

Article

Technical Feasibility of a Two-Cylinder Entablature Steam Engine with a Parallel Motion Crosshead: An Analysis from Mechanical Engineering

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Abstract: In this research, we present the results of analyzing the technical feasibility of an old invention by Henry Muncaster from the perspective of mechanical engineering, specifically focusing on the resistance of materials. The invention is a two-cylinder steam engine with a parallel motion crosshead, for which plans were published in the *Model Engineer* magazine in 1957. This complex device, composed of 76 elements and lacking descriptive information, has been the subject of a recent article that illustrated its design through the engineering drawing discipline and a 3D CAD model. To provide reliable information and conduct a comprehensive study of its technical feasibility, an extensive linear static analysis was performed. This analysis considered two critical positions of the piston inside the cylinder: upper dead center and lower dead center. We determined the optimal range of working pressures necessary to achieve a safety factor within the optimal design range of two to four. The results include von Mises stresses, displacements, and safety factor distributions, confirming that the optimal working pressure range for steam intake is between 1.885 and 3.550 MPa. This ensures that the safety factor values remain between 2.01 and 3.78.

Keywords: Henry Muncaster; steam engine; two-cylinder entablature; parallel motion crosshead; computer-aided engineering; finite-element method



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1. Introduction

The study and analysis of ancient and earlier inventions fall under the domain of industrial archeology [1], which examines material culture and the memory of labor. Specifically, technical historical heritage, encompassing old machinery or devices across various productive sectors, is being extensively scrutinized [2,3]. However, comprehensive records detailing the operation of these machines or devices are often lacking, necessitating a thorough examination based on historical information, plans or drawings, models, and any related data [4].

Furthermore, within the realm of industrial archeology, understood as a discipline grounded in material culture and the memory of labor, technical heritage is frequently analyzed. This analysis particularly focuses on ancient and earlier inventions (machines or devices) that were pivotal in the technological development of certain productive sectors.

This trend is supported by the numerous publications and international conferences dedicated to the design of machines and mechanisms. Notable among these conferences are the International Symposiums on Science of Mechanisms and Machines, on the Education in Mechanism and Machine Science, and on the History of Machines and Mechanisms. The topics covered range from the educational approach, which involves working with catalogs of mechanisms [5], or the science of machines and the history of mechanisms [6], to the practical application perspective, showcasing the developments of a country [7,8], or analyzing the design of mechanisms at the end of the 19th century [9]. Additionally, the

social perspective aims to benefit the general public by developing real or virtual models in science or technology museums [10].

This research focuses on one such example of technical historical heritage: the inventions related to steam engines, which were widely used for locomotion and integrated into ships, trains, and motor vehicles [11]. In particular, it analyzes a stationary steam engine designed by the esteemed English engineer Henry Muncaster [12], who authored numerous publications on the application of such engines in various industrial sectors, including steel, coal, and mining [13].

Steam as a source of energy has been extensively studied from various perspectives including fluid mechanics [14,15], thermodynamics [16], and innovations in piston design [17].

The steam engine has been the subject of numerous studies throughout history. To provide the reader with a better context, a brief historical overview is presented, highlighting the main machines and devices powered by steam.

Initially, the first thermal machine was discovered by the engineer and mathematician Heron of Alexandria in the 1st century AD, known as Eolipile. Later, around 1550, a Muslim scientist, Taqi al-Din Muhammad Ibn Ma'ruf, described the design and functioning of a theoretical steam turbine. In 1629, the Italian architect Giovanni Branca presented a conceptual design called "Le Machine," which was essentially a steam turbine consisting of a paddle wheel driven by a jet of steam [18].

However, it was not until the late 17th century that the Englishman Thomas Savery developed a useful steam engine for pumping water from mines and wells, essentially a water pump. In 1712, the English inventor Thomas Newcomen introduced an improved version of Thomas Savery's steam engine, which for the first time featured a piston inside a cylinder connected to a beam, with the other end of the beam linked to a water pump [18]. This steam engine was also used by another famous Spanish engineer, Agustín de Betancourt, in 1789 when he designed the double-acting steam engine, which has been analyzed from the perspectives of engineering graphics [19] and mechanical engineering [20].

Around 1765, James Watt enhanced the efficiency of Newcomen's steam engine by incorporating a condensing chamber to prevent significant steam loss within the cylinder. He later added improvements such as automatic control valves and the centrifugal governor, which regulated the amount of steam admitted. These advancements made Watt one of the pioneers of the Industrial Revolution. Subsequently, new improvements were introduced, including double-acting pistons and rotary motion [21].

In the early 19th century, these steam engines were integrated into commercial ships for cargo transport, such as the one designed by the American Robert Fulton in 1807, and steam-powered railways. They were also incorporated into vehicles, such as Richard Trevithick's 1803 design of a mail coach called "The London Steam Carriage" [22].

Finally, the steel industry also benefited from steam engines, with for example, the development of the renowned Creusot hammer in France, built in 1877, which allowed for molding tons of iron. By the early 20th century, steam engines were gradually replaced by other energy sources such as fossil fuels and nuclear power [18].

However, the introduction of the piston inside the cylinder marked a technological advancement that has been widely studied, and many steam engine inventions have focused on varying geometries, the number of cylinders, and configurations. Examples include single oscillation [23], double oscillation [24], simple slide valves [25], horizontal stationary engines [26], vertical engines [27], simple and compound engines [28], speed and gear control [29], and framed designs for increased rigidity and stability [30].

The invention analyzed in this paper is a two-cylinder entablature steam engine with a parallel motion crosshead. This stationary engine, featuring double-acting cylinders, operates independently of time and has an entablature structure with a central plane of symmetry and a speed governor.

There was no descriptive information available on the operation or conditions (such as working steam pressures) of the invention, except for plans published in the 1957 magazine

Model Engineer which were reproduced in 2017 [31]. Therefore, the initial step was to conduct research based on the engineering drawing discipline to study its design and plausibly explain its operation, providing the reader with detailed assembly information and component functionality, ultimately yielding an accurate 3D CAD model [32].

Despite its significance in the development of steam engine technology, there is no global research on this particular steam engine from a mechanical engineering perspective, underscoring its originality, novelty, and scientific interest.

It is also necessary to emphasize that there is no actual model of the analyzed invention from which experimental data can be obtained to verify the results of the simulation performed on the numerical model. Therefore, the simulation conducted is the only method available to understand the operating conditions of the invention and to ensure its safe behavior from the perspective of material strength.

Although this research does not constitute a contribution to current science, it is nonetheless an important contribution in characterizing the technical feasibility of a historically significant invention from the standpoint of mechanical engineering, which is the primary focus of this investigation.

Thus, the primary goal of this study is to evaluate its technical feasibility by performing a linear static analysis [33] using the finite element method (FEM) [34] to determine the optimal working pressure range, ensuring safe operational behavior based on material strength criteria.

The article is structured as follows: Section 2 outlines the initial materials used and the research methodology, including the operation of the steam engine. Section 3 presents the results and a discussion thereof, and Section 4 offers the main conclusions and future developments.

2. Material and Methods

The starting material for this investigation was the 3D CAD model obtained using Autodesk Inventor Professional 2024 software [35], which was utilized to conduct the linear static analysis.

Although the detailed operation of the invention has been thoroughly described in a recent publication [32], a brief overview of its functioning will be provided to give the reader a better understanding of the invention. Additionally, each of the preliminary stages before executing the linear static analysis will be detailed. This analysis examines von Mises stress, displacement, and the safety factor, as these variables mechanically characterize the behavior of the assembly, ensuring that its operation under real conditions is completely safe.

2.1. Operation of the Machine

Figure 1 presents both front and rear views of the final assembly of the invention, highlighting the most notable elements.

This steam engine consists of two double-acting cylinders, meaning that steam enters and exits on both sides of the piston. While steam admission occurs on one side of the piston, exhaust occurs on the opposite side. This process is facilitated by the slide valves located within the valve chests (Figure 2). These slide valves operate through their connection to the crankshaft via the eccentric strap and eccentric sheave.

Moreover, the presence of two cylinders allows for simultaneous steam admission in one cylinder and exhaust in the other. The advantage of having two cylinders instead of one is that steam expansion occurs more smoothly, preventing abrupt movements that could cause vibrations.

Additionally, it is important to note that the parallel motion mechanism, with the crosshead being the most prominent component, converts the reciprocating motion of the pistons into the rotational motion of the crankshaft. This mechanism also ensures that the pistons' movements are opposed and symmetrical.

Figure 3 provides cross-sectional views, both axonometric and frontal, of the invention, allowing for a clearer observation of the components that constitute the engine.

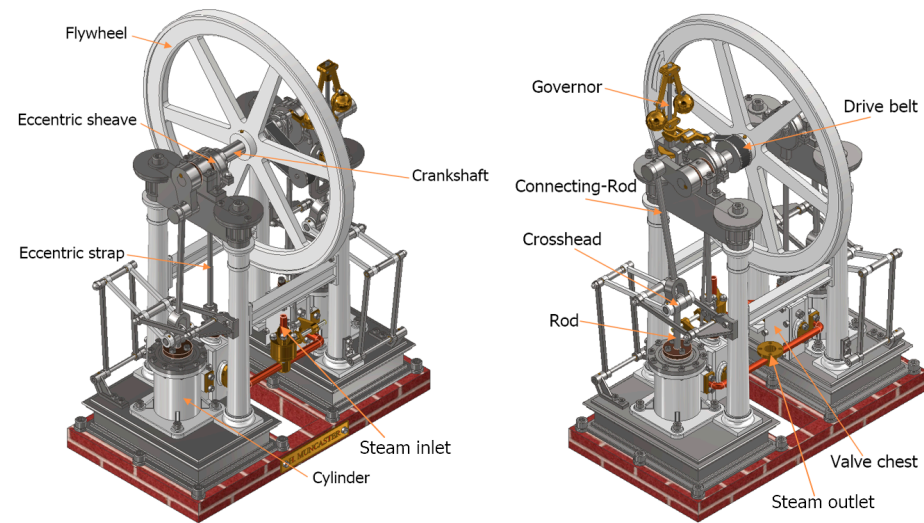


Figure 1. Assembly of the steam engine: front view (left) and rear view (right).

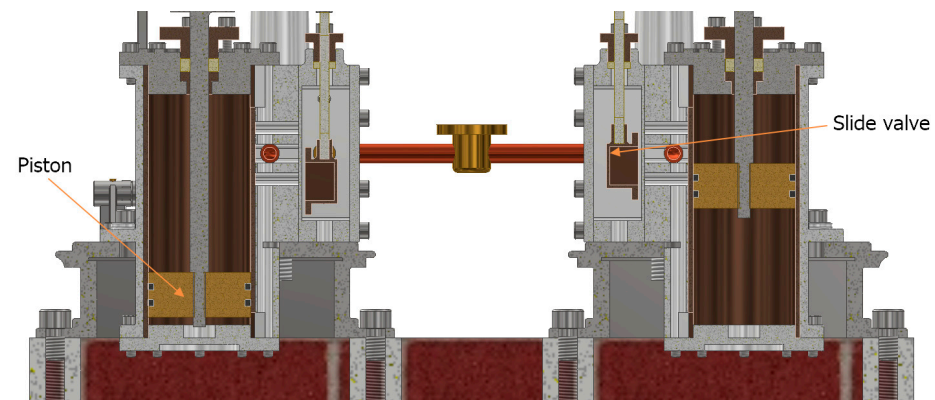


Figure 2. Cross section showing the piston and the slide valve.

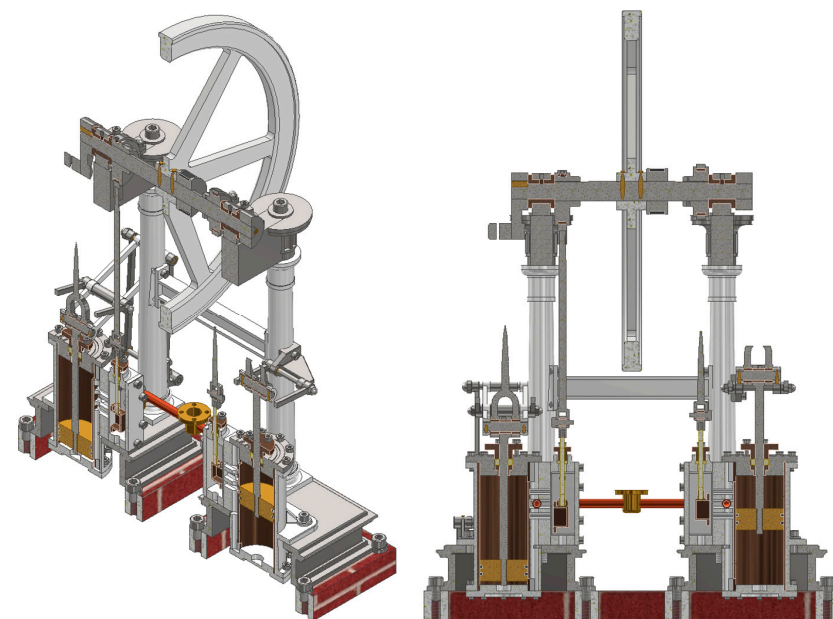


Figure 3. Sectioned views of the assembly: axonometric view (left) and front view (right).

Regarding steam admission, the system features a speed control system (Figure 4) that allows for the regulation of steam quantity according to the required needs. This regulation is made possible by the presence of a circular-section steam supply control valve (throttle valve), which rotates (altering its inlet section) due to its connection with the steam supply control rod, which is assembled to a governor control yoke. The ultimate goal of this control system is to reduce or increase the crankshaft speed via the drive belt and its connection with all other parts of the system.

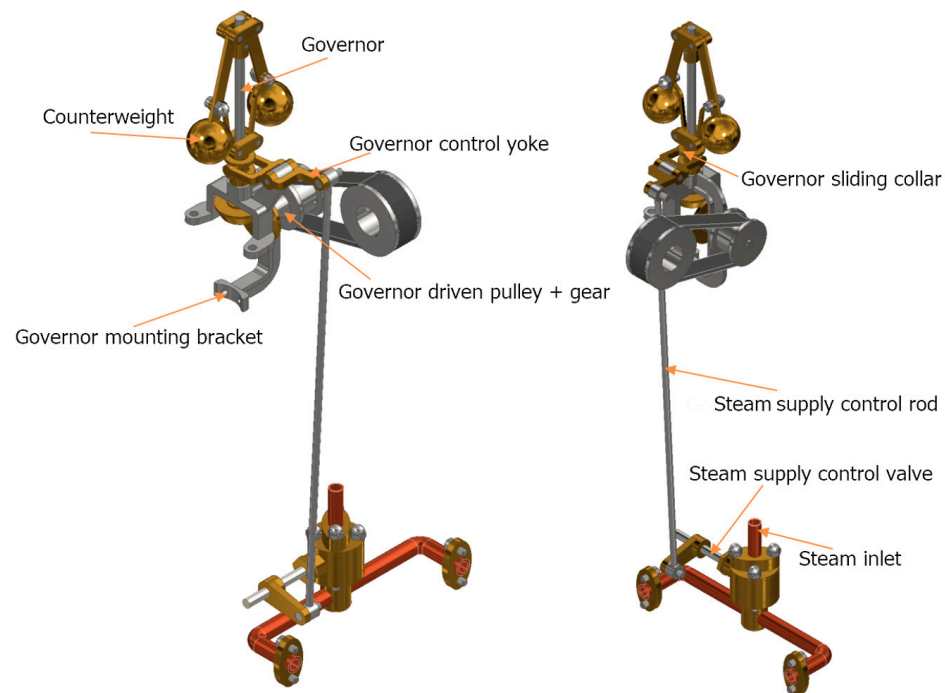


Figure 4. Speed control system with all its elements.

2.2. Analysis from a Mechanical Engineering Standpoint

The subsequent sections outline the comprehensive stages involved in performing a linear static analysis using FEM, alongside the associated working assumptions. These stages include the following:

- **Preprocessing:** This stage entails the meticulous preparation of input data.
- **Assignment of materials:** In this step, the appropriate materials are assigned to each component of the engine. This includes specifying the mechanical properties such as Young's modulus, Poisson's ratio, and density, ensuring that the simulation accurately reflects the real-world behavior of the materials under various load conditions.
- **Application of contacts:** When interactions between different parts of the model are present, it is essential to define the type and nature of contacts. This includes specifying whether the contacts are frictional, bonded, or sliding, which significantly influences the accuracy of the stress and strain results.
- **Boundary conditions:** This stage involves the establishment of boundary conditions and applied loads. Constraints such as fixed supports, symmetry conditions, and applied forces or moments are defined to mimic the real operational environment of the model. Proper application of these conditions is crucial for realistic simulation results.
- **Discretization (meshing):** The model geometry is divided into finite elements, which facilitates numerical analysis. This process, known as meshing, involves creating a mesh of elements and nodes to approximate the geometry of the model. The quality of the mesh, including element size and type, affects the accuracy and convergence of the analysis.
- **Identification of critical positions, determination of the deformation envelope and execution of the modal analysis and linear static analysis:** The simulation of the

model's behavior under various loading conditions is performed in this stage. It includes identifying critical stress and strain locations, determining the envelope of deformations, and conducting modal analysis to evaluate the natural frequencies of vibration of the system. Linear static analysis is then executed to assess the response of the model under static loads, providing insights into potential areas of failure and overall structural integrity.

By adhering to these stages, a thorough and precise linear static analysis using FEM can be achieved, leading to reliable predictions of structural performance and critical insights for design optimization.

2.2.1. Preprocessing

A comprehensive simulation of the entire steam engine could be conducted; however, the computational cost and the time required for such a simulation would be considerably high due to the large number of components involved. For this reason, elements that would not significantly influence the results have been excluded from the analysis, as their operating conditions are not critical.

In this investigation, fastening elements have been excluded because they typically require a very fine mesh and because there are numerous such elements in the assembly. Therefore, some of these elements (mainly cap screws and nuts) are replaced by a 'bonded' contact relationship, effectively simulating a welded joint. It is important to note that a welded joint is more rigid than a bolted connection, so if critical stress values are detected in an area where a fastening element or contact zone should be, those initially excluded elements must be reconsidered. Additionally, the speed governor and its associated elements, such as the drive belt and steam supply control system, are excluded from the simulation because their operation is dynamic, not linear static. The critical startup and shutdown positions will be studied, where the speed governor has no angular velocity.

Figure 5 shows the simplified assembly that will support the linear static analysis.

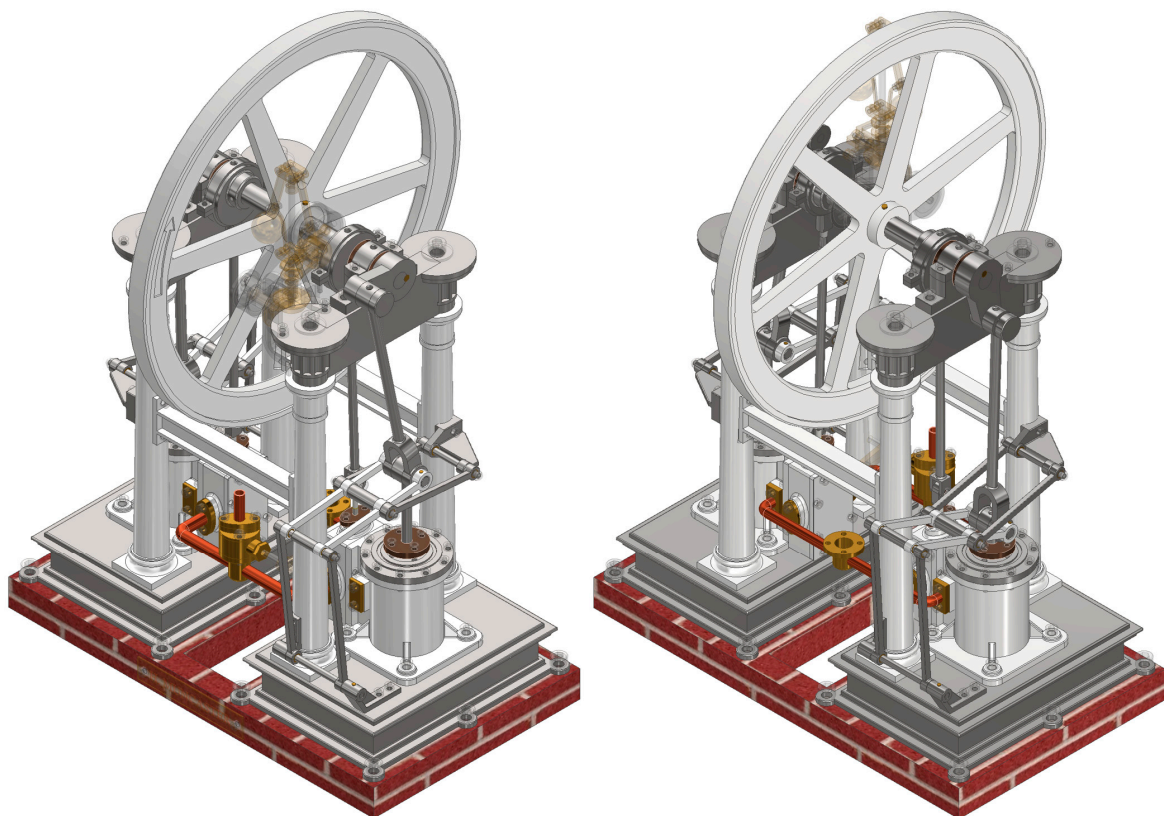


Figure 5. Isometric view of the simplified assembly: front view (left) and rear view (right).

2.2.2. Assignment of Materials

At this stage, materials are assigned to each element of the 3D CAD model to endow the solid with the mechanical properties corresponding to each material. This step is crucial as the material directly influences the behavior of the elements. For this investigation, materials from the Autodesk Inventor Professional 2024 material library were assigned, with the properties of the brick material being modified.

This modification is necessary due to the nonlinear behavior of brick, as it is typically a brittle and anisotropic material, meaning its mechanical properties can vary depending on the load direction. Additionally, bricks may not follow a linear stress–strain relationship, particularly as they approach their yield strength. This contradicts the assumption of linearity in a linear static analysis, which presumes materials behave in a linearly elastic manner. Specifically, the properties of mild steel were assigned, simulating the bedplate as being made of this material.

Table 1 lists the materials and their main mechanical properties used in the analysis.

Table 1. Properties of each material used in the analysis.

Material	Young's Modulus (MPa)	Poisson Coefficient	Density (kg/m ³)	Yield Strength (MPa)	Shear Modulus (MPa)
Aluminum 6061	68,900	0.330	2700	275.00	26,000
Brass	109,600	0.331	8470	103.40	37,000
Cast Bronze	109,600	0.335	8870	128.00	37,000
Cast Iron	120,500	0.300	7150	758.00	58,000
Mild Steel	220,000	0.280	7850	207.00	82,000
Nylon	2930.00	0.350	1130	82.75	1000.00
Stainless Steel	193,000	0.300	8000	350.00	80,000

2.2.3. Application of Contacts

To conduct an appropriate simulation using FEM, it is crucial to identify the contacts between the components of the assembly. Autodesk Inventor Professional 2024 provides a feature for automatically establishing these contacts, which is particularly useful for assemblies with numerous components, as manually defining the contacts would be highly labor-intensive and time-consuming. The following is a brief description of the types of contacts that the software allows:

- **Bonded Contact:** Simulates a welded joint, ensuring no relative motion between the connected surfaces.
- **Separation/No-Slide Contact:** Allows the surfaces to separate but not slide relative to each other.
- **Separation/Sliding Contact:** Permits both separation and sliding between the surfaces.
- **Frictional Contact:** Includes frictional effects between the surfaces, allowing for both sliding and separation, with resistance based on the friction coefficient.
- **Weld Contact:** Similar to bonded contact but specifically simulates welds with the appropriate material properties.

Furthermore, each type of contact can be configured as either symmetric or asymmetric. In symmetric contact, the nodes of one mesh cannot penetrate the nodes of the adjacent mesh, whereas in asymmetric contact, such penetration is possible. The symmetric contact was chosen to ensure that the meshing is performed independently on each element, enhancing precision.

A drawback of the automatic generation of contacts is that the software only creates them as bonded contacts, which may not accurately reflect the real behavior of the model, given that different types of contacts exist within the assembly. To address this issue, all contacts were initially entered automatically as the most numerous type (bonded contacts). Subsequently, the incorrect contacts were manually changed to the correct contact types.

2.2.4. Boundary Conditions

Constraints have been applied to limit the movement of the six degrees of freedom on the bottom face of the brick base and the flywheel (Figure 6). For the bottom surface of the base (Figure 6, left), all translations and rotations have been restricted to accurately simulate the connection between the base and another solid or the ground. Regarding the flywheel (Figure 6, right), the external surface has been constrained to simulate critical conditions, such as the flywheel being locked at the start of the machine's operation, as well as all its degrees of freedom, since this is the only way to transmit torque to the engine. Therefore, in both cases, translations in all directions and rotations around the local reference system axis of each solid have been restricted.

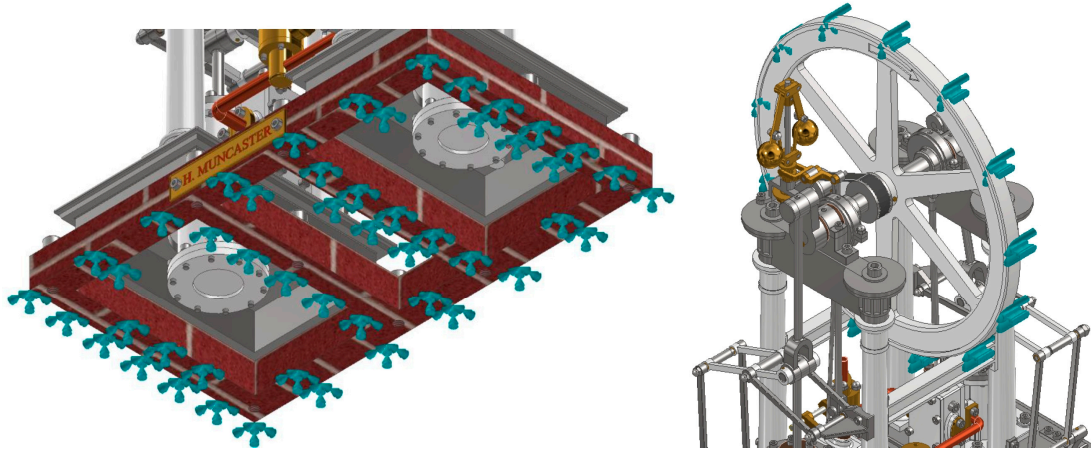


Figure 6. Brick base fixing (left) and flywheel fixing (right).

2.2.5. Discretization

To perform the analysis, it is essential to discretize the model, which involves meshing. This procedure entails dividing the model into multiple elements defined by nodes. The analysis variables will be calculated at these nodes, and values will be interpolated between them to determine the value within each element. The accuracy of this interpolation depends on the number of nodes within an element; a higher number of nodes (i.e., smaller element size) results in a more precise solution. Following this brief explanation of FEM and its importance in addressing the problem, the automatic discretization provided by the software can be observed (Figure 7).

Beginning with the mathematical foundation underlying discretization, the equilibrium equations for a continuous system can be formulated as partial differential equations. This equation describes the behavior of the model, specifically for a linear static analysis:

$$\nabla \sigma + f = 0 \quad (1)$$

where σ is the stress tensor and f is the applied force vector [36].

In the finite element method, the principle of virtual work is used:

$$\int_{\Omega} \delta \epsilon : \sigma d\Omega = \int_{\Omega} \delta u \cdot f d\Omega + \int_{\Gamma_t} \delta u \cdot t d\Gamma \quad (2)$$

where $\delta \epsilon$ is the virtual variation of strain, δu is the virtual variation of displacement, Ω is the body domain, and Γ_t is the boundary with applied traction t [36].

Now, continuing with the discretization process itself, the continuous domain Ω is divided into a finite number of elements e , such that:

$$\Omega \approx \bigcup_{e=1}^{n_e} \Omega_e \quad (3)$$

Each Ω_e is a subregion with simple geometry; in this study, it will be tetrahedral [37].

Within each element, displacements u , are approximated using shape functions N_i :

$$u^e(x) = \sum_{i=1}^{n_{en}} N_i(x) u_i \quad (4)$$

where u_i are the nodal displacements of the element and n_{en} is the number of nodes per element [36].

The relationship between nodal forces and nodal displacements is expressed through the elemental stiffness matrix K^e :

$$K^e = \int_{\Omega_e} (B^e)^T D B^e d\Omega \quad (5)$$

where B^e is the strain displacement matrix and D is the constitutive matrix depending on the material [36]. This elemental stiffness matrix K^e is assembled into a global stiffness matrix K :

$$K = \sum_{e=1}^{n_e} K^e \quad (6)$$

Similarly, the nodal force vectors are assembled to obtain the global force vector F . The discrete problem is reduced to solving a system of linear equations:

$$F = Ku \quad (7)$$

where u is the vector of nodal displacement.

Finally, the system can be solved using numerical methods such as Gaussian elimination, iterative methods (e.g., conjugate gradient), among others [36].

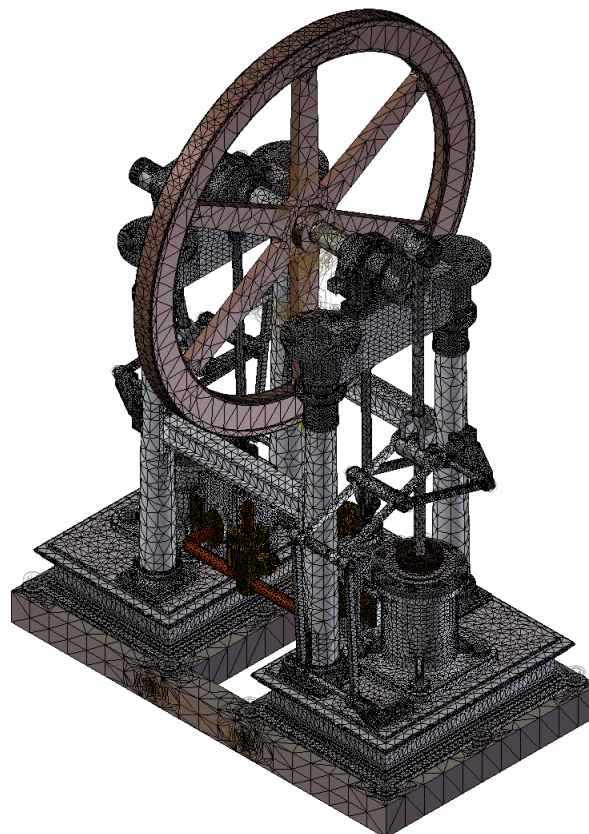


Figure 7. Automatic mesh of the model.

As mentioned previously, in this study, quadratic interpolation elements, specifically parabolic (tetrahedral) elements, were used as they better conform to the model, particularly in curved regions, to achieve results that are as accurate as possible. The software automatically generates a coarser mesh in larger areas and a finer mesh in smaller areas,

allowing the grid to better fit the geometry of the part and improve result accuracy. For this investigation, a configuration was defined with an element size of 0.005 m, a maximum element growth ratio of 1.1, a maximum turn angle of 65° , and a minimum element size of 0.001 m, providing greater control over the mesh. Additionally, the mesh was refined in specific areas where problems are anticipated to conduct a more precise study. This study utilized a mesh comprising 6,530,981 elements and 9,960,107 nodes.

2.2.6. Critical Positions

Examining critical positions in engineering, such as those of a steam engine, is essential for understanding the system's behavior under extreme conditions. In a steam engine, two critical positions are typically highlighted: the lower dead center and the upper dead center of the piston within the cylinder after closing the steam inlet. These positions reflect moments when the engine experiences maximum pressure variations across the piston, leading to significant stresses and deformations.

By locking the flywheel, the engine's operation can be simulated, allowing observation of how the force is transmitted from the piston to the other components. Since this engine is symmetric and admits the same amount of steam at the same pressure in both cylinders, the results obtained at the upper and lower dead centers of one cylinder should, in principle, be the same as those for the other cylinder.

For verification, and given the lack of information about the engine's operating conditions, an arbitrary pressure of 2 MPa is initially tested to observe the von Mises stress, displacement, and safety factor results. This pressure accounts for atmospheric pressure, which is the pressure at which the engine exhausts, necessitating the use of gauge pressure values.

As previously mentioned, the same amount of steam enters each piston at the same pressure, resulting in the same load, although the resulting values differ for each position and cylinder. While one piston compresses or expands the steam, the other is mid-stroke, meaning that steam expands on one side of the piston and exhausts to atmospheric pressure on the other. This condition will be verified at the upper and lower dead centers of each cylinder (Figures 8 and 9) with the applied pressure load, also including the force of gravity.

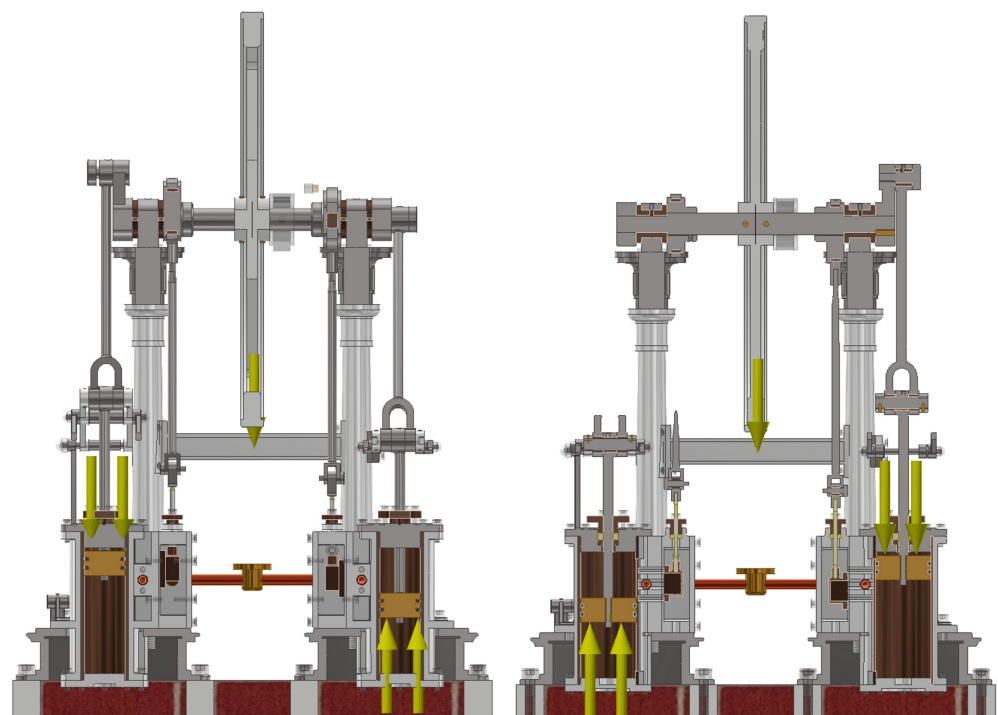


Figure 8. Upper dead center position: left cylinder (left) and right cylinder (right).

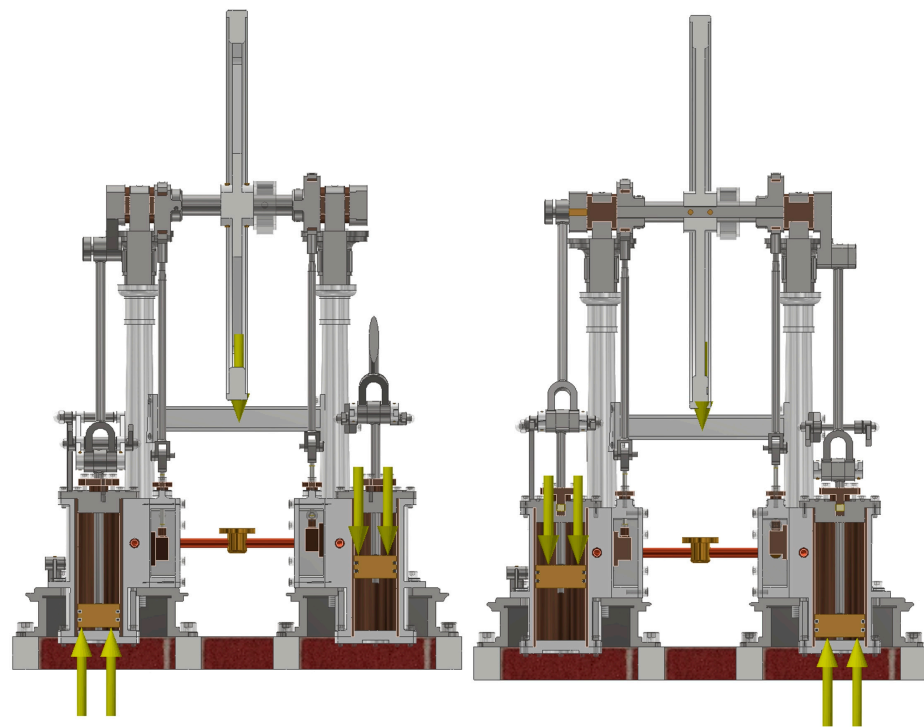


Figure 9. Lower dead center position: left cylinder (left) and right cylinder (right).

A comparison of the results for von Mises stress distribution, safety factor, and displacement at the upper dead centers of both cylinders can be seen in Figure 10, Figure 11, and Figure 12, respectively.

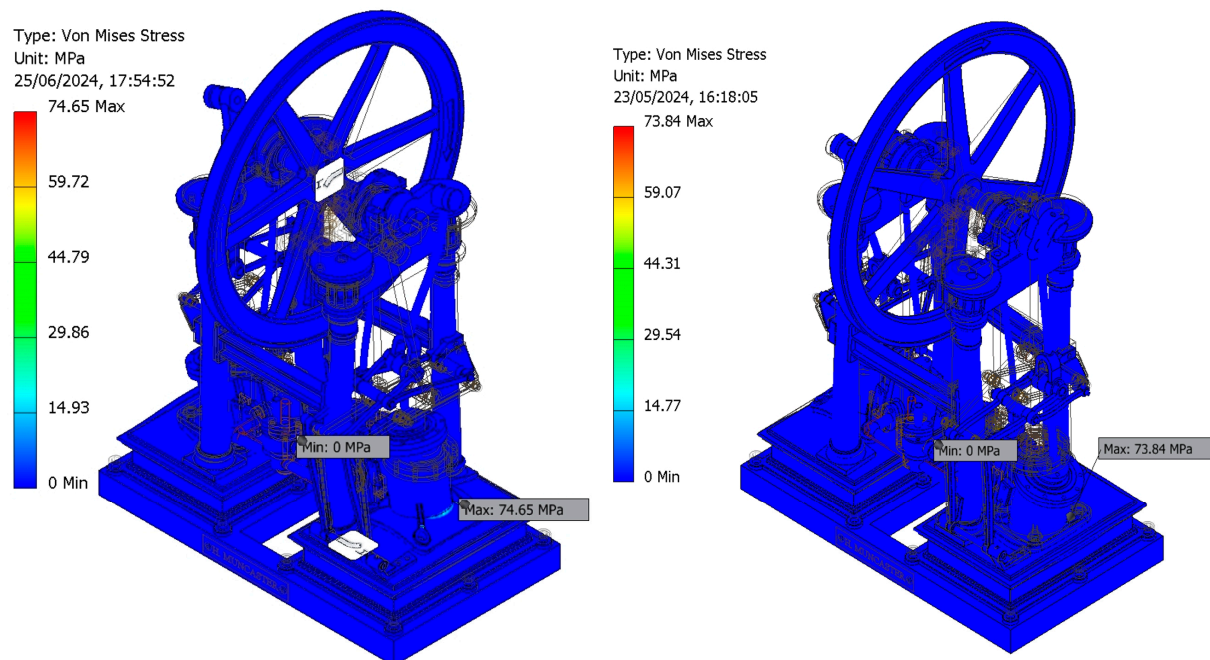


Figure 10. Von Mises stress distribution at upper dead center: left cylinder (left) and right cylinder (right).

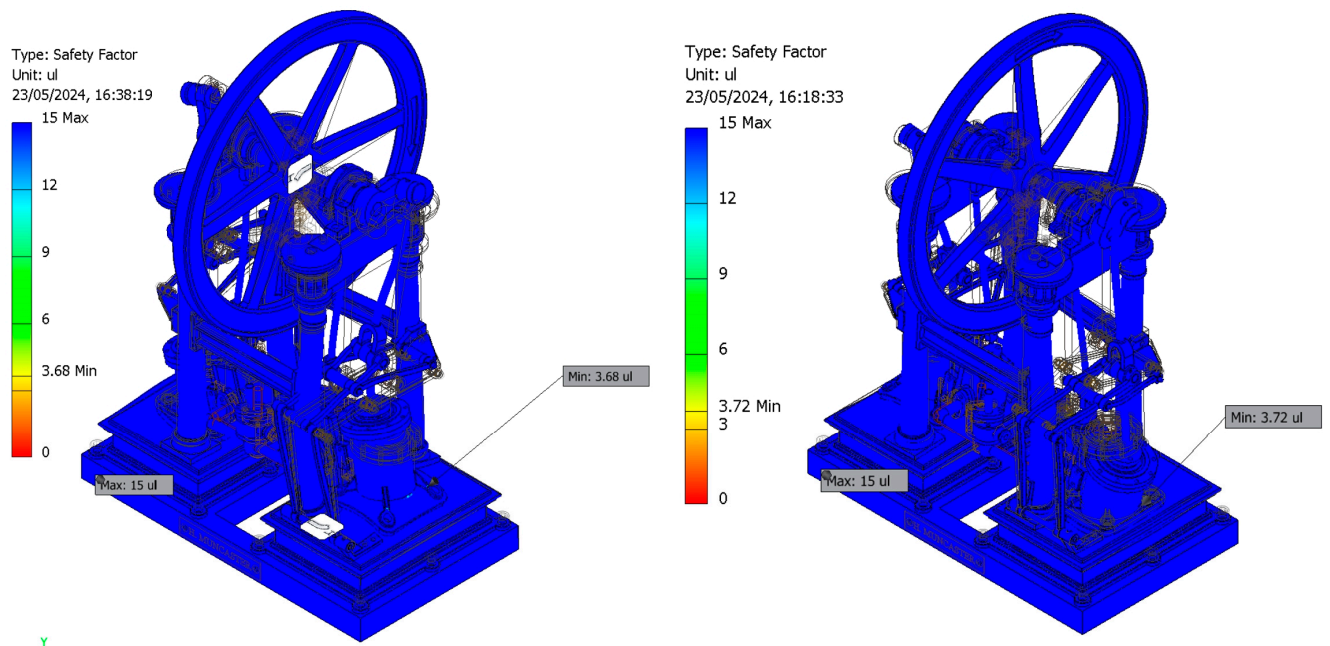


Figure 11. Safety factor distribution at upper dead center: left cylinder (**left**) and right cylinder (**right**).

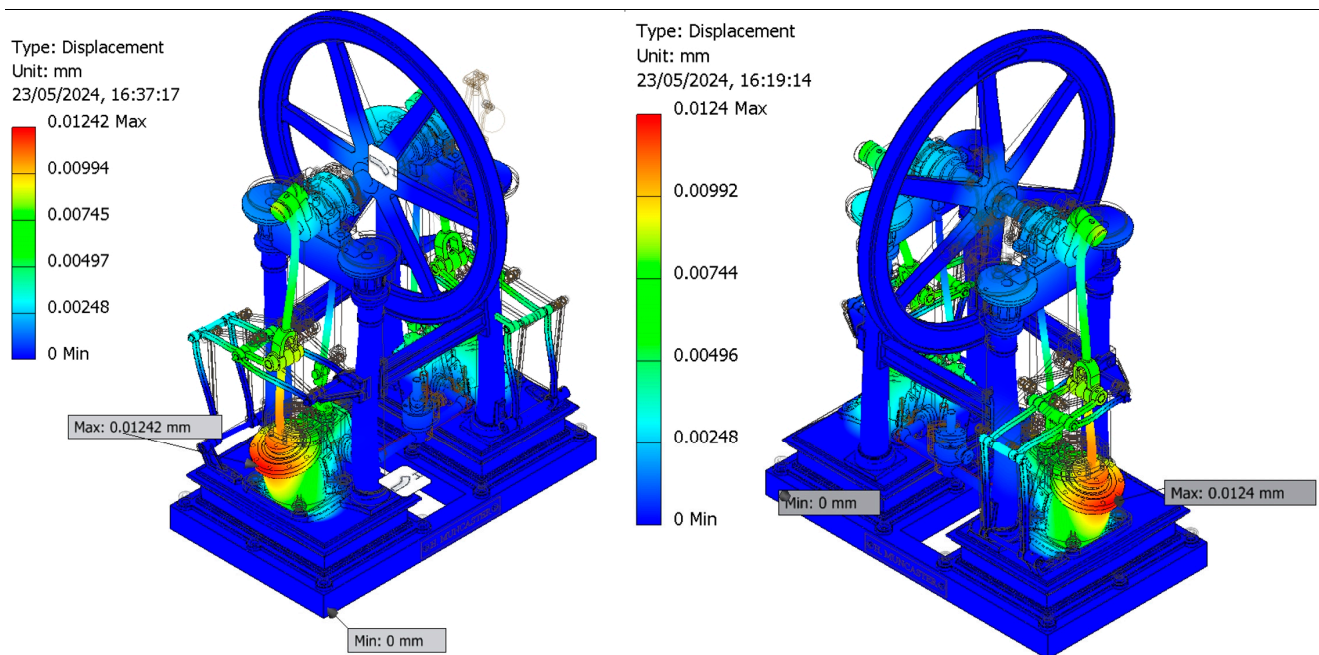


Figure 12. Displacement distribution at upper dead center: left cylinder (**left**) and right cylinder (**right**).

Similarly, a comparison of the results for von Mises stress distribution, safety factor, and displacement at the lower dead centers of both cylinders can be observed in Figure 13, Figure 14, and Figure 15, respectively.

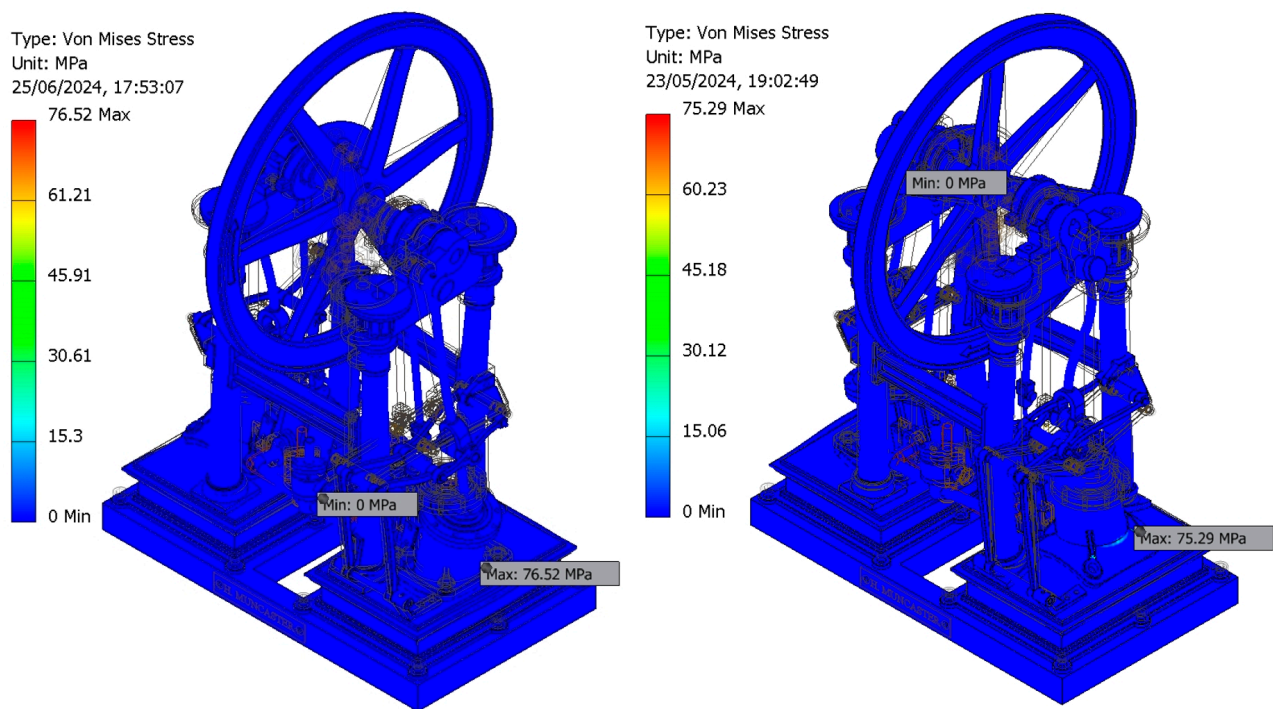


Figure 13. Von Mises stress distribution at lower dead center: left cylinder (left) and right cylinder (right).

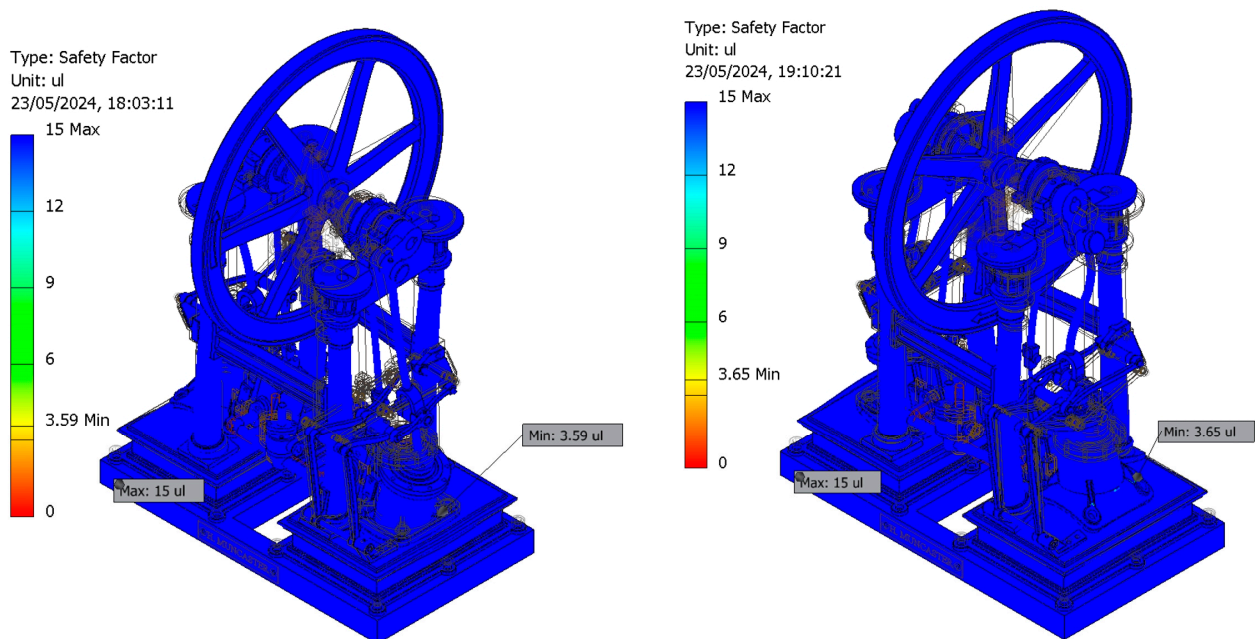


Figure 14. Safety factor distribution at lower dead center: left cylinder (left) and right cylinder (right).

From these figures, it can be observed that the maximum stresses always occur in the right cylinder. Additionally, the results for both dead centers in both cylinders are similar, although the loads applied at the dead centers of the left cylinder are slightly higher, indicating a more restrictive scenario. Therefore, the analysis will be focused on the left cylinder with new loads, as the safety factor is far from the ideal value of 2, which is considered optimal from a design perspective.

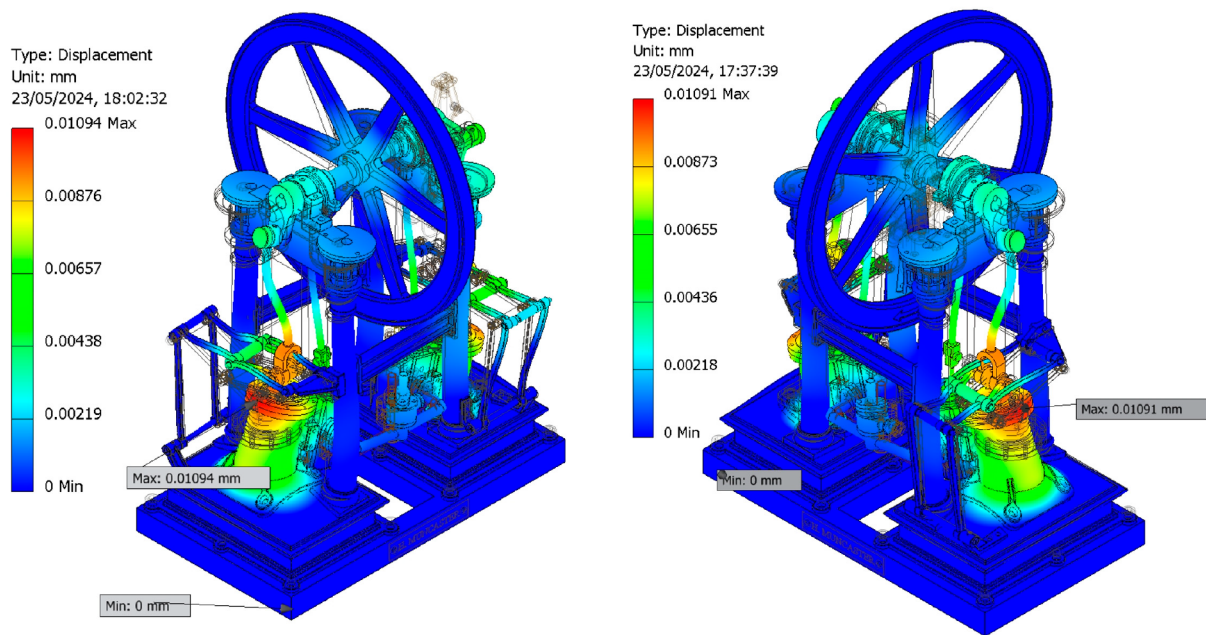


Figure 15. Displacement distribution at lower dead center: left cylinder (left) and right cylinder (right).

Thus, it is necessary to achieve a safety factor value as close to 2 as possible to maximize the admission pressure, thereby increasing the engine's efficiency and ensuring that the steam engine operates within safety factor values between 2 and 4, guaranteeing safe and proper functioning.

Figures 16 and 17 provide visual representations of the machine in these critical positions.

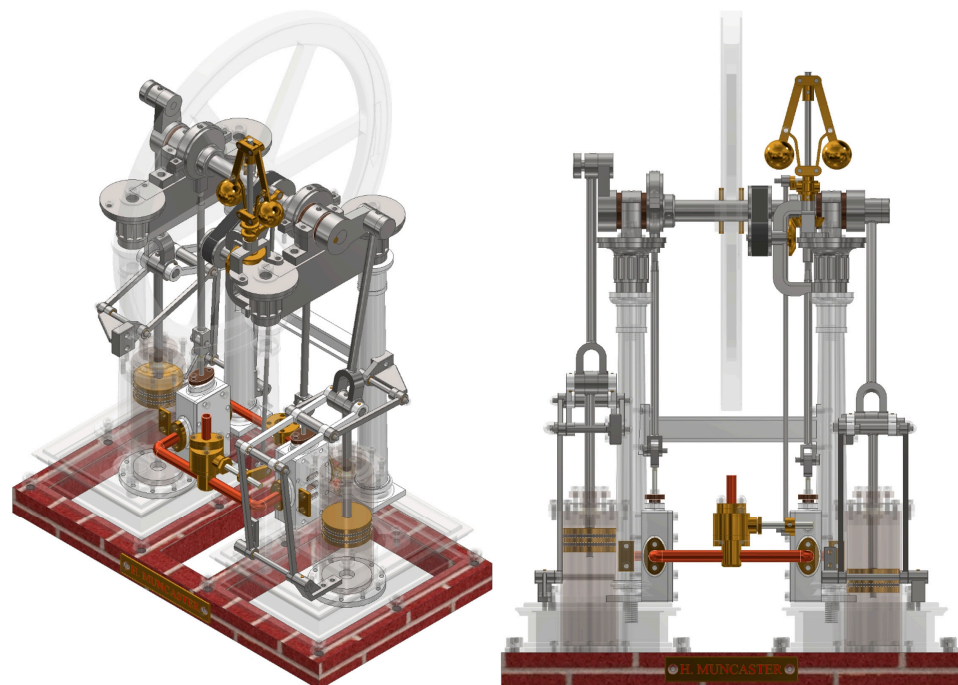


Figure 16. Critical position 1: axonometric view (left) and front view (right).

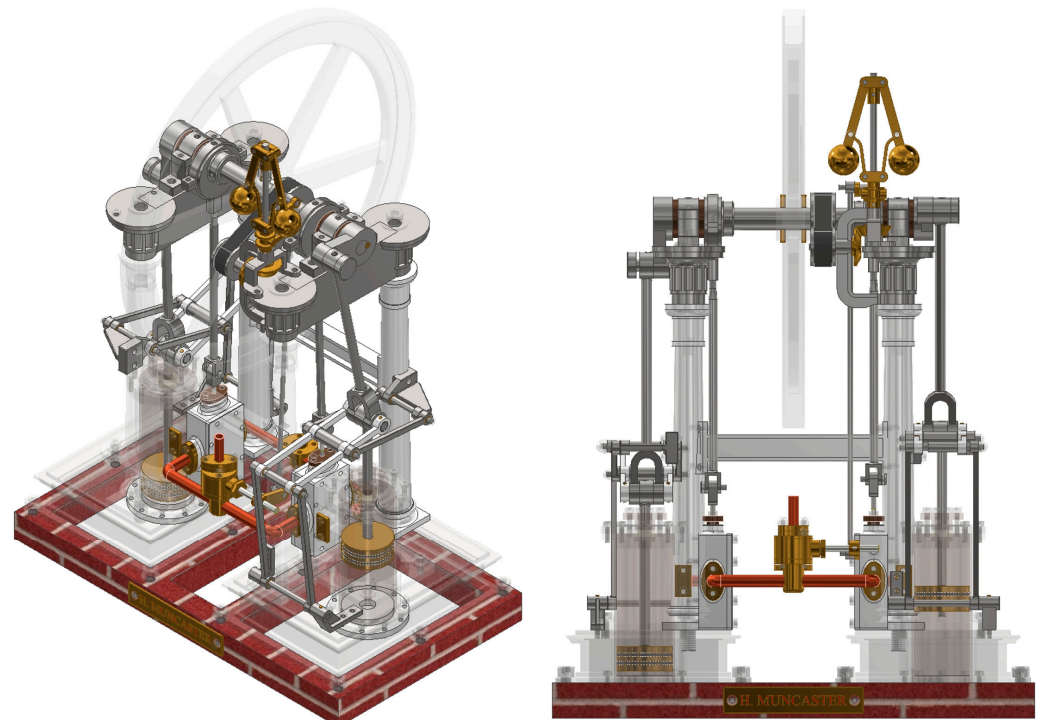


Figure 17. Critical position 2: axonometric view (left) and front view (right).

2.2.7. Modal Analysis

Modal analysis is a fundamental technique in structural and mechanical engineering that aims to explain the dynamic behavior of a system composed of multiple interconnected components. This approach is employed to determine the natural frequencies and vibration modes of the system, providing essential information about its stability, response to dynamic loads, and potential resonance issues. If the natural frequencies are non-zero and distinct from each other, it can be concluded that the system does not behave as a mechanism, and consequently, a linear static analysis can be performed.

2.2.8. Linear Static Analysis

Once it has been verified that none of the natural frequencies of the vibration modes correspond to 0 Hz, a linear static analysis is performed. To achieve results that closely approximate reality, it is essential to ensure mesh convergence until the maximum von Mises stress value shows no significant variations.

Additionally, accurately determining the stress envelope is crucial for conducting a precise analysis and avoiding incorrect results. The steam conditions at the cylinder inlet depend on the machine's geometry, the dimensions of the piston and chamber, and its intended application. The desired operating pressure should be as high as possible, theoretically achievable by increasing the number of molecules, raising the steam temperature, and reducing the space, i.e., decreasing the cylinder diameter.

This study is based on several fundamental assumptions and mathematical formulations similar to those in other studies [38].

Despite the absence of specific operational data for the machine, an initial simulation was conducted assuming a working gauge pressure of 2 MPa, which was arbitrarily chosen to begin obtaining results. This pressure was applied to both cylinders but on the opposite faces of the pistons. Additionally, the selected gauge pressure considers the atmospheric pressure in the opposite chamber. Thus, as the steam expands, the exhaust valve opens, releasing steam at atmospheric pressure, while steam is admitted at working pressure in the other chamber.

After applying a working pressure of 2 MPa, the machine's performance was evaluated. It was confirmed that the minimum safety factor was not less than one, indicating that

the yield strength was not exceeded, which would otherwise cause the steam engine to fail under static load. Consequently, iterative analyses are necessary to identify a pressure value that results in a maximum von Mises stress below the yield strength and a safety factor greater than one to ensure safe operation.

The goal was to define the range of working gauge pressures that provide safety factors for both critical positions as close as possible to the values of 2 or 4, which is the optimal range in modern machine design. After simulations, it was found that the most restrictive position for obtaining the maximum operating pressure (3.550 MPa) is the lower dead center of the left cylinder, yielding a safety factor close to 2. In contrast, to achieve the minimum operating pressure such that the safety factor does not exceed 4, the most restrictive position was determined to be the upper dead center of the left cylinder, with a pressure of 1.885 MPa.

However, this study focuses on the results corresponding to the maximum allowable pressure (3.550 MPa), as this value yields the highest possible engine efficiency. The results for the minimum allowable pressure (1.885 MPa) were also checked to determine the acceptable pressure range since it may not always be possible to operate at maximum or minimum admission pressure, depending on the required needs.

Figure 18 shows the application of pressure loads when the engine is in both critical positions.

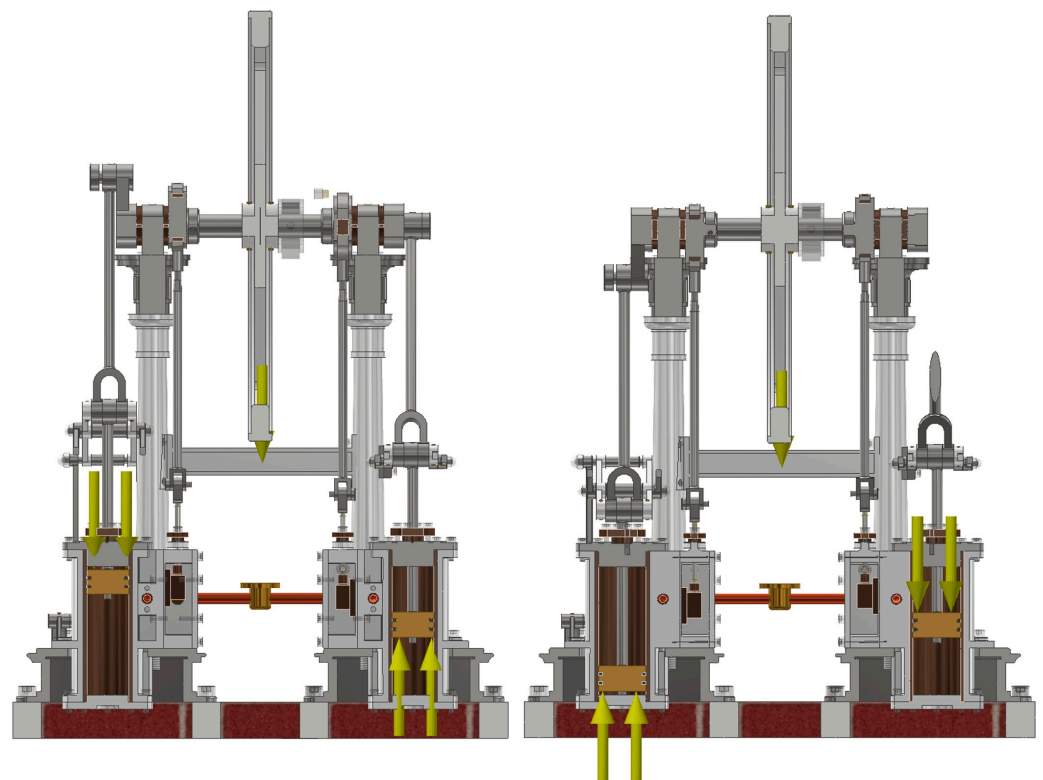


Figure 18. Front view of pressure loads at the critical positions: critical position 1 (left) and critical position 2 (right).

After applying the working pressure loads, a linear static analysis was conducted. This required a convergence analysis along with the steam engine's response, characterized by the von Mises stress, displacement, and safety factor values.

Additionally, a manual mesh convergence analysis was carried out to ensure that the results were reliable, given that the numerous components in this engine led to a complex mesh. This manual analysis involves an initial mesh with a predetermined global size followed by simulation. Finally, the location where external deformations generate the

maximum von Mises stress is identified, recording its value, the iteration number, and the element size.

Subsequently, local mesh refinement is performed in critical areas by reducing the finite element size, achievable through vertex, edge, surface, or solid refinement.

In this case, the connection between the eccentric strap and the slide valve was completely refined, as it is a critical area (though not the worst), as well as the surface of both cylinders, since the maximum von Mises stresses occur in the ribs connecting the main body of the cylinder to its base. Thus, after reducing the element size, the assembly was re-meshed, and the simulation was repeated iteratively until the relative error in the von Mises stress values between consecutive iterations was below a predefined threshold, specifically 10%.

3. Results

3.1. Critical Position 1: Upper Dead Center (Left Cylinder)

3.1.1. Modal Analysis

Following the application of the necessary boundary conditions, the simulation was conducted. The graph in Figure 19 displays the mode number on the horizontal axis and the corresponding natural frequency (Hz) on the vertical axis. This graph reveals that the first mode of vibration has a distinct frequency above 0 Hz, specifically 319.04 Hz, which is the lowest natural frequency associated with this mode. Consequently, as there was no zero natural frequency, a linear static analysis could be performed. This is because the system will not behave as a mechanism, in accordance with the simulation conditions established for the piston plunger's position at the upper dead center of the left cylinder.

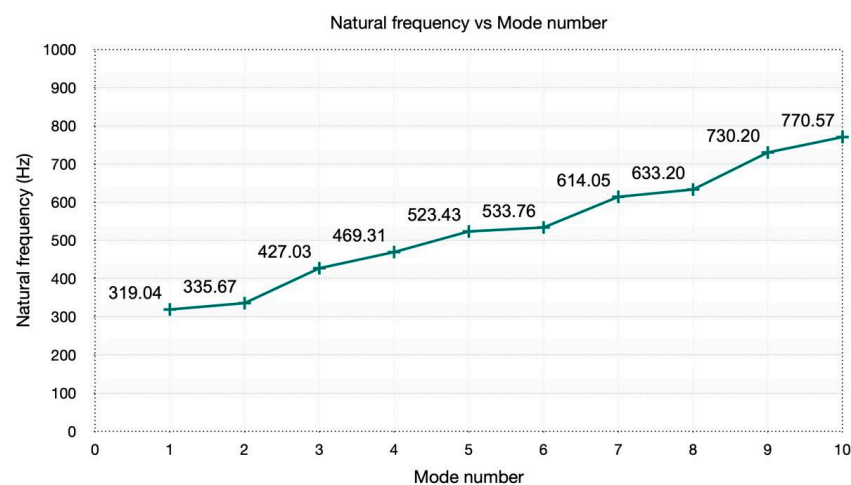


Figure 19. Natural frequency vs. mode number.

3.1.2. Linear Static Analysis

Presented here are the results for the critical position at the upper dead center of the left cylinder. As previously described, a linear static analysis was first performed under a working pressure gauge of 2 MPa, yielding a safety factor significantly greater than one. However, from a machine design perspective, an optimal range of safety factors between two and four is recommended to ensure proper operation.

Consequently, it can be asserted that the machine would not fail if the steam pressure at the inlet were 2 MPa, but since the safety factor is substantially higher than one, there is room for optimization to bring it closer to two, which would necessitate another analysis with a higher working pressure.

These simulations identify the location where the safety factor is minimal, allowing for mesh refinement. In this case, it was found at the junction between the cylinder body and its base, specifically at the ribs connecting them. The potential failure of the slide valve spindle was also examined.

Additionally, after conducting the analysis with various mesh element sizes for the slide valve spindle and the cylinder (on both the left and right sides of the engine), the mesh convergence analysis graph (Figure 20) was obtained, along with data on element size, von Mises stress, relative error (%), and the iteration number (Table 2).

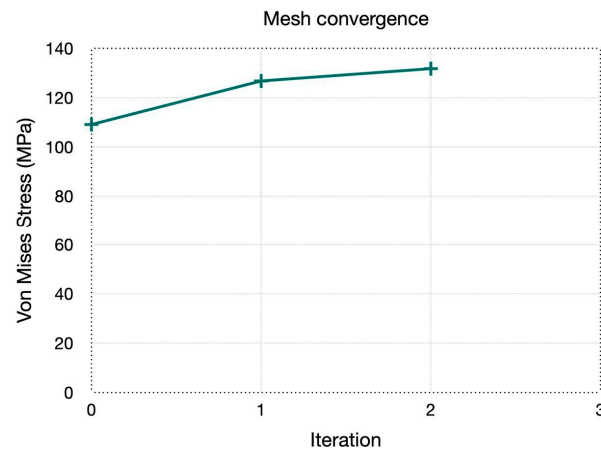


Figure 20. Mesh convergence graph.

Table 2. Mesh convergence analysis.

Element Size (mm)	Von Mises Stress (MPa)	Relative Error (%)	Iteration
1.00	109.1	Not Available	0
0.70	126.8	16.22	1
0.50	131.8	3.94	2

In Table 2, it can be observed that mesh convergence is achieved with an element size of 0.5 mm, resulting in a relative error of 3.94% in the second iteration.

Additionally, Figure 21 illustrates the distribution of von Mises stress, while Figure 22 shows the location of the maximum von Mises stress value (131.8 MPa), specifically at one of the ribs connecting the right cylinder body to its base.

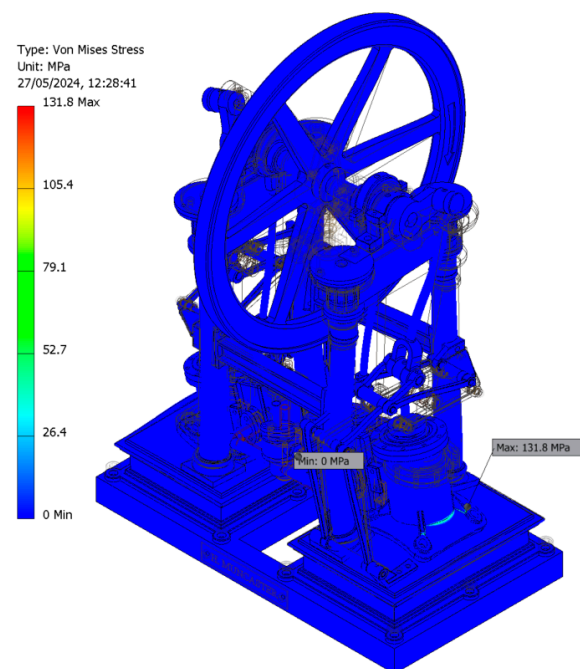


Figure 21. Von Mises stress distribution.

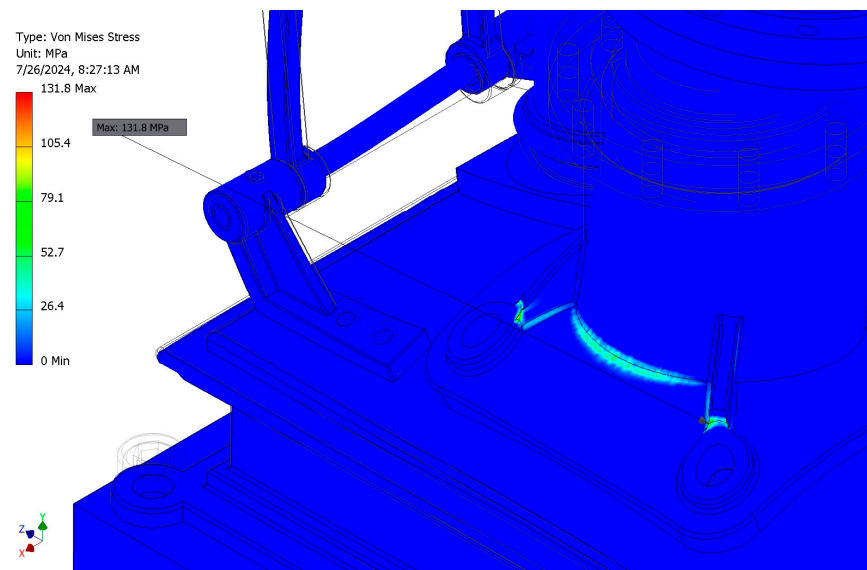


Figure 22. Location of the highest value of von Mises stress.

Moreover, the displacement distribution has been obtained (Figure 23), displayed with a scaling factor of $\times 2$ for improved visibility, though this is not the actual representation as the displacement values are negligible.

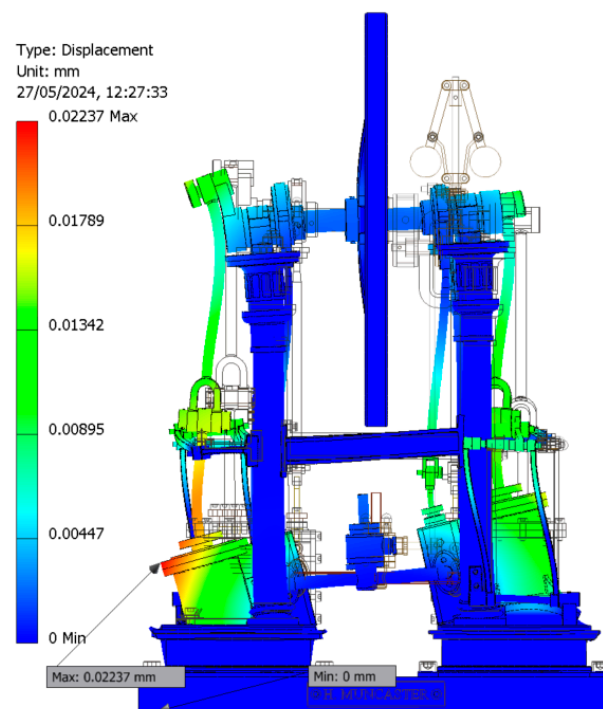


Figure 23. Displacement distribution with a $\times 2$ scale factor.

Finally, Figure 24 presents the distribution of the safety factor, while Figure 25 highlights the location of its lowest value (2.09), situated at the connecting rib between the right cylinder body and its base.

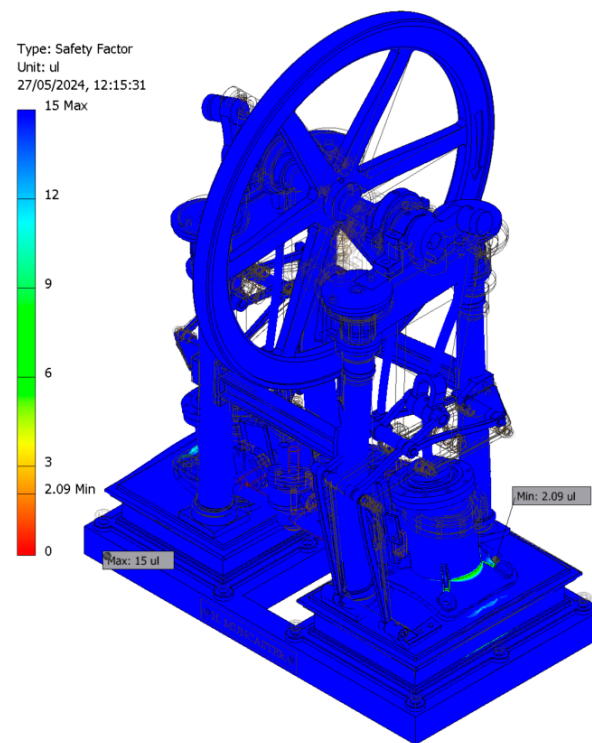


Figure 24. Safety factor distribution.

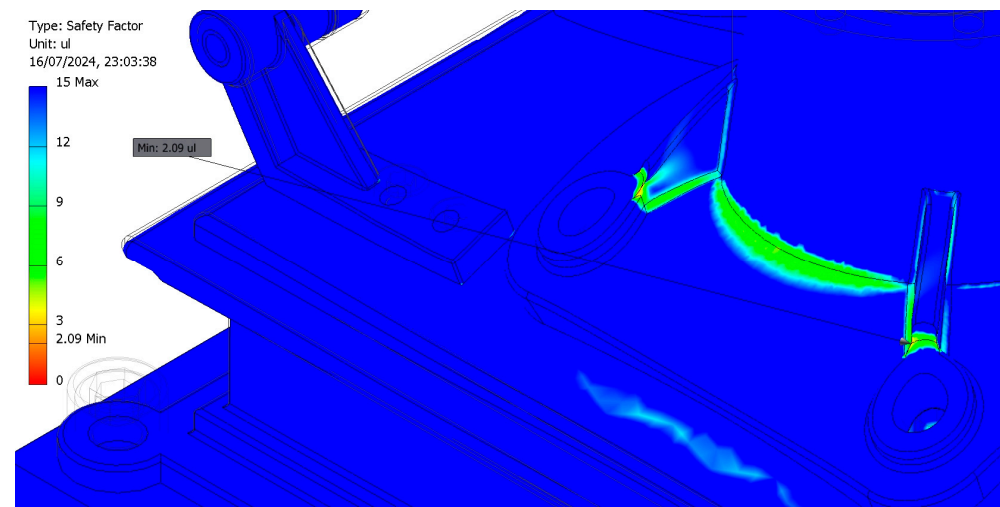


Figure 25. Location of the minimum safety factor.

3.2. Critical Position 2: Lower Dead Center (Left Cylinder)

3.2.1. Modal Analysis

Figure 26 depicts a diagram of the modal analysis performed for critical position 2, following a methodology analogous to that of critical position 1. It was confirmed that none of the natural frequencies were zero, indicating the suitability for conducting a linear static analysis. Under the simulation conditions, at the critical position associated with the piston plunger's position at the lower dead center of the left cylinder, the behavior as a mechanism is ruled out.

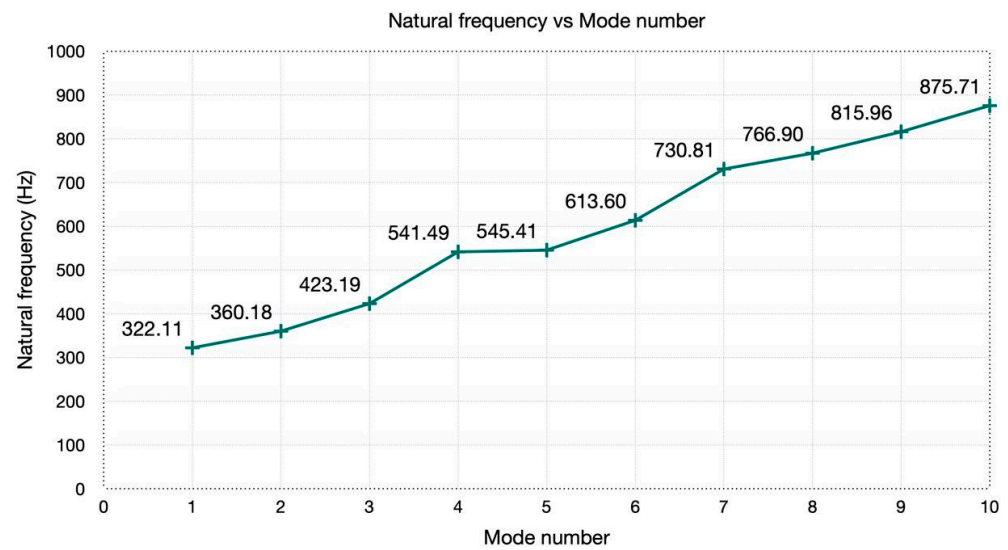


Figure 26. Natural frequency vs. mode number.

3.2.2. Linear Static Analysis

Similar to the analysis at the upper dead center position, the working pressure was set at 3.550 MPa. However, it was essential to verify all parameters, especially ensuring that the safety factor is greater than one. After performing the analysis with different mesh sizes, a mesh convergence analysis graph was obtained (Figure 27), along with detailed information on element size, von Mises stress, relative error (%), and the iteration number (Table 3).

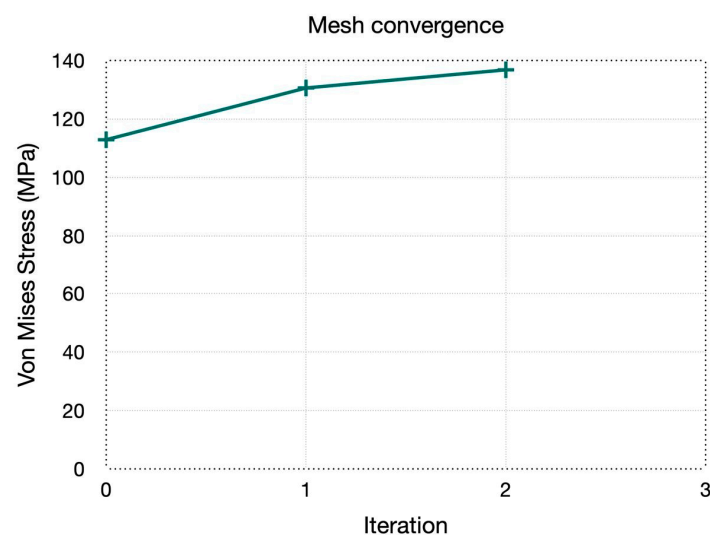


Figure 27. Mesh convergence graph.

Table 3. Mesh convergence analysis.

Element Size (mm)	Von Mises Stress (MPa)	Relative Error (%)	Iteration
1.00	112.9	Not Available	0
0.75	130.6	15.68	1
0.50	136.8	4.75	2

From Table 3, it can be observed that the relative error for an element size of 0.50 mm is 4.75% in the second iteration, which is considered sufficiently acceptable to validate the accuracy of the analysis.

Figure 28 shows the distribution of von Mises stress, and Figure 29 indicates the location of the maximum stress value (136.8 MPa), specifically at the connecting rib between the right cylinder body and its base.

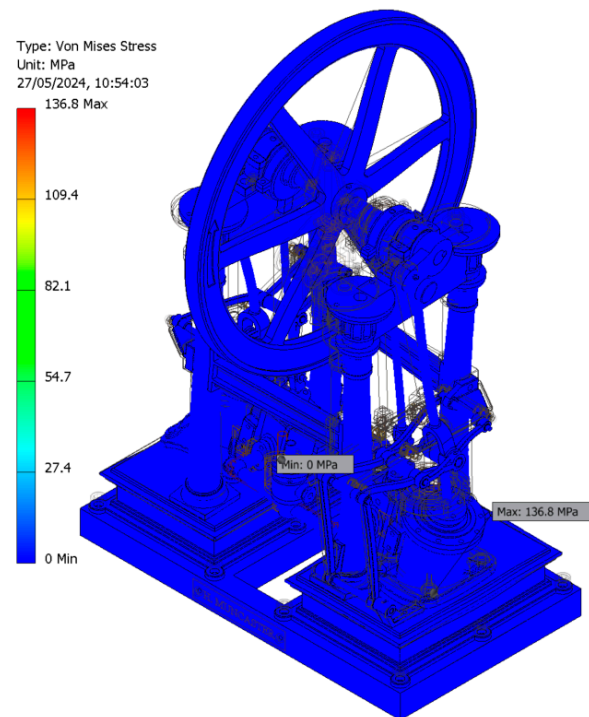


Figure 28. Von Mises stress distribution.

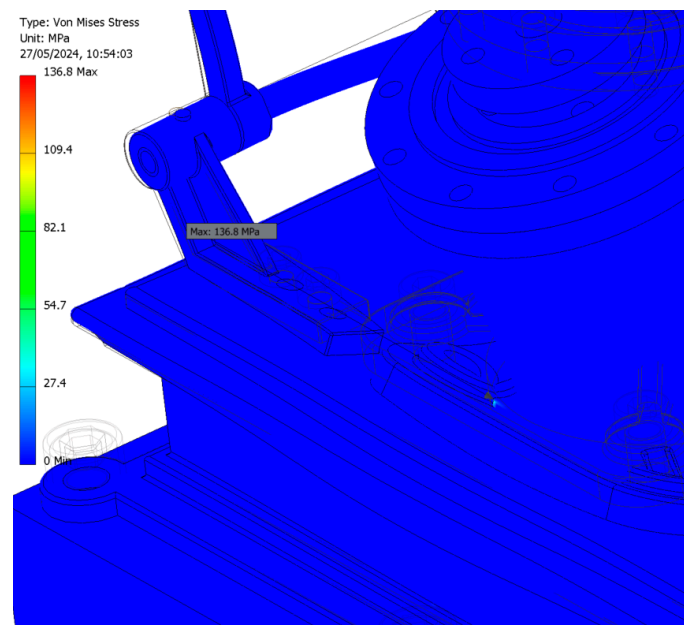


Figure 29. Location of the highest value of von Mises stress.

Additionally, the displacement distribution was obtained (Figure 30), which has been displayed with a scaling factor of $\times 2$ for better observation, though this is not the actual representation as the displacement values were negligible.

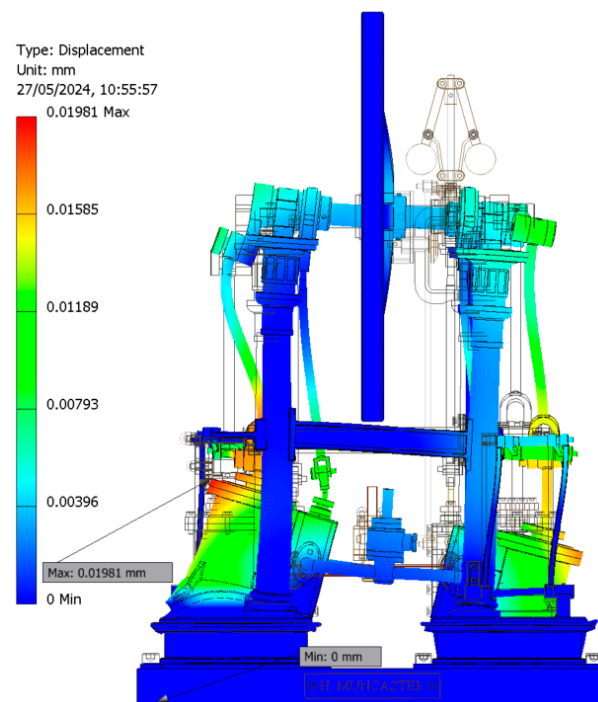


Figure 30. Displacement distribution with a $\times 2$ scale factor.

Ultimately, Figure 31 illustrates the distribution of the safety factor, while Figure 32 highlights the location of the minimum value (2.01), which is situated at the connecting rib between the right cylinder body and its base.

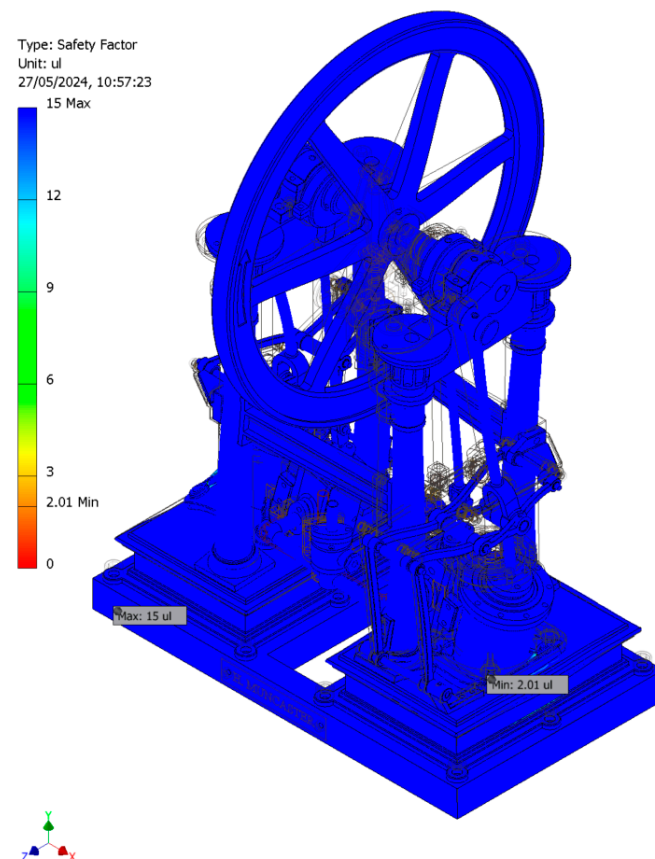


Figure 31. Safety factor distribution.

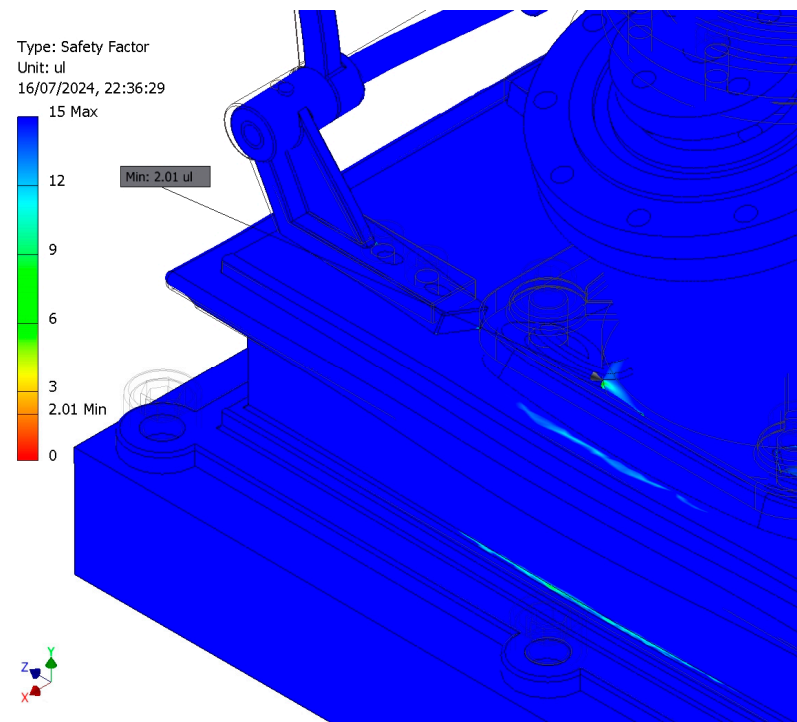


Figure 32. Location of the minimum safety factor.

4. Discussion

Our findings reveal that the peak von Mises stress is located at the lower dead center position of the left cylinder. For displacement analysis, a scaling factor of two was applied to better visualize the deformed configuration of the steam engine at both critical positions (lower and upper dead center of the left cylinder). This adjustment is consistent with the observation that maximum displacements occur in the left cylinder due to compression and tension forces acting within it. However, the displacements obtained are negligible, which supports the efficient operation of the steam engine under this pressure load. Moreover, the absence of significant deformations reduces friction between components.

As intended, the safety factor exceeds unity. Consequently, the optimal allowable steam pressure at the intake (working pressure) for this engine has been identified as 3.550 MPa, ensuring a safety factor within the range of two to four. This value can also be used to determine the working pressure for other similar engines. The working pressure value obtained, considering the lower dead center as the most critical position, represents the maximum pressure at which both compression and tension in the left cylinder maintain a safety factor above two. Additionally, the pressure value that produces a safety factor of 4 has been determined, thus establishing the optimal range of working pressures. The minimum pressure value was determined by observing the upper dead center, which is the most restrictive position, yielding a value of 1.885 MPa (Figure 33). This ensures that any intake pressure above this value would result in a safety factor of less than four.

As a verification measure, the aforementioned pressure was also applied to the lower dead center position, confirming that the safety factor is below four (3.78) (Figure 34).

Consequently, it can be concluded that the operational steam intake pressure range should be between 1.885 MPa and 3.550 MPa to ensure that the safety factor remains within the range of two to four.

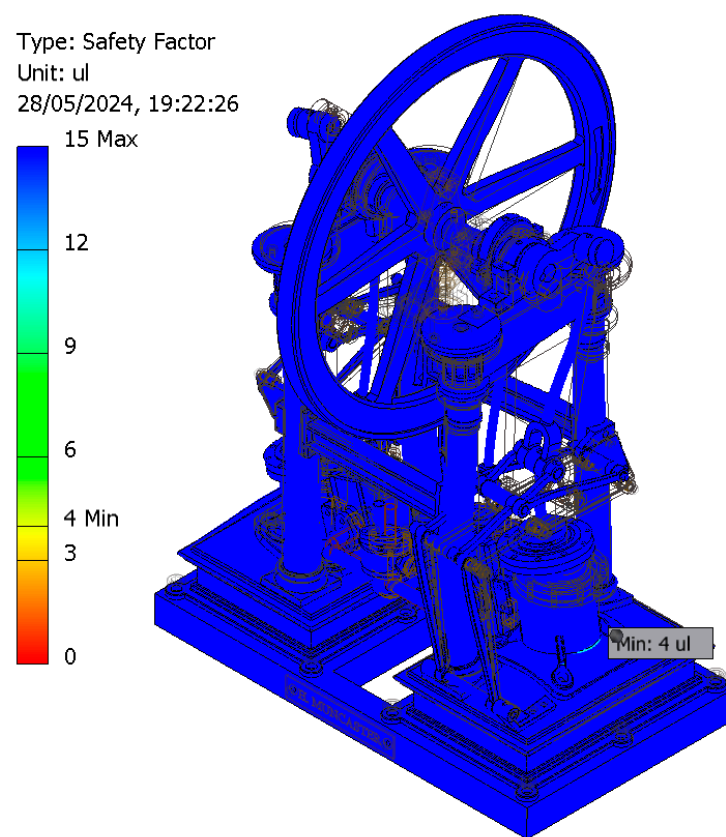


Figure 33. Safety factor distribution with a pressure of 1.885 MPa at the upper dead center position.

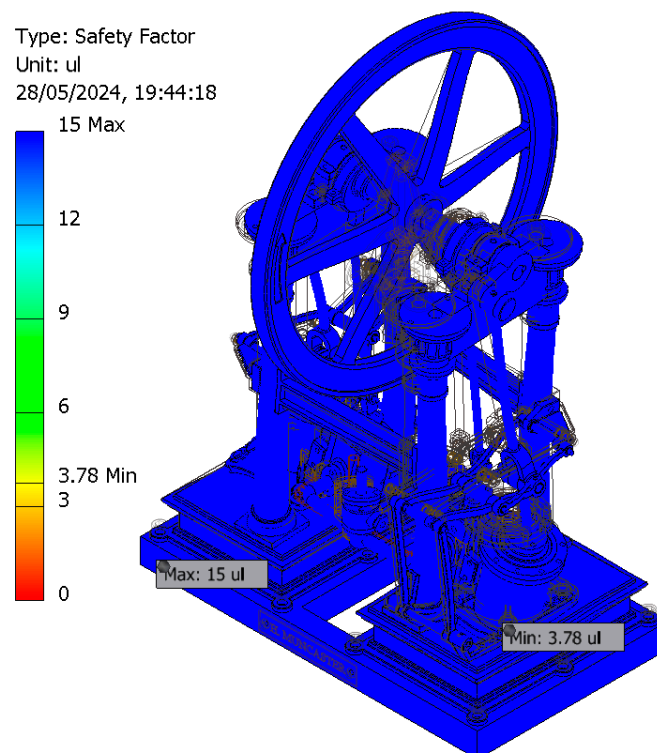


Figure 34. Safety factor distribution with a pressure of 1.885 MPa at the lower dead center position.

5. Conclusions

In this research, the technical feasibility of a two-cylinder entablature steam engine with a parallel motion crosshead has been analyzed. This analysis was based on an accurate 3D CAD model of the historical invention, created using Autodesk Inventor Professional 2024 software, which helped to explain the assembly and understand its operation.

The 3D CAD model underwent a linear static analysis using FEM, focusing on two critical positions in the left cylinder (where steam admission occurs, being the most restrictive): upper dead center and lower dead center.

However, since no information regarding the operating conditions (working pressures) was available, it was necessary to determine the working pressure values (steam admission pressures) to ensure the safe operation of the engine from a materials strength perspective. This process established the optimal range of operating pressures to ensure its technical feasibility. The simulations were performed ensuring that the safety factor values were between two and four, which is considered to be the optimal range from the perspective of modern machine design.

A manual refinement of the mesh was carried out, followed by a mesh convergence analysis, resulting in a relative error of 4.75% in the second iteration for an element size of 0.50 mm at the lower dead center position and 3.94% in the second iteration for the same element size at the upper dead center position. Ultimately, the results of the various simulations determined that the optimal working pressure range (steam admission pressure) was between 1.885 and 3.550 MPa, providing safety factor values between 2.01 and 3.78.

Additionally, the maximum von Mises stress was found to be 136.8 MPa at the lower dead center position, located at the connection rib between the right cylinder body and its base.

Lastly, this research suggests that future work could involve conducting a dimensional analysis with geometrically similar machines, relating dimensions such as the flywheel diameter, piston diameter, or cylinder length.

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