Study and Analysis of Vibration Characteristics of Selected Peristaltic Pump

Mr. Shakil H Choudhari
Research Scholar
Department of Mechanical Engineering
Dr. J. J. Magdum College of Engineering, Jaysingpur

Mr. M. V Kharade
Associate Professor
Department of Mechanical Engineering
Dr. J. J. Magdum College of Engineering, Jaysingpur

Abstract

Vibrations basically are the displacement of a mass back and forth from its static position. In every pump, dynamic forces of mechanical and hydraulic origin are present and a certain vibration and noise is therefore inevitable. To ensure the safety of the pump and associated plant components, the vibration and noise must be kept within certain limits. The objective of the paper is to identify the fundamental natural frequency of peristaltic pump. Effect of vibration on performance of peristaltic pump will be studied to find out the natural frequency and mode shapes of peristaltic pump by numerical method by use of software ANSYS Workbench.

Keywords: Dynamic Forces, Hydraulic, Peristaltic Pump, Natural Frequency

I. INTRODUCTION

Modal analysis is the process of determining the inherent dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes, and using them to formulate a mathematical model for its dynamic behaviour. The formulated mathematical model is referred to as the modal model of the system and the information for the characteristics are known as its modal data. The dynamics of a structure are physically decomposed by frequency and position [1]. This is clearly evidenced by the analytical solution of partial differential equations of continuous systems such as beams and strings. Modal analysis is based upon the fact that the vibration response of a linear time-invariant dynamic system can be expressed as the linear combination of a set of simple harmonic motions called the natural modes of vibration. This concept is akin to the use of a Fourier combination of sine and cosine waves to represent a complicated waveform. The natural modes of vibration are inherent to a dynamic system and are determined completely by its physical properties (mass, stiffness, damping) and their spatial distributions [2]. Each mode is described in terms of its modal parameters: natural frequency, the modal damping factor and characteristic displacement pattern, namely mode shape. The mode shape may be real or complex. Each corresponds to a natural frequency. The degree of participation of each natural mode in the overall vibration is determined both by properties of the excitation source(s) and by the mode shapes of the system.

Finite element method is used to find the natural frequencies and mode shapes. The CAD model created in 3D CAD modelling software was imported. For finite element analysis the ANSYS Workbench module was used.

II. THEORETICAL BACKGROUND

A. Causes of Excessive Vibration

There are many potential vibration excitation sources.

1) Resonance Response

Amplified response causes of vibration are generally more complex to analyze. They result from operation at speeds close to a mechanical or hydraulic resonant frequency of a major pump, foundation or pipe component. This is of special concern with variable speed, large multistage horizontal and/or vertical pumps. A margin of safety should be provided between the pump/vane pass speed/frequencies, and major structural (and/or pipe hydraulic) natural frequencies. Typical acceptable margins are 15-25%. The amplitude of the vibration response can be amplified 2.5 times or higher at or near a component natural (critical) resonant frequency.

2) Another Excitation Source (Bearing Wear)

Worn rolling element bearings can also be an excitation source. They have distinct vibration frequency signatures based on the number of bearing balls or rollers. These exciting frequencies can be determined from the bearing manufacturer.

3) Field Vibration Problem Example

The writer was recently asked to investigate an excessive vibration problem on three new replacement vertical sewage lift pumps, which had a different design then the original pumps. The original pumps had not experienced any vibration problems. The first step in the analysis was to distinguish between the problem being of "excitation" or "response" (natural frequency) origin, and then differentiate between the root cause being mechanical or hydraulic.
4) Mechanical

Roller cage unbalance excitement issues normally show up at one times the running speed. However, the analysis of these subject pumps did not show excessive vibration at any specific running speed. This was a variable speed application. The only trend is that the vibration generally increases with speed. This could be explained by the fact that the pump “Suction Energy” also increases with pump speed. Instead, these results strongly suggest that the primary exiting force is not mechanical in nature, which then opens the door to a possible hydraulic exciting force.

5) Suction Pressure Pulsations

To further confirm that the reported high pump vibration is mainly caused by this High Suction Energy Cavitation / Suction Recirculation hydraulic excitation, the “Suction Pressure Pulsations” were also measured, flow rates to show if it tracked with the vibration trend of higher amplitudes occurring at reduced flow rates, which it does.

From above different causes the study is carried out. To find the effect of resonance response it is necessary to carry out the modal analysis.

It is used to determine the natural frequencies and mode shapes of structure being studied.

B. Assumptions

- Valid for structural and fluid degrees of freedom (DOFs). Electrical and thermal DOFs may be present in the coupled field mode-frequency analysis using structural DOFs.
- The structure has constant stiffness and mass effects.
- There is no damping, unless the damped eigen solver (MODOPT, DAMP or MODOPT, QRDAMP) is selected.
- The structure has no time varying forces, displacements, pressures, or temperatures applied (free vibration).

This analysis type is used for natural frequency and mode shape determination. The equation of motion for an undamped system, expressed in matrix notation using the above assumptions is

\[ [M]\ddot{\mathbf{u}} + [K][\mathbf{u}] = [\mathbf{0}] \]  

(1)

Note that \([K]\), the structure stiffness matrix, may include prestress effects (PSTRES,ON). For a discussion of the damped Eigen solver (MODOPT,DAMP or MODOPT,QRDAMP).

For a linear system, free vibrations will be harmonic, of the form as,

\[ [\mathbf{u}] = [\mathbf{\phi}]_i \cos \omega_i t \]  

(2)

Where:

- \([\mathbf{\phi}]_i\) = eigenvector representing the mode shape of the \(i\)th natural frequency
- \(\omega_i\) = \(i\)th natural circular frequency (radians per unit time)

Thus, Equation (2) becomes,

\[ (-\omega_i^2[M] + [K])\mathbf{\phi}_i = \mathbf{0} \]  

(3)

This equality is satisfied if either \([\mathbf{\phi}]_i = \{0\}\) or if the determinant of \([K] - \omega^2\mathbf{M}\) is zero. The first option is the trivial one and, therefore, is not of interest. Thus, the second one gives the solution:

\[ |[K] - \omega^2[M]| = 0 \]  

(4)

This is an eigen value problem which may be solved for up to \(n\) values of \(\omega^2\) and \(n\) eigenvectors \([\mathbf{\phi}]_i\) which satisfy Equation (4) where \(n\) is the number of DOFs. The eigen value and eigenvector extraction techniques are discussed in Eigen value and Eigenvector Extraction.

Rather than outputting the natural circular frequencies \{\(\omega\)\}, the natural frequencies \(f_i\) are output; where:

\[ f_i = \frac{\omega_i}{2\pi} \]  

(5)

Where:

\(\omega_i\) = \(i\)th natural frequency (cycles per unit time) [3].

III. MODAL ANALYSIS USING FEA

A. Finite Element Analysis of Selected Parts

The commercial ANSYS package is used for the FE meshing, modeling and analysis module. The general structure of a Finite Element Analysis involves the following three steps

- The description of the geometry, the physical characteristics and the mesh (pre-processing)
- The application of finite element analysis. (solution)
- The visualization and interpretation of the results of the solution. (post processing)

1) Pre-Processing

The pre-processing module is used for entering all the information necessary to define the problem. This data relates to the discretization of the structure and the representation of its physical behavior.

The pre-processing module must accomplish the following three functions

- Description of the geometry of the object in terms of the chosen element types.
- Mesh generation.
- Definition of loading and boundary condition

Fig. 3.1: (a) shows meshed model of Peristaltic pump assembly and (b) Boundary conditions

In the solution part, computer solves the unknowns in the finite element problem, i.e. it solves the linear or non-linear system of equations based on the variation or the projective formulation. Input to solution module is the finite element model, the physical characteristics and the boundary conditions (pre-processor file). Its output is the value of the unknown quantity at each of the nodes of the grid.

2) Post-processing
The large amount of data generated in the solution phase (several thousands of nodal values in complex problems). These are often too much to be understood without further processing. The Post-Processor presents the output of the problem in a manner which is easily understood and interpreted by the user.

The Post-processor performs two tasks
- Extraction of significant information
- Synthetic presentation of the numerical data via graphics facilities.

B. Results of Modal Analysis of Peristaltic Pump

Table below indicates six mode shapes of Peristaltic Pump assembly and respective frequencies.

Table – 1

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Mode Shape</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>665.63</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>672.28</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>862.79</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>973.45</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>1315</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>1654.6</td>
</tr>
</tbody>
</table>

Fig. 3.3: Mode Shape No.1 and 02
IV. CONCLUSION

From this paper various causes of vibration in peristaltic pump are found out. From the modal analysis the fundamental frequency of the pump assembly is 665.63 Hz. The operating maximum frequency of the pump is 120 rpm or 12.56 Hz. The operating frequency and the natural frequency are far away and hence there is no chance of resonance. It is now necessary to test the other possibilities for vibration in pump.

REFERENCES