Journal of Sustainability Science and Management
Volume 14 Number 4, August 2019: 73-80
eISSN: 2672-7226

CONCEPTUAL DESIGN OF A SAGO DRYING MACHINE AND STRUCTURAL SIMULATION USING FINITE ELEMENT ANALYSIS

SHAHROL MOHAMADDAN*, LEW HAO XIAN, MOHD FADZLI ASHARI, MARINI SAWAWI AND MOHAMAD SYAZWAN ZAFWAN MOHAMAD SUFFIAN

Department of Mechanical and Manufacturing, Faculty of Engineering, Universiti Malaysia Sarawak, Kota Samarahan, Sarawak, Malaysia.

**Corresponding author: mshahrol@unimas.my*

Abstract: The increasing demand for sago flour indicates the need to enhance its production. The objective of this research is to propose a new design of a sago drying machine and conduct a finite element analysis (FEA) simulation on its structure. The sago drying machine was developed based on a tray-mixer concept. Mathematical modelling and simulation analysis were conducted on three critical parts; the tray, support plate and mixer. The maximum stress for the tray bending was 6.51 MPa with manual calculation and 7.37 MPa with simulation analysis. The tray-bending deflection was 0.52 mm with manual calculation and 0.53 mm with simulation analysis. For support plate bending, the maximum stress was 17.33 MPa with manual calculation and 12.87 MPa with simulation analysis. The deflection for support plate bending was 1.06 mm with manual calculation and 1.04 mm with simulation analysis. Lastly, the maximum stress acting on the mixer was 1.09 MPa and maximum deflection was 0.0035 mm. The error between mathematical calculation and FEA simulation was based on different assumptions on the model and limitation of dimensions covered, for example, the 2D and 3D views. Data from this research could be used to improve the structural design of a new sago drying machine.

Keywords: Sago, drying machine, conceptual design, FEA simulation, structural analysis.

Introduction

Sago is a starch extracted from the pith of the sago palm (*Metoxylon sagu*). The name of the genus Metroxylon is a combination of two Greek words, meaning heart wood, which describes the large proportion of pith contained in the trunk of the palm. Sago is a staple in the lowlands of New Guinea and Moluccas islands (Vijay *et al*., 2017). In Malaysia, it is an inexpensive and reliable source of starch although it is not a staple food (Kamal *et al*., 2007). The plant is grown as a crop by natives in Sarawak and because of its high starch content, it has been dubbed the "starch crop of the 21st century" (Kamal *et al*., 2017). One palm tree can yield around 150 to 300 kg of sago (Lal, 2003).

The sago planted in Sarawak is called *Rumbia* and the harvest period depends on soil fertility. On average, it may be harvested for between 12 and 13 years if grown in normal soil, but when planted in peat soil, it may be extended to between 15 and 18 years. Sago is harvested shortly before or early in the palm tree's flowering stage, when starch content is highest. The trunk is cut into sections and the pith is extracted and pulverised, before water is used to wash out the starch kernels. Sago starch can be used to make biscuits (tebaloi), cakes, noodles and crackers.

Traditionally, the freshly extracted starch kernels is dried under the sun or on wood-fired stoves for large quantities of harvest (Pendita *et al*., 2014). However, the use of wood-fired stoves is not efficient and contributes highly to greenhouse gas emission, which is the cause of global warming. Besides the sun and wood-fired stoves, sago can also be dried using hot air and solar hybrid methods.

Sun-drying alone is time consuming and subject to weather, where rain and cloud may disrupt the process. A hybrid dryer combines the use of solar and electrical energy to dry the sago. Hot air drying, on the other hand, takes a long time to complete. For example, at 50 °C to 60 °C, the process takes 90 to 120 minutes to decrease moisture content from 40 % to 10 % in tapioca. However, at 30 °C to 35°C, the moisture content may drop from 30 % to 10 % in 60 minutes (Pandian & Meenambal, 2017).

The design of modern dryers aims to resolve existing problems in traditional drying methods. For example, it was suggested that the optimal drying temperature be kept at 50 °C to maintain moisture content and quality (Kamal *et al*., 2017). If the drying temperature is too high, the sago will turn brown due to formation of polyphenol oxidase compounds, which are hard to remove after the drying process (Dina *et al*., 2015).

For safe storage, agricultural products should be dried to between 9 % and 13 % of their moisture content (Tiwari *et al*., 1994). The drying technology should operate in the optimal temperature range economically. The dryer should also be designed as feasible solution for farmers operating in remote areas, where machine maintenance may be difficult. Ergonomic factors should also be considered to ensure that it is easy to operate.

The ventilation system is also another important factor, which needs to operate efficiently to remove humidity inside the drying chamber. For example, one research showed that the air volume flow rate of 0.0338 m^3/s produced the best quality of dried bananas (Hegde *et al*., 2015). Typically, an airflow rate of 180 to 300 mm/min is used for vegetable pieces with dry bulb air temperature of 90 °C to 100 °C and wet bulb temperature of 50 °C (Mohammed, 2013). This study presents a new sago drying machine design and its structure is tested using the finite element analysis (FEA) simulation.

Materials and Methods

This section is divided into idea generation and mathematical modelling. Idea generation discusses the process involved in the development of the machine's conceptual design. It was a long process that applied several methods. Thus, this paper would be discussing them only briefly. On the other hand, the mathematical modelling part describes the equations used to manually calculate the stress test on the machine's structure. The manually calculated stress values were compared with simulation results produced by the ANSYS Version 16.0 software. For this paper, only three critical parts of the machine were tested; the tray, support plate and the mixer, which were main components of the drying machine.

Idea Generation

Conceptual design began with initiating several possible solutions in drying the sago and they were narrowed down to a single best solution. It was also called a feasibility study, where the steps included defining problems, gathering information and generating and evaluating a concept.

Problem definition involved identifying the users' needs (in this case, the sago farmer or producer), such as machine performance, reliability and durability. All design parameters, variables and constraints were established using the House of Quality (HOQ) concept. HOQ could map the engineering characteristics in a comprehensive way. Based on the characteristics, information on a sago drying machine were gathered from journals, patents, surveys and other sources. The information included machine functions, the construction material and analysis methods.

The TRIZ theory was used to generate the machine's concept. The method consisted of 40 principles and 39 engineering parameters for general design suggestion (Altshuller as cited in Jafari *et al.,* 2013). Lastly, in concept evaluation, three criteria were compared within the generated conceptual design. The criteria included design feasibility, technology readiness and customer requirement screening. The axiomatic design (AD) concept was also applied to support the evaluation process.

Mathematical Modelling: Tray Bending Analysis

The maximum stress acting on the tray was calculated using equation 1, while equation 2 was used to calculate the maximum deflection.

$$\sigma_m = \frac{3pr^2}{4t^2} \quad (1)$$

$$y_m = \frac{0.171pr^4}{Et^3} \quad (2)$$

where,

p = stress acting on the tray (N/m^2)
r = radius of the plate (m)
t = thickness of the plate (m)
E = elastic modulus (Pa)

Mathematical Modelling: Support Plate bending analysis

The maximum stress acting on the support plate was calculated using equation 3 while equation 4 was used to calculate maximum deflection.

$$\sigma_m = \frac{pb^2}{2t^2[0.623\left(\frac{b}{a}\right)^6 + 1]} \quad (3)$$

$$y_m = \frac{0.0284pb^4}{Et^3[1.056\left(\frac{b}{a}\right)^5 + 1]} \quad (4)$$

Mathematical Modelling: Mixer Vibration Analysis

The vibration analysis was set by the boundary condition for the cylindrical support and the moment or torque experienced by the mixer shaft. The harmonic response was investigated between 1 and 1000 Hz.

Results and Discussion

Conceptual Design

Figure 1 shows the new design of a sago drying machine based on the tray-mixer concept. Table 1 shows the dimension of the subassembly with the dimensions and selected materials.

Tray Bending

The tray bending analysis was performed using the free body diagram in Figure 2. The stress acting on the tray (P) was 385.61 Pa, the thickness of the plate (t) was 0.003 m and radius (r) was 0.45 m. There were several assumptions

Figure 1: Conceptual design of a new sago drying machine

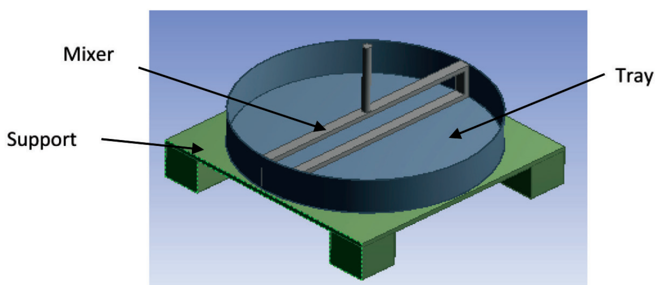

Table 1: Dimension and material selection of each subassembly

Subassembly	Dimension	Material
Tray	20 cm height x 90 cm diameter Thickness 1 mm	Stainless steel (standard part)
Support plate	100 cm x 100 cm x 20 cm	Mild steel (standard part)
Mixer	20 cm height x 85 cm long	Stainless steel (standard part)
Drying chamber	110 cm x 110 cm x 150 cm	Polycarbonate, Polyethylene

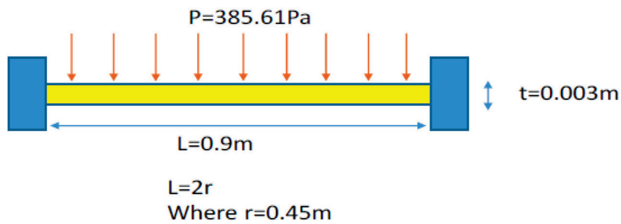

Figure 2: Free body diagram of the circular tray

in the analysis: The tray was a circular plate, the sago flour on the tray exerted a uniform load on the surface (uniform loading), the edges of the circular tray were clamped and the maximum stress was acting on the centre. Based on the calculations, the maximum stress (σ_m) acting on the tray was 6.51 MPa and the maximum deflection (y_m) was 0.52 mm.

The simulation results for tray stress and deflection were shown in Figure 3. The fixed support of the tray was highlighted in blue, which was the tray side face. The movement of the tray was assumed to be restricted by the table and chamber around it. A force equivalent to 25 kg (or 245.25 N) weight of the sago flour was acting uniformly on the internal surface. The elastic modulus used in this case was 190 GPa (for stainless steel). Based on the simulation, the maximum stress (σ_m) acting on the tray was 7.37 MPa and the maximum deflection (y_m) was 0.53 mm.

Table 2 shows the comparison between calculated and simulation results of the tray. There were errors of around 11.7% between the maximum stress and 1.8% between deflection. The errors could have occurred due to the different views (2D and 3D) applied in the calculation and simulation analysis. However, the results were considered acceptable since the maximum stress acting on the tray was less than the maximum elastic modulus of stainless steel (190 GPa). The results indicated that the stress had not exceeded the loading limit and, therefore, the design was sturdy and the tray would not break.

Support plate bending

Figure 4 shows the free body diagram of the support plate to calculate bending stress. The stress acting on the support plate (P) was 598.17 Pa, the thickness (t) was 0.003 m and the radius (r) was 0.46 m. The assumptions in this analysis

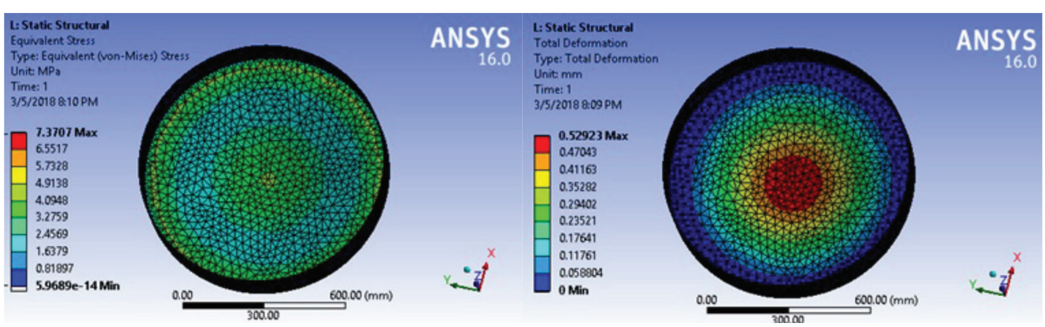

Figure 3: Stress and deflection simulation of tray

Table 2: Calculation and simulation results

Parameter	Manual	Simulation	Error (%)
Maximum Stress (MPa)	6.51	7.37	11.7
Deflection (mm)	0.52	0.53	1.8

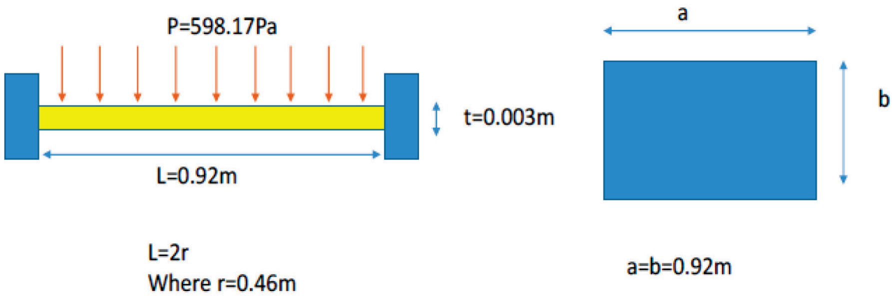

Figure 4: Free body diagram of the support plate

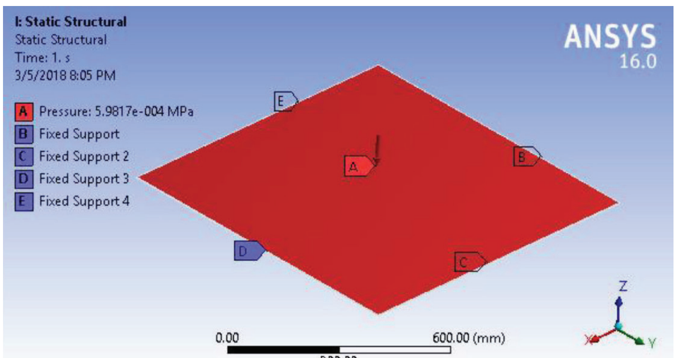

Figure 5: Simulation setting for the support plate

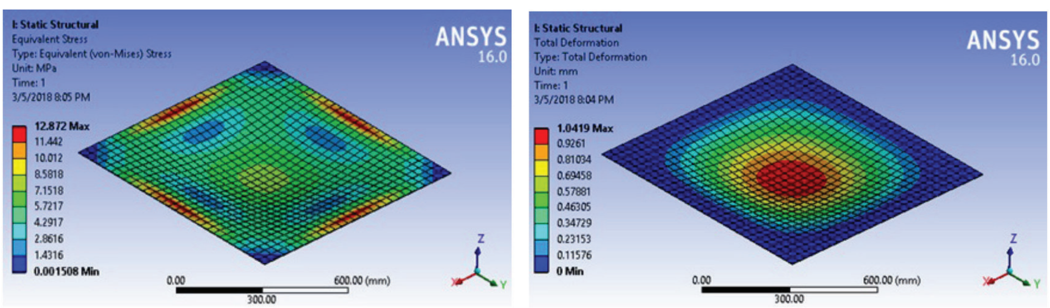

Figure 6: Bending analysis of the support plate

were the support plate was flat, the stress from the tray was uniformly distributed across the base area, the edge was clamped while load was uniformly distributed along the support plate surface and the maximum deflection was acting on the centre of the plate. Based on the calculation, the maximum stress (σ_m) acting on the tray was 17.33 MPa and the maximum deflection (y_m) was 1.06 mm.

Figure 5 shows the analysis setting of the support plate. Force was assumed to be uniformly distributed over the red area and the value acting on point A was set at 598.17 N, which was the total weight of the sago flour and tray. The support plate had four sides, namely B, C, D and E, which were fixed to the table legs and allowed no movement of the plate. The elastic modulus of mild steel was set at 207 GPa.

In Figure 6, the maximum stress acting on the support plate was 12.87 MPa, which was less than the ultimate tensile strength of Carbon Steel 1018 (341 MPa) (Budynas & Nisbett, 2015). The results indicated that the support plate would not fail in withstanding the load exerted from the sago and tray. The maximum deflection was 1.04 mm, which occurred at the middle of the support plate indicated by the red zone (Figure 6, right). Small deflections indicated that the bottom of the support plate did not need extra reinforcements because the bending was considered small and negligible.

Table 3 compares the calculation and simulation results of the support plate bending. There was an error of around 25.7% between the maximum stresses. This occurred because the manual calculation considered only one direction of stress acting on the plate surface (tensile stress). However, both results showed that the maximum stress acting on the plate was within the safety limit (341 MPa). Both deflection values obtained by the calculation and simulation were close to each other with only 1.6% error. This indicated that the simulation results were valid and acceptable.

Mixer vibration (Harmonic response)

The mixer vibration results are shown in Figures 7 and 8. The analysis could enhance the reliability of a system by preventing mechanical resonance and extreme vibration amplitudes that might damage the machine structure (Dive *et al*., 2017). There were three main inputs, namely the Young's Modulus, Poisson Ratio and Mass Density of the material. The vibration analysis was conducted to investigate the vibration response on the mixer.

Figure 7 shows the cylindrical support which was free to rotate at Z direction. The drag force acting on the mixer surface and a moment was in negative Z direction. The drag force was assumed to act only on one face of the mixer and the cylindrical rod movement was restricted by bearing.

The simulation results in Figure 8 (left) show a maximum deflection of 0.0035 mm, which was considered small and the mixer had low tendency to fail. The most critical point was at the edge of the mixer as the stress was concentrated there. The maximum stress on the mixer shown in Figure 8 (right) was 1.09 MPa, which was less than the ultimate tensile strength of Stainless Steel 304 (568 MPa). Therefore, the mixer could safely operate without breaking during the sago flour mixing operation.

Table 3: Analysis results for support bending

Parameter	Manual	Simulation	Error (%)
Maximum Stress (MPa)	17.33	12.87	25.7
Deflection (mm)	1.06	1.04	1.6

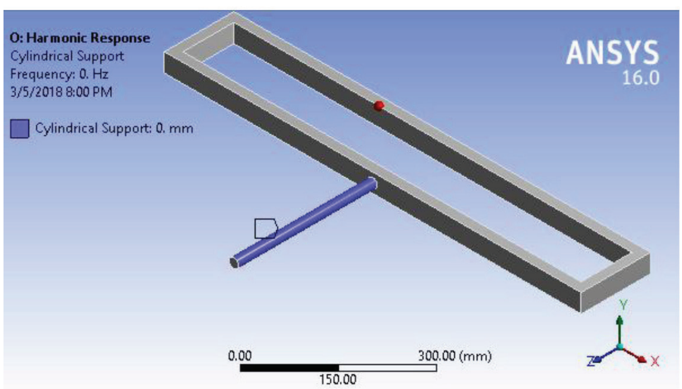

Figure 7: Cylindrical setting of the mixer

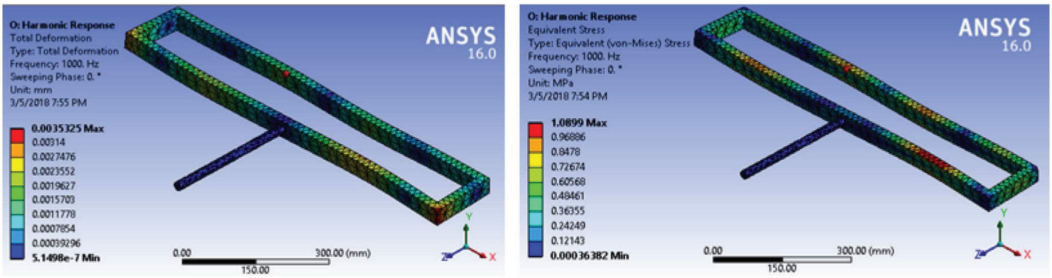

Figure 8: Vibration simulation results for mixer

Conclusion

In this paper, the conceptual design of a new sago drying machine was generated and determined. Structural analysis was conducted at the three critical parts; the tray, support plate and mixer. The mathematical modelling and calculation results showed that the selected material and design were acceptable and would not fail during the operation. The modelling and calculation were further enhanced using finite element analysis (FEA) simulation. The errors between mathematical calculation and FEA simulation was based on some assumptions on the model and limitation of dimension covered. It is hoped that the data of this research could contribute to the development of a safe and robust sago drying machine.

Acknowledgements

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