Arming Mechanism for Aerial Target

Abhishek Anand Bapat

ABSTRACT

The design and development of a turbine-driven mechanical arming mechanism for use in aerial target drones and other defense applications is the main goal of this project. The work's main goal was to develop a completely mechanical arming system that eliminates the dangers of electrical or pyrotechnic devices, like interference, hacking, and accidental activation. The mechanism works on the basis of aerodynamic energy conversion, in which a bucket-type impulse turbine is driven to produce rotational motion by air entering through a pressure valve.

A multi-stage gear reduction system transfers this rotational motion, increasing torque and decreasing speed before powering a lead screw-striker assembly that completes the arming procedure. The turbine and transmission components were analyzed using structural simulations and mechanical calculations, confirming that stresses on all major parts remain within safe limits for the chosen materials. The design exhibited high mechanical reliability, minimal deformation, and efficient power transfer. Overall, the project demonstrates an innovative and tamper-proof arming mechanism that functions solely through mechanical means, ensuring operational safety and reliability in demanding defence and aerospace environments. The concept successfully combines aerodynamic activation with mechanical precision, offering a secure and maintenance-friendly solution for modern unmanned aerial systems. [1][3]

CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1.1 INTRODUCTION TO KAMIKAZE DRONES

Kamikaze drones, also known as loitering munitions, are a type of unmanned aerial vehicle (UAV) designed for self-destructive missions. Unlike traditional drones that return after completing surveillance or airstrike operations, kamikaze drones seek out a target and destroy themselves upon impact, much like a guided missile. These drones are equipped with explosives, advanced guidance systems, and real-time target acquisition technology, making them highly effective for striking high-value enemy assets. Their ability to loiter over a battlefield before executing a precise attack provides a significant tactical advantage, especially in asymmetric warfare and contested environments.[2][4]

Kamikaze drones have become a game-changer in modern combat, offering a low-cost yet highly effective alternative to conventional missiles and aircraft. They are particularly valuable for neutralizing enemy radar systems, armoured vehicles, and troop concentrations, often without the need for direct human intervention. With their ability to operate autonomously or under remote control, these drones have blurred the lines between traditional weaponry and artificial intelligence-driven warfare. Every kamikaze drone, be it of any variety, needs a robust, highly reliable arming mechanism. Arming mechanisms incorporating electrical-mechanical-based safety and arming (SNA) mechanisms are being highly developed in all countries that currently deploy loitering munitions. A purely electrical-based SNA will help reduce the overall weight to a great extent, but it will substantially decrease reliability due to the possible risk of failures, which can damage the entire system.

1.2 KNOWLEDGE GAINED FROM THE LITERATURE

The safety-and-arming (S&A) device, as one of the core components of the weapon system, is applied to regulate the energy transfer process of the energy material. Specifically, it is used to detonate controlled explosions and prevent accidental explosions. The function of the S&A device significantly concerns the safety, reliability, and lethality of the weapon. Thus, the development of high-performance S&A devices is of great importance in modern warfare and aligns with national strategic priorities. In this project, a miniature mechanical SAD (Safety and Arming Device) with arming delay was developed for actual munitions application. Reliable arming delay performance was achieved by applying concepts of energy and motion transfer, gear-train motion translation, and finally the electro-thermal energy transfer attained during striking of the limit switch.

Recently, India used loitering munitions to destroy the Pakistani air-defence system and radars during Operation Sindoor in May 2025. India itself manufactures loitering munitions through Defence Research and Development Organisation (DRDO) with the collaboration of various private companies like Tata Defence, Larsen and Toubro Precision Systems and Engineering, and many other firms. Worldwide use of kamikaze drones is seen in all the conflicts in this decade. Therefore, it's essential to develop a national strategy to develop these highly reliable drones. Israel, the United States, China, Turkiye, Azerbaijan, and Iran are the leading manufacturers and exporters of kamikaze drones. They are also referred to as “suicide drones” due to their self-destruction capability. [1][4]

1.2.1 TYPES OF SAFETY AND ARMING DEVICES

Safety and Arming Devices (SADs) are of 3 types primarily [3][1][5]

a. Purely Electrical
b. Purely Mechanical
c. Electro-mechanical

1.2.1.1 PURELY ELECTRICAL SAFETY AND ARMING DEVICE

A purely electrical arming mechanism means the system relies only on electrical/electronic components (no gears, springs, or mechanical linkages) to activate, control, and ensure safe arming.

Key Features of Purely Electrical Arming Mechanisms:

1. Electronic Control Circuits – Microcontrollers, sensors, and logic circuits manage the sequence of arming.
2. Power-Dependent Operation – Requires a reliable and continuous power source (battery, capacitor, or external supply).
3. Switching Devices – Uses relays, transistors, or solid-state switches to enable/disable current flow.
4. Safety Interlocks – Implemented via software (logic checks, codes, timers) or hardware (fuses, safeties).
5. Signal-Based Activation – Arming triggered by electrical signals (voltage threshold, coded pulse, wireless command). [6][7]

1.2.1.2 PURELY MECHANICAL SAFETY AND ARMING DEVICE

A purely mechanical arming mechanism is one that works entirely through physical motion, forces, and linkages (like gears, cams, springs, shafts, levers, latches), without depending on any electrical or electronic system.

Key Features of Purely Mechanical Arming Mechanisms

1. Energy Source: Uses stored mechanical energy (springs, flywheels, compressed air, or turbine-driven shafts).

2. Transmission: Gear trains, cams, ratchets, and shafts transfer motion to drive the arming sequence.
3. Safety Interlocks: Physical locks, detents, or shear pins ensure the mechanism doesn't arm prematurely.
4. Triggering: Activated by motion, pressure, wind flow, or manual action (not by electrical signals).
5. Fail-Safe Design: If motion stops or something breaks, the device remains unarmed (no power = no risk). [6][11]

1.2.1.3 ELECTRO-MECHANICAL SAFETY AND ARMING DEVICES

An electro-mechanical arming mechanism is a hybrid system that combines electrical/electronic control with mechanical motion and safeties. [5] It uses electrical power to initiate or control the arming process, but the final action (locking, releasing, rotating, etc.) is carried out mechanically.

Key Features of Electro-Mechanical Arming Mechanisms

1. Electrical Input, Mechanical Output
   i) Electrical signals trigger motors, solenoids, actuators, or electromagnets.
   ii) These drive gears, cams, or linkages that complete the arming sequence.
2. Redundancy & Safety
   i) Multiple safeties: electrical interlocks + physical latches.
   ii) Often requires both electrical power and mechanical alignment to arm.
3. Controlled Flexibility
   i) Logic circuits or microcontrollers can enforce arming conditions.

Mechanical components ensure robustness and fail-safe action.

1.2.2 COMPARISON ACROSS DIFFERENT VARIABLES

Table 1.1 Comparison across variables

Aspect	Purely-Electrical	Purely Mechanical	Electro-Mechanical
Power Source	Batteries, capacitors, or an external electrical supply	Wind, springs, gear trains, physical motion	Electrical power drives actuators + mechanical motion
Control Method	Circuits, microcontrollers, relays, sensors	Gears, cams, latches, ratchets	Hybrid: circuits trigger motors/solenoids that move mechanical locks
Reliability	Vulnerable to EMI/EMP, moisture, and power failures	Very high in harsh environments (dust, moisture, EMI-proof)	More reliable than pure electrical, but still dependent on power
Flexibility	Highly programmable, logic-based control is possible	Fixed sequence, hard to modify once built	Moderate programmable logic with mechanical enforcement
Size and Weight	Heavy	Light	Heavier than electrical, lighter than mechanical
Safety	Software/hardware interlocks; risk of false signals	Physical locks, inherently fail-safe	Dual safety: requires both a correct signal and physical alignment
Applications	Modern smart fuzes, electronic arming	Old munition fuzes, timers, turbines	Missiles, aircraft safety

1.3 GAPS IDENTIFIED IN THE LITERATURE

The gaps in the literature on the importance of mechanical safety and arming devices in loitering munitions primarily revolve around the following points:

1. Limited research on cost-effective design and production methods for safety and arming devices, given the high development and production costs of loitering munitions. There is a need for affordable solutions that do not compromise safety and reliability to make the technology accessible to countries with smaller defence budgets.
2. Insufficient detailed studies on the integration of advanced AI-driven safety mechanisms with mechanical safety and arming devices. Most literature focuses separately on autonomy and electronics rather than hybrid approaches combining mechanical and electronic safety layers.
3. Lack of comprehensive evaluation of safety device performance in varied operational environments such as GPS-denied or communication-contested zones, which are common in modern warfare scenarios. There is a gap in empirical data and testing protocols under realistic battlefield conditions.
4. Sparse analysis of the implications of swarm technology on safety and arming device design. Coordination among multiple loitering munitions introduces new complexities and risks for mechanical safety that remain underexplored.
5. Few studies address the long-term maintenance and reliability challenges of mechanical safety and arming devices, specifically in loitering munitions. Given their autonomous nature and extended loitering time, ensuring consistent safe operation over time is a key research gap.
6. Ethical and legal considerations associated with safety device failure modes and malfunction in autonomous strike systems lack in-depth research, including liability and accountability frameworks.
7. Emerging technologies like MEMS-based safety and arming devices show promise but require much more validation, especially regarding scalability and integration into full-scale loitering munitions.

These gaps indicate several important directions for future research to improve the safety, affordability, and operational reliability of mechanical safety and arming devices in loitering munitions. [6][1]

1.4 PROBLEM DEFINITION

Arming mechanisms are critical safety devices that ensure controlled activation of systems only under desired conditions. Conventional purely electrical mechanisms, though compact and programmable, are highly vulnerable to failures arising from power loss, electromagnetic interference, and environmental factors such as moisture, dust, and vibration. Electro-mechanical mechanisms, while addressing some of these concerns by combining electrical control with mechanical locks, inherit vulnerabilities from both domains, including power dependency and increased design complexity. In contrast, purely mechanical arming mechanisms provide a fail-safe, power-independent, and interference-immune solution; however, they are often considered bulky and heavy compared to electrical alternatives. The core problem is to design a mechanical arming mechanism that achieves high reliability, robustness, and immunity to electrical vulnerabilities while minimizing weight and bulk through optimized materials and compact configurations.

A purely mechanical arming mechanism is often regarded as the most dependable solution when safety and reliability are critical. Since it operates solely on physical principles such as gears, springs, cams, and latches, it is completely independent of external power sources, making it highly reliable over long storage periods and in environments where batteries or electronics may degrade. Mechanical systems are inherently immune to electromagnetic interference (EMI/EMP), signal spoofing, and hacking, which pose serious risks in purely electrical mechanisms. Their operation is simple, predictable, and easy to verify through physical inspection, reducing the chances of hidden failures. Furthermore, mechanical designs can be made robust enough to withstand shock, vibration, extreme temperatures, and exposure to dust or moisture, ensuring functionality in the harshest conditions. Unlike electro-mechanical systems, which combine electronics with moving parts and therefore inherit weaknesses from both, purely mechanical mechanisms maintain a straightforward design philosophy with minimal failure modes. While they may be bulkier and slower than electrical counterparts, their fail-safe nature, immunity to interference, and long-term reliability make them the preferred choice where absolute certainty of safe arming is required. [7][5]

The concern of bulkiness in purely mechanical arming mechanisms can be effectively addressed through modern design approaches. The use of lightweight yet durable materials such as aluminum alloys, titanium, or engineering plastics significantly reduces overall weight without compromising strength. Compact gear arrangements, like planetary or compound gear trains, allow the same motion transfer in a smaller space, while multifunctional components help integrate roles and minimize part count. Additionally, hollow or stepped shafts and optimized geometries achieved through finite element analysis and advanced manufacturing further contribute to weight and size reduction. With these strategies, a purely mechanical arming mechanism can retain its reliability and fail-safe nature while overcoming traditional limitations of bulk and heaviness. [8]

1.5 OBJECTIVE OF THE PROJECT

1. Develop a purely mechanical arming mechanism that ensures safe and reliable operation without reliance on electrical power.
2. Eliminate risks of electromagnetic interference (EMI/EMP), signal spoofing, and power failures commonly associated with electrical and electromechanical systems.
3. Design a compact and lightweight gear-based system by optimizing materials, geometry, and motion transfer to counter traditional bulkiness.
4. Ensure fail-safe operation through mechanical locks and interlocks that prevent premature or accidental arming.
5. Validate robustness and durability of the mechanism under harsh environmental conditions such as vibration, dust, and temperature variations. Provide a simple, predictable, and easy-to-inspect system with minimal failure modes for long-term reliability.

Objective No	Details
1	Designing a safety and arming device only and not the electrical components after the impact on the trigger switch
2	Quick, robust arming using mechanical components only to prevent a threat of cyber attacks and making the system extremely reliable.

Table 1.2 – Project core objectives

1.6 MOTIVATION

The use of loitering munitions in Operation Sindoor directly connects to the motivation for designing a safety and arming device for such systems. Operation Sindoor demonstrated the critical operational advantage provided by loitering munitions in delivering precise, autonomous strikes in complex and sensitive environments. However, the success of these missions hinges on reliable mechanical safety and arming devices that prevent accidental detonations, ensure controlled activation, and safeguard both military personnel and civilians. By focusing on designing effective safety and arming mechanisms, this project aims to enhance the trustworthiness and operational reliability of loitering munitions, enabling their responsible and safe deployment in real-world missions like Operation Sindoor. Thus, the project is motivated by the need to support next-generation autonomous warfare technologies with robust safety solutions, ultimately contributing to safer and more effective defence capabilities.

CHAPTER 2
METHODOLOGY AND EXPERIMENTAL WORK

2.1 PROJECT EXECUTION STAGES

Stage-1 : The first step involves the construction of a mind map of how the entire safety and arming process will go along. This includes the components which will be used to provide for effective energy and motion transfer.

Stage-2 : The second stage involves the calculations relating to velocity, dimensions and tolerances of the components that will fit in the fuselage. The calculation must be detailed and must make sure that the component does not fail. Accordingly, the material can be chosen based on the its yield stress limits.

Stage-3 : The third stage involves the first geometric modelling of the mechanism in AutoDesk Fusion 360. All components including gears, shaft, slot and key, bearings, pin must be modelled to bring the mind map to a model. Analysis for the static stress failure can be performed on Fusion 360 or Ansys Workbench.

Stage-4 : The fourth stage involves ideation and critical thinking of how the design can be improved to reduce complexity, if any. Slowly, the model must be in such a way that it is ready for manufacturing and assembly. Tolerance must be added to several components keeping real life manufacturing in mind. Ready made parts are available in the market or Misumi and can be ordered to reduce the lead time.

Stage-5 : Once the final model is ready, the manufacturing process can be started with the parts which were ordered. As the model is detailed, the assembly won’t take a lot of time and will be completed with ease. One must make sure that minimal rework and machining must be needed.

2.2 TECHNICAL SPECIFICATIONS

The final mechanism thought of is completely mechanically operated and comprises of:

- A turbine, with bucket on the blades, resembling an ideal Pelton turbine.
- A reduction compound gear train giving a velocity reduction of 64 : 1 ratio.
- A shaft with slot and keys on which the gears will be mounted.
- A pressure relief valve 0.1 psi at the opening of the pitot tube to allow the air to enter at the specific value.
- A lead screw with a custom flange which houses the striker.
- A trigger switch supported by an aluminium bracket.
- Multiple pin collar bearings to support the individual rotating shafts.

2.3 FLOW DIAGRAM OF THE ENTIRE PROCESS

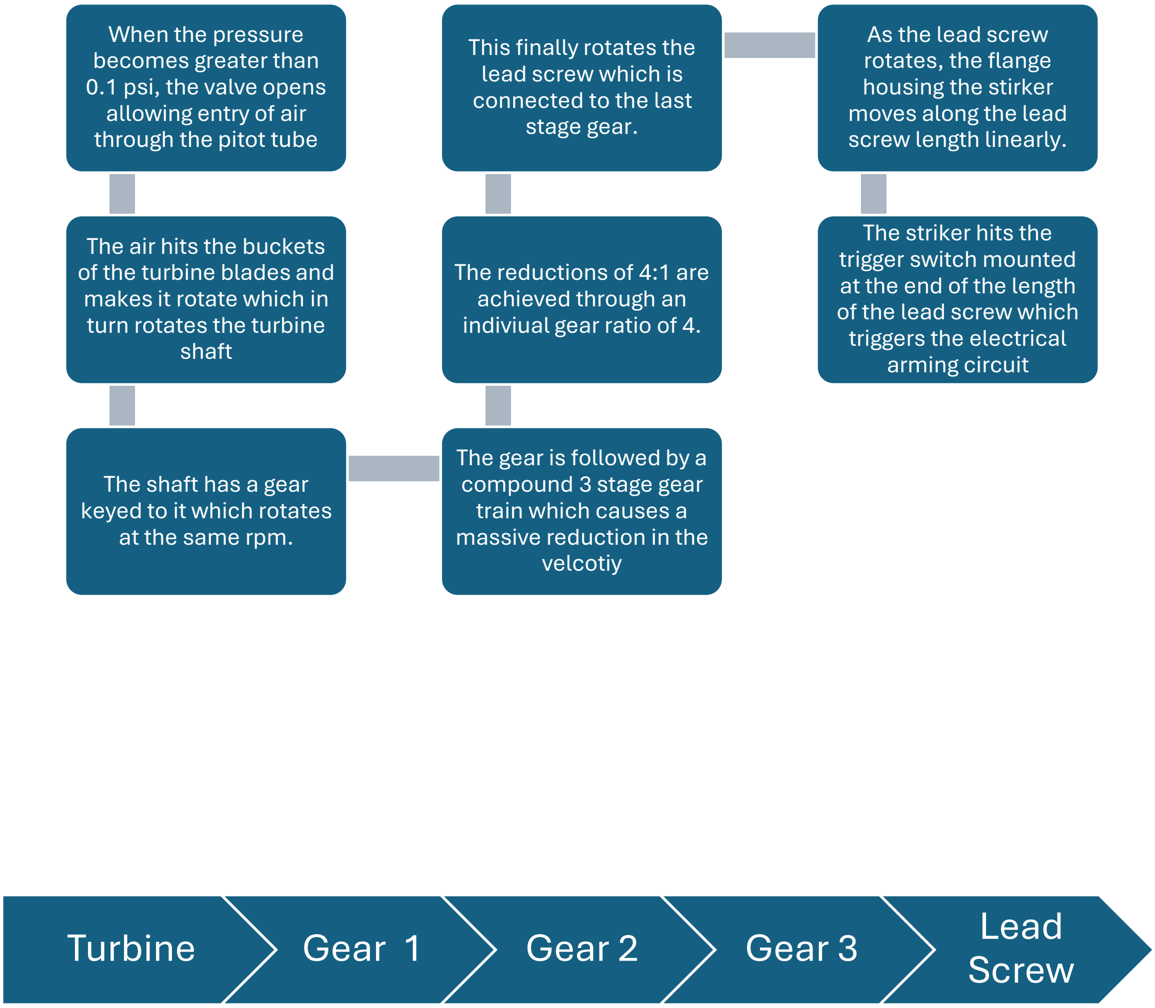

Figure 1.1 Flow diagram of the process (top) , Schematic of the component (down)

2.3.1 MATHEMATICAL ANALYSIS AND FEASIBILITY

Mathematical analysis and feasibility checks are critical in mechanism design because they ensure that the concept will actually function as intended in real-world conditions. While a design may look correct on paper or in CAD, without calculations, there is no guarantee that the mechanism will produce the required motion, speed, or force. For example, a gear train designed without analyzing the gear ratios might rotate too slowly or too quickly to achieve the desired output, rendering the design ineffective.

Another major reason for performing these analyses is to prevent mechanical failure. Every component in a mechanism is subjected to forces, moments, stresses, and deflections. Mathematical calculations help predict whether parts such as shafts, gears, or beams can withstand these loads without breaking or deforming excessively. This includes checks for shaft strength, gear tooth stresses, beam bending, and fatigue life under repeated loading, ensuring that the mechanism remains safe and reliable during operation.

Feasibility checks also help optimize the design. Through careful calculations, engineers can select the appropriate materials, dimensions, and configurations, avoiding both overdesign and underdesign. Overdesign adds unnecessary weight and cost, while under design risks failure. For instance, determining the minimum shaft diameter that prevents excessive deflection allows for a lighter, more efficient design without compromising strength.

Several calculations for dimensioning, fatigue stress checks were done before finalizing this mechanism.

2.4 MATERIAL SELECTION

Several materials were chosen to be suitable for the mechanism. Due to high torsion and force, a material with a higher Yield strength was chosen UHSS (Ultra high strength steel) with a yield stress of 1200-1800 MPa. It was also suitable to be used for the manufacturing of the turbine, shaft, and the gears over Aluminium alloys (7075-T6), Carbon Fibre composites (CFRP), Tool steel (D2,O1), and titanium alloys (Ti-6Al-4V) due to the following reasons : [2]

Table 1.3 Material Selection Based on Parameters

Material	Density	Yield Strength (MPa)	Fatigue & Impact Resistance	Machinability & Cost	Thermal Stability	Selection Status
UHSS	7.8	1200-2000	Excellent fatigue resistance, high toughness under vibration	Moderate machining, but can be machined efficiently by CNC	Very stable up to 500 °C	SELECTED
Aluminium alloys (7075-T6)	2.1	500-600	Good fatigue resistance but poor impact strength	Very easy to machine, low cost	Unstable after 150°C	REJECTED
Carbon Fibre composites (CFRP)	1.6	600-1200	Can severely cause a brittle failure	Expensive fabrication	Extremely stable	REJECTED
Tool steel (D2,O1)	7.8	900-1500	Good strength but brittle and low impact resistance	Hard machining, high cost	Stable upto 400°C	REJECTED
Titanium alloys (Ti-6Al-4V)	4.5	900-1100	Excellent strength-to-weight ratio, good fatigue	Poor machinability, very expensive	Stable upto 400°C	REJECTED

In the above table, yield strength, machinability, and impact resistance are the most important properties needed in the material that is to be chosen for the suggested mechanism.
Therefore, UHSS was chosen because –

- Superior strength-to-fatigue ratio → essential for rotating shafts and small turbine buckets.
- Toughness and impact resistance ensure no crack propagation under shock loading (gusts, debris).
- Dimensional stability under temperature variations and speed fluctuation.
- Economical for small precision parts compared to exotic alloys.
- Easy to surface finish and balance for dynamic components.

As UHSS is also moderately difficult to manufacture, to minimize the usage, the lead screw (connected to the last gear in the train) is made of aluminium. As the lead screw is rotating at the least rpm, aluminium can be used as vibrations will significantly decrease.
Reasons for choosing aluminium –

- Maximum torque that aluminium shaft can withstand >> required torque
- Sufficient torsional strength
- Lightweight (reduces inertia & critical speed issues)
- Easy and low cost machining
- Adequate fatigue for low loads

2.5 TURBINE AND GEAR TRAIN MATHEMATICAL CALCULATIONS

Arming must take place at the time of launch so as to keep the vehicle independent of air speed while in flight. The launch will be a "catapult launch" at a speed of v=30m/s. There will be a pressure valve (0.1 psi ~ 32-33 m/s) at the entry of the pitot tube to allow air to come in only after the launch and not during the transportation phase. After the entry into the pitot tube, it will hit the bucketed turbine and rotate thus, signalling a rotation of the other components too.

2.5.1 Turbine and shaft

Assumptions for the turbine radius and the number of buckets can only be made; rest of the components, such as the shaft diameter, gear thickness, and gear pitch diameter, must be calculated according to the yield stress limits.

Step-1 : Calculating the dynamic pressure (at v= 30 m/s)
Based on the initial assumptions, the turbine radius has been fixed at 0.15 m, and the bucket radius being 0.035 m. The formula relating dynamic pressure (q), density of air (ρ), and velocity of air at inlet (V)

$$q = \frac{1}{2}\rho V^2$$

density of air at sea level = 1.225kg/m^3
velocity of air entering the pitot = 30 m/s
bucket frontal area = $\pi(0.035)^2$
Upon calculation, we get the value of the dynamic pressure component as 551.25 Pa

Step-2 : Calculating the tangential force on the turbine blade
The formula relating tangential force (F), dynamic pressure (q), projected area (πr^2= A)

$$F_t = C_T \mathrm{q}\mathrm{A}$$

Dynamic Pressure (q) = 551.25 Pa
Bucket frontal Area (A) = 0.0038 m^2
$(Constant)\ C_t = 1.5$ is called as the thrust coefficient. The value has been taken as 1.5 as it is most efficient for a Pelton bucketed turbine.

$F_t = 1.5 \times 551.25 \times 0.0038 = 3.6945$ N ; therefore the value of tangential force acting on the bucket is 3.6945 N.

Step-3 : Calculating the torque generated by the turbine

The formula relating torque ($T_{(turbine)}$), tangential force (F_t), and turbine radius (r)

$$T_{(turbine)} = F_t \times r$$

Tangential force (F_t) = 3.6945 N
Turbine radius (r) = 0.15 m
Substituting the values-
$T_{(turbine)}$ = 3.6945×0.15=0.5564 Nm; Therefore the torque generated by the turbine is 0.5564 Nm.

Step-4 : Turbine angular speed

Power developed by the tangential force

$$F_t \times V$$

Substituting the values we get,
Power developed = 3.6945 × 30 = 110.825 W

P = $T_t \times \omega_t$
Therefore,
The formula relating angular velocity to torque and power is

$$\omega_t = P/T_t$$

$\omega_t = 110.835/0.5564$
= 199.20 rad/s ~ 200 rad/s
There, the angular velocity of the rotating turbine is 200 rad/s
The turbine revolutions per minute (N)

$$N = \omega_t \times \frac{60}{2\pi}$$

The turbine rpm is calculated to be ~ 1910 rpm
Step-5 : Centrifugal, bending , and combined stress analysis

Overall Torque (Maximum) = $n \times F_r \times r$ = 6 x 3.6945 x 0.15 = 3.325 Nm ;
n = Number of buckets = 6
Force per bucket = 3.6945 N ~ 3.7 N
Stress on the bucket due to the tangential force only ;

$$\frac{F_t}{A_{(frontal)}}$$

Frontal projected bucket area = $\pi(0.035)^2$
Therefore, the stress on the bucket is 959.99 ~ 960 Pa
This stress is really small compared to the yield stress of UHSS (upper and lower) ~ 1400 MPa
Three main stress acting on the turbine frame and blades

a. Bending stress – Tangential air impact on the bucket
b. Centrifugal Stress – High rotational speed
c. Shear Stress – Torque Transmission to shaft

1) **Bending stress on bucket root**

l_b = point where the force acts on the bucket root = 0.035 m

$\boldsymbol{M = F_t \times l_b}$ = 3.7 x 0.035 = 0.00453 N-m

$$Z = \frac{6M}{0.015 \times (0.005)^2} = 6.25 \times 10^{(-8)}$$ m^3

$$\sigma_b = M/Z = \frac{0.1295}{6.25 \times 10^{(-8)}} = 2.072\ MPa$$

This is the stress per bucket which is really safe as per UHSS.

2) **Centrifugal stress on the disk (solid disk)**

$$\sigma_c = \frac{3 + v}{8} \times \rho \times \omega^2 \times r^2$$

Value of Poissoon ratio = 0.3
Angular Velocity = 200 rad/s
Radius of turbine = 0.15 m
Centrifugal stress (solid disk) = 2895750 N/m2 = 2.8 MPa

This is really safe for UHSS stress limits

3) **Torsional shear stress limits in shaft hub**

The shaft diameter is 0.07m

$\tau_t = \frac{16\,T}{\pi d^3}$ = 0.49 MPa

This is again really safe from UHSS
THEREFORE, THE ENTIRE TURBINE DESIGN IS SAFE

Step-6 : Static stress analysis on Autodesk Fusion 360

The aim of this analysis was to determine the stress distribution on the turbine blades and hub when subjected to aerodynamic impact forces generated at a flight velocity of **30 m/s**. This analysis helps verify whether the chosen material (**UHSS**) and geometry can safely withstand the operating loads without structural failure or excessive deformation.

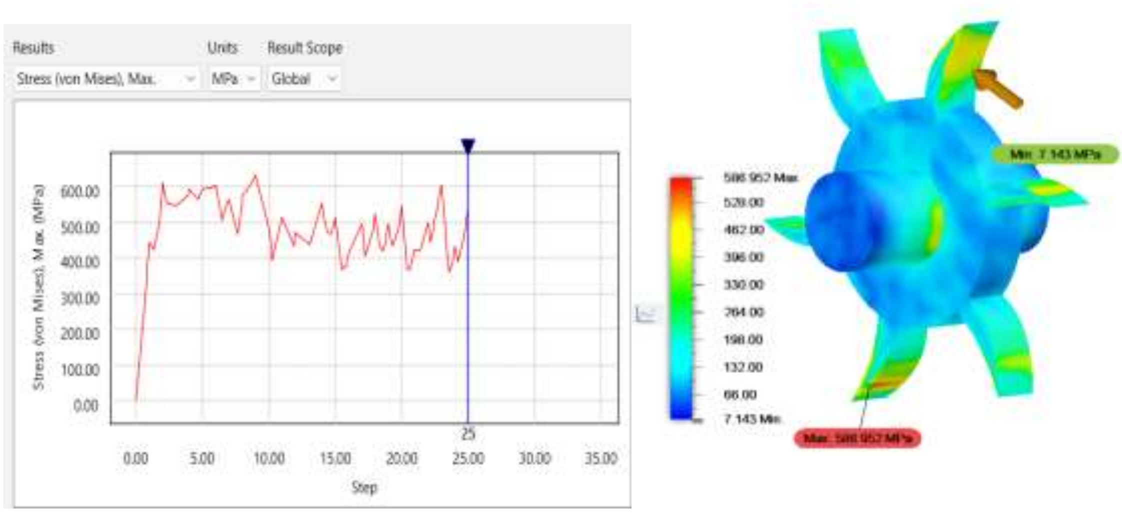

Figure 1.2 – Stepwise stress impact on the turbine (left), Turbine stress profile (right)

The following analysis on Autodesk Fusion 360 for static stress, shows that the turbine is exposed to various levels on stress throughout its body as well as the shaft hub section. The maximum stress is 586.9 MPa, which acts at the point where the air will be striking will the bucket. High local stress due to dynamic pressure and curvature. This stress is a cumulative stress due to the bending stresses, centrifugal stresses and the tangential force which acts on the turbine blade normal to the frontal blade area.
The minimum stress is acting on the turbine shaft of 70 mm diameter. This stress accounts for torsional stress, centrifugal stress and the shear stress at the shaft turbine hub.
Overall, the entire turbine-shaft design is safe considering the yield stress limits of UHSS.

2.5.2 Gear train, gear tooth and shafts

Step-1: Determining the required gear module and face width
Bore hole >= 40mm
Hub diameter D = 1.5 (40) = 60 mm
Pitch diameter of the pinion >= 60mm
Taking 60mm →
mZ =60 , m is the module and Z is the number of teeth. Due to high stresses and high rpm values we will take a higher module so assuming the module as 3.
Therefore, the number of teeth on the pinion is 20. As the gear train must be of a gear ratio of 1 : 4, therefore the number of teeth on the main gear is 80; and the main gear must be of a pitch diameter of 240mm.

Table 2.1 – Component parameter information

Component (Gear)	Face Width(considering high T)	Number of teeth
Gear	30mm	20
Pinion	30mm	80

Step-2 : Stage wise gear speeds and torques
As the input torque for the first stage is 3.325 N-m, the torque keeps increasing by a multiple of 4 across each stage and the angular velocity and rpm decreases by the same amount.

Table 2.2 – Stage wise torque values

Stage	1	2	3	End of 3 (final)
Torque (N-m)	3.325	13.3	53.2	212.8
rpm	1910	477.5	119.4	29.85

Step-3 : Gear tooth calculations
The main forces and stresses acting on the gear tooth are as follows

1. Bending stress
2. Tangential force
3. Contact stress between the gear tooth

1) Bending stress acting on the gear tooth (Lewis equation) and Tangential force

$$\sigma_b = \frac{F_t}{bmY}$$

F= Tangential force acting on the teeth
b= Gear face width
m= module
Y= Lewis factor (0.320 for 20 pinion teeth and 20° pressure angle)
The tangential force and bending stress for each stage can be calculated as follows

$$F_t = \frac{2T}{d_p}$$

T= Input torque , d =pitch diameter of the pinion
Calculating for each stage ,

Table 2.3 – Tangential force and bending stress per stage

Stage	1	2	3
Tangential force (N)	110.8	443.3	1773.3
Bending stress (MPa)	3.847	`16.4	65.7

2) Calculation of the contact stresses at the surface of the gear tooth

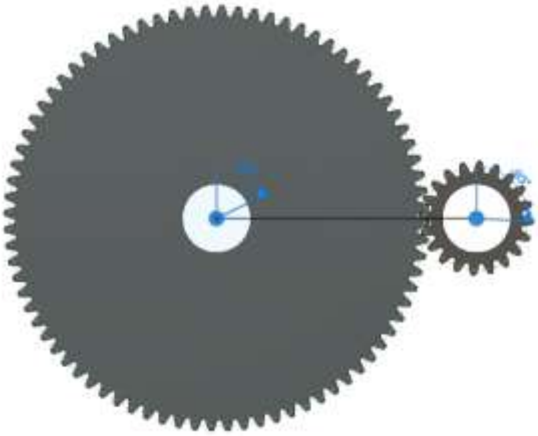

Figure 1.3 – Gear meshing pinion and main gear (20/80 =1/4 gear ratio)

To calculate the Hertzian contact stresses, the following formula is used

$$\sigma_H = Z\sqrt{\frac{F_t \times K_1 \times K_2 \times K_S}{b d_p Z_1}}$$

Z = elastic coefficient (191 for steel – steel contant) , b = 30mm , dp = 60mm

K1 = 1.25

K2 = 1.1

K3 = 1.0

Z 1 = 0.12

F_t= Tangential force at different stages

The calculated value for the denominator is 216 , and the numerator of the fraction is 1.375

Table 2.4 – Contact stress values for each gear stage

Stage	Fraction value	Square root * 191	Stress Value (MPa)
1	0.705	160.4	160.4
2	2.82	320.9	320.9
3	11.28	641.8	641.8

Step-4 : Shaft torsional stress check

$$\boldsymbol{\tau = \frac{16T}{\pi d^3}}$$

All shafts are of 40mm diameter, therefore calculating the values for torsional stress for all stages

Table 2.5- Torsional stress (in MPa) per stage

Stage	Torque (N-m)	Torsional Stress (MPa)
1	3.325	0.264
2	13.3	1.058
3	53.2	4.23

The output torque on the lead screw is 212.8 N-m, and the torsional stress acting on the lead screw is equal to 16.99 MPa. All the values are within the safety limits of UHSS yield stress limits.

Step-5 : Safety (Static stress) Analysis of the shafts across all the stages (Fusion 360)

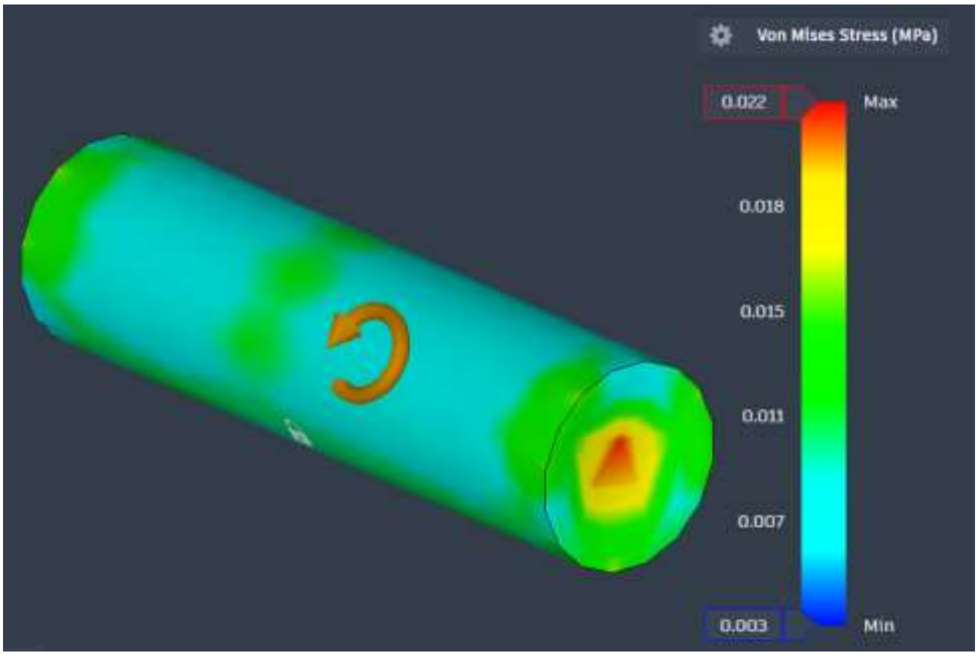

Figure 1.4 – First stage shaft analysis

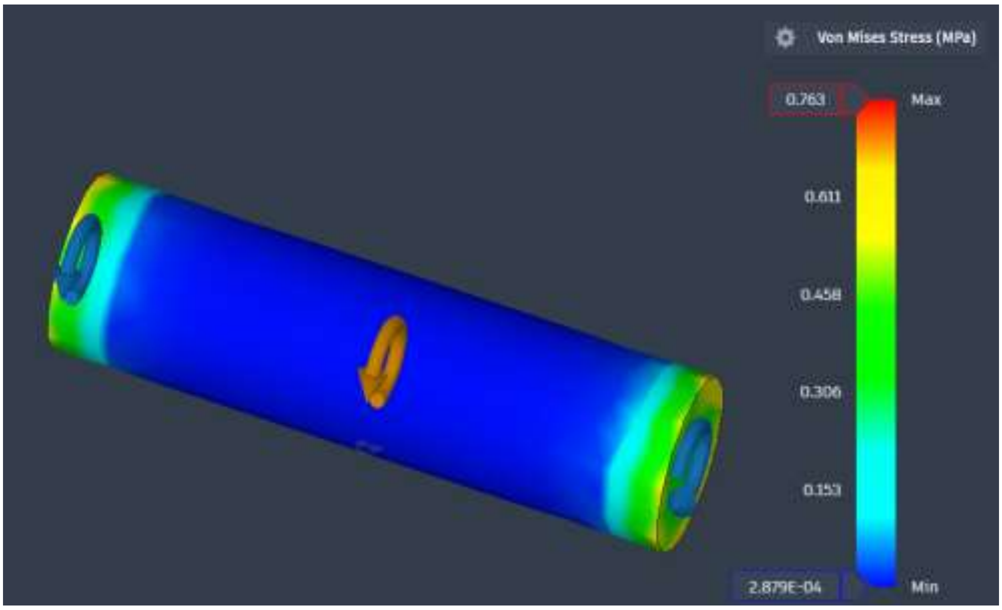

Figure 1.5 – Second stage shaft analysis

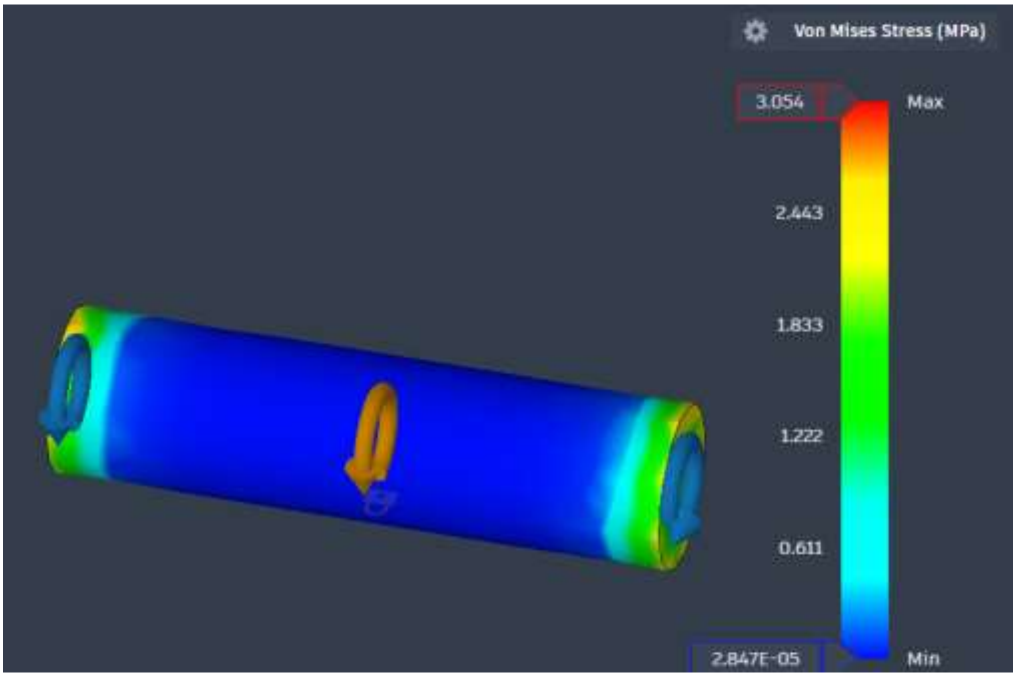

Figure 1.6 – Third stage shaft analysis

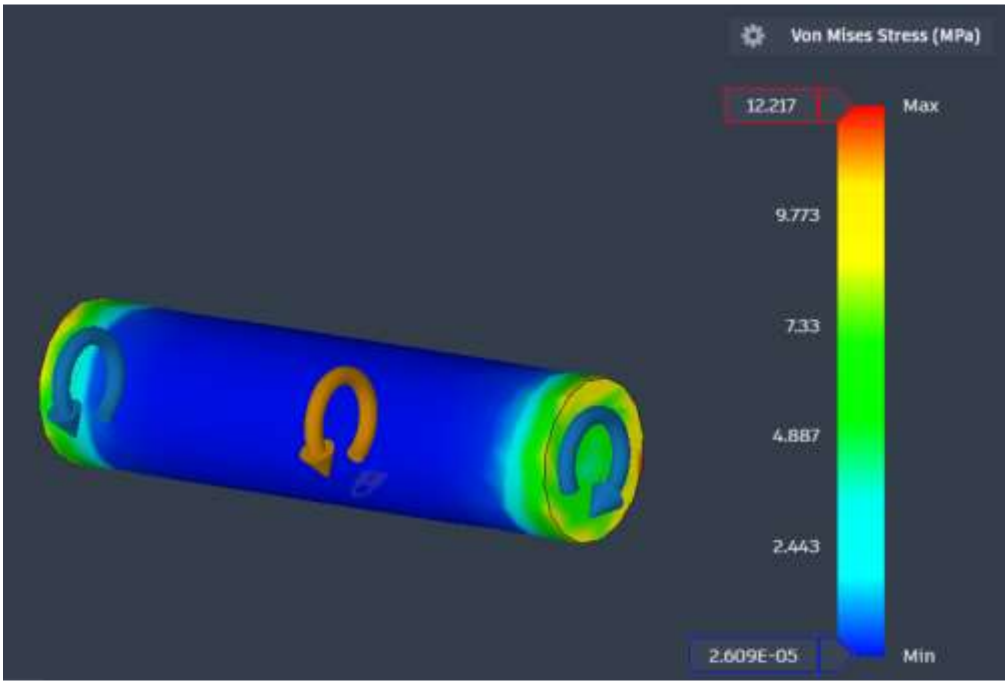

Figure 1.7 – Fourth stage shaft analysis

Step-6 Shaft deflection (angle of twist) mathematical analysis

$$\theta = \frac{TL}{GJ}$$

G = Shear modulus of the material ~ 80 GPa

J = Polar moment of inertia = $\frac{\pi d^4}{32}$

Solving the equation we can find the angle of twist at each gear stage. However finding only the last stage angle of twist is important as that stage has maximum torque value 0.091° which is negligible. Therefore stable as per UHSS stress limits.

Table 2.6 – Safety analysis of the components

Component	Safety Analysis result
Gear Tooth	SAFE
Shaft (Torsion)	SAFE
Shaft (angle of twist)	SAFE

2.6 LEAD SCREW AND STRIKER FLANGE & SHAFT KEYWAYS

2.6.1 Lead screw and striker flange

The final step in the arming process is achieved through a lead screw and striker assembly, which transforms the rotational motion from the final gear stage into a controlled linear movement. The screw mechanism provides a 10 mm linear stroke, enough to push the striker and trigger the arming switch within a short, smooth sequence. This approach eliminates the need for electrical actuation, resulting in a purely mechanical and interference-free system.

The lead screw is manufactured from Aluminium 6061-T6, selected for its light weight, high machinability, and excellent balance between strength and stiffness.[13] It has a mean diameter of 12 mm and a pitch of 3 mm, producing 3 mm of linear motion per revolution. At the gear train's output speed of around 29.8 rpm, the screw generates a linear velocity of 1.49 mm/s, completing the 10 mm travel in roughly 6.7 seconds. This duration is ideal for steady and reliable arming. If faster operation is needed, a coarser pitch or a smaller reduction ratio could be used to shorten the arming time. The screw moves through a bronze or PTFE nut, providing low-friction and wear-resistant operation. Its self-locking thread geometry ensures that once the striker is extended, it remains securely in position without any back-driving. The assembly requires minimal lubrication, operates quietly, and is designed to function repeatedly without performance degradation, making it suitable for long-term use.

Torque from the final gear-shaft is transmitted to the lead screw through a custom-machined aluminium flange that ensures perfect alignment and strong torque transfer. The flange, made from Aluminium 6061-T6, is mounted on the 40 mm output shaft via a keyed fit and M6 bolt set. It features an outer diameter of 70 mm, a bolt circle diameter of 60 mm, and a thickness of 10 mm, providing a strong and compact coupling. Four equally spaced M6 bolts secure the flange, giving it a torque capacity of about 720 N-m, which is well above the operating torque of 213 N-m.

The striker pin is attached to the nut on the lead screw's moving end, and the flange maintains coaxial alignment between the rotating screw and the striker assembly. This precise alignment minimizes vibration and mechanical noise while reducing bending loads on the screw. The design also allows for quick disassembly during maintenance or testing. By keeping the loads centered along the shaft axis, the flange contributes to smoother operation and longer component life.

2.6.2 Shaft keyways

The shafts transfer torque between the turbine-driven gears, collars, and couplings. To make sure there is a secure and slip-free connection between each rotating shaft and its mounted component, a parallel key and keyway system is used. This setup provides a firm mechanical link that prevents the gear or collar from rotating independently of the shaft, while still allowing easy assembly and disassembly when needed for maintenance. Compared to press fits or set screws, the key and keyway arrangement is simpler, stronger, and far more reliable for handling high torque loads.

All shafts are made of Ultra High Strength Steel (UHSS) and have a diameter of 40 mm. Based on standard machine design references such as ISO 773, BS 4235, and IS 2292, the appropriate key size for this shaft diameter is 12 mm wide and 8 mm high. The key itself is made from hardened steel, such as EN8 or C45, to ensure long-term durability and resistance to wear under repeated torque cycles. The keyway slots are precisely machined into both the shaft and the mating gear hub with tight tolerances to provide a smooth sliding fit during assembly.

These dimensions were chosen to safely transmit the maximum output torque of 213 N-m from the gear train while maintaining a safety factor greater than 3. Since the hub of each gear is 30 mm long, the key length was also taken as 30 mm, ensuring full contact along the hub surface. This design choice guarantees efficient torque transfer, long service life, and straightforward maintenance , essential for reliable operation of the turbine arming mechanism.

2.7 PIN COLLAR BEARING & OUTER CASING

2.7.1 Pin collar

A pin collar joint (or pin collar bearing) is a simple mechanical support arrangement where a shaft is supported and restrained axially by a collar and a transverse pin.

It is used in places where:

- Moderate radial and thrust loads are present,
- Compactness is essential,
- And the rotation speed is not extremely high.

Essentially, it acts as a bearing and axial restraint in one compact element.

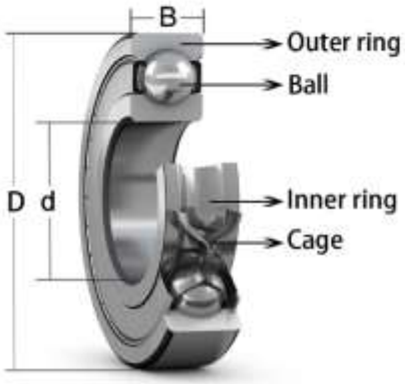

Figure 1.8 – Ball bearing

The shaft rotates freely inside the bearing housing or bush. A collar is fixed to the shaft (either by press fit, key, or set screw). A transverse pin passes through both the collar and the shaft, locking them together. This prevents axial motion while allowing rotation.

Table 2.7 – Dimensions of the pin collar bearing

Component	Symbol	Description	Value
Shaft diameter	d	Actual rotating member	40 mm
Collar outer diameter	D	Provides bearing contact surface	60-65 mm
Collar thickness	t	Ensures shear strength & rigidity	15 mm
Pin diameter	dp		8 mm

		Resists torque & axial thrust	
Pin material	-	Hardened steel (AISI 1045 or EN8)	Yield stress greater than 400 MPa

2.7.2 Outer casing

The turbine and three-stage gear train's external enclosure is a cuboidal shell made of aluminum 6061-T6 with a consistent wall thickness of 6 mm. Aluminium 6061-T6 was chosen for prototype work due to its superior stiffness-to-weight ratio, good machinability for bosses and bearing housings, corrosion resistance (which can be further enhanced by anodizing), and ease of manufacturing. The three inline gear stages (each with a center distance of about 150 mm), 30 mm face width gears, 40 mm diameter shafts, and bearing housings are all easily accommodated inside the 550 × 120 × 300 mm internal working envelope, which also provides service clearance for lubrication and inspection.

Outer length:

$$L_o = L_i + 2t = 550 \text{ mm} + 2(6 \text{ mm}) = 550 + 12 = 562 \text{ mm}$$

Outer width:

$$W_o = W_i + 2t = 120 \text{ mm} + 12 = 132 \text{ mm}$$

Outer height:

$$H_o = H_i + 2t = 300 \text{ mm} + 12 = 312 \text{ mm}$$

$$L_o \times W_o \times H_o = 562 \text{ mm} \times 132 \text{ mm} \times 312 \text{ mm}$$

CHAPTER 3

RESULTS AND DISCUSSION

3.1 Design overall safety and manufacturability

The turbine-based arming mechanism was designed, analysed, and tested through detailed mechanical calculations and simulation results. The turbine, operating under an airflow of 30 m/s, reached a speed of about 1910 rpm and produced 3.325 N·m of torque. This torque was transmitted through a three-stage spur gear system with a total reduction ratio of 64:1, increasing the final torque to around 213 N·m while reducing the speed to nearly 30 rpm. This rotational output was delivered to a lead screw through shafts supported by pin-collar bearings, converting the rotary motion into linear motion to push the striker and arm the circuit. With a 12 mm mean diameter and a 3 mm pitch, the lead screw achieved a striker motion of 3 mm in 2 seconds meeting the arming time target effectively.

The static stress simulation of the turbine in Autodesk Fusion 360 showed that the maximum stress (586.9 MPa) occurred near the bucket tips, while the minimum stress (7 MPa) was seen near the hub. Since the UHSS turbine material can withstand up to 1400 MPa, the stress levels were well within safe limits, giving a healthy safety factor of around 2.4. Similarly, stress checks on the shafts and gears showed that both bending and contact stresses remained far below the allowable limits. The aluminium casing, sized 562 × 132 × 312 mm with a 6 mm wall thickness, proved strong yet lightweight, weighing roughly 9 kg. Together, the analytical and simulation results confirm that the overall system is structurally safe, efficient, and capable of reliable operation under the intended aerodynamic loads.

CHAPTER 4
CONCLUSIONS
4.1 Contributions to the literature

The design and analysis of the turbine-driven mechanical arming mechanism were successfully carried out and confirmed to be both functional and feasible for real-world use. The chosen design featuring UHSS turbine blades, a three-stage 64:1 gear train, 40 mm shafts, and an aluminium lead screw worked together seamlessly to meet the system's performance goals while maintaining strong safety margins. The turbine produced sufficient torque at an airflow speed of 30 m/s to drive the entire mechanical setup through the compact gear reduction system. The lead screw achieved the required arming action within two seconds, showing that the system responds quickly and efficiently without relying on any electrical components, making it inherently secure against hacking or interference. Further structural and material analyses confirmed the reliability of the overall design. All the main components like gears, shafts, bearings, and the turbine which operated well below their stress limits, showing excellent stress distribution and minimal deformation. The pin-collar bearing setup provided smooth rotation and strong axial support, while also simplifying assembly. The lightweight aluminium casing added both stiffness and corrosion resistance, helping keep the mechanism durable yet easy to maintain.

4.2 Future scope

While the current turbine-driven mechanical arming mechanism performs reliably and meets all design objectives, there are several technical areas that can be explored to further improve its performance and make it more adaptable for future applications. One important direction is to enhance the aerodynamic design of the turbine. Using Computational Fluid Dynamics (CFD) simulations, the blade curvature, pitch angle, and bucket geometry can be optimized to generate higher torque even at lower airspeeds. Material optimization is another promising area — replacing UHSS with lightweight composites like carbon-fiber-reinforced polymer (CFRP) or titanium alloys could reduce the system's mass and rotational inertia, improving both responsiveness and efficiency.

Future work can also focus on refining the gear train design to make it more compact and efficient. The use of planetary or harmonic gear systems could offer smoother motion, reduced vibration, and better space utilization compared to conventional spur gears. Similarly, the lead screw assembly could be upgraded to a ball-screw mechanism to increase linear motion efficiency and precision at higher speeds. To improve safety and monitoring, non-intrusive mechanical sensors or limit switches could be integrated to detect the exact arming position without affecting the purely mechanical nature of the system.

The aluminium casing design can be further optimized using Finite Element Analysis (FEA) to identify areas where material can be reduced without sacrificing strength. The use of additive manufacturing (metal 3D printing) could also be explored to create lightweight structures with built-in reinforcements or lubrication channels. Implementing these improvements would help develop a more compact, efficient, and production-ready version of the arming mechanism, making it suitable for advanced drone systems and aerospace defense applications.

REFERENCES

1. SteelPro Group. "Ultra High Strength Steel | UHSS." Details tensile strength range from 980 MPa to over 1500 MPa with high durability and toughness. (2024)
2. Klöckner Metals. "Types of High Strength Steels in Cars." Notes steels with tensile strength exceeding 780 MPa categorized as UHSS and automotive UHSS grades called "GigaPascal steel." (2024)

3. World Auto Steel. "Advanced High-Strength Steel (AHSS) Definitions." Defines UHSS with tensile strength starting at 780 MPa and highlights grades over 1000 MPa used in automotive applications. (2024)
4. Phase Transformations Group, University of Cambridge. "Mechanical and Technological Properties of High Strength Steel." Reports UHSS plates with yield strengths over 900 MPa and good toughness.
5. ScienceDirect. Research articles on "Phase transformation and mechanical properties of ultrahigh-strength steels" showing tensile strengths from 1400 to 2000 MPa with reasonable ductility and toughness.
6. Wikipedia (2005). "Torsion (mechanics)." This resource explains shear stress in rotating shafts, with formulas relating torque, shear stress, and shaft dimensions. It includes an example calculation for a steam turbine shaft, highlighting the relationship between torque, power, and shear stress. It also discusses how the highest shear stress occurs at the shaft surface and design considerations like surface polishing for durability.
7. Lin et al. (2024). "Investigation on Mechanical Properties of MAR-M247 Superalloy." This study focuses on turbine blades, analyzing mechanical properties including strength and stress distribution through finite element simulation. It covers the impacts of centrifugal, thermal, and aerodynamic loads on turbine blade strength.
8. Zhang et al. (2018). "Dynamic analysis of offshore wind turbine blades under fluid-structure interaction and wind shear." This paper analyzes wind shear effects on turbine blades, showing how shear stresses affect displacement and stress distribution along the blade.
9. Helix Linear Technologies: Aluminum lead screws are ideal for lightweight and non-magnetic applications found in medical (MRI machines), aerospace (airline seat actuators, drone flight control), and transportation (bus and train doors). Aluminum lead screws offer weight reduction and cost benefits while maintaining precision and performance, sometimes coated with Teflon or anodized for durability.
10. Madearia.com: Lead screws, including those made from aluminum, convert rotary motion to linear motion. Aluminum lead screws are chosen based on material properties for durability, strength, and cost-effectiveness in high-force transmission applications such as CNC machines, automation, and aerospace. Material selection is critical for performance under load.
11. PowerTransmissionWorld.com: Aluminum lead screws combine lightness with strength and magnetic neutrality, suitable for precise injector drive units in MRI contrast agent delivery systems. This highlights specialized uses where aluminum's properties are advantageous.
12. Eichenberger.com: Cold-formed aluminum lead screws are used as mechanical drive components in automotive door systems and other applications where lightweight and precise linear actuation is required. This cold-forming process strengthens aluminum for effective use in lead screws.
13. These sources collectively emphasize aluminum lead screws' lightweight, non-magnetic properties and application in medical, aerospace, and transport industries, with precision and durability through coatings and manufacturing methods.