

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL

**COMMON GEARBOX DESIGN FOR PUSHER AND TRACTOR
TYPE PROPULSION SYSTEMS IN UNMANNED AIR
VEHICLES**

İBRAHİM ETHEM KAZANCI

**A THESIS OF MASTER OF SCIENCE
DEPARTMENT OF ENERGY TECHNOLOGIES
IN APPLIED PROPULSION SYSTEM DESIGN AND
ENGINEERING FOR AEROSPACE TECHNOLOGIES
PROGRAM**

ADVISOR: PROF. DR. FEVZİ BEDİR

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T.C.
GEBZE TEKNİK ÜNİVERSİTESİ
LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ

İNSANSIZ HAVA ARAÇLARINDA İTİCİ VE ÇEKİCİ
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İBRAHİM ETHEM KAZANCI

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JURY

MEMBER

(THESIS ADVISOR) : Prof. Dr. Fevzi BEDİR

MEMBER

: Prof. Dr. Süleyman KARSLI

MEMBER

: Prof. Dr. Muhammet KAYFECİ

APPROVAL

Gebze Technical University Graduate Administration Board

...../...../..... date and/..... numbered decision.

SIGNATURE/SEAL

*To my
Family and
Wife*

ABSTRACT

Unmanned Aerial Vehicles (UAVs) have many military and civilian areas of usage and provide superiority, especially in military operations. UAVs discharge tasks such as reconnaissance, surveillance, and attack by providing access to challenging and hazardous areas in military areas without risking human lives. Due to long flight durations, payload capacities, and operational costs, UAVs equipped with piston engines are frequently utilized. These types of UAVs generally operate with pusher and tractor type propulsion systems. In both types of propulsion systems, gearboxes are used as power transmission systems between the engine and the propeller. Fundamentally, gearboxes enable efficient and controlled flight under variable load and, altitude conditions by a specific transmission ratio. Additionally, as with each system, cost, maintenance, and maintainability are critical considerations for gearboxes. In line with these needs, a common gearbox design has been developed to integrate with the same motor and propeller in both pusher and tractor type propulsion systems. Initially, the literature review was conducted and information was provided about UAV types and propulsion systems in the study. Pusher and tractor type propulsion systems are detailed and system layouts, comparisons, advantages, and disadvantages are emphasized. During the preliminary design stage system requirements, motor and propeller specifications, system assumptions, and material properties have been determined. In the light of this data, design of gearbox started with the creation of system kinematics using the KISSsys program. After that, the design and selection of gears, bearings, shafts, and other elements were studied using the KISSsoft program. Detailed information is given about input-output and parameters to be considered during the design phase, and criteria for the selection of bearings and other equipment are specified. In final phase, the gearbox casings have been designed in accordance with equipment specifications and the motor-propeller assembly interface. The requirement of commonality has been taken into consideration in all studies and the gearbox that can work pusher and tractor type propulsion system has been completed.

Keywords: UAV, Propulsion, Pusher, Tractor, Gearbox

ÖZET

İnsansız Hava Araçları (İHA'lar) askeri ve sivil birçok kullanım alanına sahip olup özellikle askeri operasyonlarda üstünlük sağlamaktadır. Zorlu ve tehlikeli bölgelere insan hayatını riske atmadan erişim sağlayarak keşif, gözetleme, taarruz gibi görevleri yerine getirmektedir. Uzun uçuş süreleri, faydalı yük taşıma kapasiteleri ve uçuş maliyetlerinden dolayı pistonlu motora sahip hava araçları sıklıkla kullanılmaktadır. Bu tip İHA'lar genellikle itici veya çekici tip itki sistemleri ile faaliyet göstermektedirler. İki tür itki sisteminde de dişli kutuları motor ile pervane arasında güç aktarma sistemi olarak kullanılmaktadır. Temel olarak dişli kutuları, değişken yük ve irtifa koşullarında belirli bir aktarma oranı ile kontrollü uçuş gerçekleştirilmesini sağlar. Ek olarak her sistemde olduğu gibi maliyet, bakım, idame edilebilirlik dişli kutuları için önem arz etmektedir. Bu ihtiyaçlar doğrultusunda itici ve çekici tip itki sistemleri için aynı motor ve pervane ekipmanlarına entegre edilmek üzere ortak dişli kutusu tasarımı çalışılmıştır. Çalışmada öncelikli olarak literatür araştırması yapılmış, İHA türleri ve itki sistemleri hakkında bilgi verilmiştir. İtici ve çekici tip itki sistemleri detaylandırılmış olup sistem yerleşimleri, karşılaştırmaları, avantaj ve dezavantajları üzerinde durulmuştur. Ön tasarım aşamasında sistem gereksinimleri, motor ve pervane özellikleri, sistem kabulleri, malzeme özellikleri belirlenmiştir. Bu veriler ışığında dişli kutusu tasarımı KISSsys programı kullanılarak sistem kinematiğinin oluşturulması ile başlamıştır. Ardından dişli, rulman, şaft ve diğer elemanların tasarım ve seçimleri KISSsoft programı kullanılarak çalışılmıştır. Tasarımlar yapılırken giriş-çıkış ve dikkat edilmesi gereken parametreler hakkında detaylı bilgi verilmiş, rulman ve diğer ekipmanların seçimi ile ilgili ölçütler belirtilmiştir. Son olarak dişli kutusu muhafazası ekipmanlara ve motor-pervane bağlantı arayüzüne göre tasarlanmıştır. Yapılan tüm çalışmalarda ortaklaştırma gereksinimi göz önünde bulundurulmuş olup itici ve çekici tip itki sistemi ile çalışabilecek dişli kutusu tasarımı tamamlanmıştır.

Anahtar Kelimeler: İHA, İtici, Çekici, Dişli Kutusu

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LIST OF SYMBOLS AND ABBREVIATIONS

W_k	: Actual base tangent length
sc	: Actual chordal tooth thickness
haP^*	: Addendum coefficient
F_A	: Axial force
W_k	: Base tangent length
sc	: Chordal tooth thickness
hfP^*	: Dedendum coefficient
C_r	: Dynamic load
DM_{eff}	: Effective diameter of ball/pin
F_e	: Equivalent force
b	: Face width
S_H	: Flank safety
β	: Helix angle
L_d	: Life of bearing
$MdR.e/i$	: Measurement over pins
W_n	: Normal force
m_n	: Normal module
z	: Number of teeth
k	: Number of tooth spanned
L_{10}	: One million revolutions life
α_{fn}	: Pressure angle at normal section
x	: Profile shift coefficient
W_t	: Radial force
ha	: Reference chordal height
d	: Reference diameter
C_{req}	: Required dynamic load
d_f	: Root diameter
ρ_{fP^*}	: Root radius factor
S_F	: Root safety
W_r	: Tangential force
d_a	: Tip diameter
ρ_{aP^*}	: Tip radius factor
Δ_{dmp}	: Tolerance for bearing bore
Δ_{Dmp}	: Tolerance for bearing outside diameter
$As.e/i$	: Tooth thickness allowance
AGMA	: American Gear Manufacturers Association
ANSYS	: Analysis system
ATA	: Air Transport Association
BAE	: British Aerospace
CAD	: Computer Aided Design
DIN	: German Institute for Standardization
FKM	: Fluorine Kautschuk Material
HRC	: Hardness Rockwell C Scale
ISO	: International Organization for Standardization
ISTAR	: Intelligence, Surveillance, Target Acquisition and Reconnaissance
NBR	: Nitrile Butadiene Rubber

OD	: Outer Diameter
PD	: Pitch Diameter
PTFE	: Polytetrafluoroethylene
PU	: Polyurethane
RPM	: Revolution Per Minute
SAE	: Society of Automotive Engineers
UAVs	: Unmanned Aerial Vehicles
UCAV	: Unmanned Combat Aerial Vehicle
VTOL	: Vertical Take-Off and Landing

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1. INTRODUCTION

In the contemporary world, unmanned aerial vehicles (UAVs) have a wide range of uses and gain day after day extensive utility. Accordingly, they are bolstering their efficacy across both military and civilian domains. UAVs discharge tasks such as reconnaissance, surveillance, and attack by providing access to challenging and hazardous areas in military areas without risking human lives.

In civilian areas, unmanned aerial vehicles (UAVs) are deployed across a myriad of diverse fields such as search and rescue, agriculture, wildfire monitoring, and natural disaster intervention and management. In this context, the types of UAVs to be selected or designed are determined based on mission activities and the desired flight profile. UAVs are detailed in the following sections according to their usage areas and other features, and in this study focused on piston-powered engines.

UAVs equipped with piston engines are capable of low fuel consumption and, therefore long mission endurance. Additionally, they are utilized for applications requiring high payload capacity. Piston engines are typically preferred in fixed-wing UAVs, necessitating gearbox systems that exhibit effective performance over wide power ranges.

Gearboxes are the power transmission element between the piston engine and the propeller. They are critical to effectively transfer the power produced by the engine to the propellers at the desired gear ratio, thereby increasing engine efficiency and optimizing fuel consumption. Gearboxes, especially for high-power engine types such as piston engines, enable more efficient and controlled flight performance under varying load conditions, different altitudes, and weather conditions. This capability allows UAVs to operate for extended periods in challenging environments, thereby enhancing operational efficiency.

1.1. Purpose, Contribution, and Content of the Thesis

The purpose of this thesis is to design a common gearbox for use in unmanned aerial vehicles (UAVs) with both pusher and tractor type propulsion systems. UAVs are

classified as either pusher or tractor types based on the configuration of the engine and propeller. The thrust forces generated by the propeller are transmitted through the gearbox to the motor and the aircraft's structural system. Since the thrust forces affect the gearbox in opposite directions, different designs are required. Instead of designing separate gearboxes for these two systems, creating a common gearbox that can be used in both propulsion systems offer significant advantages in terms of simplifying the engineering and manufacturing processes. The goal of this design is to develop a unified structure that can be utilized in UAVs with both pusher and tractor type propulsion systems.

The work conducted will contribute to UAV technology and the engineering literature. The findings of the thesis will serve as a reference point for future UAV designs and contribute to the development of engineering solutions in the aviation industry. The gearbox will provide design optimization for different aircraft and reduce the design workload. This modular design approach will contribute to the development of cost-effective solutions in the aviation industry by reducing production costs and simplifying manufacturing processes. Additionally, it will significantly improve maintenance and maintainability processes. The use of a single type of gearbox will facilitate spare parts management, reduce inventory costs, and standardize maintenance procedures, thereby shortening maintenance time. This will enable UAV operations to continue uninterrupted and efficiently.

The thesis consists of four main sections. First, the general characteristics and applications of UAVs are briefly discussed, and the existing gearbox designs in the literature are reviewed. In the second section, the classification of UAVs is detailed, and information about pusher and tractor type UAVs. The third section specifies the design criteria and requirements for a common gearbox design for pusher and tractor type propulsion systems. Basic engineering principles related to gears and other subsystems are discussed. Detailed analyses are conducted on design parameters related to gear design, gear profiles, shaft arrangements, and the bearings of components. The system kinematics of the gearbox is conducted using the KISSsys program. Subsystem components such as gears, shafts, and bearings are designed and analyzed using KISSsoft software. Based on the designed components, the housing and other parts are designed. As a result, a common gearbox design for two different types of UAVs is completed. In the final section, the findings of the thesis are

summarized, and the contributions of these findings to UAV technologies and gearbox design are evaluated.

1.2. Literature Survey

UAVs technology is rapidly advancing, and these vehicles are used in a wide range of military, civil, and commercial applications. Gearboxes are mechanical systems widely used in various fields, particularly in automotive and aviation industry. These systems have the function of reducing or increasing engine speed and are used to transmit the rotational motion and torque produced by engines to other machine elements such as wheels or propellers at the desired speed. There are many studies in this scope, and some are summarized below.

The design and analysis of a single-stage reduction gear system for radio-controlled aircraft has been studied. The gear system was designed to reduce the propeller speed corresponding to the torque produced by the engine. In this study, a 7.5 cc single-cylinder engine was used. A specific pinion and wheel gear model was created and integrated with the engine assembly. Bending and contact stress analyses were performed and compared with the calculated stress values. The bending and contact stresses of the gears were compared with the theoretical calculated values and found to be in agreement. Torque requirements for propellers of different sizes were analyzed. The importance of the propeller and thrust in the gearbox design for aircraft was emphasized. In the design of gearboxes for aircraft, thrust was also considered in the calculations and designs. Additionally, a quantitative analysis of the amount of thrust before and after using the reduction gear system was performed. The developed system provided higher thrust while minimizing engine wear and offering additional load-carrying capacity [Maulik et al., 2020].

A gearbox providing better performance and minimum weight for a single-seat off-road vehicle has been developed. The study aimed to reduce weight to achieve more performance from a 10 HP engine. Various comprehensive studies related to the gearbox were conducted. First, a detailed simulation model that includes vehicle dynamics was created. Based on the simulations and analyses, a gear ratio of 6.25:1 was selected for the gearbox design. The gear ratios were determined using a two-stage gear system. The gear diameters were shaped according to the center distance

requirements between the input and output shafts. Gear widths were minimized by calculating static and fatigue failures to ensure contact and bending safety factors. AISI 8620 steel was chosen as the material, and face widths were determined according to safety factors. As a result, the newly designed gearbox achieved approximately a 40% weight reduction compared to previous designs [**Perez and Bachman, 2020**].

The use of modern software such as SolidWorks, Ansys, and KISSsoft in gearbox designs allows for faster and more accurate designs. In recent years, the use of such programs has been increasing. In this context, the design process of the two-stage gearbox was carried out using KISSsoft software. The design process began with material selection and the determination of input parameters. In the gear design, gear diameters and face widths were optimized. During this process, static and fatigue failures due to contact and bending stresses in the gearbox were calculated. The gear ratios and face widths obtained from the calculations were determined according to safety factors. After the gear design, shaft designs were made and analyzed in detail. The CAD model of the gearbox was created using SolidWorks software, and gear deformation analyses were performed [**Patel and Gupta, 2016**].

The design and optimization of a three-stage gearbox for industrial use have been carried out. The aim of the study is to enhance the performance of the gearbox while minimizing its weight and cost. During the design process, gear ratios, material selections, and the mechanical properties of gearbox components were examined in detail. First, the materials required for the gearbox design were selected based on criteria such as strength, durability, and cost. Comprehensive analyses were conducted using KISSsoft software for the design of gears and shafts, and the designs were carried out in accordance with AGMA standards. Gear ratios were determined by balancing optimal performance and weight. The gearbox was designed as a three-stage system, with each stage optimized separately. During the optimization phase, the performance of the gearbox under various load and speed conditions was analyzed. The results showed that the optimal design of the three-stage gearbox provided high efficiency and met the desired performance criteria [**Patil et al., 2022**].

Computer-aided tools are widely used to evaluate the load-carrying capacity of all major components, such as bearings, shafts, and gears, in simple gear systems. In this research, a time-saving, automatic, and systematic approach for designing weight-optimized helical gearboxes is introduced. This method is derived from existing norms

and guidelines for the design and arrangement of shafts, bearings, and gear wheel bodies. The results of the study demonstrate that the proposed model can design a weight-optimized gearbox consisting of gear meshes, shafts, and bearings **[Fürst et al., 2021]**.

A two-stage gearbox with a low weight and compact design was developed for use in a vehicle for the Baja SAE competition. In the study, AISI 4340 steel, known for its high tensile strength and toughness properties, was used for the gears and shafts. For the gearbox housing, Al6061 T6 aluminum alloy was chosen due to its light weight and good machinability. CAD modeling was performed using SolidWorks, and structural analyses were conducted with ANSYS software, including flow analysis for oil flow. Parameters such as gear ratios, minimum number of teeth, and bearing forces were optimized, and the design was validated with load testing, vibration analysis, and thermal analysis. The results showed significant improvements in the performance and durability of the gearbox, with a 15% reduction in weight and optimized costs **[Reddy and Mahato, 2016]**.

Spur gears are simpler and more economical compared to helical gears, making them easier and cheaper to manufacture. This study focuses on the design and optimization of spur gears to be used in a single-stage reduction gearbox. To this end, the dimensions of the components were calculated and optimized using Fusion 360 software. The gearbox is aimed to be lightweight, cost-effective, and durable. AISI 4130 steel was selected for its high strength, toughness, and oxidation resistance. For the housing material, Al 7075 T6 alloy, known for its high strength and good machinability, was chosen. The gear ratio was set at 4, and gear dimensions and load calculations were performed. Following the gear design, shaft diameters and load calculations were conducted using various formulas. Accordingly, bearing elements were determined. The analysis results showed that the gears and shafts could operate safely under the specified loads and conditions. With topology optimization, the weight of the gears was reduced by 40%, resulting in a weight savings of 1.15 kg without compromising performance **[Kishore et al., 2022]**.

2. TYPES OF UNMANNED AIR VEHICLE

Unmanned Aerial Vehicles (UAVs) are classified by their propulsion systems, engine types, and propeller configurations besides parameters such as weight, altitude, endurance limits and ranges, areas of usage, etc. Such classifications are typically made according to specific needs and operational areas of usage. They are also based on the performance characteristics and specifications of the UAVs [Austin, 2010].

Most of the companies working in the aviation industry are making effort to developing UAVs according to military mission requirements. These requirements have created various types of UAVs that classified in accordance with mission capabilities as Intelligence, Surveillance, Target Acquisition and Reconnaissance (ISTAR), Combat (UCAV), Vertical Take-Off and Landing (VTOL), Multi-Purpose, Radar and Communication Relay, Aerial Delivery and Resupply [Moschetta and Namuduri, 2017].

UAVs by weight are classified into five groups including from micros to super heavies as indicated **Table 2.1**.

Table 2.1: UAV classification by weight [Agostino et al., 2007].

Designation	Weight range (kg)	Example
Super Heavy	>2000	ANKA-III, Global Hawk
Heavy	200-2000	ANKA, A160 Hummingbird
Medium	50-200	Şimşek, AAI RQ-7 Shadow
Light	5-50	BAE Systems Silver Fox
Micro	<5	RQ-14 Dragon Eye



Figure 2.1: TAI Anka-III [Web 1, 2024].

Unmanned aerial vehicles (UAVs) requirements are determined by considering parameter such as endurance, range, and altitude. The operating hour of the UAVs is a determinant parameter for reconnaissance, surveillance, and attack and also maximum altitude provides a wide surveillance area and makes it possible to avoid radar detection. Additionally, range refers to the maximum radius at which the UAV can operate, starting from its initial activation point. However, by using satellite communication technologies, these limitations have been overcome and global operations have become possible. These factors enable UAVs to be effectively utilized in a wide range of civilian and military applications and are classified into two groups as indicated **Table 2.2**.

Table 2.2: UAV classification by specifications [Coban and Oktay, 2018].

Category	Endurance (h)	Range (km)	Altitude (m)	Example
Low	>24	>1500	>10000	ANKA, Aksungur
Medium	5-24	100-400	1000-10000	RQ-5 Hunter, TB2
High	<5	<100	<1000	Dragon Eye



Figure 2.2: Baykar TB2 [Web 2, 2024].

Unmanned Aerial Vehicle (UAV) technology has made significant advancements, particularly in propulsion systems that determine flight performance. UAVs have three main types of propulsion systems: fuel-powered, hybrid fuel-electric, and pure electric. Among fuel-powered propulsion systems, piston engines and turbine engines hold an important place. Piston engines, as internal combustion engines, convert chemical energy into mechanical energy by using fuel. These engines are known for their mature technology, simple structure, and low cost. Despite limitations such as low power-to-

weight ratio and poor performance at high altitudes, they operate on a four-stroke cycle, making them suitable for low-speed and medium-altitude UAVs. In addition to piston engines, turbine engines are also widely used in UAVs. Turbine engines include turbojet, turbofan, turboprop, and turboshaft engines. Turbojet engines offer high speed and altitude performance, while turbofan engines provide higher efficiency and longer lifespan. Turboprop and turboshaft engines, on the other hand, offer high efficiency in low-speed applications.

Hybrid fuel-electric propulsion systems enable the combined operation of fuel engines and electric motors to enhance energy efficiency and reduce emissions. These systems are classified into parallel, series, series-parallel, and complex hybrid structures. Pure electric propulsion systems use only electric motors as the power source and offer advantages such as low carbon emissions, low cost, and high energy efficiency. Future technologies include high-power motors, converters, and energy management systems. [Zhang et al., 2022].

2.1. Pusher Type

Pusher-type propulsion systems represent a unique configuration in the realm of aircraft design, where the propulsive engines are mounted behind the payload or cockpit area, pushing the aircraft forward as indicated **Figure 2.4**. In other words, the thrust configuration is defined based on the placement of the engine relative to the aircraft's center of gravity. In a pusher configuration, the engine is located behind the aircraft's center of gravity. This configuration stands in contrast to the more traditional tractor-type systems, where engines pull the aircraft from the front [Raymer, 2018].



Figure 2.3: TAI Anka [Web 3, 2024].

UAVs with pusher type propulsion systems have several advantages over tractor type UAVs. Firstly, positioning the engine at the rear of the aircraft shifts the center of gravity backward, providing better longitudinal control and requiring less elevator deflection during takeoff. This enhances flight safety by providing better longitudinal control during takeoff and landing. Additionally, having the engine and propeller at the rear allows for a wider and unobstructed field of view for the front-mounted camera, sensors, and other equipment. This makes pusher type UAVs extremely valuable for missions such as reconnaissance, surveillance, and mapping. The wide field of view enables clearer and more comprehensive tracking of targets.

The pusher configuration, wherein the propeller is located at the rear of the aircraft, inherently generates less aerodynamic drag within the propeller disc area, thereby enhancing overall efficiency. Considering these various factors, pusher type UAVs present themselves as an optimal design choice for specific mission profiles. These configurations are highly advantageous for long-range missions due to their efficiency in fuel consumption and aerodynamic performance. Moreover, pusher type UAVs are well-suited for operations that necessitate a wide field of view, as the rear-mounted propulsion system minimizes visual obstructions. Additionally, this configuration offers superior longitudinal control, making it ideal for situations that require precise maneuverability and stability. Therefore, pusher type UAVs stand out as an excellent design option, especially when mission requirements include extensive range, broad observational capabilities, and enhanced control dynamics [Sadraey, 2013].

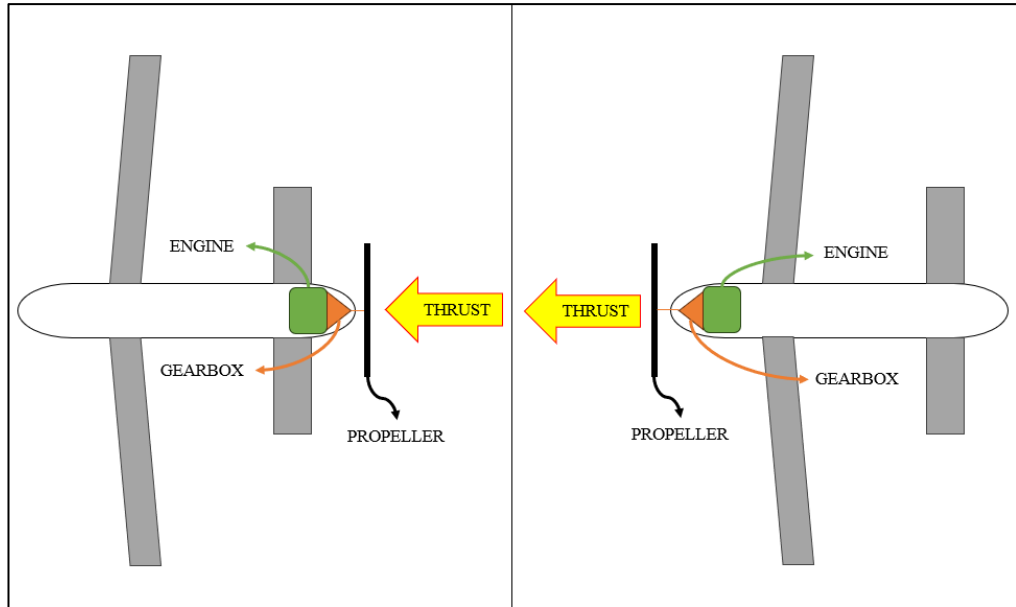


Figure 2.4: Pusher and tractor type propulsion configuration.

2.2. Tractor Type

Tractor-type propulsion systems, characterized by the engine and propeller being mounted at the front of the aircraft, represent the most traditional and widely adopted configuration in aviation as shown **Figure 2.4**.

In a tractor configuration, the engine is located front the aircraft's center of gravity. This design allows the propeller to pull the aircraft through the air, offering distinct advantages in terms of simplicity, effectiveness, and aerodynamic efficiency [Marshall et al., 2016].



Figure 2.5: TAI Aksungur [Web 4, 2024].

Positioning the engine and propeller at the front of the aircraft shifts the center of gravity forward, providing better directional stability for the aircraft. This configuration offers improved controllability, especially when flying in windy conditions and at low speeds. Having the propeller at the front allows for direct and efficient thrust, resulting in slightly higher net thrust compared to the pusher configuration. However, there are also disadvantages such as restricted field of view and structural complexity.

Additionally, in tractor type UAVs, the engine and propeller create less aerodynamic drag behind the fuselage or wings, leading to higher speed and fuel efficiency. These advantages make tractor type UAVs preferred for various missions, particularly those requiring long-range and high-speed flight. Considering all these factors, tractor type UAVs offer significant advantages in terms of reliability, controllability, and performance **[Sadraey, 2013]**.

3. COMMON GEARBOX DESIGN

The propulsion systems that are explicated intricately in Section 2, take place from the sub-systems as engine, transmission, cooling, exhaust, etc. Of these, the engine is the main ATA chapter of the propulsion system that converts heavy fuel energy into mechanic energy, and the work part is the part that converts the mechanic energy obtained from the engines into work energy. The engine provides constant torque because the nominal speeds and powers of the engines are invariable. On the other side the requirement of the thrust, which is an important parameter of the propulsion system, needs different torques under changing operating conditions. Owing to the power of the engine being constant, different torque needs can only be met by changing the RPM of the engine [Jadhav et al., 2016].

Generally used as a mechanical-based power and motion transmission element in machine construction; gear mechanisms, belt pulley mechanisms, friction wheels and chain mechanisms are used. These mechanisms have advantages and disadvantages compared to each other in terms of construction and price. Taking these into consideration, the designer selects the most suitable mechanism for the purpose in construction as the power and motion transmission element.

For this purpose, gearboxes that we call power and motion transmission elements are used between the engine and the work machine, which transmit the power of the engine and also change the value and/or direction of the rotation speed [Myszka, 2012].

Gear mechanisms, which are the main components of gearboxes and provide power and motion transmission, are generally divided into two groups: single-stage and multi-stage. The main difference between these two groups is; In single-stage gear mechanisms, a single output speed can be obtained for a single input speed, and in multi-stage gear mechanisms, more than one output speed can be obtained for a single input speed. In other words, the cycle ratio (ratio between input and output speeds) of single-stage gearboxes is a single value, while in multistage gearboxes it is not a single value but has more than one value [Brecher et al., 2017].

In this study, a single-stage gearbox design was carried out because it is the most widely used as a power and motion transmission element and meets the design

requirements. The design process of the gearbox, including relevant arguments, sketches, and designs using the parametric solid model program Siemens NX, the gear design program KISSsys, and the design of gearbox system program KISSsoft, are included in the continued chapters.

3.1. System Requirements and Assumption

System requirements and assumptions play a crucial role in the development and designing process of gearbox that subsystem of propulsion systems. These establish the technical specifications and parameters that bring the development of the gearbox to a successful conclusion. Identifying system requirements and assumptions before commencing gearbox design ensures the design process progresses correctly and results in a product that meets operational needs.

System requirements are shaped according to the scope and expectations of the product. Product features, operating conditions, mounting configurations, mission profiles, environmental factors, performance, lifetime, weight, maintenance, and interchangeability are among the elements determined through these requirements. On the other hand, system assumptions define the limits within which the product can be designed and operated. Assumptions such as specifications of engine and propeller, connection conditions, manufacturing methods, and selection of standards that are used contain critical information that shapes the design process. [Holder et al., 2016].

In summary, when establishing system requirements and assumptions, the needs, and expectations of all relevant teams, including designers, manufacturers, and operational users, should be considered. Additionally, realistic, and achievable goals must be set for successful results.

3.1.1. Engine and Propeller Specifications

The propulsion system engine is a diesel engine that has four cylinders in line and a four-stroke. The operation of the direct diesel injection engine is based on the common-rail technique and is turbocharged. The propeller is driven by an integrated gearbox on the engine. In addition, the engine is equipped with an electric starter and alternators.

The propeller of propulsion system is a composite propeller that has three blades. The role of the propeller within propulsion systems is critical for the operation and efficiency of Unmanned Aerial Vehicles (UAVs). The propeller is an essential component that converts rotational energy from the engine into thrust. Thrust directly influences an aircraft's performance, efficiency, and maneuverability. The assumptions considered in gearbox design are stated in the **Table 3.1**.

Table 3.1: Engine and propeller specifications.

Description	Specification
Engine maximum power	114 kW @3900 rpm
Propeller maximum power	114 kW @2300 rpm
Propeller revolution tolerance	$X \pm 50$ rpm
Gearbox ratio	1.69
Direction of rotation – engine and propeller	Clockwise
Oil operating temperature	50 °C – 140 °C
Coolant operating temperature	60 °C – 105 °C
Propeller weight	19 g

3.1.2. Mounting Specifications

When initiating the design of a gearbox, it is essential to determine the characteristics of the engine and propeller as well as the connection interfaces. The gearbox housing is designed to transmit the power generated by the engine to the propeller. Therefore, the accurate positioning and robust integration of the mounting points are of paramount importance. The engine connection interface ensures that the engine is securely and vibration-free aligned with the gearbox. Additionally, it enables the torque produced by the engine to be transmitted to the gearbox efficiently and without loss.

The propeller connection interface transmits the rotational motion from the gearbox to the propeller, generating thrust. This thrust enables the UAV to move through the air. Proper placement of the propeller mounting points is crucial for balanced operation and maximizing thrust force. Incorrectly positioned propeller mounts can lead to vibrations and unbalanced thrust, which can negatively affect the UAV's performance.

Additionally, the positions and rotational directions of the motor and propeller directly influence the body geometry of the gearbox. Furthermore, the shaft lengths and the

distances between gears are determined based on these positions. The positions and connection interfaces related to the motor and propeller are specified in **Table 3.2**.

Table 3.2: Engine and propeller location and mounting holes.

		Y	Z
Engine output position		0	0
Propeller flange position		205	208
Gearbox and engine mounting holes (Ø9)	1	-96	74
	2	-121	0.9
	3	-104	-63
	4	-35	-116
	5	35	-116
	6	104	-63
Gearbox and propeller mounting holes (Ø11.5)	7	121	10
	1	205	245
	2	205	224
	3	205	184
	4	205	165
	5	205	187
	6	205	227

3.1.3. Material Selection

When designing a gearbox for aviation, material selection is crucial due to the requirements for high reliability, durability, and weight considerations. Selecting the gear material involves balancing multiple factors such as performance, cost, and ease of manufacturing. Gear materials used in aviation applications must possess properties like high strength, hardness, and fatigue resistance. Additionally, these gears must withstand extreme temperatures and corrosive environments, ensuring they perform reliably under challenging operating conditions [Chatterjee and Chakraborty, 2013], [Delibaş et al., 2017].

A variety of materials, such as cast irons, powder-metallurgy materials, nonferrous alloys, and plastics, are utilized in the manufacturing of gears, each offering distinct advantages depending on the application. Cast irons, for instance, are valued for their good machinability and wear resistance. Plastics, on the other hand, are lightweight and can be used in applications where noise reduction and minimal lubrication are important. However, steels are the most widely used gear materials in heavy-duty, power transmission applications. This is primarily due to their high strength-to-weight

ratio, which provides robust performance without significantly increasing the weight of the gear system. Their ability to be heat-treated to achieve a variety of mechanical properties further enhances their suitability for demanding environments **[Davis, 2005]**.

Specifically, steels such as 20MnCr5 and 18CrNiMo7-6 are commonly used for high-performance gears. The heat treatment processes applied to these steels enhance the gears' surface hardness while maintaining the core's toughness, resulting in gears with high load-carrying capacity. The carburizing process used in gear manufacturing increases surface hardness, which enhances the wear and fatigue resistance of the gears. Additionally, these processes typically provide the high hardness and durability required for aerospace gears **[Blech et al., 2023]**.

The steel 18CrNiMo7-6 is considered ideal for high-performance gear applications because it provides significant improvements in surface integrity parameters. Specifically, features such as the reduction in average grain size during operational cycles, increased surface microhardness, and optimized residual stresses contribute to the material's enhanced fatigue life. Additionally, through heat treatments like carburizing and tempering, 18CrNiMo7-6 steel offers high hardness, durability, and wear resistance. This steel is also notable for its ability to control surface roughness 6542gear applications **[Wu et al., 2023]**.

Additionally, the AA7050 aluminum alloy is known for its high strength-to-weight ratio, making it particularly suitable for aerospace applications. It allows for the reduction of the weight of aerospace components without compromising strength. Additionally, AA7050 offers excellent corrosion resistance, enhancing the durability and reliability of parts exposed to harsh environmental conditions. Overall, these advantages make AA7050 an ideal material for high-performance aerospace applications, balancing superior mechanical properties with manufacturing efficiency **[Chen et al., 2017]**.

The gear and shaft material has been selected as 18CrNiMo7-6 steel due to its reliability in aerospace and other critical applications. This steel offers excellent performance characteristics, making it a suitable choice for high-stress environments. For the housing, AA7050 aluminum has been chosen because of its advantages in

weight, machinability, and strength. These properties make AA7050 an ideal material for applications where reducing weight without sacrificing structural integrity is crucial.

3.2. Geartrain Layout

The design of gearbox components is carried out after establishing the system kinematics. The elements within the KISSsys - Element Box are used to create the gearbox kinematics. These elements are classified according to function on the gearbox such as shaft groups, shaft elements, calculations, system elements. Each component is added to the system with the program's default settings. When creating the layout, the shaft element is selected from the groups in sequence, and couplings, bearings, and gears from the shaft elements are added to the shaft. For shaft design, a shaft calculation is added from the calculations section. A shaft group is created for each gear, and the necessary components are positioned on the shaft. The relationship between the created gears is defined with the three gears constraint, and the three gears constraint calculation is added to the system as a sub-element for gear design.

Finally, input and output definitions are made in the boundary conditions section for the kinematic calculation. As a result, as shown in **Figure 3.1**, the types of components that make up the system are identified.

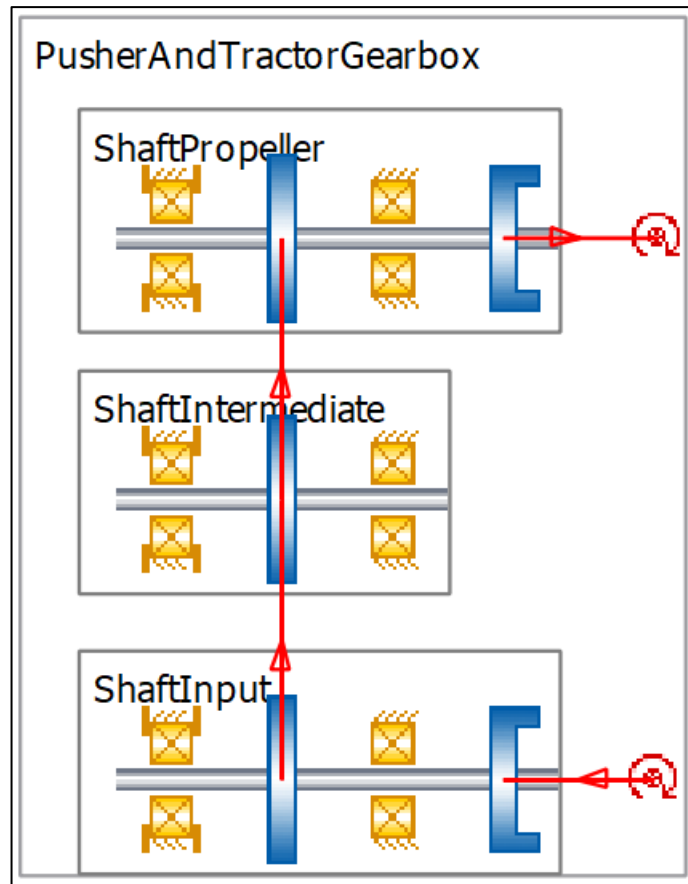


Figure 3.1: Gearbox kinematic.

3.3. Gears

Gears are of critical importance for transmission systems and are among the components that require priority completion in the design stages of a gearbox. The main purpose of gear design is to ensure power transmission at a specific transmission ratio, to achieve gears that operate without deformation throughout the lifecycle of the gearbox, to be low weight product for UAVs and to provide cost-effectiveness. To begin the design of gears, it is necessary to have prior knowledge of several values and details such as the type of gear system, configuration, engine parameters, input, and output data, etc. Parameters such as module, number of teeth, safety factors, and material must be selected based on this information.

In Section 3.1, the requirements and assumptions are specified in detail and the design details for each gear are studied. Additionally, the relation between the gears and other components of the gearbox, such as shafts and bearings, in terms of their design and selection has been emphasized. In this section of the study, gear designs were carried

out using the KISSsoft software. Afterward, information on the main gear parameters, manufacturing tables, and applied loads have been provided.

3.3.1. Gear Types

Gears are fundamental components in the common gearboxes studied in this thesis, essential for transmitting torque and motion between shafts. The classification of gears is typically based on their geometry and the orientation of their axes.

According to the axis types, gears are categorized into three groups: parallel, nonparallel coplanar, and nonparallel noncoplanar. In this study, since the motor output shaft and the propeller are positioned parallel to each other, parallel axis gears have been used. The simplest types of gears are those designed to connect parallel shafts. These gears are generally easier to manufacture and can transmit large amounts of power with high efficiency.

Parallel axis gears are more efficient in power transmission compared to any other type or form of gearing. Additionally, their simple design allows for easier alignment and maintenance, further enhancing their reliability and performance in demanding applications [Radzevich, 2012].

According to their geometries, gears are divided into four main groups: spur, helical, bevel, and worm gears, as shown in **Figure 3.2**. Spur and helical gears operate in a parallel axis configuration. The spur gear features teeth on the outer surface of a cylinder, with the teeth oriented parallel to the cylinder's axis. The spur gear design makes it the most common and widely used type of gear. Helical gears can be used for the same applications as spur gears and, when used in this manner, they are less noisy due to the more gradual engagement of the teeth during meshing.

Additionally, despite the inclined teeth of helical gears generating thrust loads and bending moments, the spur gears do not generate [Mott and Timmerman, 2024], [Tata, 2012].

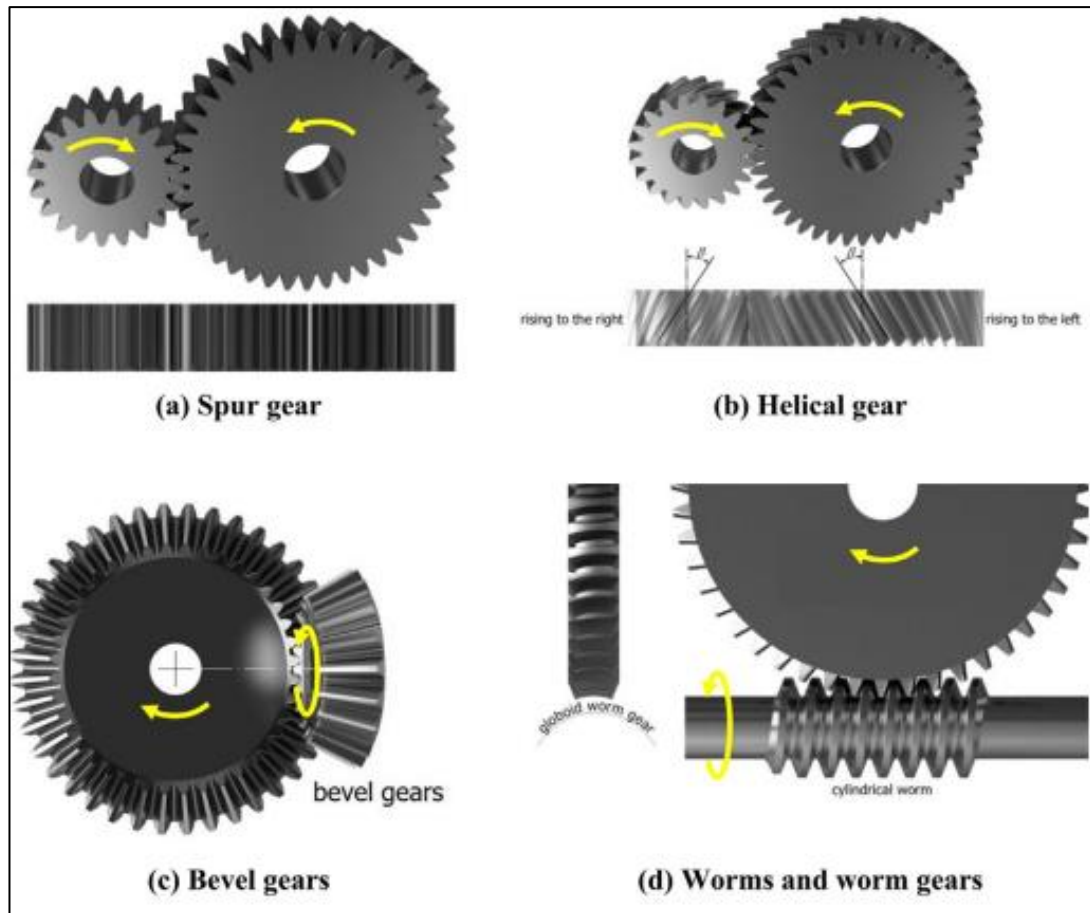


Figure 3.2: Main gear types [Mott and Timmerman, 2024].

In the gearbox designed for use in both pusher and tractor type propulsion systems, the thrust force generated by the propeller acts in both directions. Additionally, helical gears generate axial forces during operation. These forces affect the two different types of UAVs in the same and opposite directions, respectively, which directly impacts the gear operation negatively. Therefore, a spur gear design has been chosen for the common gearbox to avoid the complications and potential issues associated with thrust force.

3.3.2. Main Parameter of Gears

Designing the spur gears starts with determining the priority decisions such as input parameters. Subsequently, the outputs that need to be evaluated accordingly are determined. The priority decisions, such as load, speed, gear ratio, safety factors, design factors, and life quality number, are the primary decision parameters. Based on these inputs, the gear design is carried out by considering design decisions such as module,

number of teeth, face width, weight, and tooth safeties. After the preliminary decisions, the module is selected. Based on the module, the number of teeth and face width are determined, and the weight is calculated. Gear safety factors are calculated according to the selected materials. If the desired values are not achieved through manual calculations, the process returns to the module selection and the associated calculations. This iterative process continues until the desired results are obtained. In modern technology, computer applications are used instead of manual calculations, allowing for much faster optimization [Budynas and Nisbett, 2020].

The advantage of using an optimization technique is that the range of possible designs can be evaluated same time. The method employed in this study starts by listing the design decision parameters. The module is a free parameter that is not subject to any constraint relations. The number of input gear teeth is a second free parameter. According to the gear ratio, the number of propeller gear teeth is defined. For continuous rotation, both the number of input and propeller gears must have integer values. The input gear pitch radius and the propeller gear pitch radius define the size of the gears. On the other hand, the size of the gears is directly related to the tooth numbers by the module. The center distance is simply the sum of the pitch radiuses. If there is a limitation about the center distance, the number of input gear teeth are not a free parameter because the module and center distance are determined. For this study, increasing the module of gears cause the decreasing of the number of gears because of the constant center distance.

Additionally, addendum and dedendum ratios, face width, material properties, etc. parameters define gear sizing, but several other parameters are determined exactly in terms of the input specifications and the free parameters. Input values are a system specification or requirements, and only two parameters can be varied over an arbitrary range of values. These two free parameters in the gear design stages are the number of teeth on the gear and the module [Savage et al., 1981].

Module that is defined ratio of the pitch diameter and the number of teeth is a critical parameter for gear design and decision. It directly affects the size and geometry of the gear. When the module value increases, the gears become thicker and wider. This change in the module size results in a reduction in the number of teeth, assuming the center distance remains constant. Consequently, for a given center distance, a larger module leads to fewer teeth on the gear. This situation leads to an increase in surface

contact stresses. At the same time, as the number of teeth decreases, the contact rate decreases by direct proportion. The contact ratio represents the average number of tooth pairs in contact on gears. A low contact ratio results in the load being distributed over fewer teeth, leading to an increase in stress on the teeth. This situation directly affects the fatigue and lifespan of the gears negatively. As a result, the contact ratio is evaluated when determining the module, and lower module values are preferred as far as possible [Budynas and Nisbett, 2020].

Additionally, increasing the contact ratio causes a crease in the pressure angle. Therefore, the contact ratio between spur gears is inversely proportional to the pressure angle. In **Figure 3.3**, the relationship between the contact ratio and the module is examined. This examination is conducted for three different tooth angles: 14.5° , 20° , and 25° . The results illustrate how the contact ratio varies with changes in the module for these specific angles. It is evident that as the module increases, the contact ratio decreases for all three angles. This observation highlights the significant impact of tooth angle and module on the contact ratio. Current values of common usage today are 20° and 25° [Murali and Prasad, 2016].

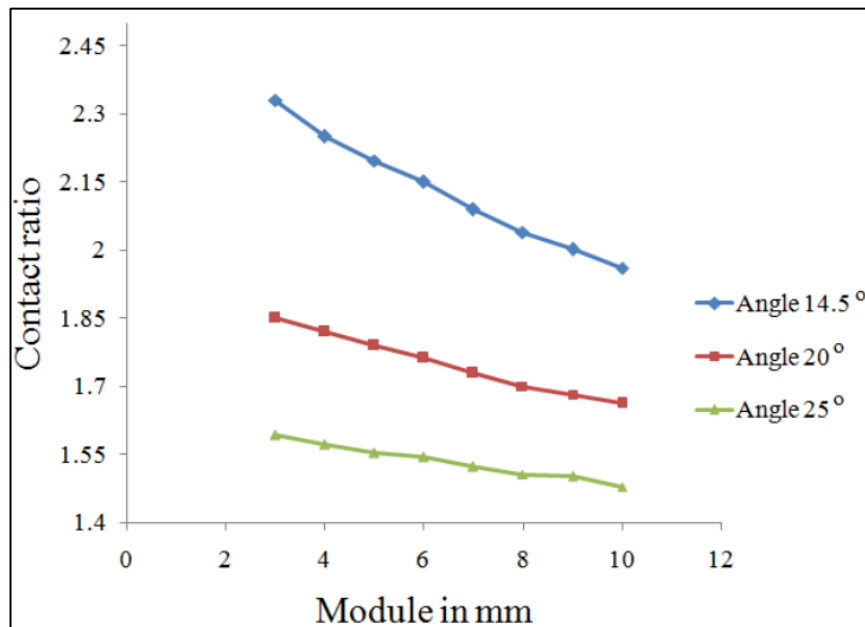


Figure 3.3: Contact ratio versus module for different tooth angles [Murali and Prasad, 2016].

The gear face width is directly proportional to the resulting strength of the gear, as long as the tooth load remains uniform. The face width is related to the number of teeth

on the gear and the module. The force on the gear's tangential component is uniformly distributed over the face width of the gears.

Increasing the face width of spur gears significantly reduces bending stress, thus increasing the load-carrying capacity. It means face width affects the root safety of the gears. The gear face width is thus determined for a given design by this limit. Generally, in gear theory, face width should be from nine to fifteen times the module [Singh and Kumar, 2020].

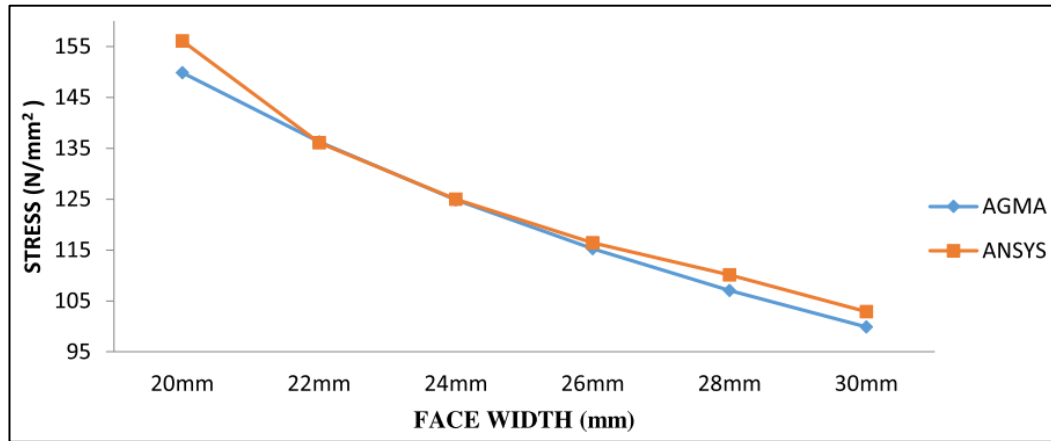


Figure 3.4: Face width versus stress [Singh and Kumar, 2020].

Root safety and flank safety are critical parameters considered to ensure the reliable and long-lasting operation of gears according to the specified design requirements. These parameters help minimize risks during the stages of gear design, manufacturing, assembly, and operational conditions. Root safety refers to the gear's safety against stresses occurring in the root region, while flank safety refers to the safety against stresses occurring on the gear tooth surface. The root region of gears is typically the weakest area and has the highest stress concentration. If not designed correctly, it can lead to crack formation in the gear root. The flank surfaces of gears are the load-bearing areas where force transmission occurs. Contact stresses and wear in these regions directly affect the performance and lifespan of the gear. High contact stresses can lead to surface damage such as wear and pitting on the gear tooth surface. Therefore, it is critical to accurately assess the stresses in these areas and apply appropriate safety factors. The standards contain a safety factor S_F guarding against bending fatigue and safety factor S_H guarding against pitting failure. The standards provide hardly any information about this. For industrial gear units, DIN 3990, Part 11 has minimum factors indicated **Table 3.3.** [KISSsoft AG., 2022].

Table 3.3: Safety factors [KISSsoft AG., 2022].

Type	Required safety
Minimum safety for root	1.4
Minimum safety for flank	1.0

3.3.3. Design of Gears

Following the system kinematic calculations, the initial gear design was completed, and the calculation for the three gear pair constraint systems was performed using KISSsoft software. Based on the system requirement and assumption given in Section 3.1, inputs were made for the Basic Data, Reference Profile, Tolerance, Rating, and Factors sections.

When designing a gearbox using any software, the operating conditions in the physical ambient such as temperature, tolerances, etc. must be accurately and completely defined in the program interface. Therefore, it is important to be familiar with the default values and parameters stored in the program and make modifications if necessary. Accordingly, the parameters that need attention were checked, and those deemed necessary were updated accordingly. The unindicated parameters in Section 3.1 were determined by the KISSsoft manual, standards, literature, and sector experience, ensuring that the program works closely to physical conditions. Values and information about important parameters in gear design are stated in **Table 3.4**.

Table 3.4: Gear design parameters.

Parameter	Description
Pressure at normal section	20°
Gear type	Spur
Face width	30 mm
Gear quality	6
Material (gear and shaft)	18CrNiMo7-6
Oil Type	Shell Spirax ATF
Lubrication type	Oil bath
Lubricant temperature	95 °C
Addendum coefficient	1.25
Root radius coefficient	0.38
Addendum coefficient	1.00
Tooth thickness tolerance	DIN 3967 cd25
Center distance tolerance	ISO 286:2010
	Measure js7
Calculation method	DIN 3990:1987
	Method B

Calculation method for micropitting	ISO TR 15144
Driving gear	First gear in the chain
Working flank gear 1	Right
Sense of rotation gear	Clockwise
Required service life	5000 hours
Application factor	1.25
Dynamic, Transverse, and Face load factor	According to calculation: ISO6336-1 Annex E
Root safety	1.4
Flank safety	1

When sizing the three-gear pair system, the distance between the gears must be provided as an input. While gear calculations can be performed using the default settings without altering the parameters in the program interface, but it is not possible to perform the calculation without inputting the distance between the gears. The foreseen schematic of the gear system is shown in **Figure 3.5**, and the design was started by determining the distance between the gears based on the formulas. Due to the motor and propeller rotating in the same direction, the idler gear is used in the geartrain system. The calculations were performed assuming that the idler gear and the input gear have the same number of teeth. The distance between the input and intermediate gears was approximately calculated to be 88.5 mm, and similarly, the distance between the intermediate gear and the propeller was approximately 119.5 mm by based on equation (3.1), (3.2), and (3.3).

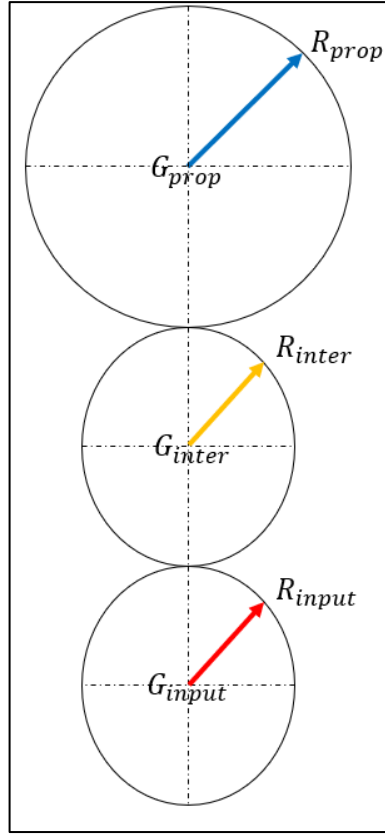


Figure 3.5: Center distances between gears.

$$i = \frac{z_{inter} z_{prop}}{z_{input} z_{inter}} \quad (3.1)$$

$$2R_{input} = mz_{input} , \quad 2R_{inter} = mz_{inter} , \quad 2R_{prop} = mz_{prop} \quad (3.2)$$

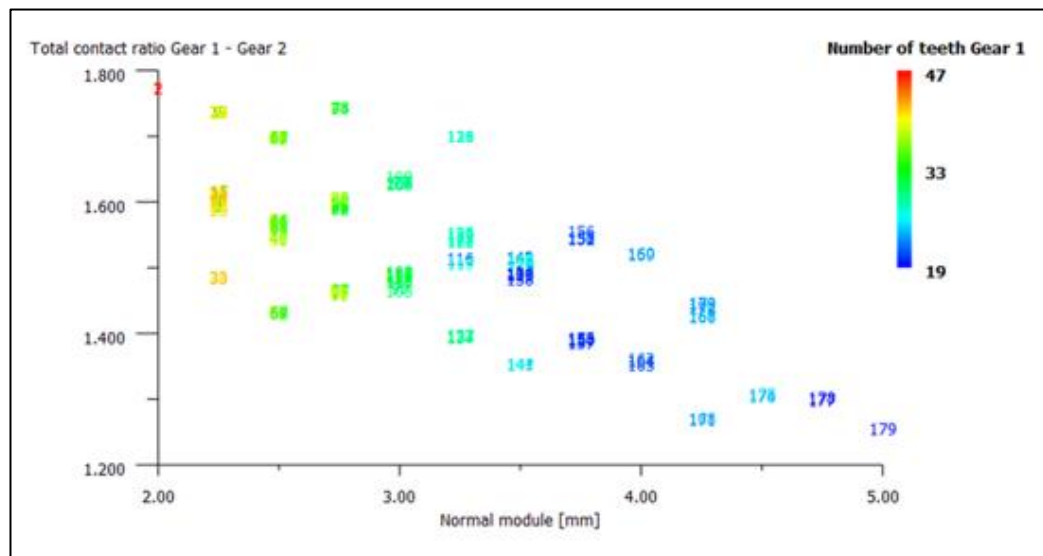
$$R_{input} = R_{inter} \quad (3.3)$$

The necessary input parameters for the Fine Sizing stage are specified in **Table 3.5**. The possible outcomes were limited using the Maximum Number of Solutions parameters. Setting a low value can negatively impact the number of possible results; therefore, if the number of results reaches the defined value, the Maximum Number of solution parameters should be increased and re-calculated. The specified gear ratio and propeller RPM tolerance from the requirements were entered into the program. Minimum, maximum, and step inputs have been provided for the module.

Table 3.5: Fine sizing parameters.

Parameter	Description
Module - min. / max. / step	2 / 10 / 0.5
Max. number of solution	1000
Nominal ratio	1.69
Nominal deviation	$\pm\%2$
Pressure angle at normal section	20
Maximum tip diameter	No limit
Minimum root diameter	0
Range of profile shift coefficient - min. / max.	-0.2 / 0.5

During the gear sizing process, the module was determined first, followed by identifying the corresponding number of teeth. In the program interface, the input gear is labeled Gear 1, the intermediate gear Gear 2, and the propeller gear Gear 3. These labels cannot be altered by the user. Contact ratio calculations and tooth numbers for the gears were performed over a wide range of modules, as indicated in **Figure 3.6** and **Figure 3.7**. A total of 179 results were listed. Results that did not meet the required root and flank safety factors were rejected.

**Figure 3.6:** Module versus contact ratio for different number of teeth (Gear 1 - 2).

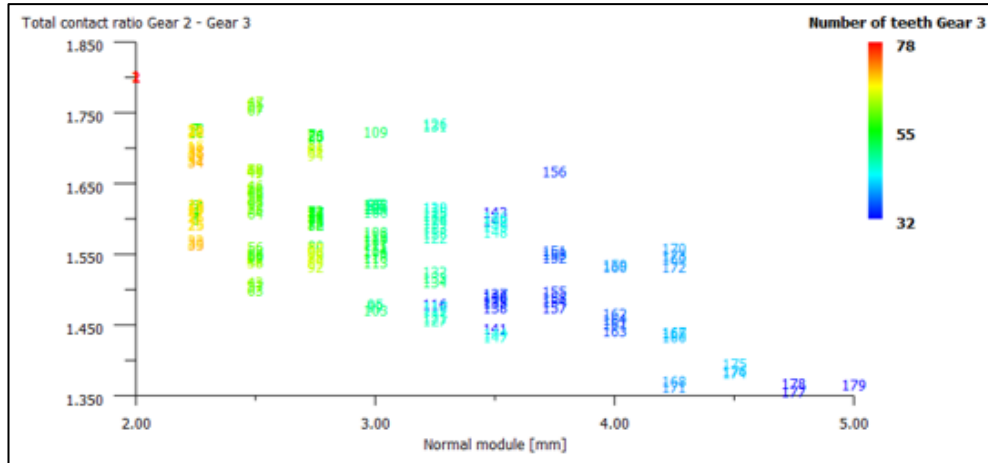


Figure 3.7: Module versus contact ratio for different number of teeth (Gear 2 - 3).

In the Normal module - Contact ratio - Number of teeth graphs, a similar distribution was observed for Gear 1-2 and Gear 2-3. In general, it has been observed that as the module increases, the number of teeth decreases and, accordingly, the contact ratio decreases. Also, the result that the gears with the same module have different contact ratio values has been seen. This situation can be explained by parameters such as the profile shift coefficient and the number of teeth, which affect the contact ratio. Since modules with a contact ratio smaller than 1.4 did not meet the requirements. For manufacturability and cost considerations, the step interval was updated from 0.25 to 0.5. The calculations were repeated for the module range of 2.5 to 3.5. Instead of evaluating each of the three gears separately, the results were plotted on a graph considering the minimum root safety, minimum flank safety, and normal module for all gears as shown **Figure 3.8**.

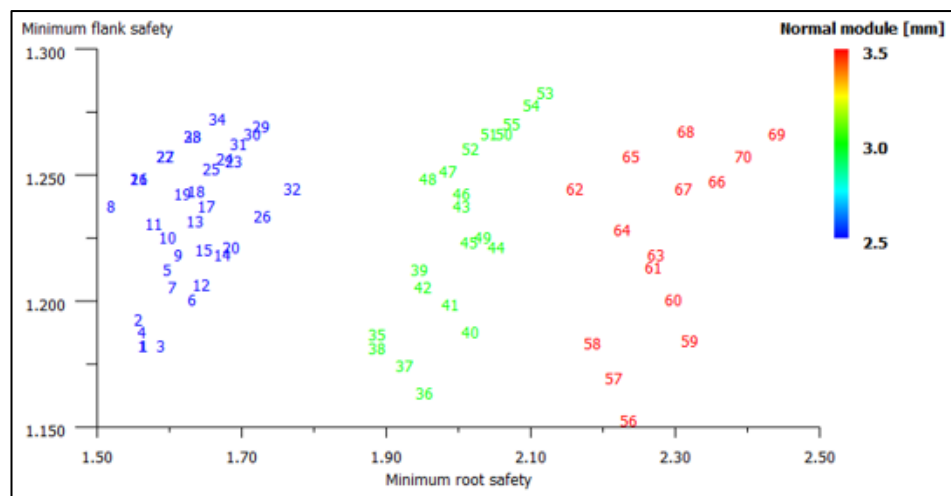


Figure 3.8: Minimum root safety versus minimum flank safety for different module.

When evaluated according to safety factors, results numbered 29, 30, 34, 53, 54, 55, 68, 69, and 70 stand out. On the other hand, the results that provide minimum safety factors can be evaluated such as 1, 2, 3, 4, 35, 36, 37, 38, 56, 57, 58. It should be noted that all the values in the graph meet the safety factor requirements. Therefore, various evaluation criteria should be considered to avoid the occurrence of overdesign. For this purpose, **Figure 3.9** was created by considering the critical importance of weight for aircraft and the interaction between the roots of gear.

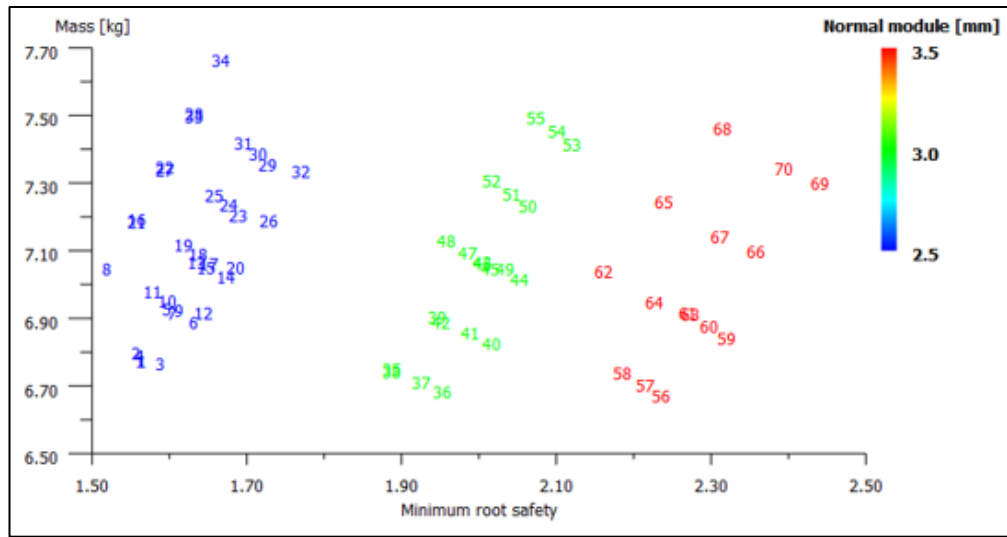


Figure 3.9: Minimum root safety versus mass for different module.

When evaluated based on weight and root safety, results numbered 1, 2, 3, 4, 35, 36, 37, 38, 56, 57, and 58 stand out. There is an approximate weight difference of 100 grams between the results numbered 1, 2, 3, 4 and those numbered 35, 36, 37, 38. This equates to a difference of approximately 1.5-2% in percentage terms and can be ignored.

Additionally, considering that the contact ratio increases as the module decreases, it would be consistent to select one of the results numbered 1, 2, 3, or 4. However, evaluating the contact ratio based solely on the module will not yield accurate results. Therefore, as indicated in **Figure 3.10** and **Figure 3.11**, the evaluation was conducted using the weight-contact ratio graph.

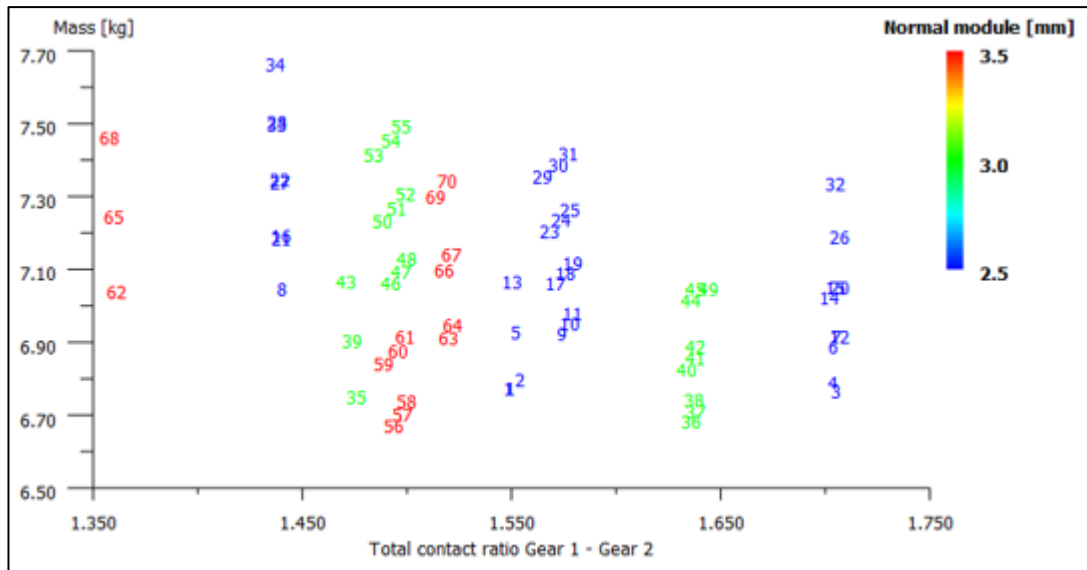


Figure 3.10: Contact ratio (Gear 1 - 2) versus mass for different module.

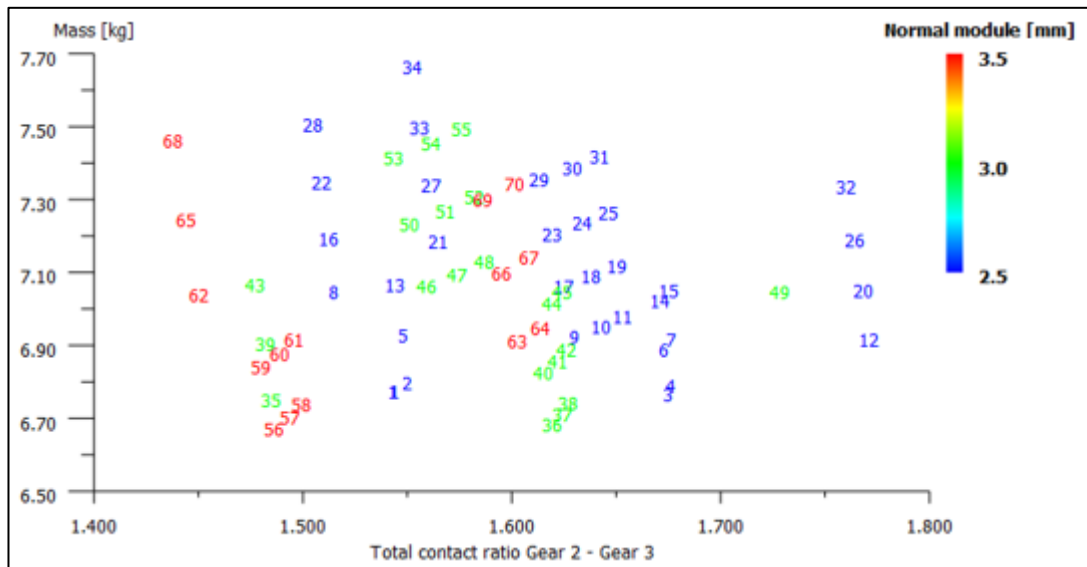


Figure 3.11: Contact ratio (Gear 2 - 3) versus mass for different module.

In conclusion, the evaluation was conducted based on the parameters of module, contact ratio, root safety, flank safety, and weight. Fine sizing was completed using results numbered 3 and 4, as shown in **Table 3.6**. The values are nearly identical, but priority was given to weight, leading to the selection of result number 3.

Table 3.6: Gear system values.

Parameter	Solution 3	Solution 4
Module	2.5	2.5
Contact ratio Gear 1-2	1.704	1.705
Contact ratio Gear 3-4	1.675	1.676
Weight	6.776	6.802
Root Safety – Gear 1	2.29	2.35
Root Safety – Gear 2	1.59	1.57
Root Safety – Gear 3	2.29	2.31
Flank Safety – Gear 1	1.20	1.21
Flank Safety – Gear 2	1.21	1.21
Flank Safety – Gear 3	1.41	1.41
Profile Shift Coefficient – Gear 1	-0.12	-0.02
Profile Shift Coefficient – Gear 2	+0.02	-0.08
Profile Shift Coefficient – Gear 3	+0.29	+0.39

3.3.4. Result of Gear Designs

In accordance with the main parameters, requirements, and assumptions detailed in previous sections, the gear design has been completed using the KISSsoft program. During the gear design process, critical parameters such as gear types, module, number of teeth, face width, and material selection were carefully considered and optimized. By completing the gear design with the KISSsoft program, all necessary details for the gear profile, dimensions, and production have been determined. Consequently, a suitable gear system has been developed that can be used for both pusher and tractor type thrust systems in UAVs. The gear profile, dimensions, and all necessary production details are specified in **Table 3.7**.

Table 3.7: Gears manufacturing data.

Parameter	Abbr.	Gear 1	Gear 2	Gear 3
Drawing or article number		0.000.0	0.000.0	0.000.0
Number of teeth	[z]	34	37	58
Face width (mm)	[b]	30.000	30.000	30.000
Normal module (mm)	[mn]	2.500	2.500	2.500
Helix angle (°)	[beta]	0.000 (0°0'0")	0.000 (0°0'0")	0.000 (0°0'0")
Hand of gear		Spur gear		
Pressure angle at normal section (°)	[alfn]	20.000 (20°0'0")	20.000 (20°0'0")	20.000 (20°0'0")
Material		18CrNiMo7-6		
Reference profile		1.25 / 0.38 / 1.0 ISO53:1998Profi l A		

Addendum coefficient	[haP*]	1.000	1.000	1.000
Dedendum coefficient	[hfP*]	1.250	1.250	1.250
Tip radius factor	[rhoaP*]	0.000	0.000	0.000
Root radius factor	[rhofP*]	0.380	0.380	0.380
Accuracy grade according DIN 3961			6	
Profile shift coefficient	[x]	-0.121	0.022	0.285
Reference diameter (mm)	[d]	85.000	92.500	145.000
		89.391	97.574	151.393
Tip diameter (mm)	[da]	(0.000 / -0.035)	(0.000 / -0.035)	(0.000 / -0.040)
		78.147	86.358 (-	140.177 (-
Root diameter (mm)	[df]	(0.192 / -0.302)	0.192 / -0.302)	0.261 / -0.398)
Tooth thickness deviation			DIN3967 cd25	
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070/ -0.110	-0.070/ -0.110	-0.095/ -0.145
Number of teeth spanned	[k]	4	5	7
Base tangent length (no backlash) (mm)	[Wk]	26.815	34.544	50.491
Actual base tangent length (‘span’) (mm)	[Wk.e/i]	26.750/26.7 12	34.478/34. 441	50.402/50. 355
Effective diameter of ball/pin (mm)	[DMeff]	4.250	4.250	4.500
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	90.063/89.9 60	98.135/98. 035	152.774/1 52.655
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[MdR.e/i]	90.063/89.9 60	98.135/98. 035	152.774/1 52.655
Chordal tooth thickness (without back lash) (mm)	[sc]	3.706	3.965	4.446
Actual chordal tooth thickness (mm)	[sc.e/i]	3.636/3.596	3.895/3.85 5	4.351/4.30 1
Reference chordal height (mm)	[ha]	2.227	2.571	3.220

3.4. Shafts and Bearings

In gearboxes, the shafts that generally have a circular cross section are a rotating element. They provide the axial rotation of elements such as gears, pulleys, flywheels, and other rotational geometries, transmitting power and motion. Primarily, the shafts transmit the rotational motion received from the engine output to the input gear, thereby driving the system. At the same time, shafts, which also provide torque transmission between gears, are critically important for ensuring proper alignment and

positioning of the gears for optimal operation. The shafts can be exposed to various axial, bending, and torsional loads that can be constant or variable combinations [Budynas and Nisbett, 2020].

The shafts that carry gears, pulleys, cams, retaining rings, etc. must be supported between bearings. The bearings assembled on the shaft take on the bearing function for the gear-shaft part, while the shafts make available the correct positioning of the bearings. The shafts that have inner and outer rings are separated by balls or rollers, and thus rolling friction is substituted for sliding friction during the motion. Generally, two bearings assemble to the shaft to provide sufficient radial support. This type of configuration allows the ability to limit shaft bending and deflection according to safety factors. Also, highly desirable and it simplifies manufacturing. On the other hand, in some of the applications, three or more bearings can be used to provide adequate support and rigidity. But for this type of uses the bearings in the supporting structure must be aligned. Shaft axial positioning and provision for carrying thrust loads normally require that one and only one bearing take thrust in each direction. In general applications, each bearing takes thrust in the direction of the mounted surface. However, this situation may vary due to assembly requirements and interfaces. The bearings can be supported in both directions and take thrust in both directions. Therefore, the shaft bearings that support the members should be sufficiently strong and rigid [Childs, 2014].

According to manufacturing types, sometimes gears are made integral with the shaft. The shafts can be manufactured as a single unite part with gears, or they can be produced separately and assembled after the manufacturing process. If there are no assembly constraints, the gear is manufactured together with the shaft. This approach minimizes tolerances and yields a gear-shaft assembled part with high precision and strength.

3.4.1. Coarse Sizing

Shafts facilitate the positioning of components such as bearings and snap rings while also transmitting forces acting on the system, such as weight and thrust, to the shaft. For initial design, shafts should be as short as possible. Also bearing should be located close to the applied loads. Thus, deflections and bending moments decrease on the

shaft. The resources of the stress raiser should be placed away from highly stressed shaft regions. The resources of the stress raiser should be placed away from highly stressed shaft regions. If not placed, for local surfaces, the shaft diameter can be increased. Stress-raiser effects can be minimized with radii and relief. In addition, the weight is critical for UAVs, and shafts should be hollowed if possible **[Juvinall and Marshek, 2012]**.

The state of the stress is caused by three types of loads that can be identified such as torque transmitted from the gears, bending of the shaft due to its weight or load, and axial forces imparted to the shaft. For the gearboxes that affect the thrust; combined torsion, bending, and axial loading are caused the stress deformation on the shaft. This effective stress can be calculated theoretically as indicated equation (3.4).

$$\sigma_{eff} = \frac{32}{\pi d_0^3} B \left\{ \left[M + \frac{F d_0}{8} (1 + \alpha^2) \right]^2 + \frac{3}{4} T^2 \right\}^{1/2} \quad (3.4)$$

The start of the deformation occurs when the effective stress exceeds the yield strength of the ductile material. Therefore, the shaft diameter (or outside diameter) can be calculated in equation (3.5) for ductile metals where local yielding at stress concentration **[Norton, 2000]**.

$$d_0^3 = \frac{FS}{\sigma_y} \frac{32}{\pi} B \left\{ \left[M + \frac{F d_0}{8} (1 + \alpha^2) \right]^2 + \frac{3}{4} T^2 \right\}^{1/2} \quad (3.5)$$

where the factor of safety FS has been indicated for the design factor of safety. The actual loads, operating environment, material strength properties, and inaccuracies of the calculation method should be considered when identifying the safety factor. According to the criticality of the applications, the range of safety factors can be determined from 1.3 to 6. But less than 1.5 is not normally recommended **[Loewenthal, 1984]**.

The calculation of the loads on the shaft, the selection of necessary factors, and other required operations can theoretically be performed. However, instead of these lengthy calculations, a computer-aided design method has been adopted, and in Section 3.2, the gearbox components have been placed on the shaft without any modifications. The

shaft design is conducted in the KISSsoft program using the shaft calculation linked to the shaft element in KISSsys. With the help of the Shaft Editor in the program, elements affecting the system such as input geometry and material data, shaft designations, the drawing, the support, boundary conditions, and external forces and torques can be defined. Additionally, machine elements positioned on the shaft such as gears, bearings, couplings, etc., can be defined. Moreover, notch geometries such as radius, chamfer, relief groove, etc., can also be specified on the shaft as indicated **Figure 3.12**.

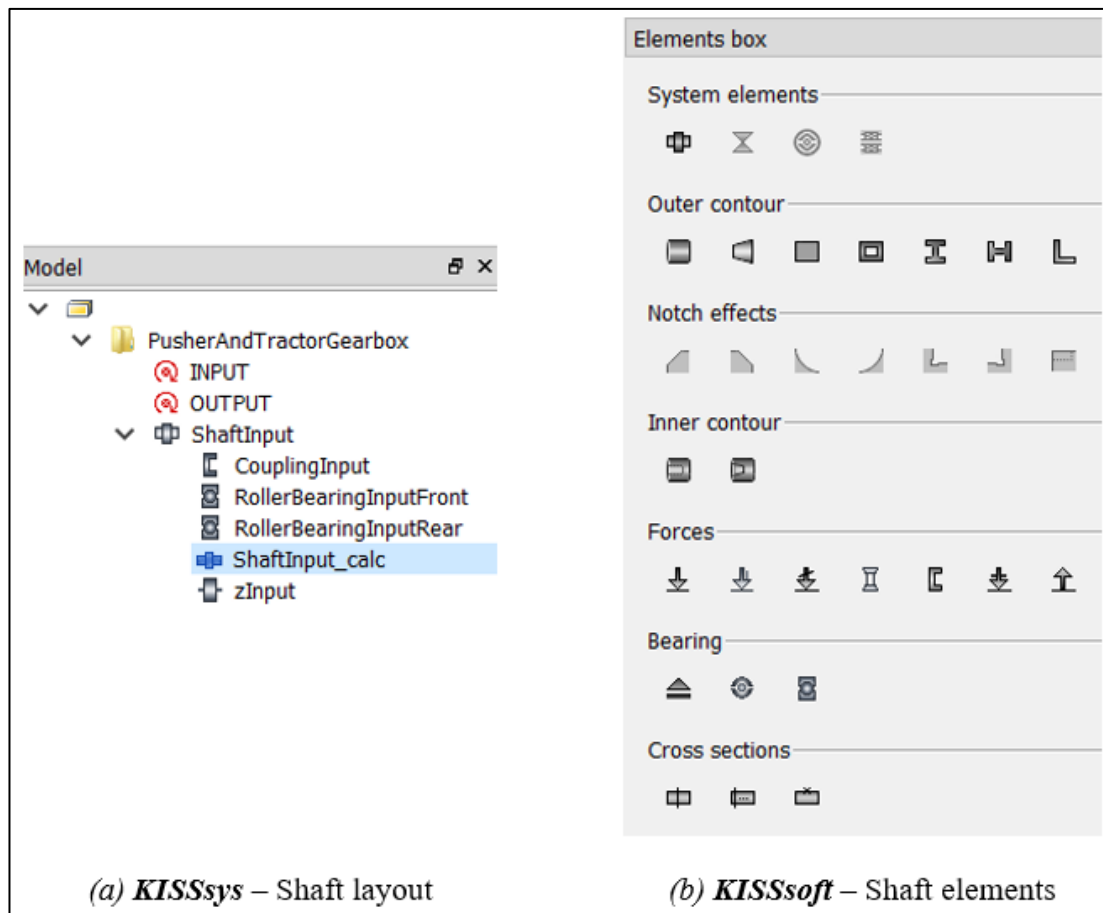


Figure 3.12: Interface of KISSsoft software [KISSsoft AG., 2022].

In the KISSsoft interface, shaft design begins with the control of module-specific setting adjustments. Subsequently, system inputs are entered, and calculation methods are determined in the Shaft Editor, Basic Data, and Strength menus. Like gear design, when designing the shaft and bearings, the default values and settings of the program should be checked. Necessary modifications should be made and defined in the system. Important parameters for the shaft and its design are specified in **Table 3.8**.

Table 3.8: Gears manufacturing data.

Parameter	Description
Non-linear shaft	OK
Reference temperature	20 °C
Failure probability	10%
Required service life	5000
Surface roughness of housing	N8 Rz=16 (Milling)
Lubrication type	Oil bath
Required system reliability	90%
Gears	Consider gears as mass and as stiffness
Rolling bearings	Rolling bearing service from inner geometry (ISO/TS16281)
Bearing lubricant temperature	95 °C
Housing material	Wrought aluminum alloy
Temperature of housing	80°C
Material (gear and shaft)	18CrNiMo7-6
Shaft operating temperature	105°C
Oil Type	Shell Spirax ATF
Lubrication type	Oil bath
Required safeties	1.5

The gear design on the shaft has been completed, and a layout without any modifications to other elements is shown in **Figure 3.13**. For the current situation, the shaft diameter can be sized using the shaft object and calculated for strength and deflection. The diameter required for strength is shown in green, while the diameter required for deflection is shown in purple. Taking the maximum value into account, the necessary shaft diameter is determined by strength requirements, and the approximate dimensions calculated are 30 mm for the input shaft, 30 mm for the intermediate shaft, and 35 mm for the propeller shaft.

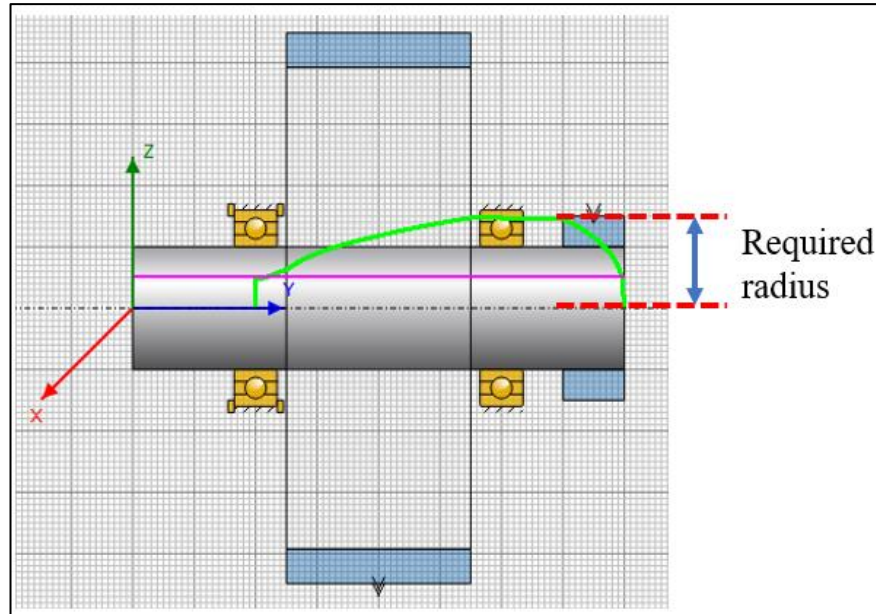


Figure 3.13: Shaft elements.

The shaft design needs to be conducted iteratively, and the shaft should be updated as components are designed. Specifically, the definition of external forces acting on the system, the selection and mounting interface of bearings, and the effects of internal hollowing for weight reduction can result in changes to the required shaft diameter. Considering these factors, the shaft design has been carried out in stages.

3.4.2. External Forces

Shaft elements such as gears, belt sheaves, pulleys, chain sprockets, etc. cause forces and deformations. The information on the methods is given for computing these forces for thesis study. In general, the principles of statics and dynamics are used to determine the forces for any element [Litvin and Fuentes, 2004].

During the power transmission, the normal force, W_n , exerted on a gear tooth is perpendicular to the involute profile. This force acts in the radial and tangential directions according to the pressure angle indicated **Figure 3.14**. The angle between the normal force and the tangential force component is determined as the pressure angle, ϕ . For most of the applications, this angle is typically 20° or 25° [Savage et al., 1981].

While the radial force, W_r , is perpendicular to the pitch diameter, PD, the tangential force, W_t , acts tangentially to the pitch diameter. Therefore, it is most convenient to

compute the tangential force that is directly from the torque being transmitted by the gear.

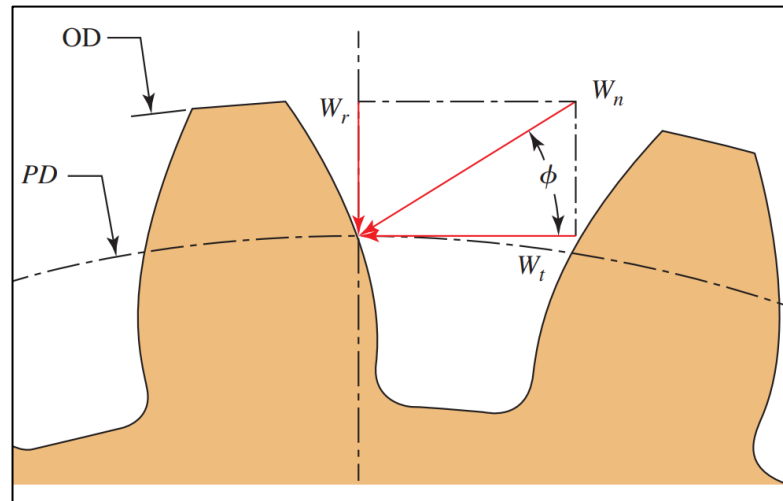


Figure 3.14: Spur gear forces [Mott et al., 2018].

Determining the forces on gears in their correct directions is important to the analysis of the stresses on the shafts. As shown in **Figure 3.15**, the forces exerted act on driven gear B by driving gear A. An important principle of mechanics states that there is an equal and opposite reaction force for each action force from gear B to gear A.

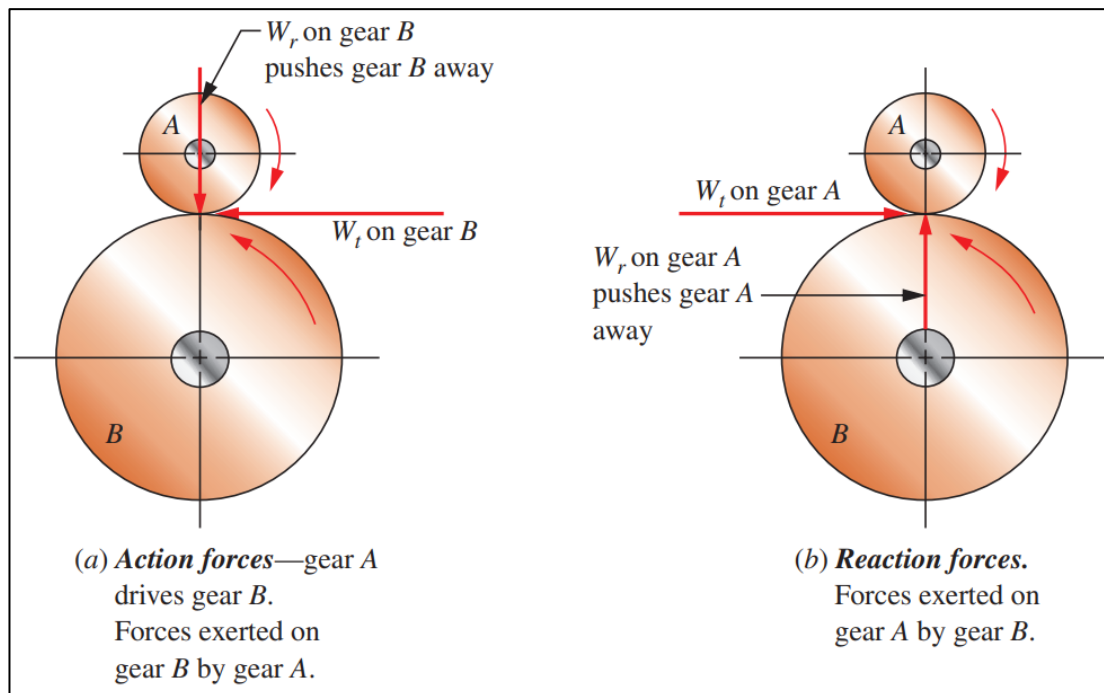


Figure 3.15: Spur gear action – reaction forces [Mott et al., 2018].

While the tangential force, W_t , of driving gear A pushes the driven gear B along the pitch diameter, the reaction force acts opposite direction. This force causes the transmission between gears. At the same time, the tangential forces induce torsional stress when making the shaft transmit torque. This stress must be considered during the shaft design to ensure the shaft can be transmitted torque without failure.

The radial force, W_r , exerted by the driving gear A, acts along the center of distance tending to push the driven gear B away. Similarly, the reaction force affects gear A. Because of the radial forces cause bending stress, leading to shaft deflection. This can result in misalignment. Also, uneven loading on the gears and bearings increases wear and reduces lifetime [Mott et al., 2018].

Although typically minimal in spur gears, axial forces, F_A , can occur due to installation errors. Also, external loads can affect the system. In this study, the gearbox has an interface between the propeller. Additionally, according to requirements, the system must transmit the power for pusher and tractor-type propulsion systems. It means thrust acts in both directions for the propeller shaft indicated **Figure 3.16**.

These forces act along the axis of the shaft. Thrust creates axial loads on the shaft, affecting the alignment and load distribution on the bearings. Proper bearing selection and shaft design are crucial to manage these loads and prevent catastrophic failure. Additionally, the shaft is subjected to both pusher and tractor forces, as well as the weight and inertia of the propeller.

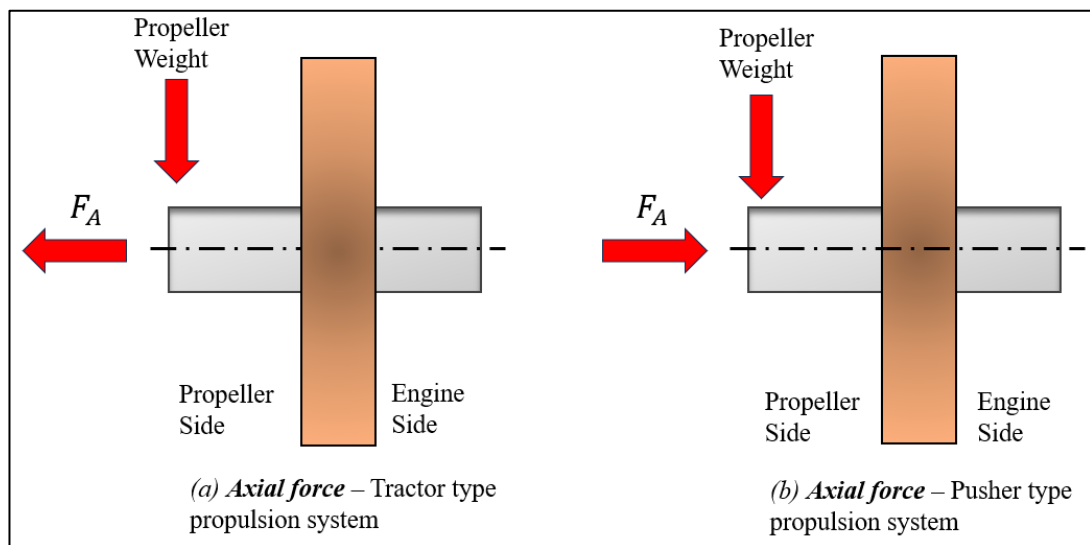


Figure 3.16: Thrust forces for pusher and tractor types.

3.4.3. Bearings

Rolling bearings enable support and guide machine elements such as shafts, axles, or wheels. The bearings are better than transmitting loads between machine components with minimal friction, rotating or oscillating compared to the sliding bearings as shown **Figure 3.17**. Rolling bearings provide high precision and low friction. Therefore, it enables high rotational speeds while reducing noise, heat, energy consumption, and wear. They are cost-effective and exchangeable machine elements that typically international dimension standards [Oberg et al., 2008].

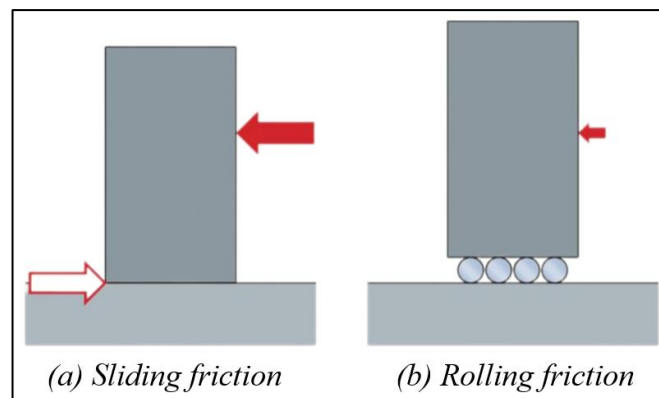


Figure 3.17: Bearing element frictions [SKF Group, 2018].

There are two types of rolling elements distinguished for rolling bearing. While balls make point contact with the ring raceways, rollers make line contact with the ring raceways indicated **Figure 3.18**. With increasing load acting on the bearing, while the ball contact point becomes an elliptical area, the roller contact line becomes rectangular. Ball bearings provide a small contact area. The small contact area provides low rolling friction and accommodates high speeds but also limits their load-carrying capacity. On the other hand, roller bearings enable a larger contact area and consequently cause higher friction, a roller bearing can accommodate heavier loads, but at lower speeds, than a same-sized ball bearing [Childs, 2014].

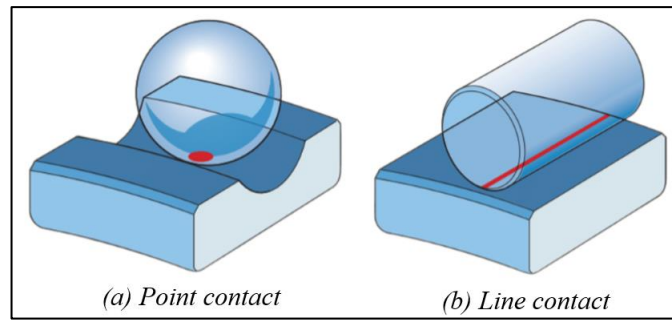


Figure 3.18: Rolling element contact types [SKF Group, 2018].

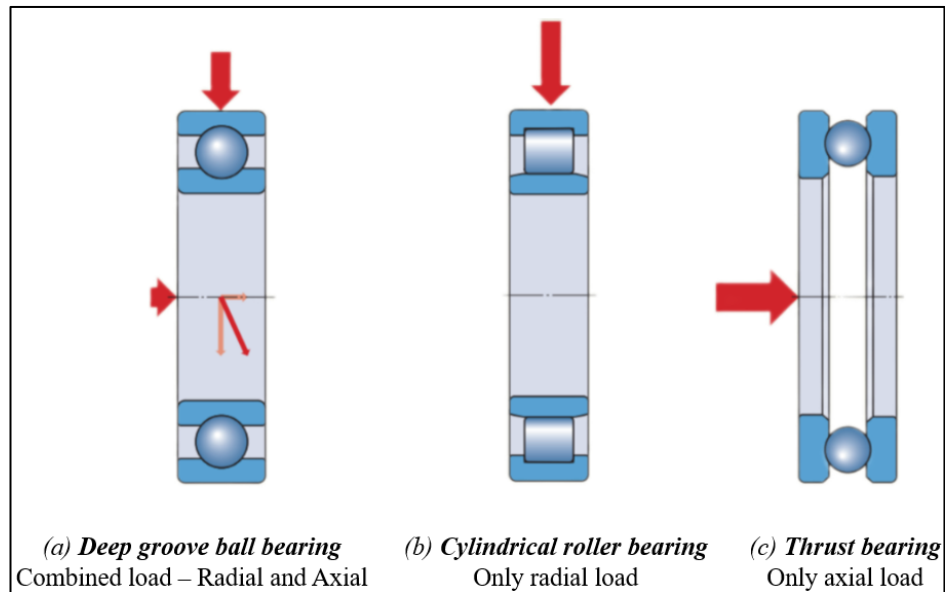
According to the configuration of the rolling elements, different types of roller bearings are used for various applications. Deep groove ball bearings are designed for load-carrying primarily, although the deep grooves provide to fairly carrying of the axial loads. Also, double row deep groove ball bearing with the second row of balls, the load-carrying capacity is increased compared with the single-row design because more balls share the load. In addition, angular contact bearing is designed for the axial loads. It allows greater thrust loads compared with the standard single-row, deep-groove bearing.

By replacing the balls with cylindrical rolling elements different types of bearings such as cylindrical roller, needle spherical roller, and tapered roller are accomplished as a result. Cylindrical roller bearings allow carrying a higher load against ball bearings, but they are not recommended that thrust load be applied. Needle bearings are cylindrical roller bearings that have much smaller diameter rolling elements. A smaller radial space is typically required for needle bearings to carry a given load than for rolling bearings. The spherical roller bearings provide relative rotation of the outer race to rollers and the inner race when angular misalignments occur. This capability supplies an excellent rating for misalignment capability. Tapered roller bearings are used against substantial thrust loads along with high radial loads and consequently give result excellent ratings on both. The bearing types so far are used for radial or combined - radial and axial - loads. In contrast, thrust bearings are designed exclusively for axial loads. Most of the thrust bearings cannot work against radial loads or can take little. For the mentioned types of bearing, a comparison of the performance relative to the others is given in **Table 3.9** [Mott et al., 2018].

Table 3.9: Bearings comparison [Mott et al., 2018].

Bearing type	Radial load	Thrust load	Misalignment
Single-row, deep-groove ball	Good	Fair	Fair
Double-row, deep-groove ball	Excellent	Good	Fair
Angular contact	Good	Excellent	Poor
Cylindrical roller	Excellent	Poor	Fair
Needle	Excellent	Poor	Poor
Spherical roller	Excellent	Fair	Excellent
Tapered roller	Excellent	Excellent	Poor
Thrust	Fair	Excellent	Poor

Briefly stated, the design, or the selection of bearings that will be used, is dependent on the type and magnitude of loads, the design life, cost efficiency, ease of supply, and maintainability. The load-carrying capacities for the most fundamental bearings with widely used areas are shown in **Figure 3.19**. In this study, primarily deep groove ball bearings and cylindrical roller bearings are selected for the gearbox because of their advantages.

**Figure 3.19:** Rolling element contact types [SKF Group, 2018].

While the sizes of the bearings are selected for the gearbox, design parameters such as the load capacity, the geometry of the bearing, the size of the shaft, requirements, etc. are considered. In addition, the bearings must have a higher load rate compared to the system load rating to provide an acceptable combination of life and reliability. Bearing applications usually require life and dynamic load calculations by using equations.

Ball and roller bearing lives are confirmed to vary inversely with approximately the third or fourth power of the load. Many manufacturers assume Palmgren's exponent as 3 for ball bearings and 3.33 for roller bearings [Shigley et al., 2004].

$$L = \left(C_r / F_e \right)^{1/k} (10^6) \quad (3.6)$$

$$C_{req} = (F_e) \left(\frac{L_d}{10^6} \right)^{1/k} \quad (3.7)$$

$$L_d = (h)(RPM)(60 \text{ min/h}) \quad (3.8)$$

According to durability tests, the life of rolling element bearings especially ball bearings is about five times the standard 10 percent failure fatigue life. It also means that this is the life in which 90 percent have not failed, or 90 percent reliability. The standard life is commonly designated as the L_{10} life and achieves a rated life (L_{10}) of 1 million revolutions (10^6 rev.). Adjustment factor for reliability is an equal to 1 for L_{10} life, therefore it has not been considered. While cylindrical roller bearings are very limited in their axial load capacity, deep groove ball bearings expose a combination of radial and axial loads. Therefore, the equivalent load is equal to radial loads for cylindrical roller bearings, whereas, for deep groove ball bearings, the equivalent load is calculated according to following equations [Juvinal and Marshek, 2012].

$$\text{For } 0 < W_t/W_r < 0.35, F_e = W_r \quad (3.4)$$

$$\text{For } 0.35 < W_t/W_r < 10, F_e = W_r \left[1 + 1.115 \left(\frac{W_t}{W_r} - 0.35 \right) \right] \quad (3.4)$$

$$\text{For } W_t/W_r > 10, F_e = 1.176 W_t \quad (3.4)$$

Since the gear system is designed with spur gears, there are no axial forces exerted on the bearings due to the gears. Therefore, cylindrical roller bearings have been selected for the input and intermediate shafts. The propeller shaft, being subjected to both radial

and thrust forces, uses a deep groove ball bearing at the front and a cylindrical roller bearing at the rear. The necessary dynamic load calculations for the bearings were performed based on the design inputs, requirements, and the loads acting on the shafts, using the specified formulas. The dynamic load for the bearings supporting the input and intermediate shafts was calculated to be approximately 20 kN. According to the SKF catalogs, bearings with a minimum inner diameter of 17 mm can support a dynamic load of 20 kN. In the preliminary shaft calculation done in KISSsoft, a diameter of 30 mm was determined to be necessary for the input and intermediate shafts. Therefore, cylindrical roller bearings with an inner diameter of 30 mm have been selected for the input and intermediate shafts, considering the minimum shaft diameter requirement.

The dynamic load values for the bearings supporting the propeller shaft have been calculated to be approximately 36 kN for the front bearing and 29 kN for the rear bearing. According to the SKF catalogs, a bearing with a minimum inner diameter of 20 mm is required for a dynamic load value of 29 kN. However, the preliminary shaft calculation determined a minimum diameter of 35 mm for the propeller shaft, leading to the selection of a cylindrical roller bearing with a 35 mm diameter. For the front bearing of the propeller shaft, which is subjected to both radial and axial loads, a bearing with a minimum inner diameter of 40 mm is necessary to meet the dynamic load requirement. Since the required bearing inner diameter is larger than the minimum propeller shaft diameter, the shaft diameter for the front bearing was updated accordingly. As a result, a deep groove ball bearing with an inner diameter of 40 mm was selected for the front bearing of the propeller shaft.

Table 3.10: Bearing positions, type and lifetime.

Bearing assembly position	Bearing	Lifetime
Input shaft – Front casing	NJ 206 ECJ	7500
Input shaft – Rear casing	NUP 206 ECP	10000
Intermediate shaft – Front casing	NJ 2206 ECJ	5700
Intermediate shaft – Rear casing	NJ 2206 ECJ	5700
Propeller shaft – Front casing	6308	6900
Propeller shaft – Front casing	NJ 206 ECJ	7500

3.4.4. Results of Shaft Design

As mentioned in previous sections, the shaft design needs to be performed iteratively, and preliminary shaft dimensions have been determined using the KISSsoft program. After determining the forces acting on the system and selecting the appropriate bearings, the shaft diameters were finalized. Successful application of the bearings requires not only selecting the appropriate bearing type and size based on load-carrying capacity and bore size but also ensuring correct installation. One bearing is fixed while the other is allowed to move axially to accommodate thermal expansion and prevent binding. The fixed bearing is held against a shoulder on the shaft and secured within the housing with a retaining ring. The movable bearing is installed with allowance for axial movement to prevent excessive restriction. Proper installation ensures that the bearing maintains correct alignment, operates efficiently, and has a long service life [Mott et al., 2018].

The shaft geometries resulting from the bearing positions and diameters have been determined in the KISSsoft program according to the gearbox operating conditions. The parameters are specified in **Table 3.11**. When performing shaft analyses, the geometry mesh structure was selected as normal. Since the shafts will be produced together with the gears, the same material specification has been defined, and the operating temperatures have been determined. As specified in the bearing calculation, the required service life has been defined as 90%.

Table 3.11: Shaft design parameters.

Parameter	Description
Node density	normal
Reference temperature	20 °C
Material (gear and shaft)	18CrNiMo7-6
Oil Type	Shell Spirax ATF
Lubrication type	Oil bath
Lubricant temperature	95 °C
Required service life	5000 hours
Bearing failure probability	10 %
Required system reliability	90 %

The final shaft designs, along with the Elements Tree used and the resulting geometries, are shown in **Figure 3.20** for the input shaft, **Figure 3.21** for the intermediate shaft, and **Figure 3.22** for the propeller shaft.

A relief groove has been added to the bearing seating surfaces to define grinding clearance, and radii have been defined in the necessary regions. With the help of KISSsoft, a Cross Section addition has been made to analyze critical sections. The bearings on the input and intermediate shafts are cylindrical roller bearings. The mounting interfaces of these bearings are considered, with those on the front housing fixed from the right side and those on the rear housing fixed from the left side. This means that the outer races of the bearings will be mounted from the inside of the housings. Mounting the bearings from the inside has been chosen for better sealing control. If they were mounted from the outer surface of the housing, additional designs such as O-rings, O-ring grooves, and covers would need to be lubricated to ensure proper sealing.

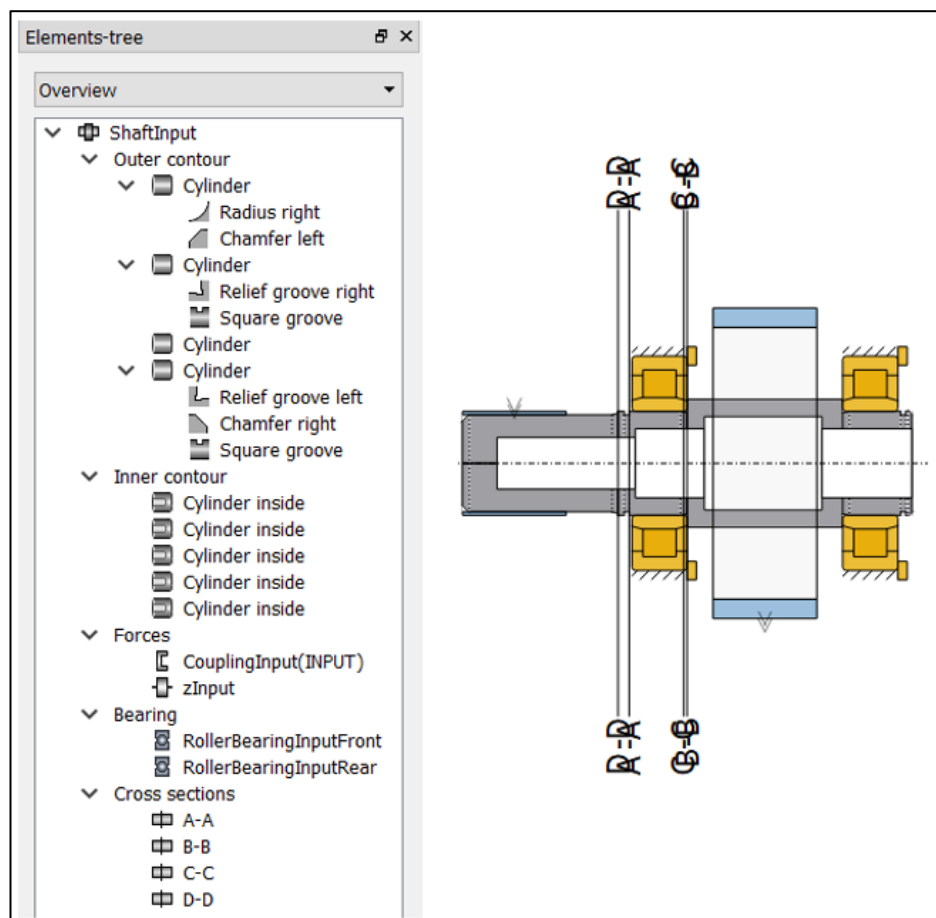


Figure 3.20: Input shaft element tree and geometry.

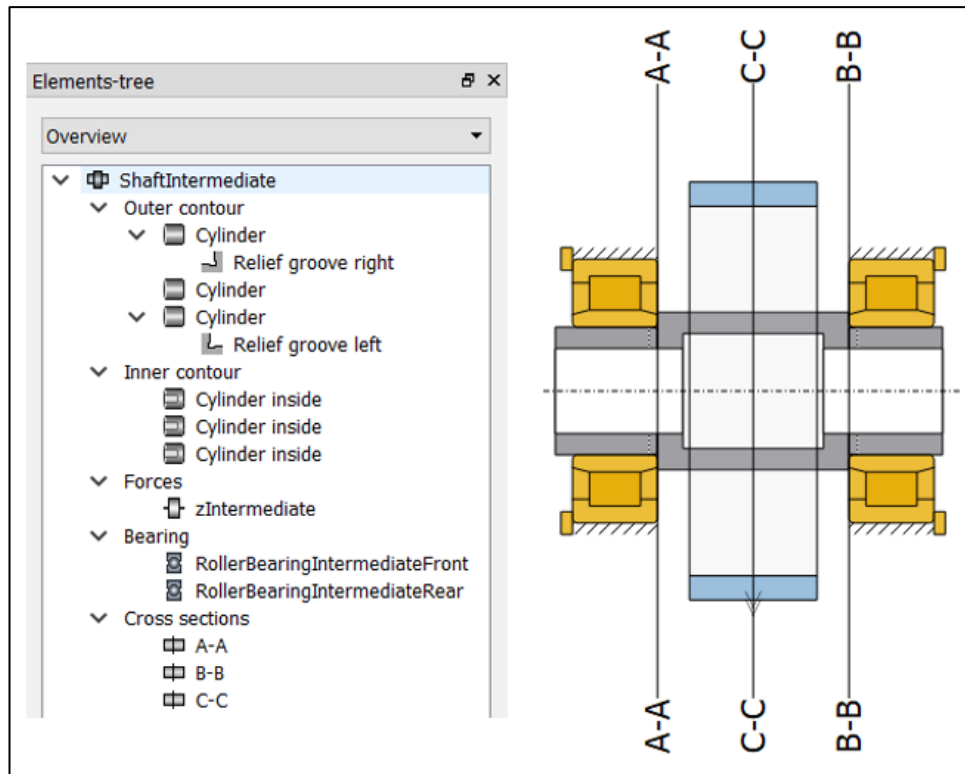


Figure 3.21: Intermediate shaft element tree and geometry.

Unlike the input and intermediate shafts, the propeller shaft is subjected to bidirectional thrust forces due to the pusher and tractor propulsion systems. These thrust forces have been defined to act in both directions, and their effects have been observed in the system by activating elements according to the type of UAV. For both thrust systems, ball bearings have been mounted from the right and left sides to effectively handle the bidirectional thrust loads.

The propeller shaft has been designed to be longer than the other shafts due to its mounting connection points and the need to accommodate the propeller's weight. Additionally, the weight of the propeller has been specified and incorporated into the design to ensure that the shaft can withstand the operational loads without compromising performance or safety. By carefully analyzing and defining the thrust forces and using ball bearings on both sides, the design ensures optimal performance and reliability for both pusher and tractor UAV configurations.

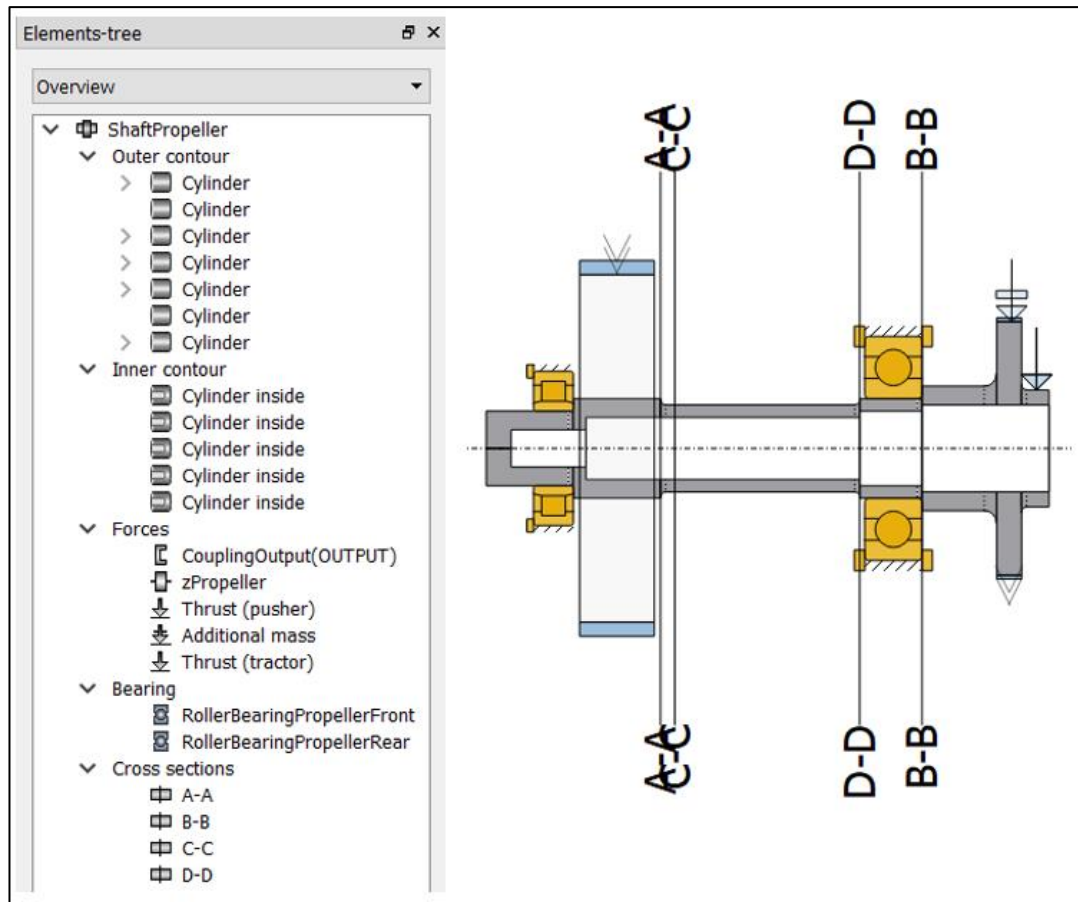


Figure 3.22: Propeller shaft element tree and geometry.

After the selection of bearings, loads, sections, and all other definitions, the final shaft analyses were conducted. The load application diagrams for the shafts are shown in **Figure 3.23**, **Figure 3.24**, and **Figure 3.25**.

The safety factors for the most critical sections on the shafts were analyzed and found to be 2.77 for the input shaft, 2.35 for the intermediate shaft, and 2.93 for the propeller shaft, as indicated in **Table 3.12**.

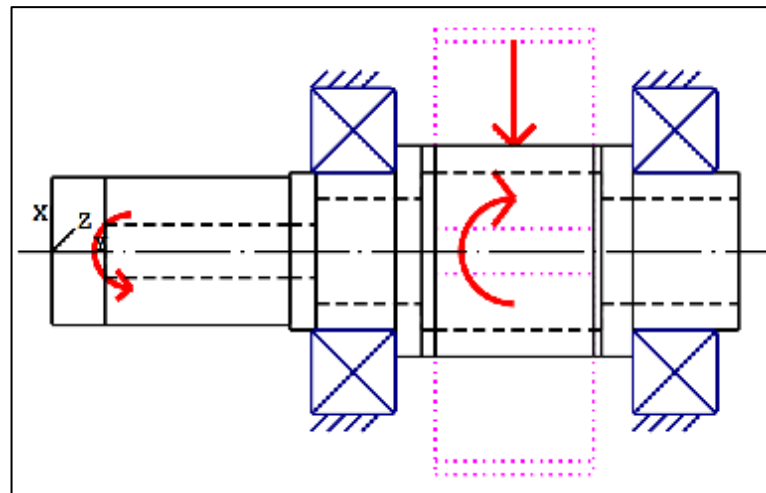


Figure 3.23: Input shaft load applications.

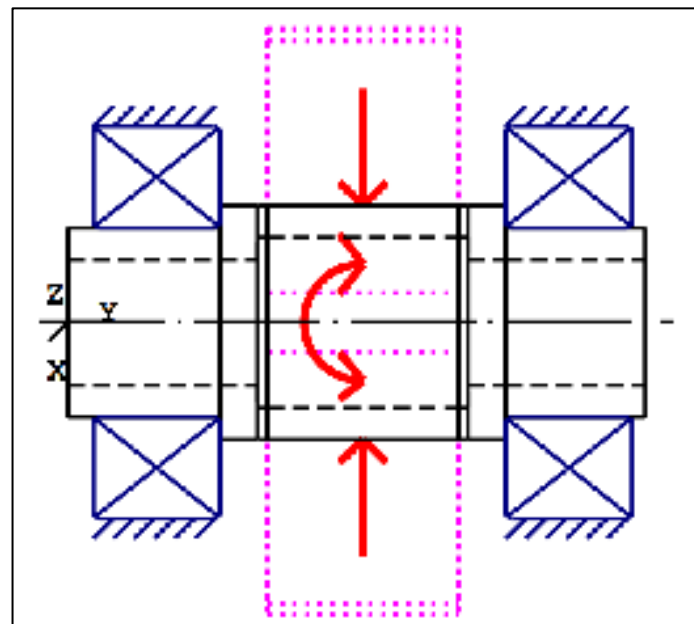


Figure 3.24: Intermediate shaft load applications.

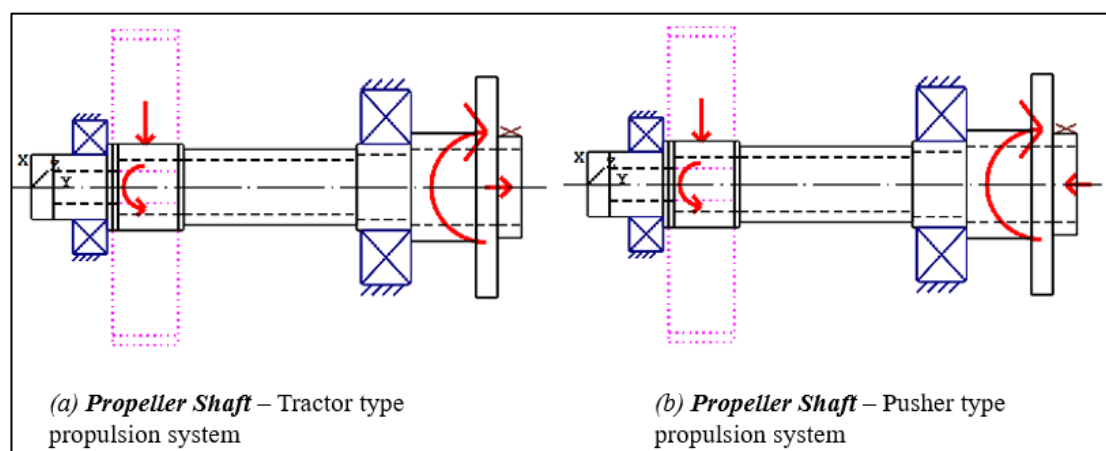


Figure 3.25: Propeller shaft load application.

Table 3.12: Shaft results.

Parameter	Input	Intermediate	Propeller
Torque (Nm)	279	303	476
Radial force (N)	2340	2500	2500
Tangential force (N)	6580	6520	6520
Axial force (N)	-	-	±2400
Length (mm)	130	91	226
Weight (kg)	0.496	0.343	1.65
Safety factor	2.77	2,35	2.93
Maximum deflection (μm)	25.7	49.6	47.3

3.5. Casing Design

In propulsion systems, a gearbox typically includes components such as gears, shafts, bearings, and casings. The gearbox casing is a critical structure that contains and positions the internal mechanical components of the gearbox. These components have been examined in detail in previous sections. One of the fundamental functions of the gearbox casing is to provide robust mechanical support to the rotating components inside, namely the gears, shafts, and their bearings. This support ensures that the components are correctly alignment relative to one another and operate concentricity. Additionally, the gearbox casing protects the internal mechanical components from adverse environmental effects, such as dirt, moisture, dust, and impacts. This protection ensures the longevity and reliable operation of the gearbox [Rizwan et al., 2019].

Additionally, the lubrication and cooling of mechanical elements within the gearbox are necessary. The casing ensures the containment of the fluid that lubricates the internal components and reduces friction. This containment must be completely sealed to prevent leaks. Adequate lubrication reduces the wear of gears and other moving components. Moreover, the cooling effect prevents overheating of the components [Patil et al., 2019].

In this context, the gearbox casing must meet several requirements. It should provide mechanical support for the main rotary transmission components through bearing mounts. Additionally, it should support selective mechanisms, actuators, cooling systems, and potential pumps. The casing must ensure proper gear engagement by adhering to the assembly tolerances of the rotary components. It also needs to

withstand the system load resulting from the transmitted torque. Furthermore, it must provide mechanical protection for the internal components of the transmission [Amaral, 2023].

After successfully completing the detailed design and optimization of the gears, shafts, and bearings using the KISSsoft program, the geartrain layout to be enclosed by the gearbox housing is shown in **Figures 3.26** and **Figure 3.27**.

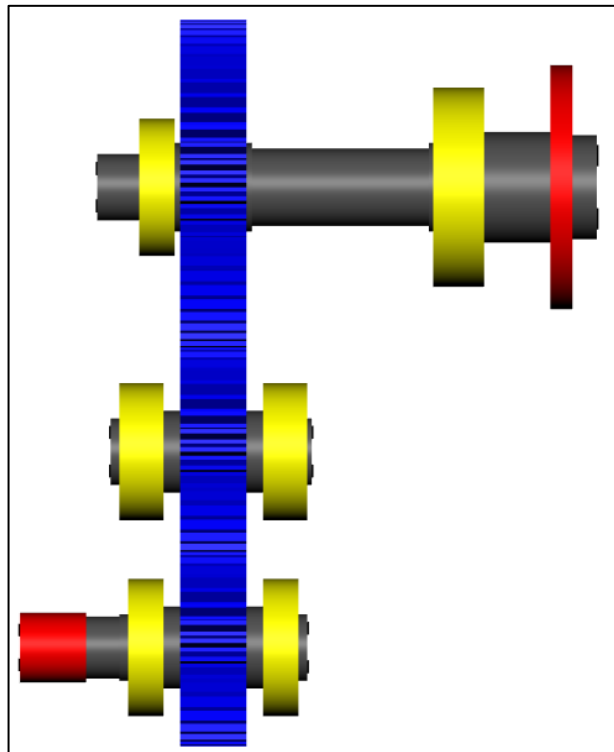


Figure 3.26: Geartrain layout side view.

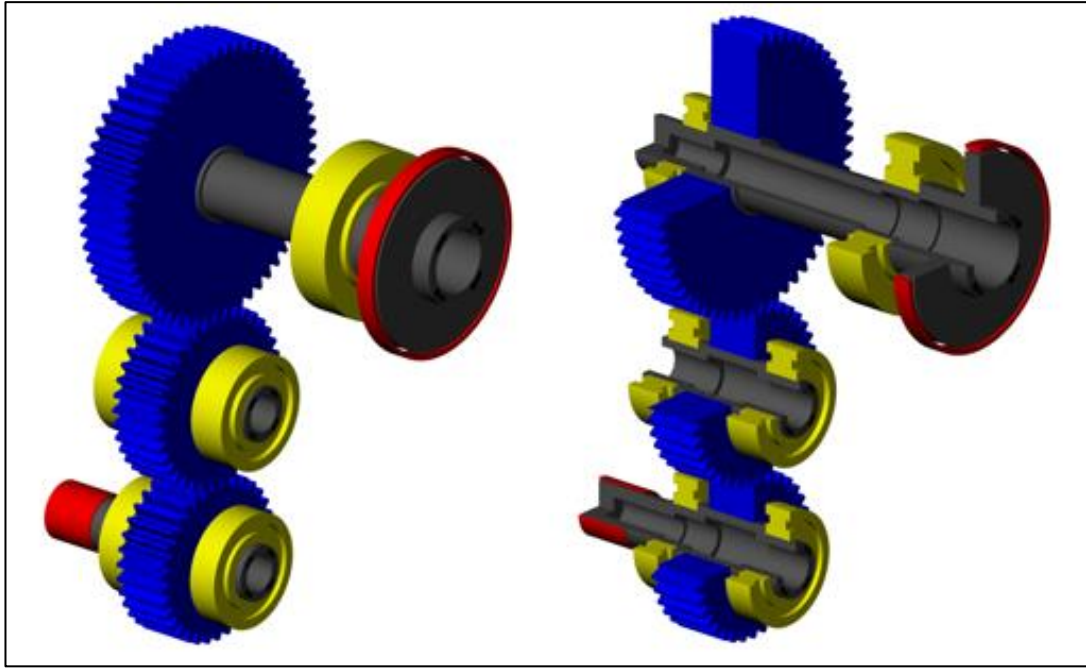


Figure 3.27: Geartrain layout isometric and section view.

In the design of the casing, the process initially focused on the placement of the bearings. In the casing design, bearings play a critical role in the overall performance and durability of the gearbox. Bearings are used to support the shafts in the gearbox and stabilize their movements. Accurately determining the tolerances between the bearings and the housing is essential for the reliability and performance of the gearbox [Maláková, 2021].

Tolerances ensure that the bearings fit correctly into the casing and provide the required load-carrying capacity. Additionally, they prevent the bearings from being overloaded, moving within the casing due to temperature changes, or experiencing stress. Therefore, the design applies a tight fit tolerance of S6 for the casing diameters and m6 for the shaft diameters. The bearing installation tolerance ranges for the casing and shafts are indicated in **Figure 3.28**, and the explanations are provided in **Table 3.13**. [Web 5, 2024].

General-purpose O-rings have an average hardness of 70 Shore, and this value has been considered for selections within the scope of this work. Additionally, O-rings should not chemically react with the working fluid. It is essential to ensure that there is no chemical reaction with oils or other chemical fluids. If a reaction occurs, the O-ring will begin to deform and lose its functionality. Fluorocarbon (FKM) has been selected as the O-ring material due to its superior physical properties, including excellent performance at high temperatures, high resistance to working fluids, and high-pressure tolerance. [Parker, 2021].

O-ring grooves can be categorized into two groups: static and dynamic applications. In the design of the common gearbox, the O-rings are subjected to static applications since they are used for casings and covers where there is no radial or axial movement. In static applications, O-ring grooves can have rectangular, triangular, or dovetail shapes. Dovetail grooves, which have a different geometry, are more costly to manufacture compared to the others. Generally, the trapezoidal geometry of dovetail grooves limits the movement of the O-ring within the groove, making it very difficult for the O-ring to come out of the groove, and also provides ease of assembly for large diameter O-rings. The three different groove geometries are shown in **Figure 3.29** and **Figure 3.30** [Web 6, 2024].

A rectangular groove design has been selected for the O-rings used between the cover and casings. For the O-rings between the casings, a dovetail groove design has been implemented since the O-rings are long and need to remain fixed during assembly.

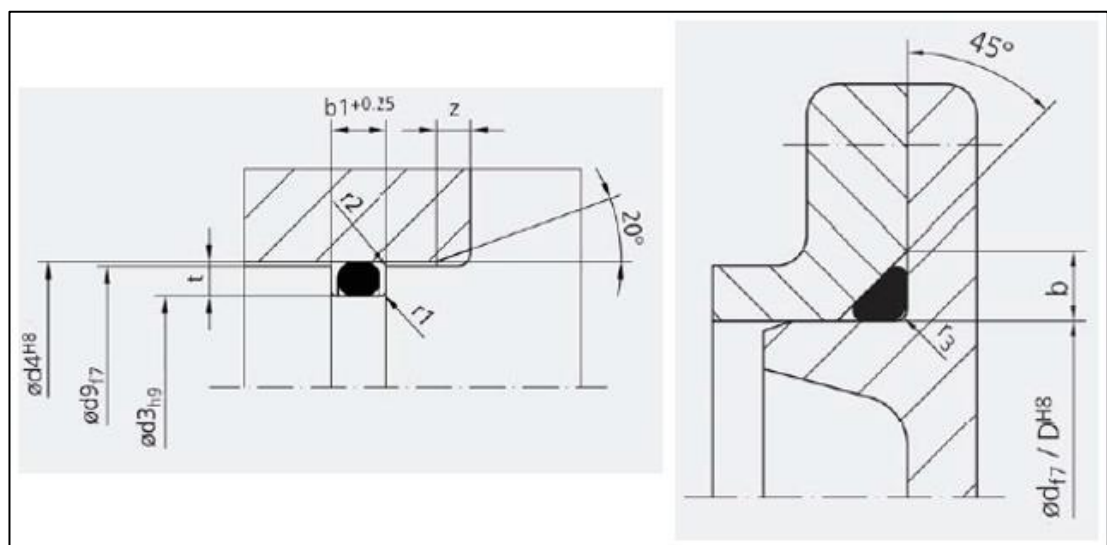


Figure 3.29: Rectangular and triangular grooves.

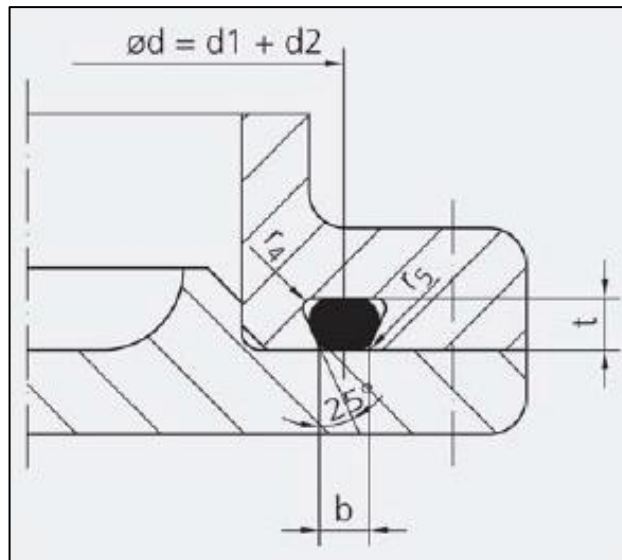


Figure 3.30: Dovetail groove.

After determining the O-ring diameter, length, and groove, the fill rate, stretch rate, and compression rate were calculated. During assembly, the O-ring is expected to stretch slightly to fit into the groove. The benefit of this stretch allowance is to restrict the movement of the O-ring within the groove, prevent unwanted O-ring extrusion, and provide a more stable assembly. The fill rate is a parameter that indicates how much of the groove is filled by the O-ring, calculated as the ratio of the O-ring cross-sectional area to the groove cross-sectional area. The compression rate is a parameter that indicates the amount of compression the O-ring experiences within the groove after installation, calculated as the ratio of the O-ring cross-sectional diameter to the height of the groove. These calculations were performed to meet the values specified in **Table 3.14 [Web 7, 2024]**.

Table 3.14: O-ring calculation parameters.

Parameter (%)	Minimum	Maximum
Gland fill	60	90
Stretching ratio	0	5
Compression ratio	5	35

While the sealing of static parts is ensured with O-rings, the sealing of rotating parts is achieved with radial seals. To ensure the reliable sealing performance and long lifespan of radial seals, the shaft and housing requirements must be carefully considered. The shaft surface roughness should be specified in the design details

according to ISO 4288 or DIN 4768 standards, with an Ra value of 0.2–0.5 μm , and the housing surface roughness should have an Ra value of 1.6–3.2 μm . The hardness of the shaft surface should be at least 45 HRC; for PTFE-lipped seals, this value should be at least 58 HRC.

Material selection in the choice of radial shaft seals is a critical step that directly affects the seal's performance and lifespan. Nitrile rubber (NBR) is a material known for its high oil resistance and suitability for use at low temperatures, although its performance can degrade at high temperatures. Viton (FKM) is renowned for its high-temperature and chemical resistance, making it ideal for environments with extreme temperatures and aggressive chemicals. Silicone provides flexibility over a wide temperature range and performs well even at low temperatures, but its mechanical strength is lower compared to other materials. Polyurethane (PU) offers high wear resistance and moderate temperature resistance, making it suitable for abrasive environments. The selected material must be compatible with the application conditions and the type of lubricant used [SKF group, 2020].

Due to the high temperatures and the use of chemical oils in the gearbox, FKM material has been selected for its superior resistance properties. The inner and outer diameters of the seals have been carefully determined based on the specific geometries of the shaft and casing to ensure a proper fit. In total, two seals are used within the gearbox to provide effective sealing and prevent leakage. One is positioned at the interface where the propeller shaft is located, and the other is placed between the motor input shaft and the casing.

The casing designs have been completed based on the gear system, as well as the outer diameters of the bearings, O-rings, and radial seals. These components have been carefully considered to ensure optimal fit and function within the gearbox. The assembly visuals of the gearbox are shown in **Figure 3.31**, and the exploded view is depicted in **Figure 3.32**.

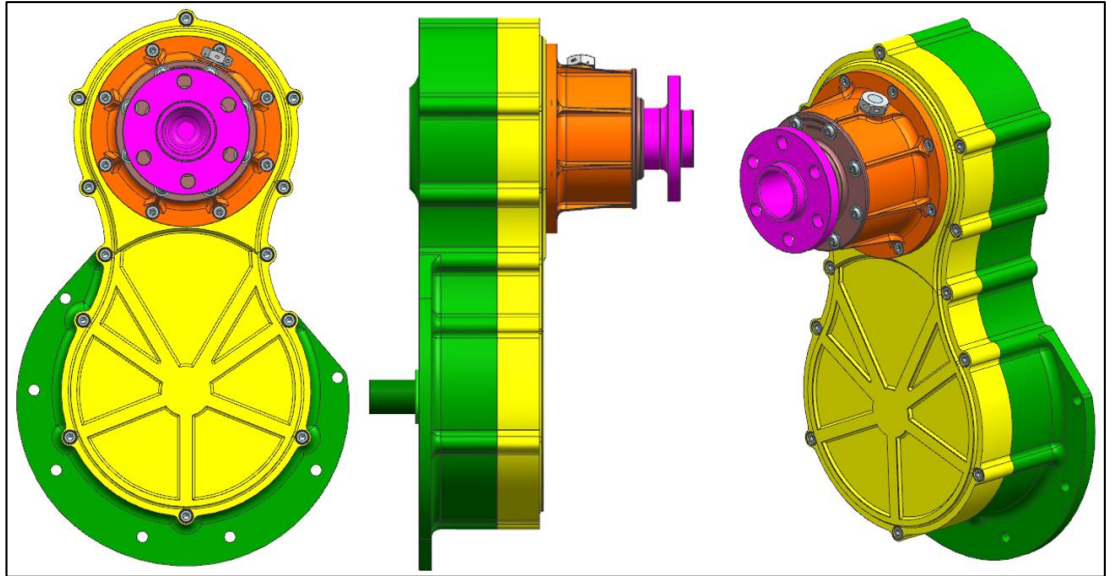


Figure 3.31: Gearbox general views.

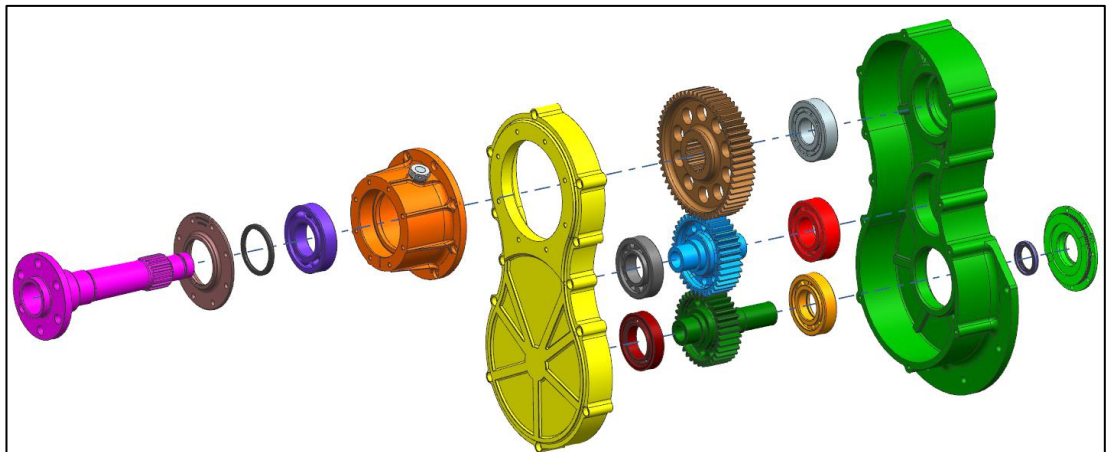


Figure 3.32: Gearbox exploded view.

4. RESULTS

In this study, a comprehensive approach to the common gearbox design for unmanned aerial vehicles (UAVs) with both pusher and tractor thrust systems has been taken. The design process was carried out using advanced software tools such as KISSsys and KISSsoft, allowing for the design and analysis of system components. Initially, a literature review specific to the gearbox was conducted. Detailed information about the classification of UAVs and the types of motors used was provided. The system requirements and assumptions for the thrust systems, which include both pusher and tractor types, were determined.

Based on this information, the gearbox design began with the creation of the system kinematics using the KISSsys program, followed by the design of gears, bearings, shafts, and other elements using the KISSsoft program. Throughout the design process, input-output parameters and critical design parameters were examined in detail. Spur gears were designed to operate without generating axial forces in both systems. The importance of parameters affecting gear design, such as module, contact ratio, root safety, and flank safety, in ensuring the reliability, longevity, and selection of gears was emphasized. The torque, moments, and forces on the shaft caused by gears, the propeller, and thrust in opposing directions were analyzed, and the necessary shaft diameters were determined, forming the initial design. Bearings were selected for supporting the shaft and its components. Minimum shaft diameters, system loads, and bearings suitable for both thrust systems, such as deep groove ball bearings and cylindrical roller bearings, were considered.

After determining the bearings, the positions of gears, bearings, and other components on the shaft were established, and the final shaft design was completed. At the end of the study, a common gearbox design for UAVs with both pusher and tractor thrust systems was finalized. The designed gearbox provides significant advantages in terms of maintenance and upkeep, and having a common production for both types of UAVs offer economic benefits.

5. CONCLUSIONS

The gearbox designed for two different aircraft with pusher and tractor type propulsion systems serves as a reference for the literature and the aviation industry. Literature reviews have examined gearbox designs for aircraft, but no detailed studies were found for both pusher and tractor aircraft.

The gear system is subjected to axial force due to thrust. The thrust force acts in opposite directions depending on the pusher or tractor configuration. Designs have been implemented with straight gears to ensure that no additional axial force affects the gear system aside from the thrust force. Gear designs can also be made with helical gears, which can result in a quieter system with smoother power transmission. However, the axial force generated by helical gear design, combined with the thrust force, will affect the lifespan of the gears and shafts. Therefore, bearing designs may need to be revised to meet the required lifespan. Shaft designs must also be considered in relation to the gears and bearings.

In gearbox design, system optimization can be achieved by considering the thrust force acting on the propeller shaft and the system, as well as the aerodynamics of the propeller. Casing designs can be detailed by considering the flow dynamics affecting the aircraft. Additionally, lubrication system flow analyses can be performed and detailed, and if necessary, pump and cooler integration can be added to the system. In addition to all these studies, optimization efforts can be made for weight, which is one of the most important parameters for unmanned aerial vehicle

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BIOGRAPHY

The author, İbrahim Ethem KAZANCI, commenced his professional journey as a Mechanical Design Engineer in the automotive industry. At present, he holds the position of Design Lead Engineer in the propulsion systems for Unmanned Aerial Vehicles.

PUBLICATIONS AND PRESENTATIONS FROM THE THESIS

Kazancı İ. E., (2024), “Common Gearbox Design for Pusher and Tractor Type Propulsion Systems in Unmanned Air Vehicles”, 3rd Bilsel International Ahlat Scientific Researches Congress, 411-427, Bitlis, Turkey, 08-09 June.