Dynamic characteristics analysis of agitator design for soy sauce cooking process

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Abstract. Soy sauce is one type of dark brown condiment, distinctive smell, salty or sweet taste, lumpy, and contains protein produced from fermented. One of the machines that can be used for cooking soy sauce is a pressure vessel as cooking pan with an agitator. Agitation is a process of mixing and stirring and are carried out by heat transfer and mass inter-phases or with external surfaces (due to outside influences). The agitator is a system used for mixing and stirring accompanied by a phase change. In the design of rotating machinery, it is necessary to predict the dynamic characteristic in bending and in torsion to avoid failure. Dynamic characteristics analysis of agitator design for soy sauce cooking process consists of mechanical vibration analysis and mass unbalance response. The stiffness method for the agitator shaft by dividing the shaft element into two elements based on the bearing position. The bearing is assumed to be roller supports and only moves in the direction of translation and rotation. Based on the dynamic characteristics of rotordynamics prediction by using finite element method both theoretical and software, the agitator in operating conditions with the rotation speed of 5 RPM according to Campbell diagram will not fail. The highest amplitude of the mass unbalance response is less than 2.5x10\textsuperscript{-2} mm.

Keywords: Soy Sauce, Agitator, Finite Element Methods, Campbell Diagram, Rotordynamics

1. Introduction
Soy sauce is one type of dark brown food product or condiment, distinctive smell, salty or sweet taste, lumpy, and contains protein produced from fermented [1] and [2]. The basic ingredients in making Indonesian soy sauce is soybeans. Soybeans are the protein source with the highest content of about 40% compared to other types of nuts [3]. Soy sauce consumption needs are increasing every year, requiring the companies to increase production capacity and quality of soy sauce products. To increase production capacity, it is necessary to design a soy sauce production process that is able to meet the needs. The flow chart of the cooking process for soy sauce, shown in Figure 1. Based on Figure 1, one step in the process of producing soy sauce is the cooking process [4]. The machine used for soy sauce cooking process is using a pressure vessel as storage and an agitator as a mixer. The function of the
agitator for soy sauce cooking process is as a stirrer, transfer and evenly distribute heat to the fluid (soy sauce) of the pressure vessel wall. An agitator is machines used in industries that process products in the chemical, food, pharmaceutical and cosmetic industries [5]. If the result of the design is not appropriate with the function of the agitator, the agitator performance does not optimal. Another result is the effect of uneven heat distribution give damage to the taste of soy sauce [4].

![Flowchart of soy sauce making process](image)

**Figure 1.** Flowchart of soy sauce making process [6]

The dynamic characteristics of rotordynamics in agitator must be analyzed to get satisfactory design performance and to minimize mechanical vibration that can cause failure in the planned operating conditions. The frequency, mass unbalance response, and mode shapes can be analyzed theoretically or numerically [7]. In this paper, the finite element method is employed for dynamic characteristics analysis of agitator. Comparisons are made between finite element analysis (FEA) theoretical and FEA software to vibration frequency in Campbell diagram and mass unbalance response.

2. **Material and methods**

2.1. Agitator design data

The agitator impeller type that has been designed based on the operating conditions in the soy sauce cooking process is anchor impeller. The viscosity of soy sauce 1900 cP, volume 6000 Liter, and density of soy sauce 1300 kg/m³. The anchor impeller geometry adjusted with the pressure vessel. The agitator design concept is shown in Figure 2, and the impeller geometry is shown in Table 1.

![Agitator design concept](image)

**Figure 2.** Agitator design concept
Table 1. Anchor impeller geometry

| Geometry                | Symbol | Value  |
|-------------------------|--------|--------|
| Pressure vessel diameter| $D_p$  | 2450 mm|
| Impeller diameter       | $D_i$  | 2352 mm|
| Clearance               | $C_w$  | 47 mm  |
| Impeller width          | $W$    | 193 mm |
| Impeller height         | $h$    | 1344 mm|
| Shaft length            | $L$    | 2360 mm|
| Shaft diameter          | $D$    | 70 mm  |

The agitator rotation speed for soy sauce cooking process of 5 RPM, the 5052-H38 aluminum alloy is the material used with the mechanical properties as yield strength of 255 MPa, density of 2680 kg/m³, and based on ANSI/NSF 51-1997 is a food grade material. The agitator components are shown in Figure 3.

2.2. Dynamic characteristics analysis

Based on the rotordynamics prediction, the rotor is part of rotating machinery including the shaft, disk, impeller, bearing, gear, coupling, and other rotating elements [8]. In this theory, the disk is assumed to be rigid and is then characterized solely by its kinetic energy, and there is no strain energy. The dynamic characteristics analysis is used to determine the ability of the agitator shaft at operation rotating speed based on natural frequency, Campbell diagram analysis, and mass unbalance response. Finite element methods both theoretical and software for dynamic characteristics analysis as follows:

2.2.1. Finite element equation for rotor disk

The agitator for soy sauce cooking process has one rotor so that the dynamic behavior of the monorotor. The equation of motion with mode shapes in Figure 4 for rotor disk as follows:

\[
\frac{d}{dt} \left( M \frac{\partial \delta}{\partial \delta} \right) - \frac{\partial T}{\partial \delta} = \begin{bmatrix}
M_d & 0 & 0 & 0 \\
0 & M_d & 0 & 0 \\
0 & 0 & I_{dx} & 0 \\
0 & 0 & 0 & I_{dz}
\end{bmatrix} \begin{bmatrix}
\ddot{u} \\
\dot{w} \\
\dot{\theta} \\
\dot{\psi}
\end{bmatrix} + \Omega \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & -I_{dy} \\
0 & 0 & I_{dy} & 0
\end{bmatrix} \begin{bmatrix}
\dot{u} \\
\dot{w} \\
\dot{\theta} \\
\dot{\psi}
\end{bmatrix}
\] (1)
2.2.2. Finite element equation for shaft

Mass matrix, stiffness matrix, and force vector for bar element. The bar element is shown in Figure 5. For this one dimensional element, the two endpoints form the joints (nodes). This element type is affected by the axial movement or axial displacement of the shaft [9].

\[
[K] = \frac{AE}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}
\]  
(2)

For the equation of mass matrix as follow: [8]

\[
[M] = \frac{\rho A L}{6} \begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}
\]  
(3)

To calculate mass matrix, stiffness matrix in the torsion element shown in Figure 6 as follows: [9]

\[
[K] = \frac{J G}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}
\]  
(4)

\[
[M] = \frac{\rho J L}{6} \begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}
\]  
(5)

Figure 7 shows a uniform beam element subjected to the transverse force distribution. In this case, the joint undergo both translational and rotational displacements.

\[
[K] = \frac{J G L}{EI} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}
\]  

(6)

\[
[M] = \frac{\rho J L}{6} \begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}
\]  

(7)

The equation of stiffness matrix and mass matrix for beam element as follow: [9]
The mode shapes for three-dimensional beam element with gyroscopic matrix shown in Figure 8 and the equation as follow: 

\[
[K] = \frac{E I}{L^3} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4I^2 & -6L & 2I^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2I^2 & -6L & 4I^2 \end{bmatrix}
\]

\[
[M] = \frac{\rho A L}{420} \begin{bmatrix} 156 & 22L & 54 & -13L \\ 22L & 4I^2 & 13L & -3I^2 \\ 54 & 13L & 156 & -22L \\ -13L & -3I^2 & -22L & 4I^2 \end{bmatrix}
\]

The mode shapes for three-dimensional beam element with gyroscopic matrix shown in Figure 8 and the equation as follow: [10]

\[
[C] = \frac{\rho \Omega}{15L} \begin{bmatrix} 0 & -3L & 0 & -3L & 36 & 0 & -36 & 0 \\ 3L & 0 & -3L & 0 & 0 & 4L^2 & 0 & L^2 \\ 0 & 3L & 0 & 3L & -36 & 0 & 36 & 0 \\ 3L & 0 & -3L & 0 & 0 & -L^2 & 0 & 4L^2 \\ -36 & 0 & 36 & 0 & 0 & -3L & 0 & -3L \\ 0 & 4L^2 & 0 & L^2 & 3L & 0 & -3L & 0 \\ 36 & 0 & -36 & 0 & 0 & 3L & 0 & 3L \\ 0 & L^2 & 0 & -4L^2 & 3L & 0 & -3L & 0 \end{bmatrix}
\]

2.2.3. Finite element equation for bearing

The equation of motion for rotor bearing as follows: [8]

\[
\begin{bmatrix} F_u \\ F_\theta \\ F_w \\ F_\psi \end{bmatrix} = - \begin{bmatrix} k_{xx} & 0 & k_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ k_{xz} & 0 & k_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u \\ \theta \\ w \\ \psi \end{bmatrix} - \begin{bmatrix} c_{xx} & 0 & c_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ c_{xz} & 0 & c_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \dot{u} \\ \dot{\theta} \\ \dot{w} \\ \dot{\psi} \end{bmatrix}
\]

2.2.4. Mass unbalance equation

The mass unbalances in rotordynamics prediction is shown in Figure 9 below. The general expression for the kinetic energy of mass unbalance and based on Lagrange’s equations for finite element method was given in equation: [11]

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \delta} \right) - \frac{\partial T}{\partial \delta} = \begin{bmatrix} F_u \\ F_w \end{bmatrix} = -m_u d\Omega^2 \begin{bmatrix} \sin \Omega t \\ \cos \Omega t \end{bmatrix}
\]
Where:  \( M_d \) = Mass of disk (kg), \( I_d \) = Inertial mass of disk (kgm\(^2\)), \( A \) = Cross section area (m\(^2\)), \( E \) = Young modulus (N/m\(^2\)), \( \rho \) = Density (kg/m\(^3\)), \( J \) = Polar moment of inertia (m\(^4\)), \( G \) = Shear modulus (N/m\(^2\)), \( I \) = Momen of inertia (m\(^4\)), \( \Omega \) = Rotation speed (rad/s), and \( m_u \) = Mass unbalance (kg).

### 2.2.5. General equation of motion

The most general case of the rotordynamics was enunciated, and it was briefly mentioned how to in general terms it should be solved. A solution of the equation to rotordynamics problems using finite elements will produce natural frequency and mode shapes. Two methods of solving this problem are the pseudo-modal method with quicker and more efficient, and the second methods are the direct method, more precise but more demanding of computational effort \[11\]. The general equation to calculate natural frequency and mode shapes as follow:

\[
[M] \ddot{x} + [C](\Omega) \dot{x} + [K]x = 0
\]

(11)

The solution pseudo-modal method as follow: \[11\].

\[
\begin{bmatrix}
0 & I \\
-m^{-1}k & m^{-1}C
\end{bmatrix}
\begin{bmatrix}
rP \\
p
\end{bmatrix}
= \frac{1}{r}
\begin{bmatrix}
rP \\
p
\end{bmatrix}
\]

(12)

### 3. Result and discussion

Dynamic characteristics analysis in rotordynamics is assumed free motion with vibration frequency and force response analysis. The impeller is assumed rigid disk, and the material properties with the shaft are same. The disk parameter to vibration frequency analysis consists of the mass of disk (Md), and inertia mass of disk of X, Y, Z axis. Mass of disk as 48.145 kg, inertia mass \( I_{dx} = 7.249 \) kgm\(^2\), \( I_{dy} = 22.196 \) kgm\(^2\), \( I_{dz} = 19.415 \) kgm\(^2\). The bearing as roller supports \[12\]. The translation and the rotation displacement of the shaft axis occur on the roller supports. To force response analysis caused by a mass unbalance when the rotor rotates then eight M14 bolts is modeled as a connector between impeller. The length of bolt 60 mm and total weight 1.6 kg. The mass unbalances position from the rotor axis is radius impeller 1176 mm. The shaft parameters needed to analyze dynamic characteristics as in Table 2.

| Parameters                        | Symbol | Value     |
|-----------------------------------|--------|-----------|
| Shaft length between two bearings | L1     | 200 mm    |
| Overhang shaft length             | L2     | 2260 mm   |
| Radius                            | R1     | 35 mm     |
| Density                           | \( \rho \) | 2680 Kg/m\(^3\) |
| Young modulus                     | E      | 70 GPa    |
| Cross section area                | A      | 3.847e3 mm\(^2\) |
| Inertia                           | I      | 1.178e6 mm\(^4\) |
| Polar moment of inertia           | J      | 2.356e6 mm\(^4\) |
| Poisson Ratio                     | \( \nu \) | 0.33      |
3.1. Frequency analysis of vibration

The main characteristic of the agitator for the cooking process is that it has intermediate support. For illustration, the shaft could be divided into two elements with three nodes as shown in Figure 10. The first node in the top bearing, the second node in the bottom bearing, and the last node in the disk. The model analysis uses FEA theoretical and FEA software. The stiffness method free body diagram for this model as shown in Figure 11 and Figure 12. The frequency analysis of vibration with FEA theoretical uses the equation of motion above.

| Parameters      | Symbol | Value |
|-----------------|--------|-------|
| Shear modulus   | G      | 26 GPa|

The frequency analysis uses FEA software, four mode shapes selected based on FEA theoretical. In this FEA software problem, the rigid disk modeled in modal analysis can use a mass point feature, by entering data in the form of mass and inertia on all axis. Roller supports as boundary condition use remote displacement feature. Element type used is tetrahedron quadratic with the number of element 14888 or mesh size 8 mm.

Figure 13 is a Campbell diagram for two bearings model as roller support based on FEA theoretical and software. Points A, C, D, and F correspond to the intersection with N/60 and 0.5 N/60 as BW (Backward Whirl). Points B and E with FW (Forward Whirl). In points, A, B, and C is the intersections between lines of F2, F3, and F4 with N/60. At the corresponding points, a frequency of the rotor equals the frequency of rotation. Points D, E, dan F is intersection points between lines of F2, F3, F4 with 0.5 N/60. At the corresponding points, a frequency of the rotor equals half of the frequency of rotation. These six points in the agitator design will occur resonance that is affected by vibration so that the application needs to be considered not to use rotational speed around the six points. The operation speed of agitator 5 RPM based on the Campbell diagram is no resonance. That, the design of an agitator with frequency analysis is still safe. Campbell diagram around operation speed is the plot in Figure 14.
Comparisons of frequency in Table 3 show good accuracy FEA theoretical and software in beam element for rotation speed is 0 and 5 RPM.

**Table 3.** The frequency associated with the previous Campbell diagram

|          | F1          | F2          | F3          | F4          |
|----------|-------------|-------------|-------------|-------------|
| 0 rotation speed FEA Theoretical (Hz) | 21.1911     | 14.1213     | 3.0916      | 2.9490      |
| FEA Software (Hz)                  | 21.0130     | 14.0950     | 3.0908      | 2.9485      |
| Error (%)                          | 0.84        | 0.19        | 0.03        | 0.02        |
| 5 RPM rotation speed FEA Theoretical (Hz) | 21.1919     | 14.1209     | 3.0920      | 2.9487      |
| FEA Software (Hz)                  | 21.0140     | 14.0950     | 3.0911      | 2.9481      |
| Error (%)                          | 0.84        | 0.18        | 0.03        | 0.02        |
3.2. Mass unbalance response

Using equation 10, to analyze the effect amplitude of mass unbalances with assuming one bearing model as fixed support. Comparison of force response analysis for FEA theoretical and software with up to a maximum speed of 1750 RPM motor is shown in Figure 15. The maximum amplitude occurs at point B, mode shape 3 or FW in operation speeds 190 RPM is 45.5 mm.

![Figure 15. Mass unbalance response](image1)

Figure 15. Mass unbalance response

Figure 16 shows the amplitude of mass unbalances response 5 RPM for agitator operation condition. The maximum amplitude in mode shape 3 is less then 2.5x10⁻² mm and can be considered very small.

![Figure 16. Mass unbalance response for 5 RPM](image2)

The rotation speed and frequency range that must be avoided when the agitator operates is based on the Campbell diagram as in Table 4.

| Points | The rotation speed range (RPM) | The frequency range (Hz) |
|--------|-------------------------------|-------------------------|
| A      | ± 160                         | ± 2.5                   |
| B      | ± 190                         | ± 3                     |
| C      | ± 680                         | ± 11                    |
| D      | ± 310                         | ± 2.5                   |
| E      | ± 430                         | ± 3                     |
| F      | ± 1150                        | ± 9                     |
4. Conclusion
Dynamic characteristics analysis of agitator for soy sauce cooking process with FEA both theoretical and software has a similar result. The stiffness method approach for the agitator shaft is carried out by dividing the shaft element into two elements based on the bearing position. The bearing is assumed as roller support. The critical speed is not in the range of the operating speed. The agitator in operating conditions with the operation speed of 5 RPM based on Campbell diagram will not fail with the highest amplitude occurring in mode shape 3.

5. Acknowledgments
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