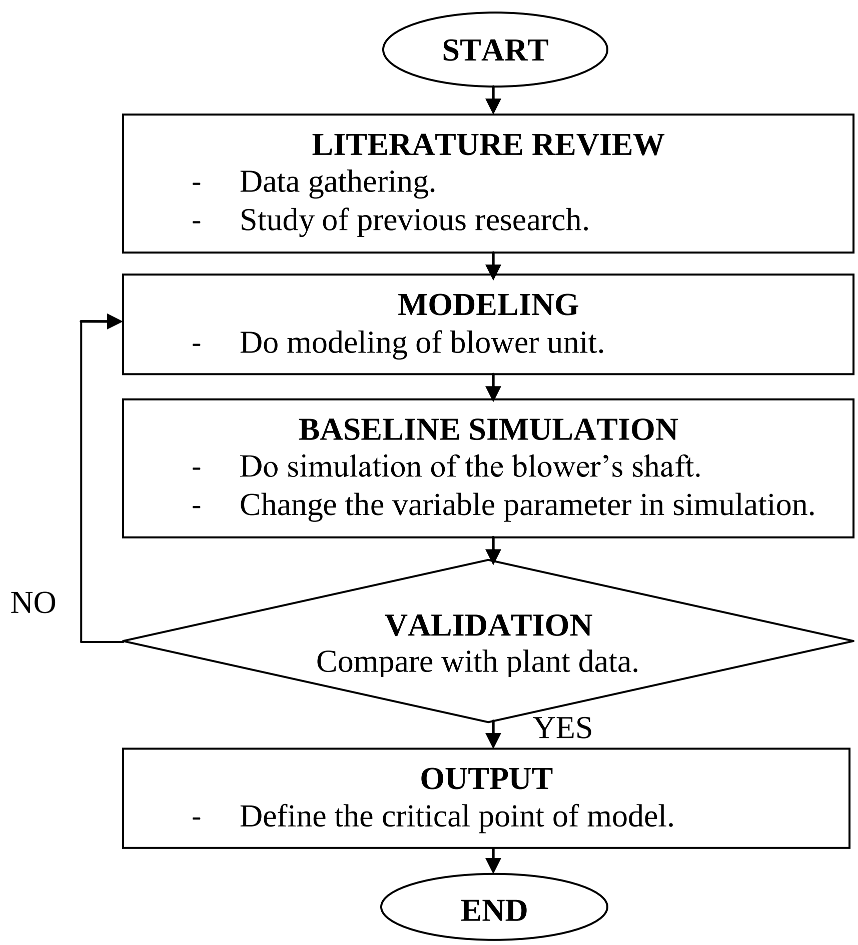


Figure 3.1: Flow chart of project.

3.2 Modeling

In order to perform the dynamic analysis, an approach of modal analysis using Ansys software is used. Firstly, data from the company such as design detail and operational record are gathered. The first main part in the project is to do modeling of blower and its assembly.

The blower unit is modeled in the Catia V5 software which consist of motor, motor's shaft, rotary piston blower and blower's shaft. The geometrical dimension of the motor is shown in the Figure 3.2. The geometrical parameters of motor's shaft, rotary piston blower and blower's shaft are using standard dimension of blower unit which are listed in Table 3.1. Figure 3.3 shows the whole model drawing.

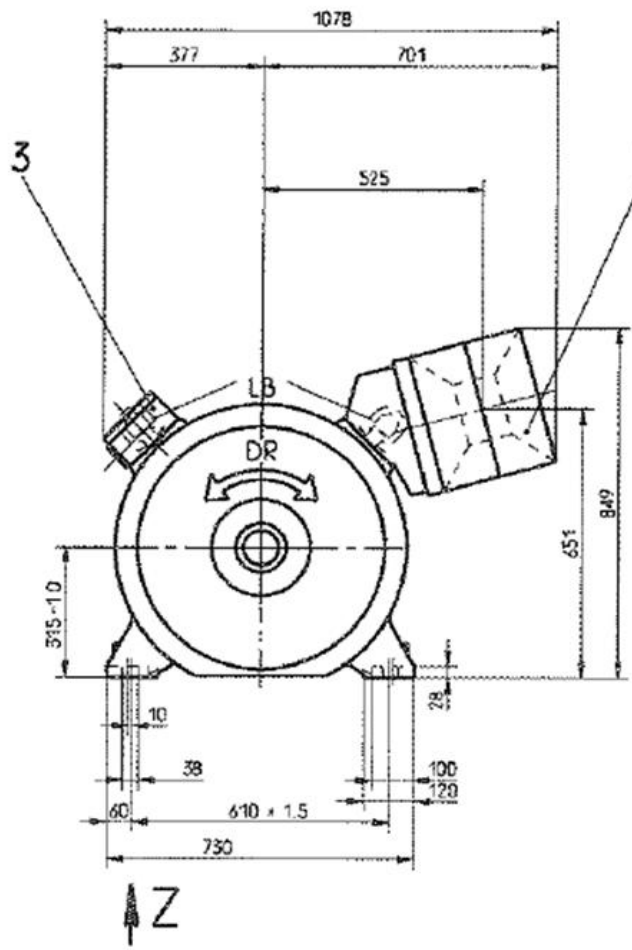


Figure 3.2: Dimension of motor [4]

Table 3.1: Dimension of components in blower

Component	Dimension	Value (mm)
Motor's shaft	Diameter	85
	Length	1800
Rotary piston blower	Diameter of lobe	39
	Length	1000
Blower's shaft	Diameter	1400
	Length of driven	1400
	Length of non-driven	1200

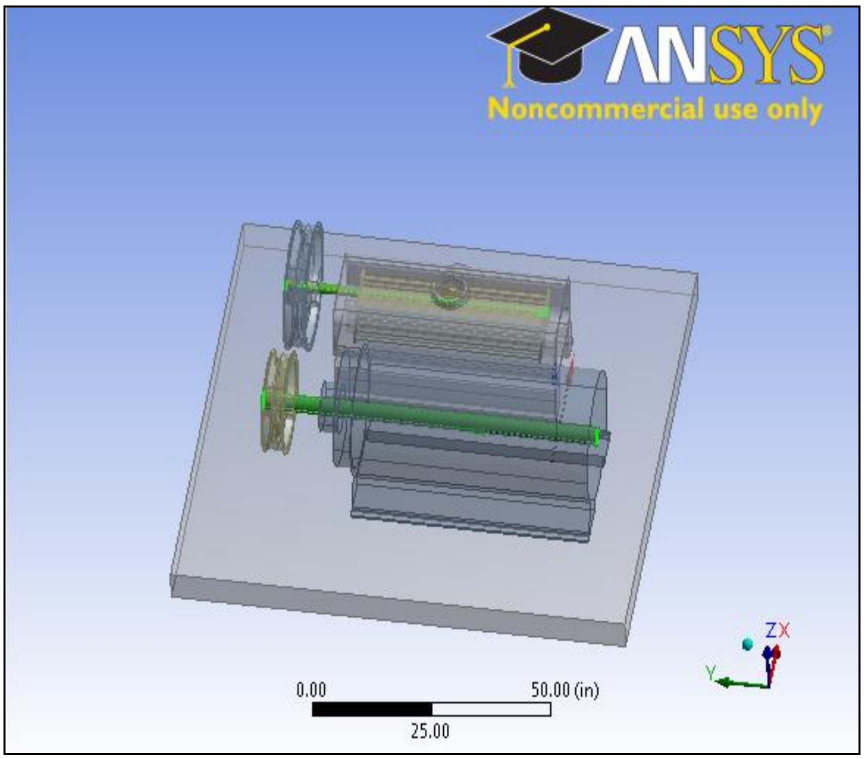


Figure 3.3: Whole model drawing

In order to achieve feasible in FEM analysis, only the most critical main component of blower’s shaft is being analyze in Ansys Workbench software.

3.3 Material Properties

The blower's shaft is exported to the Ansys Workbench software in order to do the simulation. The next step is to define the material properties of the model. Based on the technical specification given by the plant, C45N material is used for the blower's shaft. C45N, a medium carbon steel is used when greater strength and hardness is desired. Extreme size accuracy, straightness and concentricity combine to minimize wear in high speed applications [17]. Material data which is used for blower's shaft is listed in Table 3.2.

Table 3.2: Material properties

Material properties	**C45N material for shaft** [18]				
	Density (Kg/m^3)	Young's modulus (MPa)	Poison ratio	Yield stress (MPa)	Ultimate stress (MPa)
	7850	210, 000	0.3	340	600

3.4 Meshing and Boundary Condition

After that, the meshing and boundary condition of the model are defined in the Ansys Workbench software. Finite element analysis is carried out for quick identification of natural frequencies. Modal analysis is performed to note the high amplitude displacement at the various operating speed.

Meshing model of the blower's shaft is shown in Figure 3.4. Elements are sufficiently refined of the analysis become mesh independent. The boundary condition is the critical factor for the correctness of calculation [19]. Figure 3.5 shows the boundary condition of the model.

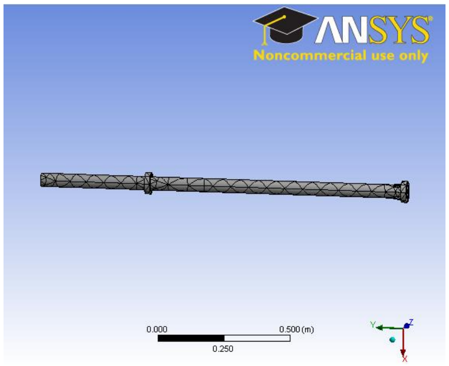


Figure 3.4: Mesh model of blower's shaft

Number of nodes: 1450

Number of elements: 698

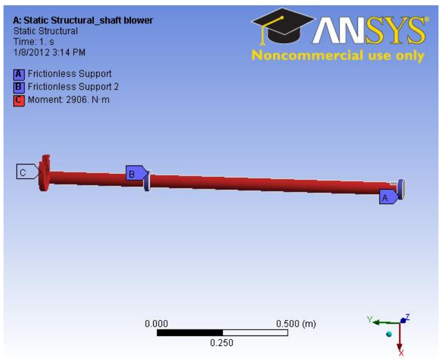


Figure 3.5: Boundary condition of blower's shaft

Frictionless support at both bearings

Moment: Various moments are applied at the shaft with respect to the speed of blower.

The output data from simulation part will be analyzed in the next chapter. The analyzing part must focus on determining the mode shape, resonance point, critical frequency, and shaft deflection.

CHAPTER 4

RESULT AND DISCUSSION

4.1 Determination of the Critical Point

The dynamic characteristic of the blower's shaft is obtained from modal analysis. The simulation is done through Ansys Workbench software. Different value of moment which indicates different value of frequency is applied to the shaft. Blower's shaft is set to rotate from 8.33Hz to 83.33Hz. Figure 4.1 shows the maximum and minimum amplitude response at different operating frequency. The maximum displacement of blower's shaft in every mode shape is calculated from FEA. The maximum displacement at different mode shape is graphically shown in Figure 4.2.

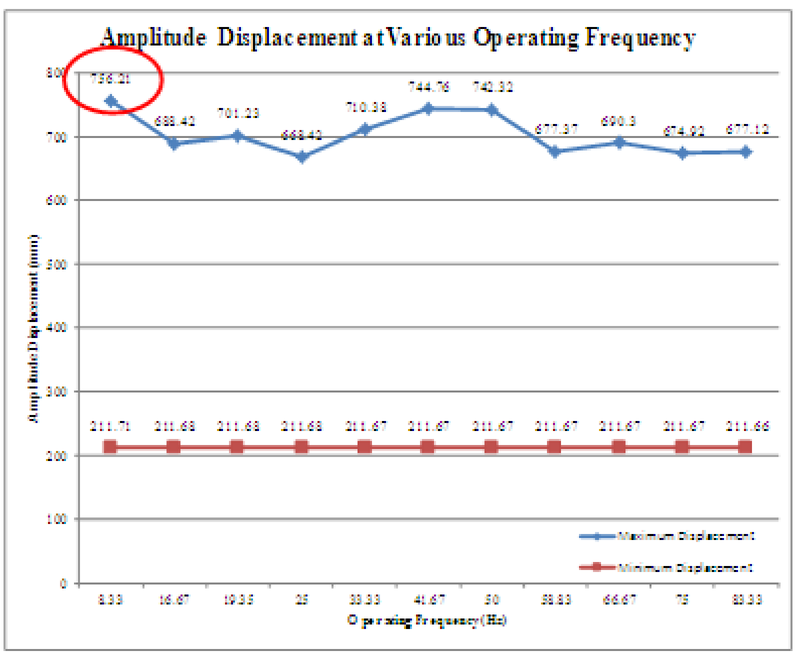


Figure 4.1: Amplitude response at difference operating frequency

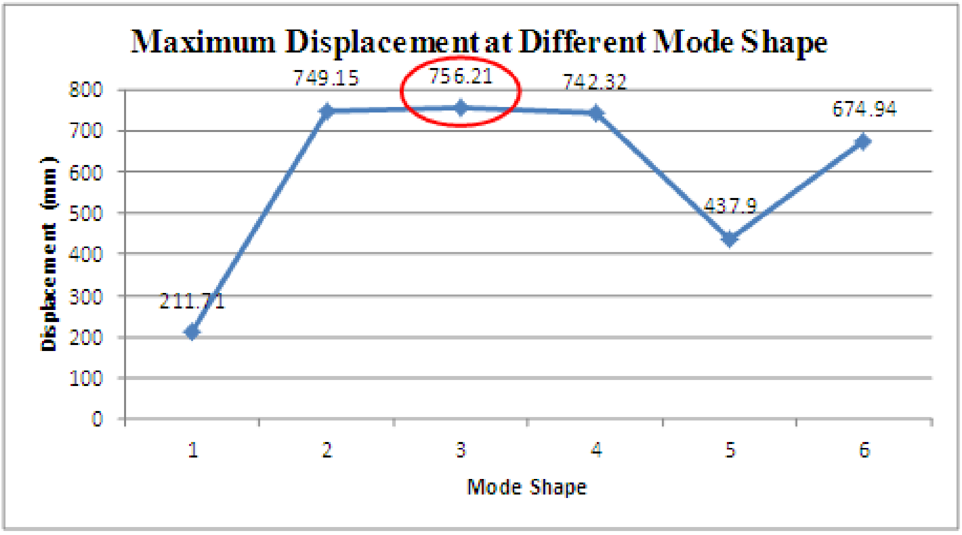


Figure 4.2: Maximum displacement at different mode shape.

As shown in Figure 4.1, large displacement of blower's shaft occurs at low operating frequency of 8.33Hz. From the simulation results and the vibrational modal shapes, it is observed that resonance frequency occurs at 8.33Hz at mode shape 3 with 756.21mm displacement at the drive end of the blower's shaft.

4.2 Mode Shape Analysis

From simulation result, it is shown that 8.33Hz is the most critical frequency of the blower's shaft. Hence, it is important to analyze the mode shape obtain from FEM to observe the various deformation of shaft deflection at 8.33Hz.

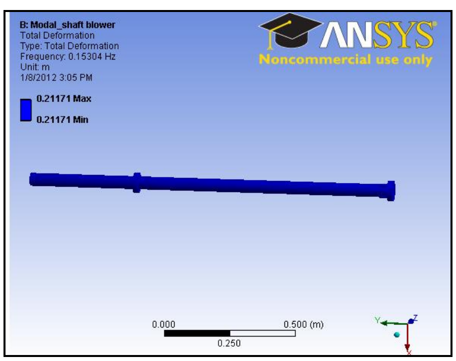
Referring to the Figure 4.3(a), the first vibration mode only has amplitude displacement in horizontal direction with no amplitude displacement in vertical direction. The maximum displacement at the horizontal direction is 211.71mm. It is important for the designer to have this tolerance value in the blower unit. As shown in mode shape 2 and 3 in Figure 4.3(b) and 4.3(c), the maximum distortion appears at the drive end of the blower's shaft with amplitude displacement of 749.15mm and 756.21mm respectively.

Based on analysis done before, it is determined that the resonance point is occurred at this mode shape 3 with 178.53Hz of natural frequency. From Figure 4.3(d) until Figure 4.3(f), it is observed that the maximum deformation start to appear at the middle of the shaft. Figure 4.3(f) shows that locations of deformation increase as the frequency increase.

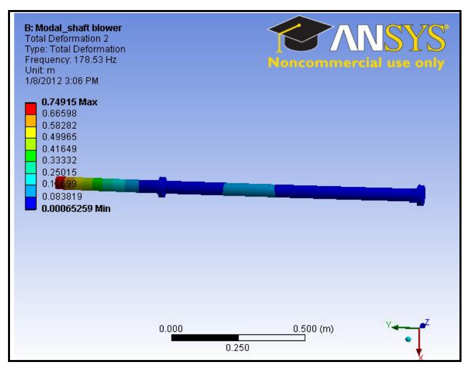
Table 4.1 below shows first six natural frequencies of the blower's shaft at the 8.33Hz of operating frequency. It is observed that the natural frequency is directly proportional as the mode shape increase.

Table 4.1: First six natural frequencies at 8.33Hz operating frequency

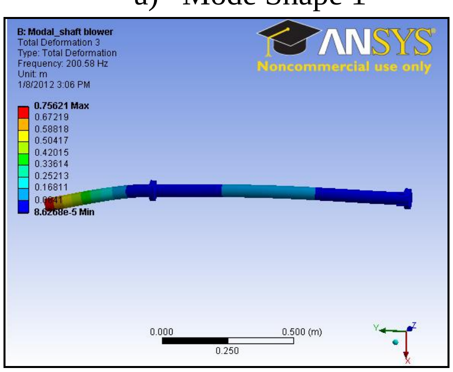
Mode Shape	**Natural Frequency (Hz)**	**Operating Frequency (Hz)**
1	0.15304	8.33
2	178.53	8.33
3	200.58	8.33
4	238.48	8.33
5	306.71	8.33
6	642.31	8.33



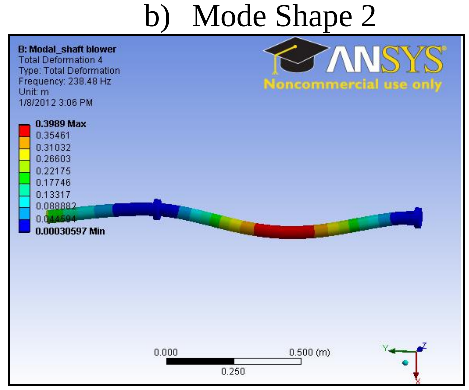
a) Mode Shape 1



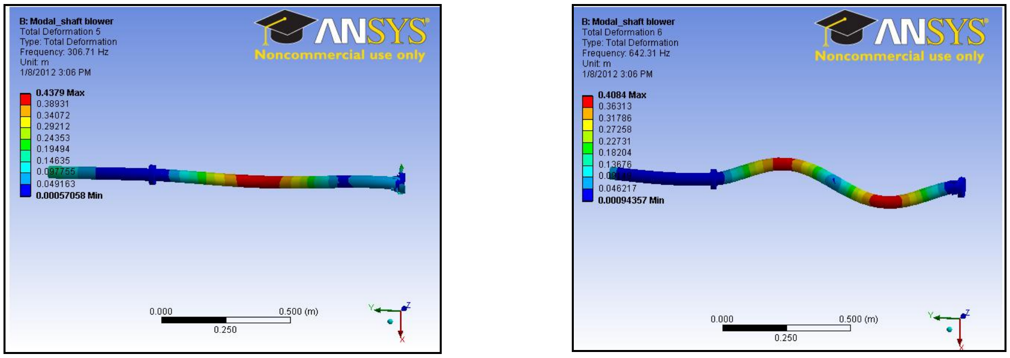
b) Mode Shape 2



c) Mode Shape 3



d) Mode Shpae 4



e) Mode Shape 5 f) Mode Shape 6

Figure 4.3: Deformation at 8.33Hz

The operating frequency is increased up to the 83.33Hz. As the operating frequency increase, it is observed that the drive end bearing will experience maximum deformation at 75Hz operating frequency. Figure 4.4 will show the deformation of the drive end bearing. This point shows that at the operator should avoid from 75Hz operating frequency to make sure that the drive end bearing not undergoes maximum deformation.

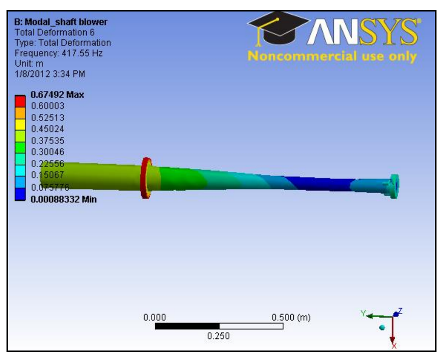


Figure 4.4: Deformation at 75Hz

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

Application of modal analysis technique for monitoring the blower unit is convenient and reliable in determining the failures in early stage as well as avoiding unscheduled shutdown and expensive repair cost. The ultimate goal of this project is to investigate the mode shape of rotary piston blower, K302 at different load condition.

In this paper, the blower's shaft is created in Catia V5 software and imported to the Ansys Workbench 14.0 software for simulation. From the FEA result, the critical parameter of the blower dynamic characteristic is defined. The maximum deformation appears at the drive end of the blower's shaft which is mainly bending under low operating frequency, 8.33Hz. The resonance frequency is very important for the plant operator to operate the machine so that any failure caused by resonance can be early evaded.

5.2 Recommendation

Further work should be done to study effect of the whole components in the blower unit. Sensitivity Analysis of blower unit and detailed FEM analysis can shed light to unrevealed aspects of dynamic response of blower unit. It is advantage if methodology includes the experimental tests to validate the results from FEM. The analysis on blower unit can give good benefit for plant reference.

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APPENDICES

A. Gantt Chart

Table A.1: Timeline for FYP I

Activity/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	FYP I (JAN 12)													
Selection of topic														
Preliminary research work														
Extended proposal defense														
Proposal defense														
Project work														
- Tutorial Ansys														
- Modeling														
Interim draft report														
Interim report														

	Process
	Mile stone

Table A.2: Timeline for FYP II

Activity/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	FYP II (MAY 12)														
Project work															
- **Modeling**															
- **Calculation**															
- **Simulation on model**															
- **Analyze the output**															
Progress report															
Pre-SEDEX															
Draft report															
Submission of dissertation (Soft bound)															
Submission of technical paper															
Oral presentation															
Project submission															

	Process
	Mile stone

B. Tools

Table A.3: Tools used in project

Tool	Function
Catia V5 software	To do modeling of blower unit and its assembly
Ansys Workbench 14.0 software	To do simulation of blower’s shaft
Microsoft Excel 2010 software	To tabulate and analyze the result