STRESS AND DEFORMATION ANALYSIS OF A COMPRESSION SPRING IN A ROTARY TYPE BRIQUETTE PRESS MACHINE USING FEA SIMULATION

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ABSTRACT
A comprehensive analysis of a helical compression spring subjected to static loading using two approaches: Finite Element Analysis (FEA) performed with ANSYS and manual calculations based on classical mechanical theory. The objective of this research is to compare and validate the accuracy of simulation results with manual computation, particularly in determining the force and deformation response of the spring.The spring model was first created using Computer-Aided Design (CAD) software and then imported into ANSYS in .STP format. Static structural analysis was selected as the simulation method, with meshing conducted using a body sizing of 1 mm to ensure sufficient detail while maintaining computational efficiency. The spring was modeled using appropriate material properties and subjected to axial forces simulating compression scenarios. Meanwhile, manual calculations were conducted using Hooke's Law. The comparison between both methods revealed a percentage difference of 4.78%**,** which falls within the generally accepted engineering tolerance of ±5%. This suggests a strong correlation between the FEA results and manual computations, thereby validating the use of FEA for

analyzing spring behavior under static loads. Furthermore, the study highlights the importance of mesh refinement and the impact of geometry setup on the accuracy of simulation outcomes.In conclusion, the findings confirm that ANSYS-based FEA can be effectively used for evaluating the mechanical performance of springs, provided that appropriate modeling, meshing, and material data are used. This approach not only enhances efficiency in analysis but also reduces potential errors associated with manual calculations, especially in complex designs or variable loading conditions.

Keywords: *ansys, compex designs, finite element analysis, hooke law, static structure.*

INTRODUCTION

Energy is recognized as a key component in the pursuit of sustainable development. Indonesia's current energy consumption heavily relies on fossil fuels, including oil, natural gas, and coal, which are non-renewable resources. These sources are finite and will eventually be exhausted over time. In contrast, renewable and environmentally friendly alternatives such as coconut shells, sawdust, and rice husks are abundantly available across various regions in Indonesia and offer promising potential for sustainable energy development. Current energy demands for daily and conventional use remain largely dependent on fossil fuels, whose reserves are gradually depleting. Fossil fuels originate from the decomposition of ancient biological matter plants and animals that lived millions of years ago.

To reduce the reliance on fossil fuels, it is essential to explore and promote alternative energy sources that are sustainable, widely available, and cost-effective. Biomass energy, such as briquettes produced from organic waste materials, represents one such alternative. As a renewable and environmentally friendly option, biomass holds strong potential to serve as a reliable and ideal source of future energy. To optimize the use of renewable resources, the development of a briquette production machine has been undertaken to enhance operational efficiency and increase production capacity. This innovation is expected to contribute significantly to improving both the volume and quality of briquette output. Despite its potential, the current market price of biobriquette poduction equipment remains relatively high, making it economically unfeasible for micro, small, and medium enterprises (MSMEs). In light of this challenge and the growing opportunity to utilize biomass as an alternative energy source.

Mechanical components are utilized in various industrial fields such as weighing systems, braking systems, vehicle suspensions, and engine valves to perform specific functions, including applying force, storing or dissipating energy, providing flexibility to mechanical systems, and maintaining consistent force or pressure. Helical springs, for example, are commonly used as the main elastic elements in many types of vibration dampers. The most widely used helical springs typically have a cylindrical shape with a three-dimensional spiral structure and a uniform curvature along their axis. However, these types of springs require a considerable amount of space both laterally and vertically. In specialized applications where space is limited in the lateral and/or vertical directions, conventional springs may become less effective, as using multiple springs can lead to an undesirable increase in stiffness. To address this issue, two types of specialized springs can be employed: first, springs with a non-circular shape to accommodate lateral space limitations, and second, circular helical springs with a non-prismatic profile to reduce vertical space requirements.

A coil spring is a type of elastic body designed to deform when subjected to a load and return to its original shape once the load is removed. It is made from an elastic material and formed into a helical coil. When exposed to compressive or tensile forces, the spring wire undergoes torsion, making the characteristics of the spring highly dependent on the material's shear modulus.

In addition to functioning as a compression or extension spring, a coil spring can also be used as a torsion spring, where the entire spring structure is twisted along its helical axis. In this condition, a bending moment arises that affects the increase or decrease of the helical radius, and the Young's modulus is required to determine the spring's response to such deformation.

This chapter focuses on experimental testing and static numerical analysis using the FEA method on the spring component, which plays a crucial role in the working system of a rotary-type briquette extruder machine. The spring is subjected to repeated compression loads during the continuous, high-pressure extrusion of biomass, affecting both the mechanical stability and compaction efficiency of the briquettes. Therefore, a simulation-based design approach is required to accurately determine the strength limits and deformation behavior of the spring. The objective of this research is to develop an optimized and structurally efficient spring component with a service life suitable for small and medium enterprises (SMEs), while also ensuring energy efficiency, ease of maintenance, and overall machine reliability.

A. FEA-Based Structural Evaluation of Compression Springs

The workflow of this research, illustrated in Figure 1, outlines the stages carried out using a FEA simulation framework. The study begins with an in-depth literature review sourced from relevant academic publications. A 3D model of the spring is then developed using CAD software and imported into a user-defined simulation environment. Subsequently, appropriate boundary conditions are applied, followed by mesh generation and loading conditions. The simulation yields output data including total deformation and von Mises stress, which are further analyzed to assess the structural performance and stress response of the spring component.

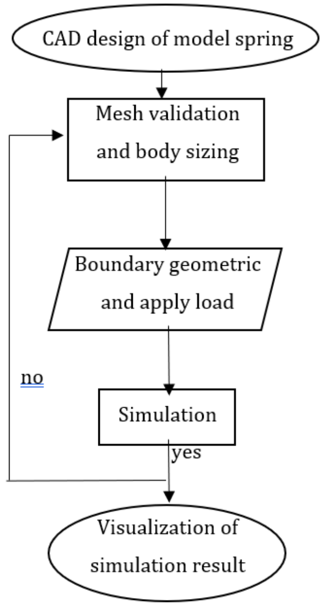


Figure 1. Process Stage of FEA Simulation

B. Mechanical Load Estimation and Analytical Force Derivation

Table 1. Boundary Condition

Boundary Condition	Value
Motor Power P	750 watt
Comparison Gearbox Reducer	1:60
Maximum Rotation Motor	1400 rpm
Gear 1 R2	20 tooth
Gear 2 R3	40 tooth
Rotary Diameter	D = 45 cm and r = 22,5 cm

The rpm generated after passing through the reducer gearbox are shown in Equation (1).

$$n_1 = \frac{1400}{60} = 23{,}3 \, . \frac{1}{s} \, rpm \qquad (1)$$

Comparison of driving gear and driven gear are shown in Equation (2).

$$\frac{n_2}{n_3} = \frac{R_2}{R_3} = \frac{20}{40} = \frac{1}{2} \qquad (2)$$

Calculated value the revolutions per second of the motor rotation based on the Equation (2) are shown in Equations (3), (4), and (5).

$$n_1 = 23{,}3 \,. \frac{1}{s} = 23{,}3 \,. \frac{1}{60} = 0.38\, rps\ (0.4\, rps) \quad (3)$$

$$n_2 = \frac{1}{2} \,. n_1 = \frac{1}{2} \,. 0.38\, rps = 0.19\, rps\ (0.2\, rps) \quad (4)$$

$$n_3 = \frac{1}{2} \,. n_2 = \frac{1}{2} \,. 0.19\, rps = 0.097\, rps\ (0.1\, rps) \quad (5)$$

Calculated value the force from the motor every 0.1 rps are shown in Equation (6).

$$T_3 = \frac{P}{n_3} = \frac{750\, watt}{0.1\, rps} = 7.500\, N.m \quad (6)$$

Calculated value of the total force on one full rotation of the rotary briquette making instrument are shown in Equation (7).

$$F = \frac{T_3}{r} = \frac{7.500\, N.m}{0{,}225\, m} = 33.333{,}33\, N \quad (7)$$

Use Hooke's law equation to determine the force required on an object, this law presents an empirical relationship between the magnitude of the applied force and the elongation of the spring. As the applied force increases, the spring's extension also increases proportionally. However, this proportionality no longer holds once the spring exceeds its elastic limit. The formula of F is shown in Equation (8).

$$F = k.x \quad (8)$$

Known

F = 33.333,33 N

For example, consider the force required to determine the displacement or deformation (either elongation or compression) from the equilibrium position, which is 5 mm or 0.005 m. So that the value of K is shown in Equation (9).

$$K = \frac{33.333{,}33\, N}{0.005\, M} = 6.666.600\, N/m \quad (9)$$

After determining the spring constant, calculate the force required for deformations of 10 mm, 15 mm, and 20 mm. The calculation value for that is shown in Equations (10), (11), adn (12).

The force required to compress the spring by 10 mm or 0.01 m

$$F = 6.666.600 \,. 0{,}01 = 66.666\, N \quad (10)$$

The force required to compress the spring by 15 mm or 0.015 m

$$F = 6.666.600 \,.\, 0{,}015 = 99.999\ N \qquad (11)$$

The force required to compress the spring by 20 mm or 0.02 m

$$F = 6.666.600 \,.\, 0{,}02 = 133.332\ N \qquad (12)$$

C. Helical Spring Modeling and Design Parameters

Helical springs are categorized into two types, namely helical compression springs and helical extension springs, based on the direction and characteristics of the force generated when the spring is deflected, In this design Autodesk Inventor software was used, utilizing the coil feature. The detailed size specifications are presented in Table 2, while the visual representation is shown in Figure 2.

Table 2. Specification of Coil Spring

Spesification	**Value**
Wire Diameter	2 mm
Pitch	4.9 mm
Revolution	9.5
Helical Free Heigh	51 mm
Helical Outer Diameter	16 mm

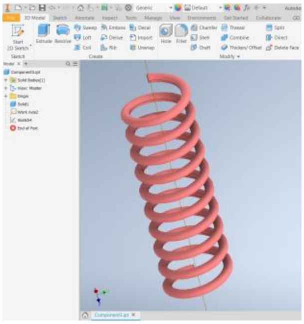


Figure 2. Spring Model

D. Static Stress Simulation and Deformation Analysis

The static analysis was carried out using ANSYS 2020 R2 software with a structural static system analysis. The design model was imported into ANSYS in STP format and then meshed. The mesh model was created with a limited number of nodes and elements to obtain accurate results. For mesh validation, a body sizing of 1 mm was used.

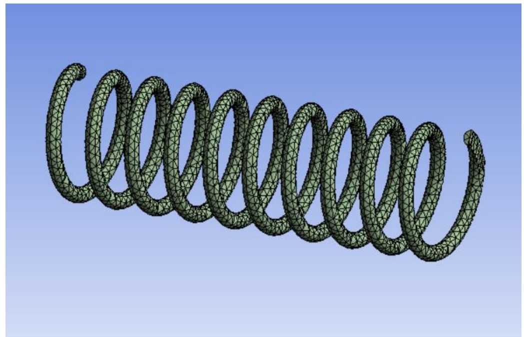
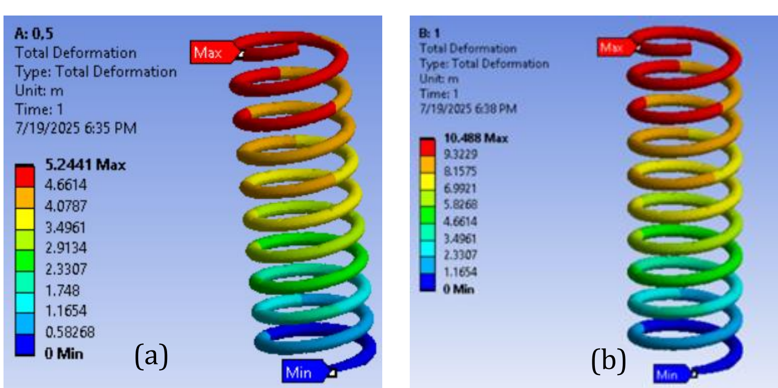


Figure 3. Meshing with Body Sizing 1 mm

After the meshing is obtained, then determine the fixed support on the spring and enter the force that has been calculated in Equation (7), (10), (11), and (12). After that, in the solution section, select total deformation to find out how much change is produced from the input force above and get some data that presented on Tabel 3 and Figure 4.

Table 3. Total Deformation

Load (N)	Total Deformation
33.333	5.2441
66.666	10.448
99.999	15.732
133.332	20.976



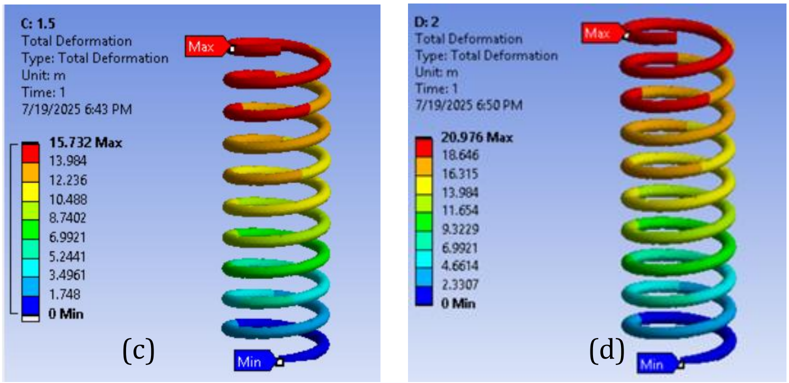


Figure 4. Total Deformation at (a) 33.333 N (b) 66.666 N (c) 99.999 N (d) 133.332 N

From the simulation results and manual force calculations, a percentage difference of 4.78% was obtained, which indicates that the calculations and simulation results are considered convergent or approaching a common point.. Some of the results indicate that the discrepancy between the FEA simulation and the experimental data is not significant and shows a good level of agreement, as the margin of error remains within the acceptable engineering tolerance of ±5%. However, in other results, the margin of error exceeds this tolerance. These differences are likely due to the varying levels of accuracy between the experimental and numerical methods. The percentage difference are shown on the Tabel 4.

Table 4. Percentage difference

Calculation Deformation	Total Deformation (mm)	Percentage Difference
5	5.2441	4.882
10	10.448	4.8
15	15.732	4.88
20	20.976	4.88

CONCLUSION

Based on the input force data and analysis testing on body sizing, it can be concluded that the greater the variation in body sizing, the more data can be obtained. As a result, the percentage difference between the manual calculation and the analysis results becomes smaller—from an initial average percentage of 4.78% to less than 4.78%. This reduction can also potentially be achieved by modifying the geometry using CAD software, by adjusting the coil settings or the spring diameter.

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