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Dynamic Force Analysis of Single Acting Compressor Based on Angular Velocity

Budi Hartono[1*], Edy Sutoyo[1], Setya Permana Sutisna[2], Hablinur Al-Kindi[1], and Rohman Suhada[1]
[1] Mechanical Engineering Department, Ibn Khaldun University, Bogor
[2] Agricultural Engineering Department, Jenderal Soedirman University, Purwokerto
[*]email : budi.hartono@uika-bogor.ac.id

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ABSTRACT– This paper presents a dynamic force analysis of a single-action piston compressor operating at various crankshaft angular velocities. The analysis was performed using Scilab and CADCAM software tools to evaluate the inertia forces acting on the crankshaft, connecting rod, and piston head. The study was conducted to investigate how variations in angular velocity significantly influence the magnitude of dynamic forces acting on compressor components, which is essential for optimizing design, performance, and durability of the compressor. Moreover, there is limited research that utilizes an integrated approach using CADCAM for modeling and Scilab for computation, offering a cost-effective and accurate alternative to experimental methods. At a crankshaft speed of 1500 rpm, the inertia force on the crankshaft was calculated as 0.03 N, increasing to 0.055 N at 2000 rpm and 0.066 N at 2200 rpm. For the connecting rod, the corresponding forces were 17.51 N, 31.12 N, and 37.66 N, respectively. At the piston head, the inertia force rose from 0.063 N at 1500 rpm to 0.11 N at 2000 rpm, and 0.14 N at 2200 rpm. The results indicate that variations in angular velocity significantly affect the magnitude of dynamic forces within the compressor components.

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Introduction

Advancements in science and technology have provide significant positive impacts and benefits in various aspects of human life through the discovery and development of more sophisticated and efficient equipment. Researchers and experts continue to conduct studies and experiments to create innovations that are both useful and practical. Technology plays a vital role in fulfilling human needs and simplifying industrial activities. One such important innovation is the compressor, a machine used to increase the pressure of fluids such as gases or air [1]. Compressors are widely applied in the industrial sector, and their performance is influenced by factors such as fluid flow rate, pressure, and temperature [2]. These variables are interrelated in the air compression process to achieve optimal performance [3].

With the rapid advancement oftechnology, the analysis of these factors can now be performed using computer-based programs, which not only increase the accuracy of the results but also reduce analysis costs compared to manual methods. Analytical techniques have been developed to understand the dynamic forces acting on single-action piston compressors [4]. Kinematic and dynamic analyses are employed in the compressor mechanism and implemented through computational methods to simplify calculations and improve efficiency [5].

Previous studies have explored method for analyzing slider-crank mechanisms, which share similarities with the mechanism used in piston compressors. These studies utilized Matlab software to analyze the dynamic response of the mechanism [6]. The use of Matlab offers both advantages and limitations. In Simulink-based model analysis, the lengths of the crank and rocker links can be modified by inputting data through an M-file, and the results are visualized in a figure window, allowing the motion of the slider-crank mechanism to be observed [7].

Moreover, most prior studies tend to treat crankshaft speed as a static input parameter, rather than exploring how variations in angular velocity influence the magnitude and distribution of dynamic forces throughout the compression cycle [8]. This omission represents a critical knowledge gap, especially as compressors are frequently subjected to non-uniform loading and fluctuating speeds in practical applications.

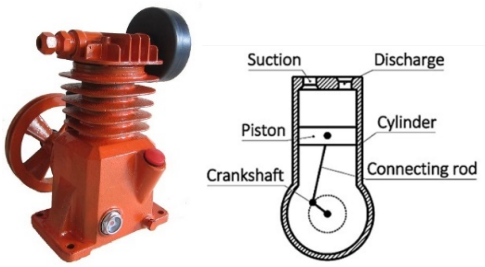
To address these limitations, the present study introduces an integrative methodology combining CADCAM-based three-dimensional modeling with Scilab-based computational analysis. CADCAM is employed to generate realistic 3D geometries of compressor components, enabling precise mass property calculations derived from material data and part dimensions. Scilab, an open-source computational tool, is

then used to perform the mathematical modeling and dynamic force analysis, offering a transparent and customizable alternative to proprietary platforms.

The primary objective of this study is to conduct a dynamic force analysis of a single-acting piston compressor by examining the effects of varying crankshaft angular velocities. Through this dual-software approach, the research aims to deliver a more accurate, cost-effective, and widely accessible analytical framework for evaluating compressor dynamics. This contribution is particularly relevant in academic and industrial settings where advanced simulations are needed but constrained by budgetary or software access limitations.

Material and Methods

This study employed an integrated computational approach that combined CAD-based geometric modeling with dynamic force analysis using Scilab. The methodology consisted of four main stages: data acquisition, 3D modeling with mass property extraction, dynamic analysis through computational scripts, and interpretation of force–torque–pressure characteristics under varying operating speeds.



(a) (b)

Figure 1. (a) Single-acting piston compressor head (b) Kinematic diagram and component single acting compressor

The physical dimensions of each compressor component—namely the crankshaft, connecting rod, and piston—were measured using precision instruments (digital calipers with ±0.01 mm accuracy) and verified against manufacturer datasheets for consistency. The primary focus of this study is the head of a single-acting piston compressor, which constitutes the key element responsible for air compression during the piston's working stroke [9].

Table 1. Compressor specifications

Brand	Matrix YAH1051
Compressor Type	Single Action
Compressor Power	¼ HP
Working Pressure	6 Bar
Compressor Speed	1400 rpm
Piston Stroke	38 mm

The structural configuration of the compressor head is illustrated in Figure 1, while Tables 1–3 present the compressor specifications, material types, and dimensions employed in the modeling process.

A personal computer equipped with CADCAM and Scilab software was used for computational analysis. Each component was modeled in SolidWorks to ensure accurate replication of geometry and mass distribution. The Mass Properties tool was then employed to compute the moment of inertia (I) about the principal axes and the center of mass (CoM) position relative to the rotational axis. These values were essential inputs for the subsequent dynamic force and torque calculations.

Table 2. Material Type of Components

No.	Components	Material Type
1	Crankshaft	Steel
2	Connecting Rod	Aluminium
3	Piston	Aluminium

Meanwhile, the dynamic torque and force analysis of the piston was performed using Scilab software. The analysis was based on the kinematic equations of a slider–crank mechanism, adapted to the configuration of a single-acting compressor. Custom Scilab scripts were developed to calculate the angular velocity (ω) and angular acceleration (α) for specified rotational speeds of 1500, 2000, and 2200 rpm, as well as the inertia force (F_i) using Eq. (1) [10].

$$F_i = mr\omega^2 F_i \quad (1)$$

Torque variation over a single crankshaft revolution was analyzed by accounting for geometric constraints. Simulations were conducted under two operating conditions: (i) the unloaded condition, with no cylinder pressure applied to represent idle operation, and (ii) the loaded condition, incorporating a 6-bar pressure effect to simulate real compression. Pressure fluctuations were computed as a function of crank angle and piston displacement, based on thermodynamic relationships between volume and pressure during the compression stroke. This approach enabled correlation between speed variation and peak internal pressures.

Table 3. Dimentions of Components

No.	Components	Ø Outer (mm)	Ø Inner (mm)	Length (mm)	Width (mm)
1	Crankshaft	33	-	26	10
2	Connecting Rod	32	22	102	22
3	Piston	51	44	50	2.5

Results and Discussion

The dynamic force analysis of the bare single-acting compressor included the evaluation of the moment of inertia and inertia forces of each component, as well as the calculation of drive torque under both unloaded and loaded (pressurized) conditions. The analysis was conducted at crankshaft angular velocities of 1500, 2000, and 2200 rpm. The results demonstrate a strong correlation between crankshaft angular velocity and the magnitude of inertia forces, drive torque, and cylinder pressure in the single-acting compressor. These findings highlight the critical influence of rotational speed as a dominant factor shaping the dynamic load profile of the compressor mechanism [11].

Moment of Inertia Analysis

To determine the moment of inertia for each component, the mass properties feature in the CADCAM software was used. By assigning the appropriate material and dimension specifications in the 3D model, the software was able to generate the moment of inertia values for each component.

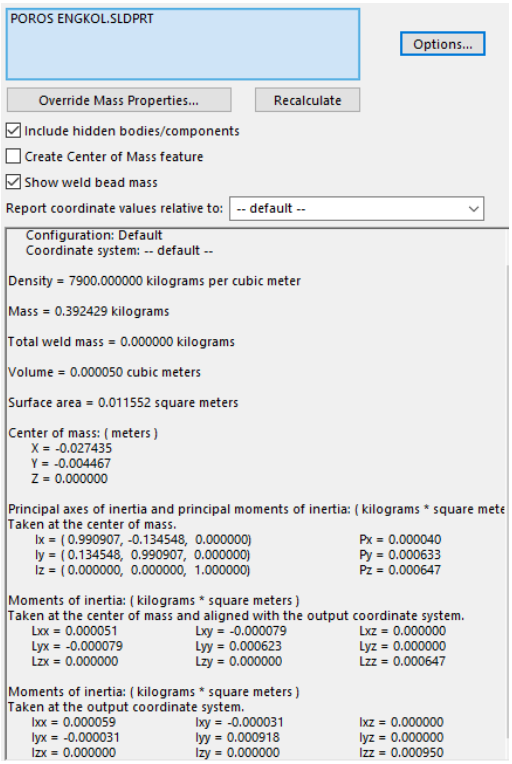


Figure 2. Mass properties of the crankshaft

As shown in Figure 2, the moment of inertia about the rotational axis (Z-axis) is 950 × 10^{-6} kg·m².

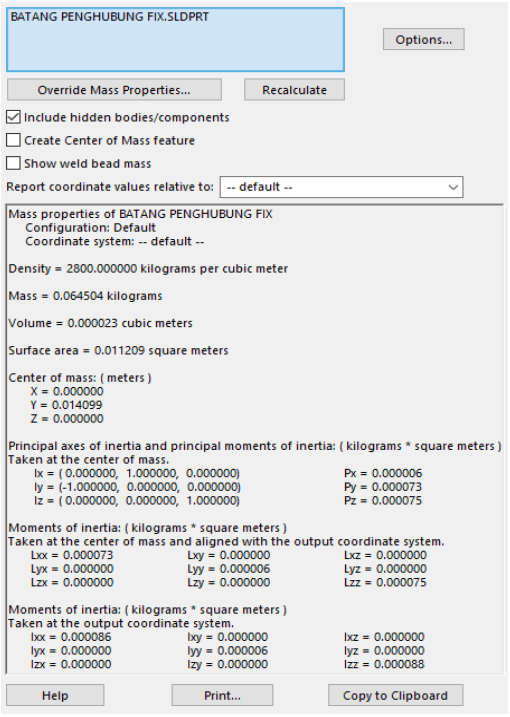


Figure 3. Mass properties of the connecting rod

Figure 3 shows that the moment of inertia about the Y-axis is 6 × 10^{-6} kg·m².

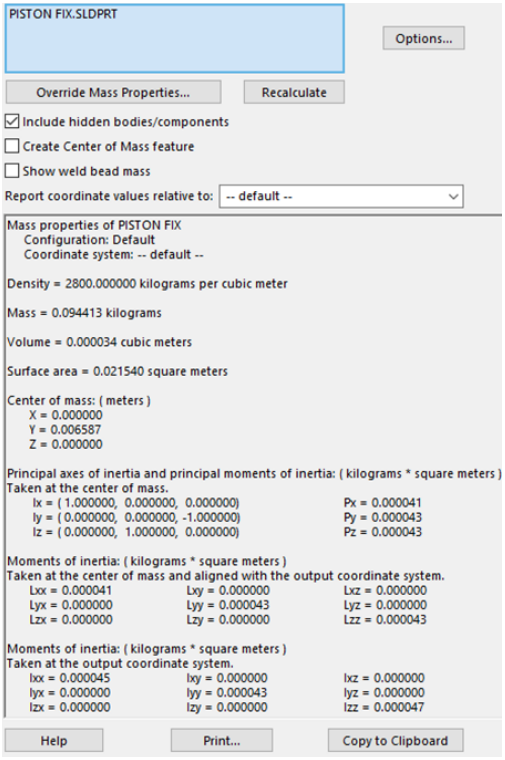


Figure 4. Mass properties of the piston

According to Figure 4, the moment of inertia for the piston component is 43 × 10^{-6} kg·m².

Inertia Force Analysis

The calculated inertia forces exhibited a quadratic relationship with rotational speed, consistent with fundamental dynamic principles. Scilab software was employed, using prewritten scripts, to perform the calculations. The results and detailed computations for each component are presented below.

Crankshaft.

The inertia force increases proportionally with angular velocity and is determined by the mass, the radius from the rotation center, and the angular acceleration of the component. Using the Scilab script, the inertia force was calculated. At 1500 rpm, the inertia force was 0.03 N; at 2000 rpm, it increased to 0.055 N; and at 2200 rpm, it reached 0.066 N.

```
m = 0.392428; // mass (kg)
r = 0.0185;   // radius (m)
RPM = [1500 2000 2200];
omega = 2 * %pi * RPM / 60;
Ti = m * r^2 * omega.^2 / 2;
Fi = Ti * r;
disp([RPM' Ti' Fi'])
```

Connecting Rod.

The inertia force depends on both the moment of inertia and the angular acceleration. Assuming a constant moment of

inertia, the force increases linearly with angular acceleration. Using the Scilab script, the inertia force was calculated. At 1500 rpm, the inertia force was 17.51 N; at 2000 rpm, it increased to 31.12 N; and at 2200 rpm, it reached 37.66 N.

```
m = 0.064505;
r = 0.011;
n = [1500, 2000, 2200];
omega = 2 * %pi * n / 60;
a = omega.^2 * r;
F_i = m * a;
```

Piston

The results indicate that higher crankshaft speeds require greater force to overcome piston inertia. Using the Scilab script, the inertia force was calculated. At 1500 rpm, the inertia force was 0.063 N; at 2000 rpm, it increased to 0.110 N; and at 2200 rpm, it reached 0.140 N.

```
m = 0.000043;
r = 0.0255;
n = [1500 2000 2200];
w = 2*%pi*n;
a = w.^2*r;
F = m*a.^2*r;
```

This observation aligns with prior studies, such as Takayama and Arase [12], which emphasize that reciprocating elements with intermediate mass and significant oscillatory motion contribute substantially to overall dynamic stress. The increase in inertia forces at higher speeds poses significant implications for fatigue life, lubrication requirements, and vibration control strategies, suggesting the necessity for robust material selection and component balancing during design.

Torque Analysis Without Load Pressure

Similarly, the torque analysis under both unloaded and pressurized conditions revealed significant fluctuations throughout the crankshaft cycle. As illustrated in Figure 5, the graph presents the driving torque pattern during crankshaft rotation.

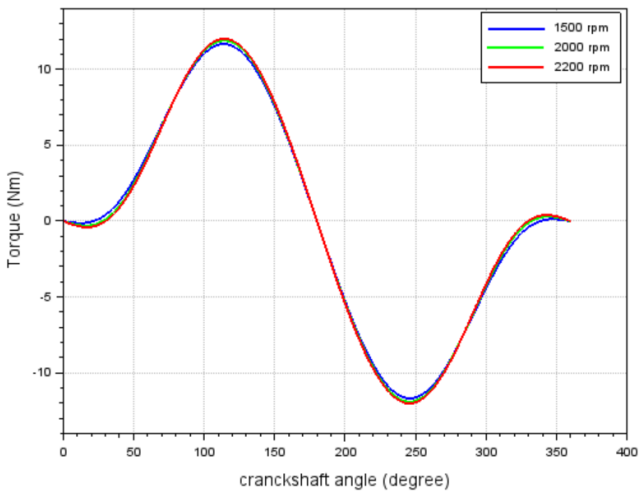


Figure 5. Changes in driving torque without compressive force as a function of angular displacement

The torque cycle represents a repeating variation within one revolution (360°), where the torque periodically rises to a maximum, decreases to a minimum, and returns to the maximum in each cycle. The graph presents three plot lines, each representing the driving torque at different engine speeds: 1500 rpm (blue), 2000 rpm (green), and 2200 rpm (red). Although the torque cycle pattern remains essentially consistent across speeds, the torque magnitude varies according to the speed parameter. Torque reaches its maximum at specific positions, such as 120° of crankshaft rotation, where the crank and connecting rod alignment produces the highest torque on the piston. Conversely, minimum torque occurs at positions such as 0° and 360°, where the crank–rod configuration is less favorable for torque generation.

Torque Analysis Under Load Pressure

In Figure 6, the drive torque ranges from 20–50 Nm at 0° crankshaft position, increasing to a peak of 45–75 Nm at 80°. Under pressurized conditions, the torque distribution exhibits significant variation, with the highest mechanical demand occurring mid-cycle. The graph presents three plot lines representing the driving torque at engine speeds of 1500 rpm (blue), 2000 rpm (green), and 2200 rpm (red). While the torque cycle pattern remains consistent across speeds, the torque magnitude scales with increasing speed. In axial piston pumps, torque fluctuations along the YY-axis typically reach about 8% of the total torque, with double peaks per cycle arising from piston–port plate pressure distributions.

This observation is consistent with the findings of Bergada et al. [13], who identified torque irregularities as a major factor influencing mechanical efficiency in reciprocating mechanisms. Furthermore, structural deformations and clearances—factors not considered in the present model—may introduce additional deviations, as noted by Chalhoub et al. [14]. Incorporating elastic deformation modeling or multi-body dynamic simulations could therefore enhance predictive accuracy under real-world operating conditions.

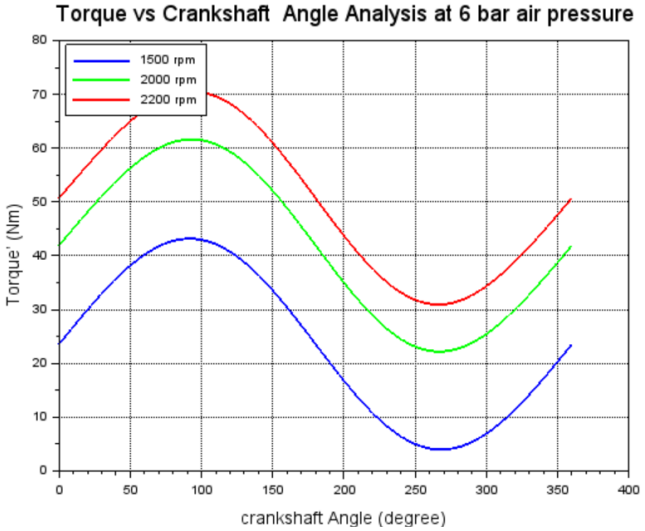


Figure 6. Changes in driving torque with compressive force as a function of angular displacement

Absolute Pressure in the Cylinder

The pressure variation inside the cylinder, as shown in Figure 7, indicates that higher crankshaft speeds lead to increased peak pressures and reduced compression duration. The blue, green, and red lines represent 1500 rpm, 2000 rpm, and 2200 rpm, respectively. The pressure profile changes dynamically with piston position and rotational speed, reflecting the effects of speed on the compressive force required during operation. While this can enhance the rate of compression, it also introduces thermodynamic inefficiencies and higher mechanical loading, corroborating the observations by Mutlu and Kilic [15]. These dynamic interactions between speed, pressure, and force magnitudes suggest that optimal compressor performance cannot be achieved solely by increasing operating speed; rather, a balanced approach is required, considering energy efficiency, mechanical reliability, and thermal stability.

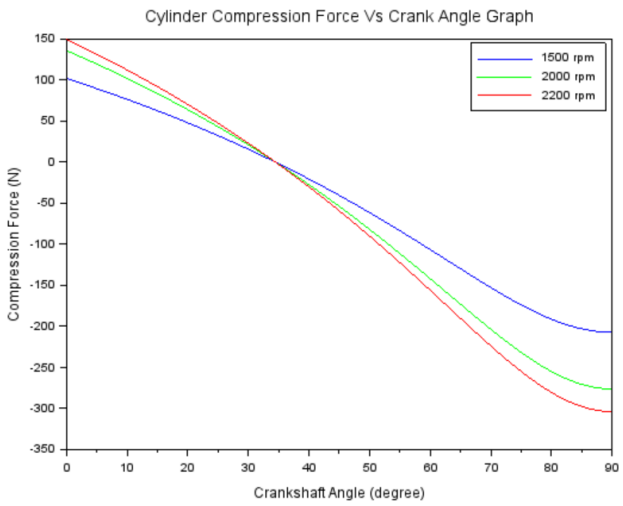


Figure 7. Absolute pressure fluctuations inside the cylinder

Conclusion

This study has successfully conducted a comprehensive dynamic force analysis of a single-acting piston compressor under varying crankshaft angular velocities by employing a dual-software approach utilizing CADCAM for 3D modeling and Scilab for computational analysis. The methodology proved effective in generating precise values for moment of inertia and inertia forces of critical compressor components—namely, the crankshaft, connecting rod, and piston—while also providing insights into the corresponding drive torque and internal cylinder pressure profiles under both unloaded and loaded conditions.

The results clearly demonstrate that angular velocity is a dominant factor influencing the magnitude of dynamic forces within the system. As crankshaft speed increases, so do the inertia forces, torque requirements, and internal pressures—indicating greater mechanical stress on components. This highlights the critical importance of dynamic analysis in the design and operational optimization of piston compressors, particularly when operating under varying speed conditions.

The proposed method enhances both the analytical accuracy and applicability of compressor design evaluation. It lays a foundation for future research in the direction of more advanced simulations, integration with thermodynamic modeling, and experimental validation, thereby supporting more resilient and efficient mechanical system designs.

Limitations and Future Work

This study was limited to the use of CADCAM and Scilab for modeling and analysis under ideal assumptions. Future work should involve experimental validation using sensor-based data acquisition and simulation of thermodynamic effects such as heat transfer and fluid dynamics. Additionally, advanced tools like MATLAB/Simulink or ANSYS can be employed to compare and extend the analysis.

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