

MODELING AND TESTING OF A PROPORTIONAL WASTEGATE FOR RPM
REGULATION OF A TURBOPUMP SYSTEM

by

Mehmet Reşat UFUK

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ABSTRACT

MODELING AND TESTING OF A PROPORTIONAL WASTEGATE FOR RPM REGULATION OF A TURBOPUMP SYSTEM

Launch vehicles use pump-fed pressurization systems, and the energy to drive the pump can be obtained in various ways. Small rockets may use electric motors, but larger liquid rocket engines often use gas generator cycles, where the turbine is driven by pre-combustion chamber products. In this study, a novel PID controller system has been proposed in which the hot gas resulting from the combustion of a hybrid rocket motor or cold gas resulting from pressurized tank is regulated before entering the turbine, similar to the turbochargers in cars. Due to the characteristics of the hybrid rocket used in the hot gas generator, the energy produced decreases over time, and therefore the system produces more energy during operation than the turbopump system requires. The required amount of the produced gas is directed to the turbopump system and the rest is exhausted using a proportional wastegate. The excess energy produced ensures that RPM stays constant during pressure fluctuations in the combustion chamber. The proposed system employs a closed-loop control strategy, where feedback from the turbine speed is measured with a distance sensor and this feedback is used to adjust the wastegate position and maintain the desired RPM. In this study, modeling and characterization of the proportional wastegate were conducted, and a turbine speed sensor was developed using a proximity sensor. System was modeled in MATLAB Simulink, and control algorithm was developed. Then, tests were conducted with both hot and cold gas generators to validate the system's performance. Both simulation and test results demonstrate that the proposed control system can effectively regulates the turbopump speed within a desired range, even under disturbances.

ÖZET

TURBOPOMPA SİSTEMİNİN RPM DÜZENLEMESİ İÇİN KULLANILACAK ORANSAL TAHLİYE VANASININ MODELLENMESİ VE TESTİ

Fırlatma araçları basınçlandırma sistemleri olarak pompa beslemeli sistemleri kullanır ve pompayı çalıştırmak için gereken enerji farklı yollarla elde edilebilir. Küçük roketler elektrik motorları kullanabilir, daha büyük sıvı roketler ise genellikle türbinin ön yanma odasının ürünleri tarafından çalıştırıldığı gaz üretici çevrimi kullanır. Bu çalışmada, hibrit roket motorunun yanmasıyla oluşan sıcak gazın veya yüksek basınçlı tanktan çıkan soğuk gazın, araçlardaki turboşarjlara benzer şekilde türbine girmeden önce regüle edildiği yeni bir PID kontrol sistemi önerilmiştir. Sıcak gaz üretici olarak kullanılan hibrit roketten dolayı sistemin ürettiği enerji zamanla azalmakta, bu nedenle sistemin operasyon sırasında turbopompa sisteminin ihtiyaç duyduğundan daha fazla enerji üretmesi gerekmektedir. Üretilen gazın gerekli miktarı turbopompa sistemine yönlendirilmekte ve geri kalanı oransal tahliye vanası kullanılarak dışarı atılmaktadır. Gaz üreticinin ürettiği fazla enerji, yanma odasındaki basınç dalgalanmaları sırasında RPM'in sabit tutulması için kullanılmaktadır. Önerilen sistemde, sensör tarafından ölçülen türbin hızının, oransal tahliye vanasının konumunu ayarlamak ve istenen RPM'i korumak için kullanıldığı kapalı devre bir kontrol stratejisi kullanılmaktadır. Bu çalışma sırasında oransal tahliye vanasının modellenmesi ve karakterizasyonu gerçekleştirilmiş ve bir yakınlık sensörü kullanılarak türbin hız sensörü geliştirilmiştir. Sistem öncelikle MATLAB Simulink üzerinden modellenmiş ve kontrol algoritması geliştirilmiştir. Ardından sistemin performansını doğrulamak için sıcak ve soğuk gaz üreteçleri ile testler icra edilmiştir. Simülasyon ve test sonuçları, önerilen kontrol sisteminin turbopompa sisteminin hızını istenen aralıkta etkili bir şekilde tutabileceğini göstermektedir.

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LIST OF SYMBOLS

A_e	Exit area of nozzle
$A_{opening}$	Area of the proportional valve
A^*	Normalized area of the proportional valve
C	Cavitation number
C_d	Discharge coefficient
d	Diameter
d_o	Orifice diameter of seat
dt	Time change
dV	Velocity change
e	Error value
g_0	Gravitational acceleration
h	Stroke
I_{sp}	Specific impulse
I_T	Total thrust
inc	Increment number
K	Inverse of cavitation number
K_d	Coefficient of derivative term
K_i	Coefficient of integral term
K_p	Coefficient of proportional term
M	Mach number
m	Mass of the rocket
$\dot{m}$	Mass flow rate
m_P	Total effective propellant mass
P_{critic}	Critic pressure for choked flow
$P_{downstream}$	Downstream pressure
$P_{upstream}$	Upstream pressure
PR	Critical pressure ratio
P_1	Inlet pressure

P_2	Outlet pressure
p_a	Ambient pressure
p_e	Exit pressure at nozzle
p_t	Total pressure
R	Gas constant
R^2	Coefficient of determination
r	Desired process value
s	Slant height
T	Thrust
T_t	Total temperature
u	Control variable
v_e	Exhaust velocity
y	Measured process value
γ	Specific heat ratio
ΔP	Pressure difference between inlet and outlet
$\Delta t_{tooth-to-tooth}$	Time required to sense other tooth on coupling
θ	Seat cone angle
ρ	Density

LIST OF ACRONYMS/ABBREVIATIONS

AMROC	American Rocket Company
CAD	Computer-Aided Design
CFM	Coriolis Flow Meter
DAQ	Data Acquisition
DELTA V	DeltaV Space Technologies Inc.
ECU	Electronic Control Unit
EPOS	Easy to use Positioning System
FS	Full Scale
HTPB	Hydroxyl-Terminated Polybutadiene
H_2O_2	Hydrogen Peroxide
LABVIEW	Laboratory Virtual Instrumentation Engineering Workbench
LED	Light Emitting Diode
LH_2	Liquid Hydrogen
LOX	Liquid Oxygen
LRE	Liquid-Propellant Rocket Engine
MATLAB	Matrix Laboratory
NASA	National Aeronautics and Space Administration
NI	National Instruments
N_2	Nitrogen
N_2H_4	Hydrazine
N_2O_4	Dinitrogen Tetroxide
PI	Proportional – Integral
PID	Proportional – Integral – Derivative
PNP	Positive-Negative-Positive
PV	Process Variable
P&ID	Piping and Instrumentation Diagram
RPM	Revolution Per Minute
RP-1	Rocket Propellant-1 (Highly Refined Form of Kerosene)

SORS	Sounding Rocket System
SP	Set Point
USA	United States of America



1. INTRODUCTION

Spacecraft execute their maneuvers using propulsion systems. These maneuvers include launching the spacecraft from Earth's surface using a launch vehicle, as well as performing space maneuvers like orbit maintenance and orbital transfers. Rocket propulsion systems are classified according to their energy sources, such as nuclear, electric, cold gas, and chemical propulsion, and are shown in Figure 1.1. Among these, chemical propulsion is the most widely used due to its significant advantages over the others. In chemical propulsion, the energy needed for thrust is generated through the combustion of onboard fuel and oxidizer.

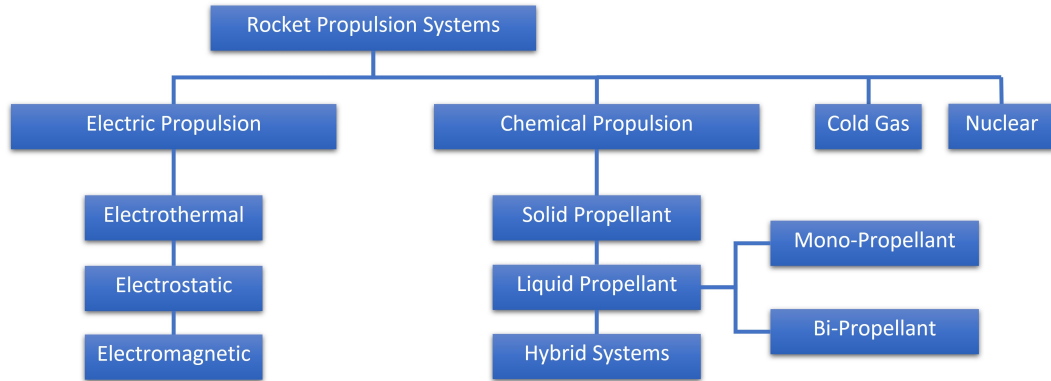


Figure 1.1. Rocket propulsion systems.

The operational principle of propulsion systems is based on the conservation of momentum. As a mass is expelled from a spacecraft, it produces thrust, which in turn causes the spacecraft to move in the opposite direction. This relationship can be expressed through Newton's Second Law which is

$$Force = m \frac{dV}{dt} \quad (1.1)$$

where *Force* is the thrust applied to the rocket, *m* is the mass of the rocket and $\frac{dV}{dt}$ is the change in velocity of the rocket. In rocket propulsion, thrust can be defined as the change in momentum. It is given by the product of the mass flow rate of the propellant

and the velocity at which the propellant is ejected from the nozzle exit plane. If we assume that the mass flow rate and exit propellant velocity are constant, and the exit velocity is uniform, then thrust can be expressed as

$$T = \dot{m}v_e \quad (1.2)$$

where T is the thrust, $\dot{m}$ is the mass flow rate of the propellant, and v_e is the exit velocity of the exhaust. The term given in Equation 1.2 represents "momentum thrust" which is the thrust calculated based solely on the mass flow rate and the velocity of the ejected propellant. This value equals the total thrust only when the nozzle exit pressure, p_e , matches the ambient pressure, p_a , outside the rocket. However, as atmospheric pressure varies with altitude during flight, a discrepancy arises between the ambient pressure and the nozzle exit pressure. This pressure difference at the nozzle exit generates additional thrust, known as "pressure thrust". Consequently, the total thrust is the sum of momentum thrust and pressure thrust [1]. The total thrust is calculated as

$$T = \dot{m}v_e + (p_e - p_a)A_e \quad (1.3)$$

where A_e is the exit area of the nozzle. Specific impulse, I_{sp} , is a key metric defined as the thrust produced per unit of propellant weight flow rate. It reflects the combined efficiency of both the rocket engine and the propellant. Specific impulse is crucial for comparing the performance of different rocket engines. Engines with higher specific impulses need less propellant to achieve the same amount of thrust. The calculation of I_{sp} is given by

$$I_{sp} = \frac{\int_0^t T dt}{g_0 \int_0^t \dot{m} dt} \quad (1.4)$$

where g_0 is the gravitational acceleration and I_{sp} varies with thrust and mass flow rates. Among chemical thrusters, liquid rockets typically exhibit the highest specific impulse, while solid rockets have the lowest. Hybrid rockets fall in between, though recent research is focused on enhancing their performance [1].

The average specific impulse (I_{sp}) of a propulsion system is determined by dividing the total impulse by the total weight of the ejected propellant as

$$I_{sp} = \frac{I_T}{m_p g_0} \quad (1.5)$$

where I_T is the total impulse and m_p is the total effective propellant mass. Specific impulse is the most widely used metric for assessing a rocket engine's performance under various conditions. Some rocket engines are optimized for operation in a vacuum, achieving higher I_{sp} in space, while others are designed to perform better during takeoff and achieve higher I_{sp} within the atmosphere [1].

1.1. Chemical Propulsion Systems

Chemical propulsion is the most commonly used type of propulsion. It generates thrust by utilizing energy and the products of chemical reactions. Typically, fuel and oxidizer are combined in a combustion chamber, where the chemical reaction occurs at high temperatures. The gaseous reaction products then expand rapidly through a supersonic nozzle, resulting in the expulsion of mass and generating thrust in the opposite direction. Chemical propulsion can be classified based on the physical phase of the propellants as solid, liquid, or hybrid.

1.1.1. Solid Rockets

Solid-fuel rockets use solid propellants that contain both the oxidizer and the fuel. In most solid rockets, the propellant is contained within a hollow cylindrical grain. A typical solid rocket motor is shown in Figure 1.2. When ignition occurs, the propellant begins to burn, increasing the pressure and temperature inside the rocket. The hot, pressurized combustion products are expanded through a convergent-divergent nozzle. Specific impulses of up to 300 seconds can be achieved with solid propellants. One of the main advantages of solid rockets is that they have few or no moving parts. Additionally, since the propellant is stored in solid form, it requires less volume and can be stored for long periods. However, a significant disadvantage of solid rockets is that once ignited, the combustion cannot be controlled or stopped. Furthermore, because both the fuel and oxidizer are contained within the solid propellant, there are inherent storage and operational risks [1].

Despite their disadvantages, solid-fuel rocket motors are used in various applications. For instance, they power large boosters, some upper-stage motors, and tactical ballistic missiles. Some sounding rockets, such as "Black Brant" and "Astrobee" employ solid-fuel rocket engines to generate thrust. Additionally, some rockets with liquid-fuel engines use solid-fuel rocket boosters due to their high thrust and suitability for volume-limited applications. Notable liquid-fuel rockets that utilize solid-fuel rocket engines include the "Atlas V", "Space Shuttle", and "Ariane 5" [1].

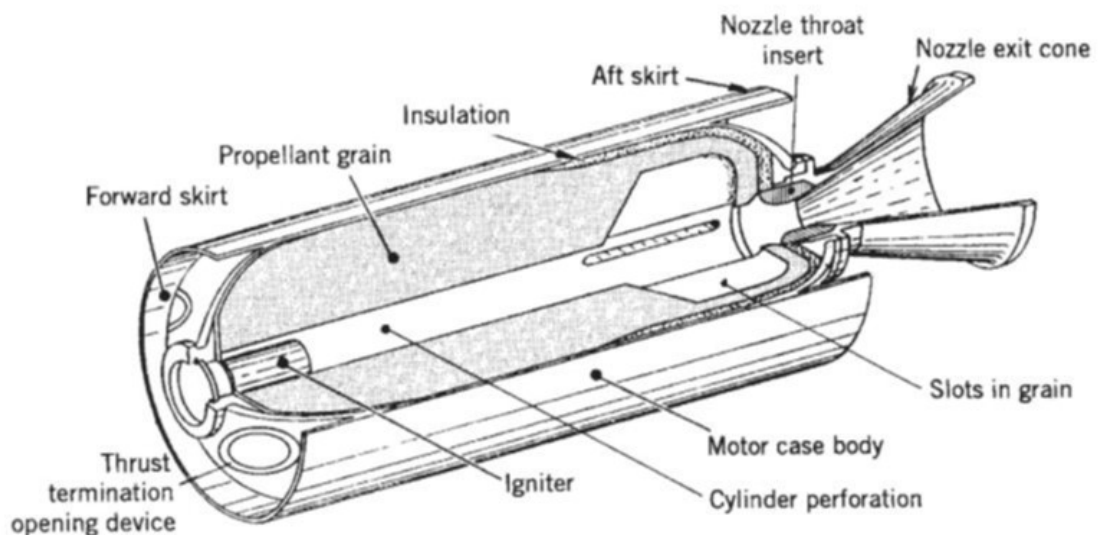


Figure 1.2. Typical solid rocket motor [1].

1.1.2. Liquid Rockets

In liquid rocket engines, the reaction of one or more liquid propellants produces high-pressure and high-temperature gases within a combustion chamber. Thrust is generated by accelerating these gases through a nozzle. Propellants can be pressurized using turbopumps or stored under pressure in tanks. The key benefits of liquid rocket engines include their throttling capability, multiple re-start, and the ability to provide higher specific impulses compared to solid rockets. Liquid rocket engines are categorized into two types: mono-propellant and bi-propellant rockets.

In monopropellant rockets, a single propellant, such as hydrazine (N_2H_4) or hydrogen peroxide (H_2O_2), is decomposed exothermically while flowing through a catalytic bed. These rockets achieve specific impulse levels of up to 240 seconds and are commonly employed for spacecraft attitude control [2].

Bi-propellants are the most common type of liquid rocket. In these rockets, the fuel and oxidizer are combined in the combustion chamber, where the chemical reaction takes place. The products of this reaction are expelled through a supersonic nozzle connected to the chamber. Propellants used in liquid propulsion systems can be either storable or cryogenic. Liquid hydrogen and liquid oxygen (LOX) are well-known examples of cryogenic fuels and oxidizers, respectively, while kerosene and dinitrogen tetroxide (N_2O_4) serve as examples of storable fuels and oxidizers. Their specific impulse values range from 270 to 465 seconds. The highest practical I_{sp} achievable with bi-propellants is approximately 465 seconds, obtained with an LOX/LH₂ combustion [3].

In liquid rocket engines, the systems that deliver fuel and oxidizer to the combustion chamber are called feed systems, and there are three different feed systems used in liquid rocket engines. These are pressure-fed, turbopump-fed, and electric-pump-fed systems. Pressure-fed systems use additional pressurization tanks to regulate the pressure inside the propellant tanks. Propellants are injected into the combustion chamber by the force created by the pressure difference between the tank and the combustion chamber. In turbopump systems, propellant is injected into the combustion chamber by the pressure created by the pump powered by a turbine. In electric-pump-fed systems, an electric motor is used instead of the turbine to power the pump [1].

Many launch systems use bi-propellant liquid rockets as their primary propulsion unit. For example, the RS-25 main engine of the Space Shuttle is designed to produce thrust using liquid hydrogen and LOX. Another notable example is the Merlin 1D engine of the Falcon 9, which uses RP-1 and LOX as propellants. However, a drawback of bi-propellant liquid rockets is that the need for a pressurization system, injectors,

valves, and regulators increases the overall complexity and cost of the rocket system compared to solid-fuel rockets. Additionally, the early mixing of fuel and oxidizer can lead to catastrophic failures, making these rockets particularly hazardous for crewed missions [1]. Figure 1.3 shows a schematic of the main parts used in bi-propellant liquid rockets, including the hot gas generator, turbopump, valves, and combustion chamber.

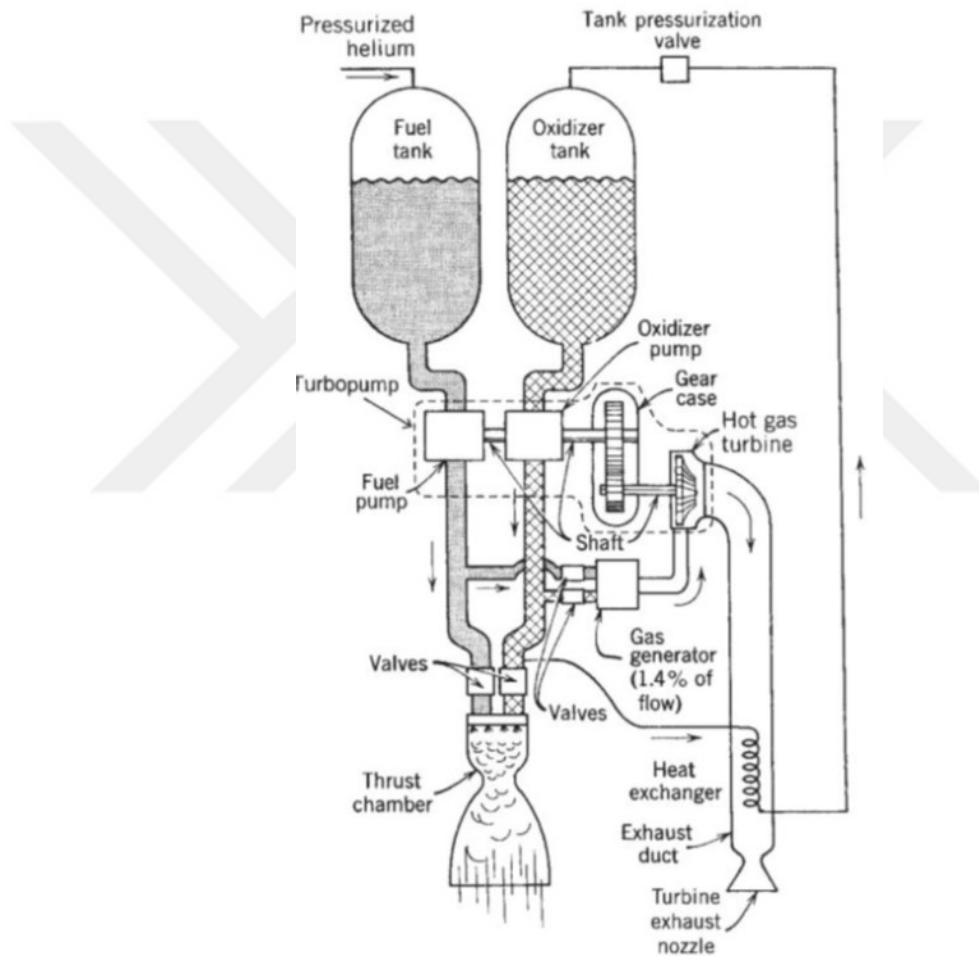


Figure 1.3. Schematic of the main parts used in bi-propellant liquid rocket motor [1].

1.1.3. Hybrid Rockets

In hybrid rocket engines, the fuel and oxidizer exist in different phases: usually, the oxidizer is in liquid form, while the fuel is solid. This approach blends elements of both liquid and solid propulsion, providing hybrid systems with benefits over each

type individually. For instance, hybrid rocket systems can be throttled due to the controllability of the oxidizer mass flow rate, unlike solid propellants, and they feature a less complex design compared to liquid propellant rockets. In hybrid rockets, similar to liquid rockets, combustion can be stopped when the main valve is closed and restarted when the valve is reopened. Additionally, hybrid rockets are inherently safer than other chemical propulsion options because the fuel and oxidizer are in different phases, which reduces the risk of mixing in the event of a structural failure. Hybrid rockets can achieve specific impulses of up to 330 seconds and offer potentially lower costs and safer operations compared to liquid and solid rocket engines [4].

The configuration of a hybrid rocket engine is shown in Figure 1.4. According to Figure 1.4, the liquid oxidizer is located in a separate tank separated from the solid fuel by an oxidizer valve and is pressurized with the help of pressurized gas.

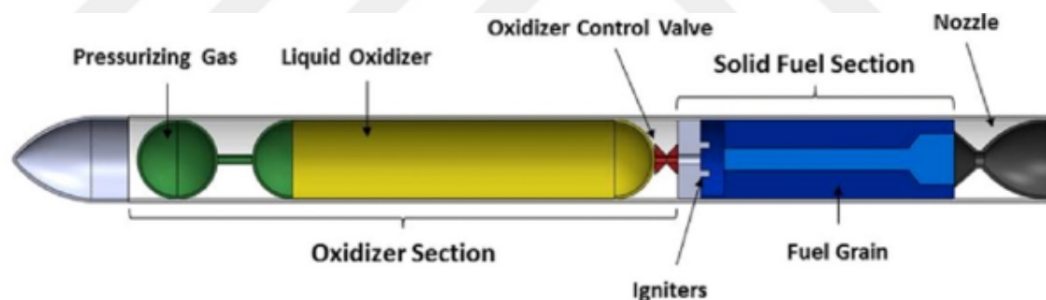


Figure 1.4. Typical hybrid rocket motor configuration [5].

Although hybrid rockets were first developed in the 1930s, they did not gain prominence in the space industry because of the low regression rate and thrust provided by solid propellants. However, in the late 1990s, a new fuel combination, such as paraffin wax mixed with specialized additives, was introduced, which offered a higher regression rate. This breakthrough drew the attention of companies to hybrid rocket propulsion. American Rocket Company (AMROC), one of the pioneers in hybrid rocket technology, was established in 1985 with the goal of developing large-scale hybrid rocket engines. The company conducted numerous hot-fire tests with hydroxyl-terminated polybutadiene (HTPB) and LOX, contributing significantly to the advancement of

hybrid rocket motor technology. The largest rocket motor developed by AMROC is the H-1800, which can deliver a thrust of 250,000 pounds using a combination of liquid oxygen and HTPB propellant [6]. A image from the test of the AMROC hybrid rocket motor is shown in Figure 1.5.



Figure 1.5. Image from AMROC H-1800 hybrid rocket test [6].

Recently, several hybrid rocket technology initiatives have emerged from various countries, including Turkey, South Korea, Germany, Australia, and others. DeltaV Space Technologies Corporation has been developing hybrid rocket propulsion in Turkey since 2017. DeltaV's Sounding Rocket System (SORS) utilizes paraffin and LOX as propellants and has been launched more than sixteen times [7]. Figure 1.6 shows an image from a static firing test conducted by DeltaV Space Technologies.



Figure 1.6. Image from DeltaV Space Technologies' static firing test [7].

1.2. Electrical Propulsion Systems

Electric propulsion systems have the highest specific impulse compared to other propulsion systems. However, their thrust production is limited. Typically, electric propulsion is used for maintaining the orbits of small satellites or for minor orbital adjustments due to its low thrust capability. Electromagnetic fields are applied to gas particles, accelerating them to high speeds before expelling them. A schematic of an electric rocket engine is shown in Figure 1.7 [8].

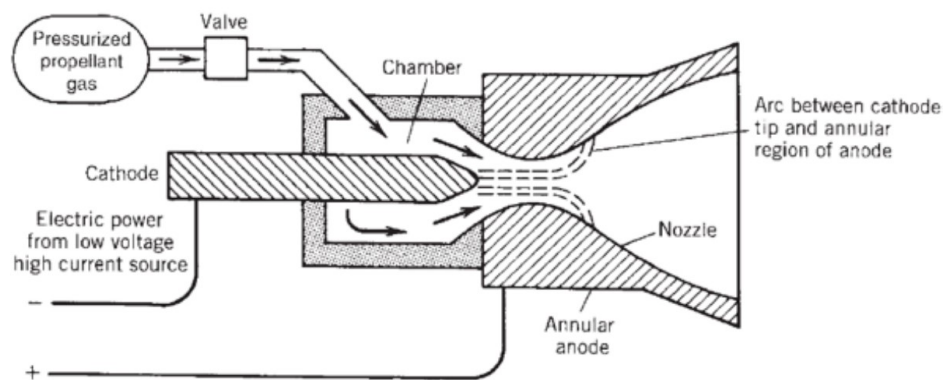


Figure 1.7. Schematic of an electric rocket engine [8].

1.3. Nuclear Propulsion Systems

Nuclear rocket engines utilize nuclear fission technology to generate thrust. Experimental tests using nuclear fission graphite reactors and liquid hydrogen were conducted in the 1960s [9]. The thermal energy from the nuclear reaction is transferred to the liquid hydrogen, which is then converted into kinetic energy through the nozzle. During tests, a specific impulse of 834 seconds is achieved, which is more than double the capability of liquid rocket engines. Although the system has abundant thermal energy, the limited volume of the propellant is a restrictive factor. Despite their great potential, concerns about radioactive contamination have hindered further development of nuclear rocket engines [1].

1.4. Feeding Systems Used in Liquid Rocket Motors

Feed systems in liquid rocket engines are responsible for delivering fuel and oxidizer to the combustion chamber, and there are three main types: pressure-fed, turbopump-fed, and electric-pump-fed systems.

1.4.1. Pressure-Fed Systems

Most pressure-fed systems typically include a separate high-pressure vessel that contains an inert gas, usually helium or nitrogen. The inert gas is used to pressurize the liquid propellants and transfer them to the combustion chamber at a specific predetermined pressure and flow rate. These systems are the most economical and simplest method for pressurizing liquid propellants. However, a significant disadvantage is that the pressure tanks need to be designed to endure high pressure, which leads to a higher structural mass. This disadvantage also constrains the maximum pressure achievable in pressure-fed rocket systems, as the need for high structural mass results in thrust that may become insufficient to support the rocket's weight. Therefore, pressure-fed systems are commonly used in missions that require low pressure, such as space propulsion and small to moderate sounding rockets [1]. Figure 1.8 shows a schematic of the

pressure-fed system used in liquid rocket motors.

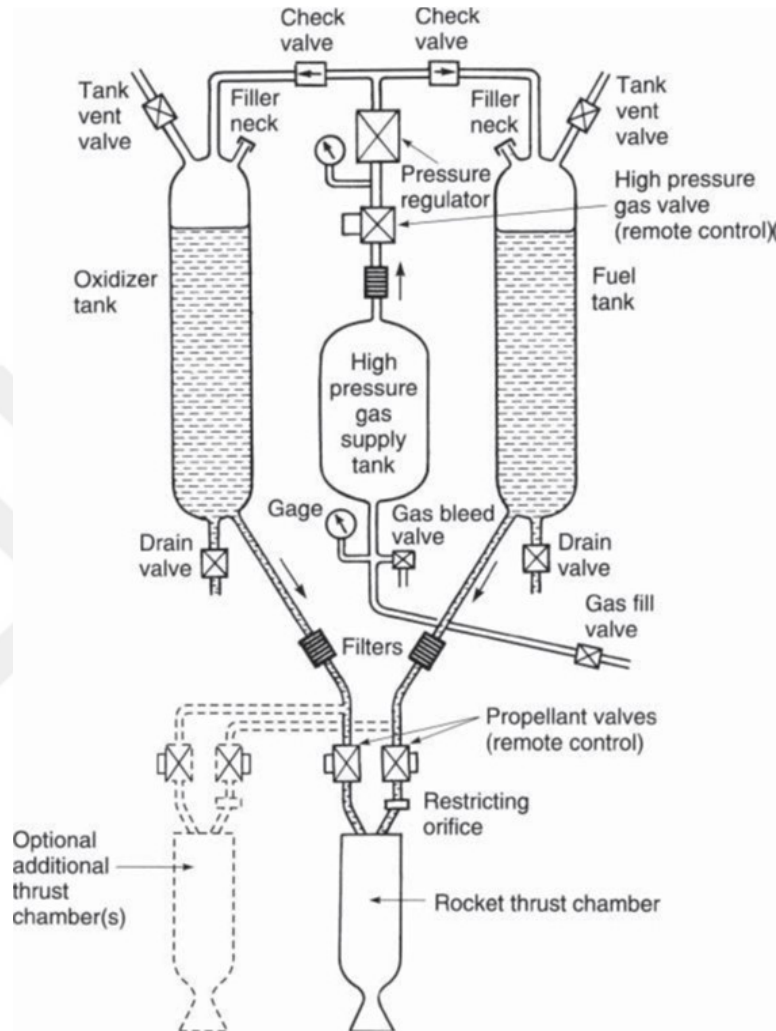


Figure 1.8. Schematic of pressure-fed liquid rocket motor [1].

1.4.2. Pump-Fed Systems

Most liquid rocket engines rely on a turbopump assembly to elevate the pressure of the fuel and oxidizer. Turbopumps are devices that combine turbines and pumps, incorporating different connection types and essential for liquid propulsion systems. Rockets that use turbopumps for pressurization are known as pump-fed systems, and they can be classified into three types: gas generator cycle, expander cycle, and staged combustion cycle, and shown in Figure 1.9.

Many liquid rocket engines, such as NASA's Saturn V (F-1 LRE), Russia's Soyuz (RD-107 LRE), SpaceX's Falcon 9 and Falcon Heavy (Merlin LRE), and Ariane 5 (Vulcan LRE) use the gas-generator cycle to power their pump-fed pressurization systems. Meanwhile, liquid rocket engines such as the Blue Origin BE-3U, and BE-7, Pratt & Whitney RL 60, and Aerojet Rocketdyne RL10 use the expander cycle in their turbopump systems. Staged combustion cycle liquid rocket motors are commonly used in space applications, including the Space Shuttle main engine, Soviet N1 rockets, Space Launch System main engine, SpaceX Starship, and the Proton rocket family [1].

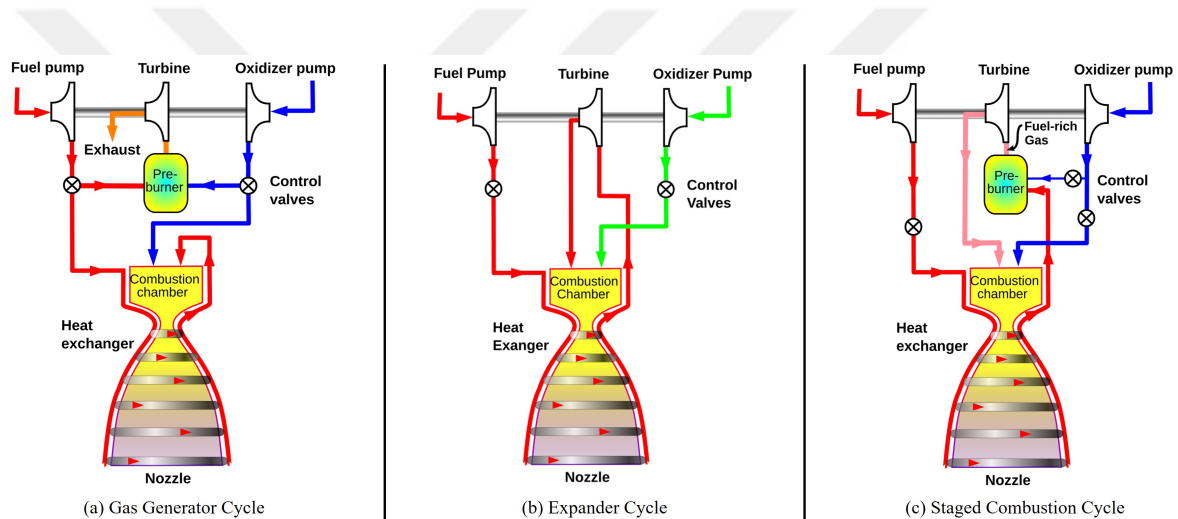


Figure 1.9. Schematics of cycles used in turbopump-fed liquid rocket motor: (a) Gas generator cycle, (b) Expander cycle, (c) Staged combustion cycle [10].

1.4.3. Electric-Pump-Fed Systems

An electric-pump-fed motor is a bipropellant rocket motor in which the fuel pumps are driven by electricity. This design allows the entire input propellant to be combusted in the main chamber without diverting any to power the pumps. This design contrasts with traditional rocket engines, where a portion of the input propellants is used to power the pumps. Figure 1.10 shows a schematic of the electric-pump-fed system used in liquid rocket motors.

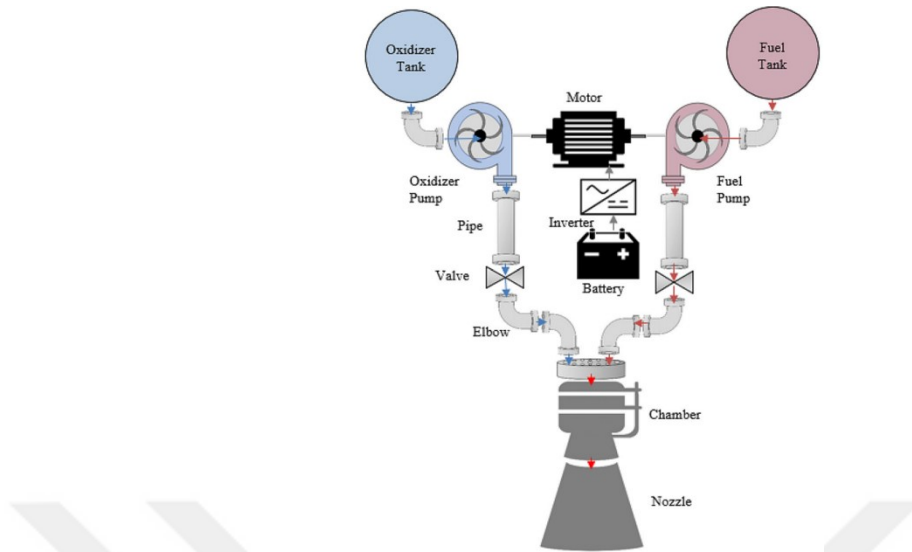


Figure 1.10. Schematic of electric-pump-fed liquid rocket motor [11].

Compared to turbopump rocket cycles like staged combustion and gas generator, an electric cycle engine may potentially have worse performance due to the added mass of the batteries. However, its mechanical simplicity, absence of high-temperature turbomachinery, and easier controllability could result in lower development and production costs [12].

Electric pumps were used in the secondary propulsion system of the Agena upper stage vehicle [13]. As of December 2020, the only rocket engines utilizing electric pump-fed systems are the Rutherford engine, which powers the Electron rocket with ten engines, and the Delphin engine, which powers the first stage of Astra Space's Rocket 3 with five engines [14, 15].

1.5. Gas Generator Cycle

The gas generator cycle, or open cycle, is among the most commonly used power cycles in bipropellant liquid rocket engines. In this cycle, propellants are burned in a gas generator or preburner, producing hot gas that powers the fuel and oxidizer pumps before being exhausted overboard, leading to its designation as an open cycle.

In the gas generator cycle, exhaust products first pass through the turbine before being expelled overboard. These gases can be released directly from the turbine or, in some cases, into the nozzle downstream of the throat for a slight efficiency boost. Since the main combustion chamber does not use these gases, the cycle is called "open." The primary drawback is that this exhaust contributes little or no thrust, as it bypasses the main combustion chamber. However, the cycle's main benefit lies in its simpler engineering design, especially when compared to the staged combustion (closed) cycle [1].

1.5.1. Previous Applications

The gas generator cycle is used in the turbopump-fed pressurization systems of many liquid rocket engines. Some of these rocket systems of historical importance are discussed below.

1.5.1.1. V-2 Rocket (Germany). The V-2 is a rocket developed in Germany during World War II, recognized as the first long-range ballistic missile. It has dimensions of 1.64 m in diameter and 14 m in length. The rocket is powered by a liquid-propellant rocket motor, with a pump-fed pressurization system driven by a gas-generator cycle turbopump. Over 3,000 V-2 missiles were produced and launched from German territories [10].

The V-2 rocket uses an ethanol-water mixture as fuel and liquid oxygen as an oxidizer, generating up to 34 tons of thrust. With a burn time of approximately 65 seconds, the missile then follows a free-trajectory path. The V-2 reaches a maximum altitude of over 90 km and has a range of around 310 km [10].

Both the fuel and oxidizer pumps are driven by a turbine that uses steam as the working fluid. The steam is produced by the catalytic decomposition of a mixture containing 80% hydrogen peroxide, water, and stabilizers. This decomposition process is facilitated by a catalyst, sodium permanganate [10].

1.5.1.2. Soyuz Rocket (USSR & Russia). The Soyuz, a rocket developed in the Soviet Union and Russia, is powered by the RD-107 and RD-108 engine family. The RD-107 is a turbopump-fed liquid rocket engine that uses kerosene as fuel and liquid oxygen as an oxidizer. This motor operates on a gas generator cycle, utilizing hydrogen peroxide to produce gas. The turbopump's turbine is driven by the catalytic decomposition of hydrogen peroxide with potassium permanganate [1].

The RD-107 motors generate 840 kN of thrust at sea level and up to 1000 kN in a vacuum. The combustion chamber pressure varies across engine models but typically is around 60 bar. The RD-107 and RD-108 engines have powered many significant Soviet space programs, including the R-7, Vostok, Sputnik, and ultimately the Soyuz [1]. Figure 1.11 shows an RD-108 engine on display in a museum.

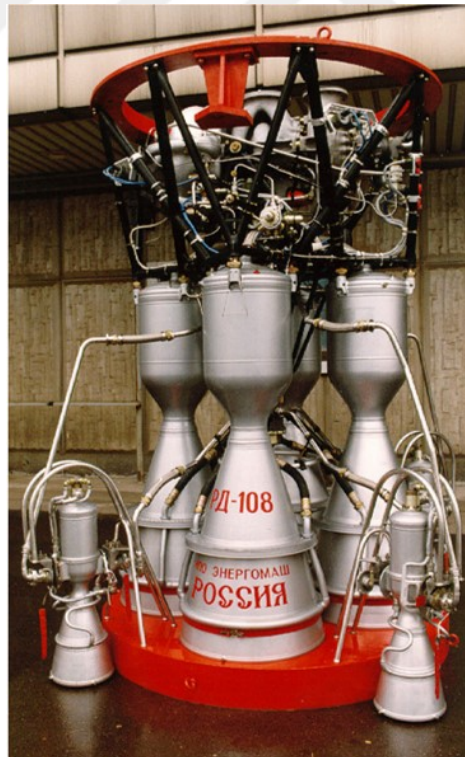


Figure 1.11. RD-108 engine [16].

1.5.1.3. Saturn V Rocket (USA). The Saturn V rocket, developed by NASA, is one of the most powerful rockets ever built. The Saturn V rocket, designed primarily

for the Apollo missions, was capable of delivering nearly 7.5 million pounds of thrust (34,500 kN) at liftoff. This thrust enabled the rocket to transport astronauts and essential equipment to the Moon. The first stage, housed five F-1 engines, which generated about 6,770 kN of thrust each. These engines were fueled by RP-1 and liquid oxygen [1].

In F-1 engine, RP-1 and LOX were combined in the gas generator to produce the necessary gas to drive the turbopump's turbine. The turbine, in turn, rotated the pump, forcing RP-1 and LOX into the main combustion chamber at high pressure. Unlike closed cycles, the exhaust from this process was not fed back into the main chamber. Instead, it was expelled directly overboard, creating visible black exhaust plumes seen in firing photos of the F-1 engine and shown in Figure 1.12.

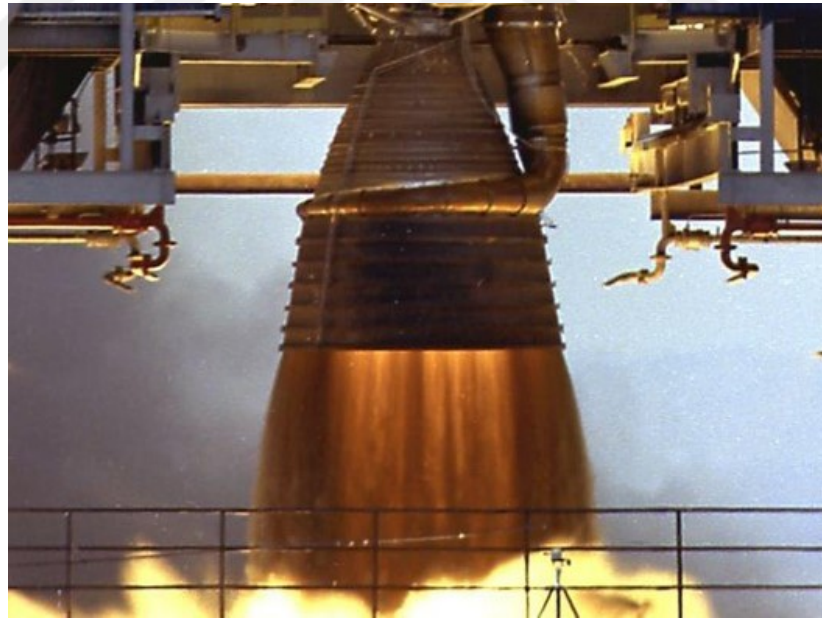


Figure 1.12. Test firing of the F-1 engine with exhaust plume of gas generator [17].

1.5.1.4. SpaceX Merlin 1D Rocket (USA). The Merlin rocket engine, developed by SpaceX, is one of the most notable engines in modern rocketry, powering the Falcon 9 and Falcon Heavy rockets. Known for its reliability, efficiency, and reusability, the Merlin 1D uses a kerosene (RP-1) and liquid oxygen propellant combination, which

makes it ideal for delivering payloads to low Earth orbit. SpaceX's Merlin 1D was designed to provide high thrust-to-weight ratio and to withstand multiple reuse cycles. Each Merlin 1D engine produces approximately 845 kN of thrust at sea level, with a vacuum variant capable of 914 kN in a vacuum. A core feature of this engine is its gas generator cycle, which is fundamental to its operation and design efficiency [18].

This cycle burns a small portion of the propellant to drive the engine's turbopumps, which supply high-pressure RP-1 and LOX to the combustion chamber. The turbine of the turbopump is powered by the hot gases produced by burning RP-1 and LOX in a dedicated, lower-pressure gas generator, separate from the main combustion chamber. While this results in a loss of some efficiency compared to closed-cycle engines, it also simplifies the system, reducing thermal stress and making the Merlin 1D lighter and easier to manufacture [1].

Figure 1.13 shows a static firing photo of SpaceX's Merlin 1D engine. Next to the main engine, the location releasing black smoke indicates the gas generator, where turbine-driving exhaust gases are expelled after powering the turbopump in the gas generator cycle.

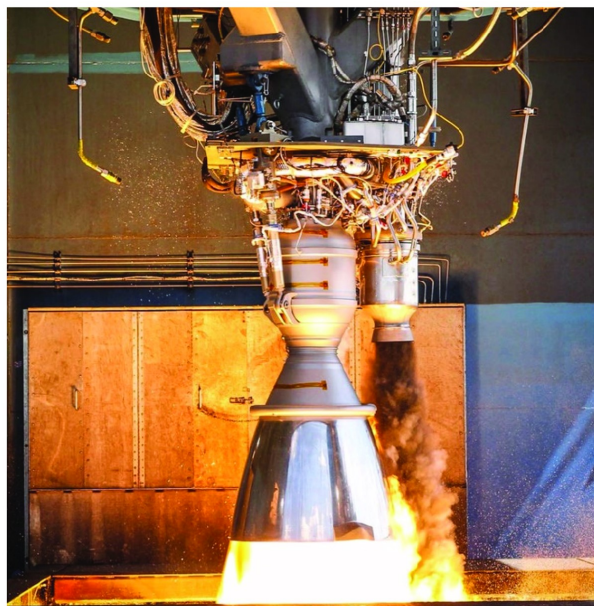


Figure 1.13. Test firing of the Merlin 1D at SpaceX's McGregor test facility [19].

1.6. Wastegate System

A wastegate is a crucial component in turbocharged internal combustion engines, playing a vital role in regulating the boost pressure produced by the turbocharger. In turbocharged engines, exhaust gases are directed to spin a turbine connected to a compressor, which in turn pressurizes the intake air, increasing the engine's power output. However, if left unregulated, the turbocharger can create excessive pressure, which could damage the engine or lead to inefficiency. The wastegate prevents this by diverting excess exhaust gases away from the turbine when the desired pressure is reached, maintaining a safe and optimal boost level [20]. Figure 1.14 shows the schematic of the wastegate system.

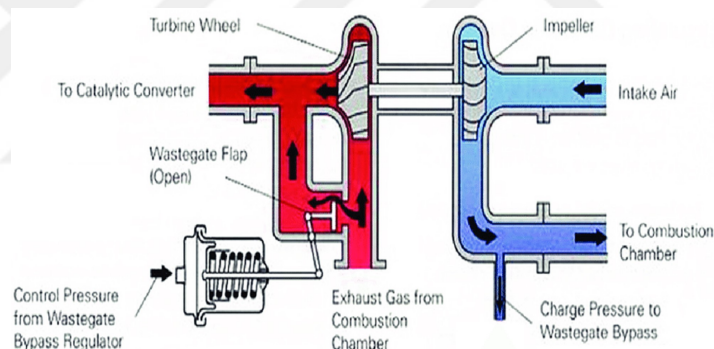


Figure 1.14. Schematic of a wastegate system [21].

1.6.1. Wastegate Types

Wastegates are generally divided into two main types: internal and external. Internal wastegates are integrated within the turbocharger housing and are typically found in smaller or factory-installed turbochargers. They are cost-effective and more compact, though they provide less control and are less durable than external wastegates.

Figure 1.15 shows the closed and open internal wastegate, respectively. As boost pressure builds, it acts on the diaphragm of the wastegate. When the pressure reaches

a specified threshold, the diaphragm moves, causing the wastegate to open. This allows excess exhaust gases to bypass the turbine, thus regulating the turbocharger's speed and maintaining the desired boost level.

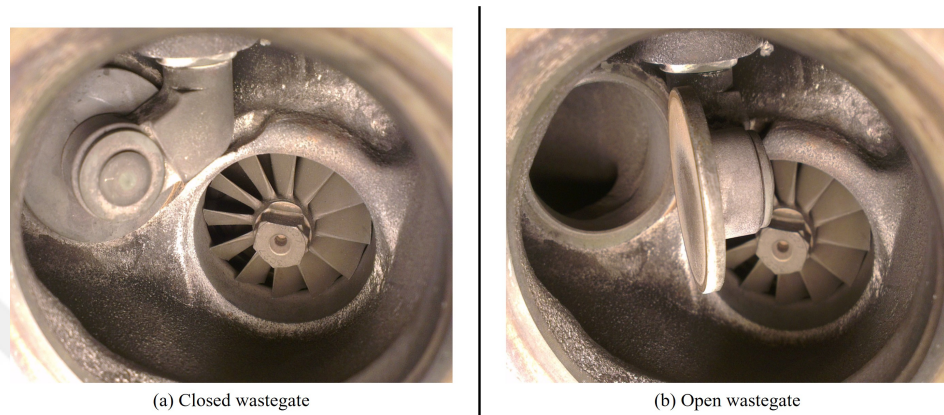


Figure 1.15. Internal wastegate system: (a) Closed wastegate, (b) Open wastegate [22].

External wastegates are mounted separately from the turbocharger. They are usually used in high-performance or heavily modified engines where precision and durability are essential. External wastegates are more versatile and offer better control over boost pressure, making them popular in racing or custom applications.

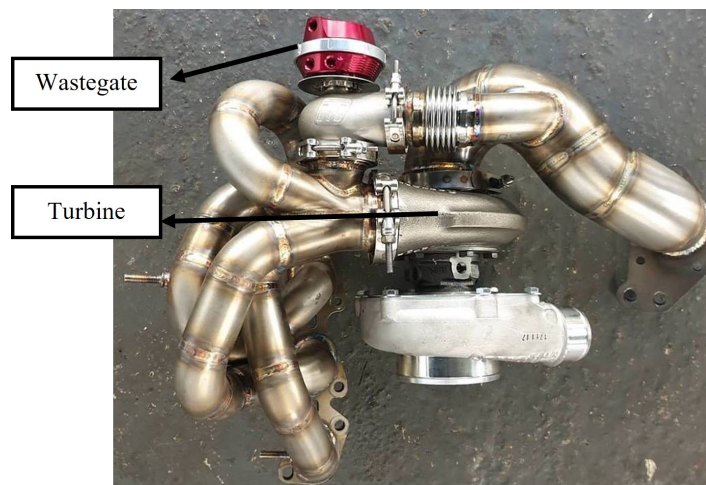


Figure 1.16. External wastegate system [23].

1.6.2. Wastegate Control Mechanisms

The control of the wastegate can be achieved using various methods like mechanical control, electronic control, and hybrid control. Mechanical control is the simplest form of wastegate control, where a spring-loaded actuator is calibrated to open the wastegate at a specific boost pressure. The spring's tension determines the boost threshold, and it can be adjusted to suit different performance needs. Adjusting the spring's tension within the actuator is the primary method for changing the boost threshold in mechanical systems. Heavier springs increase the boost threshold, allowing the turbocharger to create more boost before the wastegate opens, while lighter springs lower the threshold [20]. Figure 1.17 shows a mechanical wastegate controller with a spring and diaphragm structure.

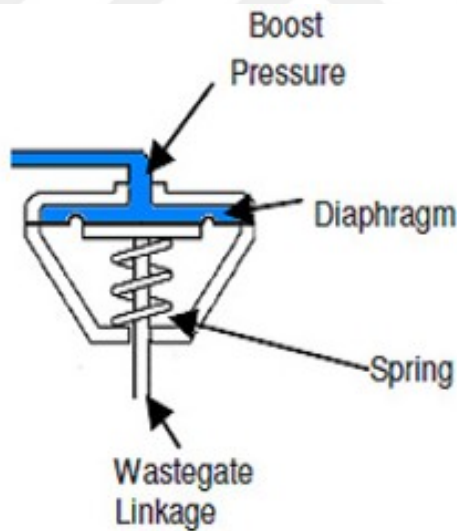


Figure 1.17. Mechanical spring-diaphragm wastegate actuator system [20].

Electronic control in advanced wastegate systems utilizes electronic actuators controlled by the engine's electronic control unit (ECU). This allows for more precise control over the boost pressure by adjusting the wastegate position based on various engine parameters, such as throttle position, RPM, and load conditions. Electronic wastegate control systems can optimize performance dynamically, improving response and efficiency [20].

Hybrid systems combine mechanical and electronic control elements, typically integrating an electronically controlled solenoid valve to modify the signal going to a mechanical actuator. By controlling the airflow to the actuator, these systems can fine-tune the mechanical wastegate's response under various conditions [20].

In examining the literature on wastegate control mechanisms and controllers, existing research primarily focuses on how turbine and compressor performance responds to wastegate adjustments. Given that the turbocharger components in automotive applications are well-established, studies typically aim to enhance the performance of these systems by utilizing available graphical data and models. These controllers often rely on turbine and compressor pressures and turbocharger rotational speed as feedback variables to analyze and optimize system performance [20, 22, 24, 25].

1.7. Contribution of this Work

In this thesis, a novel PID controller system has been proposed in which the hot gas resulting from the combustion of a hybrid rocket motor or cold gas resulting from pressurized tank can be regulated before entering the turbine, similar to the turbochargers in cars. Pressure-fed hybrid motors are used as hot gas generator and operate in blow-down mode which means that as the propellants flow out of the tank, the pressure in the tank decreases, which in turn reduces the flow rate. Therefore, the gas generator system produces more energy than the turbopump system needs during operation. Required amount of generated gas is diverted to the turbopump system, and the rest is expelled using a proportional wastegate. The excess energy produced by the gas generator ensures that RPM stays constant during pressure fluctuations in the combustion chamber, particularly when using a hybrid rocket motor. Since testing with a hot gas generator is difficult and costly, nitrogen held at high pressure was used as a cold gas generator to drive the turbine. In the cold gas generator system, similar to the hot gas generator, the energy going to the turbine will decrease over time as the pressure of the tank decreases. For this reason, excess energy was given to the system at the beginning, so that the turbine would receive the energy it needs

to rotate at the desired speed until the end of the test, and then the remaining excess gas was expelled with the proportional wastegate. Figure 1.18 shows two different gas generator cycles used to drive the turbine in the turbopump system and the proposed closed-loop control mechanism.

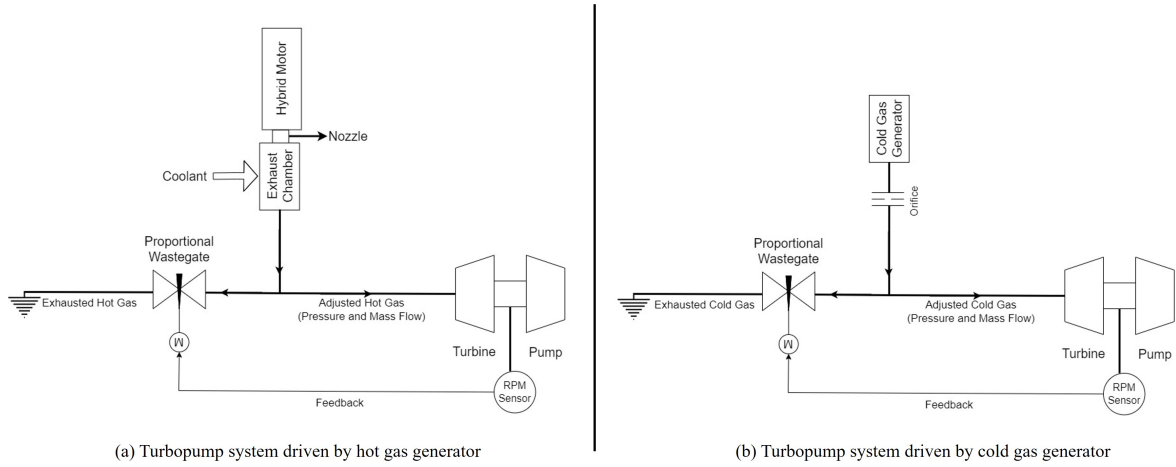


Figure 1.18. Schematic of gas generator cycles used to drive the turbopump system:
(a) Hot gas generator, (b) Cold gas generator.

The proposed system employs a closed-loop control strategy, where feedback from the turbine speed is measured with the help of a sensitive distance sensor and this feedback is used to adjust the wastegate position and maintain the desired RPM. The proportional wastegate used here was modeled based on measuring and calculating the movement of the electric motor and gears, and initially, the system was simulated with Simulink in MATLAB using a PID control algorithm. Simscape was used in Simulink to build the model of the test setup including turbopump and gas dynamics. Afterward, the system was tested with hot gas and cold gas and the model is updated with the parameters obtained from the test. Both simulation and test results demonstrate that the proposed control system can effectively regulate the speed of the turbopump system within a desired range, even in the presence of disturbances and variations in load. This study highlights the potential of PID control for the turbopump system's speed regulation using a proportional wastegate.

In Chapter 2, the design criteria and features of the proportional wastegate used are discussed. Then, the orifice area of the valve is calculated analytically and the opening area of the valve at different positions is obtained. Afterwards, the tests and the results obtained were discussed to validate the accuracy of the analytical data and to further characterize the proportional wastegate. Finally, the discharge coefficient of the proportional wastegate was characterized as a function of the valve opening ratio and cavitation number, providing a comprehensive understanding of its behavior under various conditions.

In Chapter 3, the selection process of the RPM sensor required to measure the rotational speed of the turbine is explained. After the RPM sensor is selected, the tests performed to see the measurement sensitivity and accuracy of the sensor are mentioned.

In Chapter 4, the closed-loop PID controller used to keep the turbopump system at the desired rotation speed is mentioned and the test setup created with the Simscape module of MATLAB Simulink used to model physical systems and the results of the simulations are examined.

In Chapter 5, integrated system tests using hot and cold gas generators are mentioned and the results of each test are examined one by one, and the causes and solutions of the errors are discussed. In Chapter 6, the results of the study and future work are discussed.

2. DEVELOPMENT OF PROPORTIONAL WASTEGATE

In this section, the development process of the proportional wastegate, which regulates the energy supplied to the turbopump system, is discussed, beginning with the design constraints and system requirements. The design of the proportional wastegate is then explained, followed by its analytical modeling. Finally, tests were carried out for the characterization of the proportional wastegate, and the discharge coefficient of the system was determined based on different opening ratios and cavitation numbers.

2.1. Design Constraints and Requirements

The turbopump system intended for use with the proposed proportional wastegate was previously developed by DELTAV. Although numerous tests have been conducted on this turbopump system, data regarding its performance under varying conditions, as commonly seen in the literature, is not available. The design and production processes of the proportional wastegate were also completed beforehand, yet this valve has not been utilized to date. Consequently, the necessary tests for system characterization and modeling were conducted within the scope of this study.

Examining the mechanical constraints reveals that the line and connection components at the turbine inlet adhere to established standards, ensuring there are no limitations on assembling the proportional wastegate in this area. Similarly, since the inlet and outlet lines of the proportional wastegate were designed and manufactured to be compatible with standard connection components, no restrictions arise for assembly with the turbine. The pintle, which controls the flow area within the proportional wastegate, has a design stroke length of 11 mm, although the valve reaches a fully open position without requiring the full stroke. The proportional wastegate is also specifically designed to withstand the high-temperature gases (ranging from 700 K to 1100 K) produced by the hot gas generator.

The proposed system requires the turbine to operate under conditions of both increased and reduced energy input to the turbopump. To meet this requirement, the narrowest area between the turbine stators, where flow choking occurs, has been calculated and found to approximate the area of a 13 mm diameter circle. Consequently, to allow the turbopump system to decelerate to the desired speed when more energy is applied, the proportional wastegate's orifice must exceed this 13 mm threshold. Upon examining the proportional wastegate, it is observed that, when fully opened, the orifice has the area of a circle with a 19.84 mm diameter. This results in an area difference of more than twice the minimum requirement, indicating no restriction to the system's performance.

Finally, the electric motor in the proportional wastegate, responsible for controlling the turbopump system's rotation speed, is operated by the avionics system. Currently, no sensor provides real-time feedback on the turbopump system's rotation speed to the avionics. Therefore, after essential modifications, a sensitive distance sensor was adapted for use as the RPM sensor.

2.2. Proportional Wastegate Design

Figure 2.1 shows the proportional wastegate designed for use. Within the proportional valve, a pintle is driven by a high-power electric motor, chosen to ensure the valve operates effectively under demanding conditions without size or energy limitations. When activated, this motor rotates a gear, which in turn rotates a pinion. The pinion's angular movement is converted into linear motion via a lead screw attached to the pintle, enabling precise linear movement of the pintle.

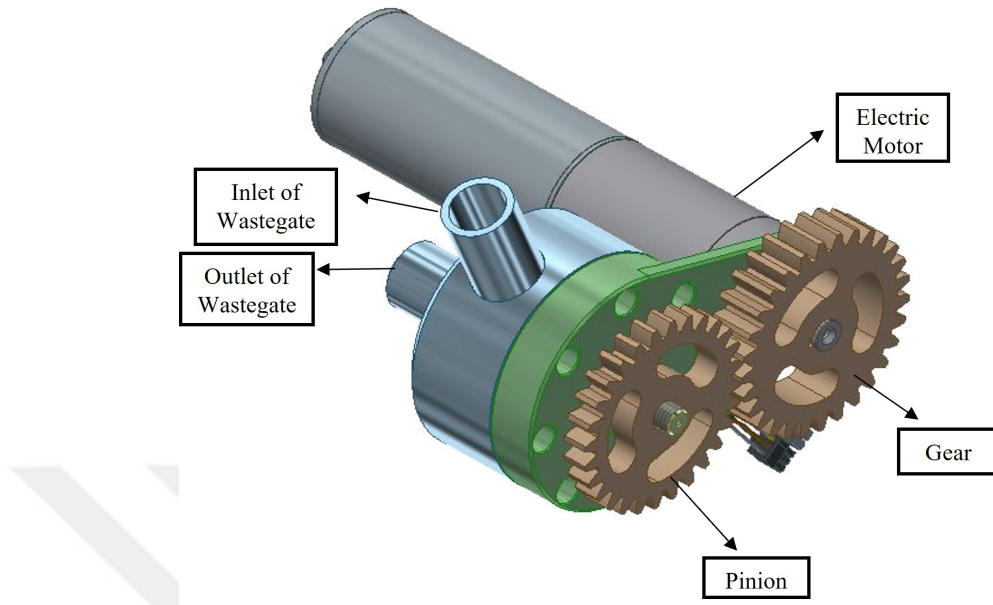


Figure 2.1. Design of proportional wastegate.

Due to confidentiality restrictions on the internal design of the proportional valve intended for testing, Figure 2.2 presents a cross-sectional view of a similar valve that shares comparable design and features. In the system illustrated in Figure 2.2, unlike the proportional valve planned for the tests, the electric motor is directly connected to the pintle, allowing the pintle to achieve linear movement through the drive screw mechanism with the motion of the stepper motor [26].

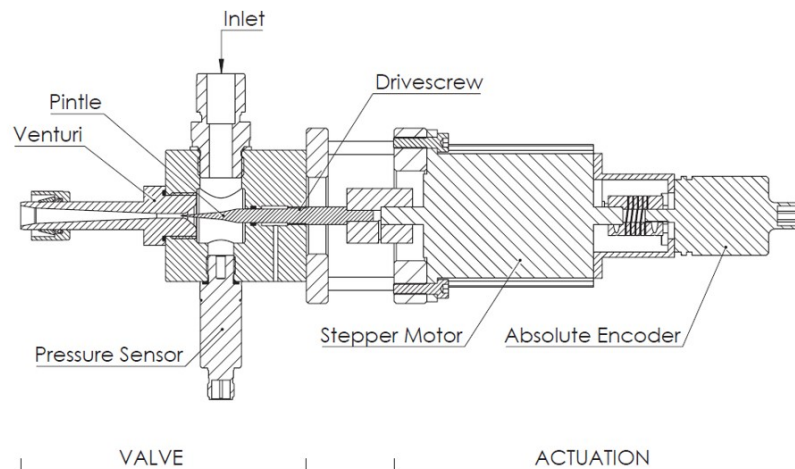


Figure 2.2. Cross-section schematic of proportional valve from literature [26].

2.3. Modeling and Analysis of Proportional Wastegate

Proportional valves serve to regulate fluid flow by adjusting the available cross-sectional area. Mass flow rate calculations depend on whether the fluid is compressible or incompressible, employing distinct formulas for each case. Precise control of the turbopump system necessitates a well-developed and accurate model of the proportional wastegate, which is why calculations have been performed for both fluid types.

2.3.1. Incompressible Fluid

The mass flow rate through an orifice for incompressible fluids—whose density remains constant during flow—is given by

$$\dot{m} = C_{d,incompressible} A_{opening} \sqrt{2 \times \rho_{fluid} \times (P_1 - P_2)} \quad (2.1)$$

where $\dot{m}$ is the mass flow rate of the fluid, C_d is the discharge coefficient of the valve, $A_{opening}$ is the narrowest area through which the fluid passes in the valve, ρ is the density of the fluid, P_1 is the pressure of the fluid entering the valve, and P_2 is the pressure of the fluid leaving the valve.

From the values in Equation 2.1; flow rate, inlet and outlet pressures are measured with the help of sensors in the tests, and the opening area of the valve is calculated theoretically. The liquid density is obtained from CoolProp, an open-source thermo-physical property library, using the pressure and temperature conditions at the valve inlet [27]. Finally, the discharge coefficient was calculated for different opening areas according to the values obtained. In the literature, there are values about the discharge coefficient in the valves that vary according to the orifice shape of the valve, and these values are compared with the results obtained after the tests.

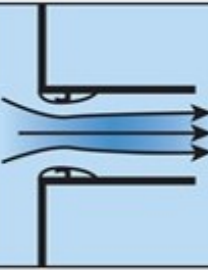
	Sharp edged	Rounded	Short tube	Borda
				
C	0.61	0.98	0.80	0.51

Figure 2.3. Orifice types and their nominal discharge coefficients [28].

2.3.2. Compressible Fluid

For the ideal compressible fluid, the flow rate through the valve is calculated with

$$\dot{m} = C_{d,compressible} \frac{A_{opening} p_t}{\sqrt{T_t}} \sqrt{\frac{\gamma}{R}} M \left(1 + \frac{\gamma - 1}{2} M^2\right)^{-\frac{\gamma+1}{2(\gamma-1)}} \quad (2.2)$$

where C_d is the discharge coefficient of the compressible fluid, $A_{opening}$ is the narrowest area through which the fluid passes in the valve, p_t is the total pressure of the fluid at the inlet of the valve, T_t is the total temperature of the fluid at the inlet of the valve, γ is the specific heat ratio of the fluid, R is the gas constant of the fluid, and M is the Mach number. According to this formula, the flow reaches its maximum when the Mach number is equal to one—that is, when the flow is choked—and the formula is modified as

$$\dot{m}_{maximum} = C_{d,compressible} \frac{A_{opening} p_t}{\sqrt{T_t}} \sqrt{\frac{\gamma}{R}} \left(\frac{\gamma + 1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}. \quad (2.3)$$

To find the pressure at which the flow is choked, it is necessary to calculate the critical pressure ratio. The critical pressure ratio changes depending on the specific heat ratio of the fluid. By using this ratio and the inlet pressure, the critical pressure at the downstream can also be calculated. This means that if the pressure of the downstream flow is below the critical pressure, the flow chokes. The critical pressure

ratio is calculated as

$$\text{Critical pressure ratio, } PR = \frac{P_{critic}}{P_{upstream}} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}. \quad (2.4)$$

If the flow is not choked and is assumed to be isentropic, the Mach number, M , is calculated as

$$M = \sqrt{\left(\left(\frac{P_{downstream}}{P_{upstream}}\right)^{-\frac{\gamma - 1}{\gamma}} - 1\right)\left(\frac{2}{\gamma - 1}\right)}. \quad (2.5)$$

When Equation 2.2 is examined, p_t and T_t values are taken with the help of sensors during the tests, and data is taken from the outlet of the valve with the pressure sensor to determine whether the flow is choked or not. The specific heat ratio, γ , and gas constant, R , values are taken from CoolProp by using the properties of the fluid. The value of $A_{opening}$ represents the smallest opening area in the valve and the discharge coefficient has been found from the literature for compressible fluids as 0.92 [29].

2.3.3. Area Calculation of Proportional Wastegate

As mentioned in the design section of proportional wastegate, the large gear rotates with the movement of the electric motor, and then the large gear rotates the pinion gear. The angular movement in the pinion was converted into linear movement with the help of the lead screw on the pintle attached to the pinion.

Since the angular movement with the gears becomes linear in the pintle with the lead screw, firstly, the increment amount that should be given by the user for one revolution of the electric motor was calculated. In this calculation, the number of pole pairs, the number of phases, the polarity number of the phases and the absolute reduction value in the selected electric motor are used as

$$\begin{aligned}
\text{Number of pole pairs} &= 8 \\
\text{Number of phases} &= 3 \\
\text{Polarity number of phases} &= 2 \\
\text{Absolute reduction number} &= \frac{1183}{18}.
\end{aligned} \tag{2.6}$$

The increment required for one revolution of the gear is calculated by multiplying the number of pole pairs, the number of phases, the polarity number of phases, and the absolute reduction number of the electric motor as

$$\begin{aligned}
\text{The increment required for one revolution of the gear} &= 8 \times 3 \times 2 \times \frac{1183}{18} \\
&= 3154.67 \text{ inc} \\
&\approx 3155 \text{ inc}
\end{aligned} \tag{2.7}$$

where 8 is the number of pole pairs, 3 is the number of phases, 2 is the polarity number per phase, and $\frac{1183}{18}$ is the absolute reduction ratio of the electric motor.

Since there are 29 teeth in the gear and 28 teeth in the pinion, the gear ratio is $\frac{29}{28}$. The amount of increment that should be given by the user for one revolution of the pinion was calculated by dividing the increment required for one revolution of the gear by the gear ratio which as

$$\begin{aligned}
\text{The increment required for one revolution of the pinion} &= 3155 \times \frac{28}{29} \\
&= 3046.2 \text{ inc} \\
&\approx 3046 \text{ inc.}
\end{aligned} \tag{2.8}$$

The lead screw used in the pintle has a thread pitch of 4 millimeters, meaning the pintle moves 4 millimeters linearly per one full revolution. This linear movement is known as a stroke. The number of increments required for the pintle to move 1 millimeter linearly is calculated as

$$\begin{aligned}
\text{Increment required for the pintle to advance linearly by one millimeter} &= \frac{3046}{4} \\
&= 761.5 \text{ inc} \\
&\approx 761 \text{ inc.} \\
&\quad (2.9)
\end{aligned}$$

As a result, the 761 incremental rotation command given to the electric motor by the user will cause the pintle to advance 1 millimeter linearly. The reason for converting the increment value to the stroke is that the area is calculated depending on the stroke while calculating the opening area in the proportional valves in the literature.

To find the narrowest area through which the flow passes in proportional valves using a needle tip, it is necessary to consider a truncated cone between the tip of the pintle and the seat where the pintle sits and calculate the lateral area of that truncated cone. In Figure 2.4, there is a schematic view of the narrowest area through which the flow passes in the proportional valve, from the side and from the top.

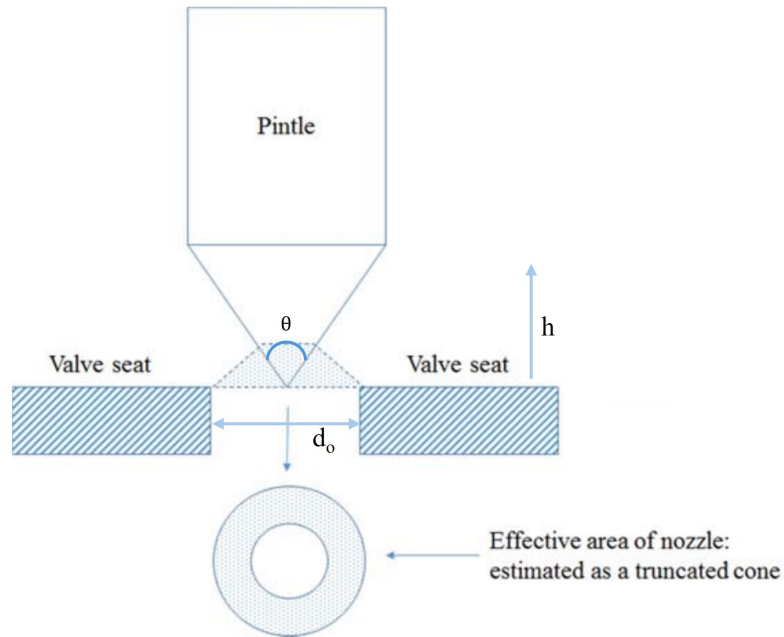


Figure 2.4. Schematic view of the pintle valve from side perspective [30].

Here, h is the stroke, θ is the seat cone angle, and d_o is the seat's orifice diameter. In these conditions, the lateral area of the truncated cone is calculated as

$$\text{Area} = \pi \times h \times \cos\left(\frac{\theta}{2}\right) \times \left(\frac{d}{2} + \frac{d - 2 \times h \times \cos\left(\frac{\theta}{2}\right) \times \sin\left(\frac{\theta}{2}\right)}{2}\right). \quad (2.10)$$

Unlike what is available in the literature, a special trigonometric formula has been produced for the lateral area of the truncated cone, which varies depending on the stroke, since the conical angle of the seat and needle is not the same. In the first section up to the end of the chamfer, the formula in the literature was used along the 45-degree chamfer made from the seat orifice diameter, and the accuracy of this formula was also verified using CAD. Figure 2.5 shows the lateral area of the truncated cone formed at any point taken from the CAD along the chamfer, and this shown area gives the narrowest area where the flow is choked.

In Appendix A, the areas of different geometrical approaches that can be used to calculate the narrowest area in the valve when the pintle has a stroke of 1.5 mm are compared, and it is seen that the narrowest area is the one given in Equation 2.10 mentioned in the literature and in the case given in Figure 2.5.

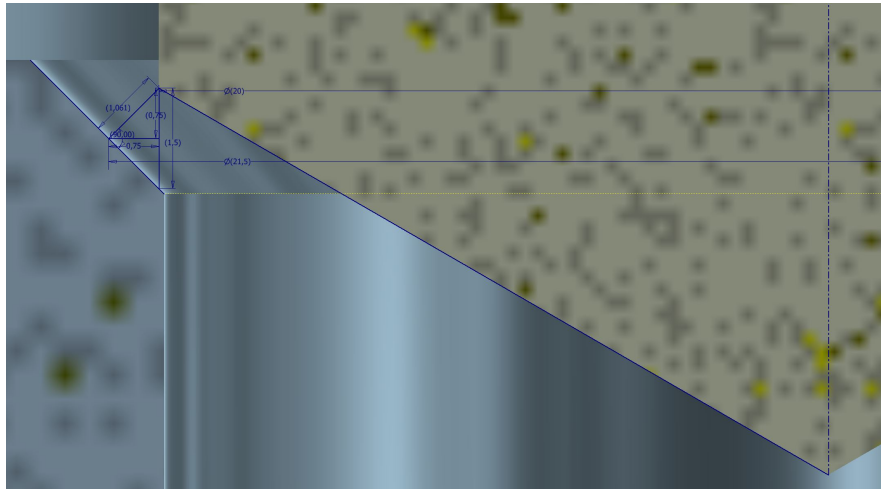


Figure 2.5. Geometry used for orifice area calculation of proportional wastegate with 1.5 mm stroke from CAD drawing.

In calculating the area of the proportional wastegate, the diameter d in Equation 2.10 varies according to the stroke movement in the initial section until the end of the chamfer. This diameter is determined as

$$d(i) = d_{pintle} \times (i - 1) \times h(i). \quad (2.11)$$

The area formula, defined based on the characteristics of the proportional wastegate, incorporates the chamfer angle as θ , set to 45° . The resulting area formula, derived using Equation 2.10, is presented below as

$$\text{Area} = \pi \times h(i) \times \cos(45^\circ) \times \left(\frac{d(i)}{2} + \frac{d(i) - 2 \times h(i) \times \cos(45^\circ) \times \sin(45^\circ)}{2} \right). \quad (2.12)$$

After the chamfered part was finished, two different area calculation formulas were used for the remaining part, and it was chosen whichever formula had the smaller area because it would determine the flow. The two field formulas used are also given as

$$\text{Area} = \pi \times h(i) \times \cos(45^\circ) \times \left(\frac{d_{chamfer}}{2} + \frac{d_{chamfer} - 2 \times h(i) \times \cos(45^\circ) \times \sin(45^\circ)}{2} \right) \quad (2.13)$$

$$\text{Area} = \pi \times s(i) \times \left(\frac{d_o}{2} + \frac{d_o - 2 \times (x(i) + 2)}{2} \right). \quad (2.14)$$

Equation 2.13 was used up to stroke value of 4.14 mm, and then Equation 2.14 was used. In Equation 2.14, s is the slant height, and x is calculated after trigonometric operations. The calculation of x and opening area of the pintle valve is shown in Appendix B, step by step, using the CAD model of the pintle valve.

The opening area of the valve obtained by using Equation 2.13 and Equation 2.14 is plotted in Figure 2.6 depending on the stroke, which is the linear movement of the pintle. As mentioned in the design section of proportional wastegate, due to the mechanical limitation, the pintle can move up to 11 millimeters in the calculations. The point where the dashed lines intersect on this graph is the area obtained when the

valve is opened 100 percent. This area is calculated from the area of the circle with a diameter of 19.84 millimeters, which is the narrowest part of the seat's orifice, and the stroke value of the narrowest area in the valve corresponding to this point is 7.145 millimeters with 309.153 mm^2 . After this step, further advancement of the pintle will not change the orifice, so this area determines the maximum flow through the valve.

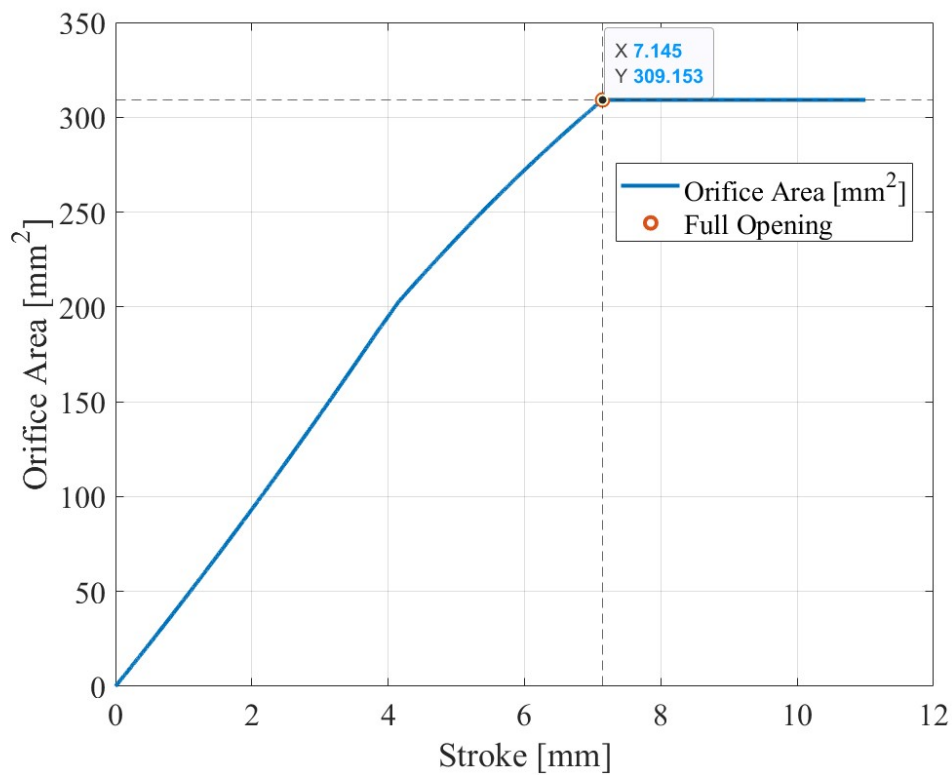


Figure 2.6. Orifice area versus stroke for proportional wastegate.

In Figure 2.7, the opening area of the valve is plotted against the increment given to the valve by the user, instead of the stroke, using Equation 2.9. The incremental value required to open the valve 100 percent was found to be 5441.

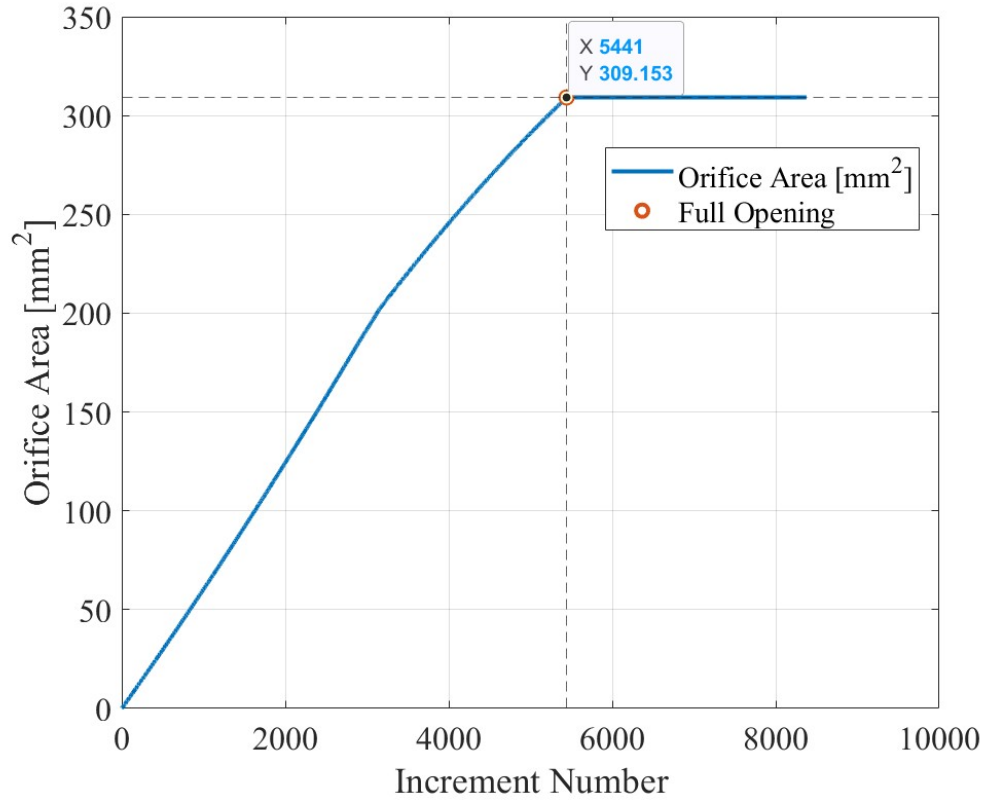


Figure 2.7. Orifice area versus incremental number for proportional wastegate.

After calculating the opening area in the pintle valve, when Equation 2.1 for incompressible fluids and Equation 2.2 for compressible fluids are examined, it is seen that the values other than the discharge coefficient are the values to be measured during the test and the discharge coefficient should be calculated according to these values. For this reason, before starting the tests, the mass flow rate in the valve was calculated according to the opening area of the valve, as if there was no loss in the valve, that is, the discharge coefficient was one, according to different pressure values.

Firstly, the mass flow rate, $\dot{m}$, in the pintle valve for incompressible fluids is calculated according to four different ΔP values, valve inlet and outlet pressure difference, by using Equation 2.1 with taking the density of water as 997 kg/m^3 and the discharge coefficient of the valve, C_d , as 1. Theoretical mass flow rate in the valve is calculated accordingly to valve opening percentage is shown in Figure 2.8.

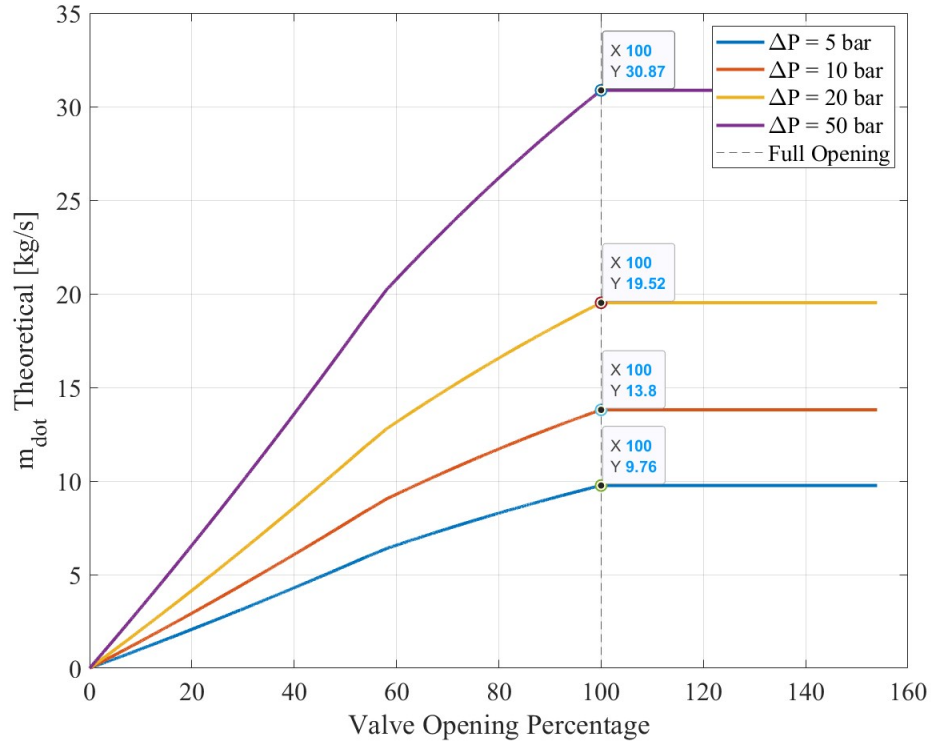


Figure 2.8. Theoretical $\dot{m}$ versus opening percentage of proportional wastegate.

According to Figure 2.8, when water is selected as the fluid, the mass flow rate at full valve opening will be 9.76 kg/s at 5 bar ΔP , 13.8 kg/s at 10 bar ΔP , 19.52 kg/s at 20 bar ΔP and finally 30.87 kg/s at 50 bar ΔP .

The mass flow rate calculation for compressible fluids passing through the pintle valve is made for nitrogen gas to be used in the cold gas generator. The calculation was made using Equation 2.2. The gas pressure at the inlet of the pintle valve is assumed to be 10 bar and the mass flow rate depending on the valve opening ratios for 5 different outlet pressures is given in Figure 2.9.

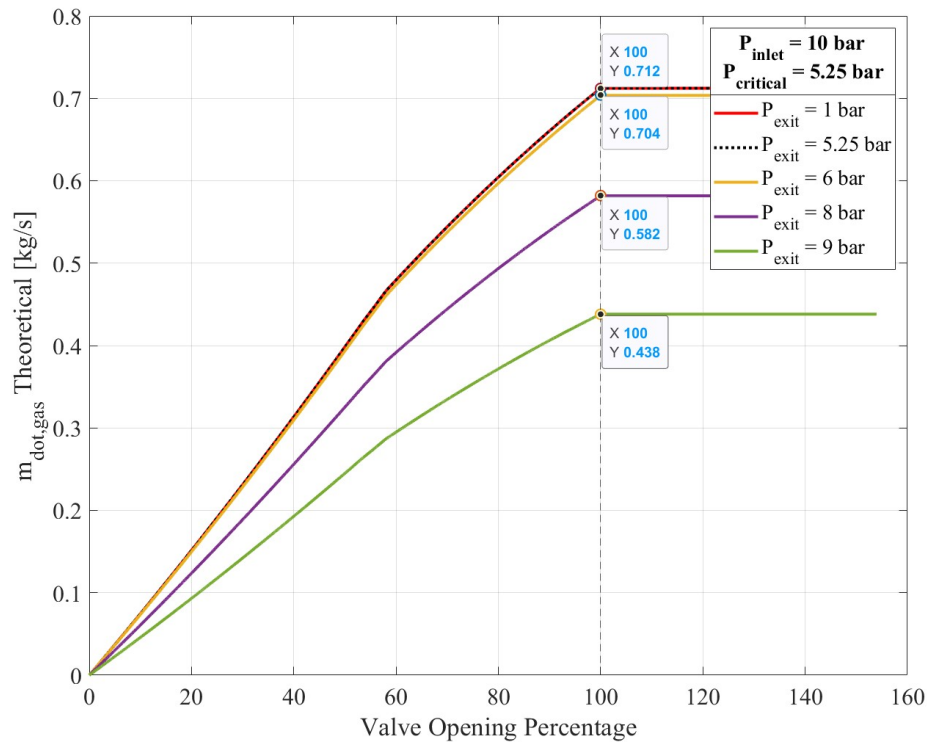


Figure 2.9. Theoretical $\dot{m}_{gas}$ versus opening percentage of proportional wastegate.

In compressible flow, since the flow is choked, which is the point where the fluid velocity reaches Mach one and the flow reaches its maximum mass flow rate, Equation 2.4 was used to find the critical pressure and the outlet pressure required to choke the flow, $P_{critical}$, was calculated to be 5.25 bar. As long as the outlet pressure is above the critical pressure, the flow is not choked, but if the outlet pressure is equal to or below the critical pressure, the mass flow rate of the fluid does not change.

When calculating the nitrogen gas flow rate passing through the pintle valve, the discharge coefficient value, C_d , in Equation 2.2 is taken as 1 and the gas properties are taken from CoolProp. According to Figure 2.9 when nitrogen gas is used as the fluid, the mass flow rate at full valve opening will be 438 g/s when the outlet pressure is 9 bar, 582 g/s when the outlet pressure is 8 bar, 704 g/s when the outlet pressure is 6 bar and 712 g/s when the outlet pressure is at or below the critical pressure of 5.25 bar.

2.4. Characterization Tests of Proportional Wastegate

Following the theoretical calculations for the proportional wastegate, a series of tests were conducted to characterize the system. This section details the sensors employed in these tests, the test procedures implemented, and the results obtained.

2.4.1. Sensors

2.4.1.1. Coriolis Mass Flow Meter. The flow in the water line is measured by the mass flow meter connected to the water tank. The mass flow meter used is a Coriolis flow meter type and measures the mass flow rate of the fluid by utilizing the Coriolis effect, which is related to the deflection of a vibrating element due to the movement of the fluid. The accuracy of the flow meter used for liquids is $\pm 0.1\%$ and the mass flow rate it can measure in nominal operation is 78000 kg/h [31]. The Coriolis flow meter used is shown in Figure 2.10.



Figure 2.10. Coriolis flow meter used in tests [31].

2.4.1.2. Pressure Transmitter. During the tests, IFM PT2402 model pressure sensors were used. These sensors have a diaphragm structure and a diaphragm-based pressure sensor measures the pressure by converting the mechanical deformation of the diaphragm into an electrical signal, which is then processed to provide an accurate pressure reading. These sensors measure the gauge pressure of the system, that is, the relative pressure value of the system with respect to atmospheric pressure. The

measuring range of the pressure sensors used in the tests is 0-68.9 bar (0-1000 psi) and the accuracy is $\pm 0.5\%$ FS (full scale) [32]. Figure 2.11 shows the pressure transmitter used in the tests.



Figure 2.11. Pressure transmitter used in tests [32].

2.4.1.3. Thermocouples. Both T and K type Omega thermocouple models were used as temperature sensors during the tests. A thermocouple consists of two different metal wires or alloys joined at one end to form a measuring junction. The other ends of these wires are connected to a reference junction. The thermocouple measures temperature by producing a voltage proportional to the temperature difference between the measuring and reference junctions. This voltage is then converted into a temperature reading [33].

The measuring range of T type temperature sensors is -250°C to 350°C , and the accuracy is $\pm 1^{\circ}\text{C}$. The measuring range of K type temperature sensors is -200°C to 1250°C , and the accuracy is $\pm 2.2^{\circ}\text{C}$. In Figure 2.12 a T type thermocouple is shown on the left and a K type thermocouple is shown on the right [33].

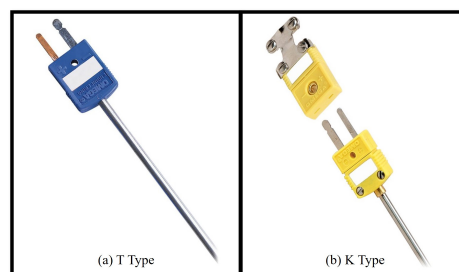


Figure 2.12. Thermocouples used in tests: (a) T Type, (b) K Type [33].

2.4.2. Software and Hardware

2.4.2.1. EPOS Studio. During the tests, the proportional wastegate's electric motor was operated using the EPOS Studio application of Maxon, the manufacturer of the electric motor [34]. In the characterization tests of the proportional wastegate with water, the electric motor's position mode was used and the user gave absolute increase values to the electric motor so that the proportional wastegate would go to different valve opening values. Before the tests, the maximum speed and acceleration values of the electric motor were also set via the EPOS Studio program and the position, speed, acceleration and current values of the electric motor were monitored during the tests. Figure 2.13 shows the user interface of the EPOS Studio application.

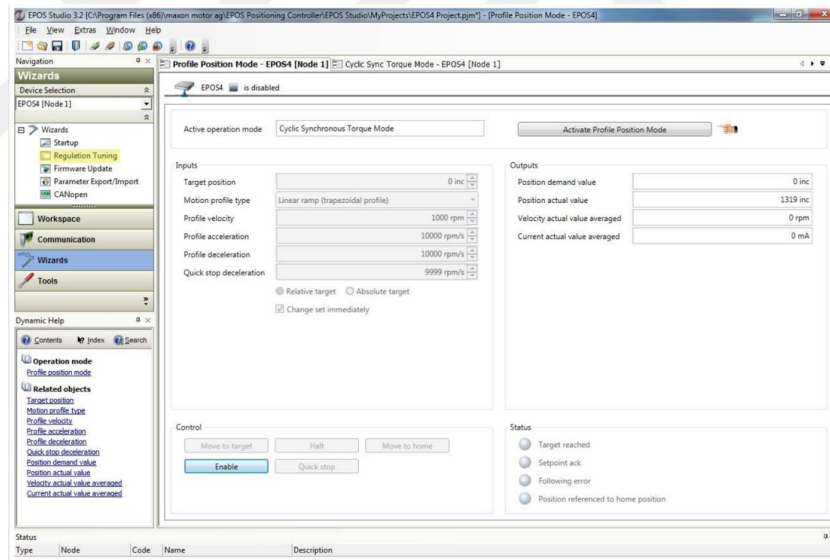


Figure 2.13. User interface of the EPOS Studio.

2.4.2.2. NI DAQ and LabView. The data obtained from the sensors in the tests were collected with NI DAQ (National Instruments Data Acquisition System) and then analyzed in MATLAB. The sensors used in the tests produce their measurements as specific voltage or current values and the analog outputs corresponding to these values. The data produced by the sensors are then collected with the DAQ system and recalculated using specific calibration coefficients in the LabView program of National

Instruments (NI), and as a result, the pressure, temperature and mass flow rate values are obtained. Figure 2.14 shows the block diagram structure where the user defines the data to be obtained in the tests in the LabView program.

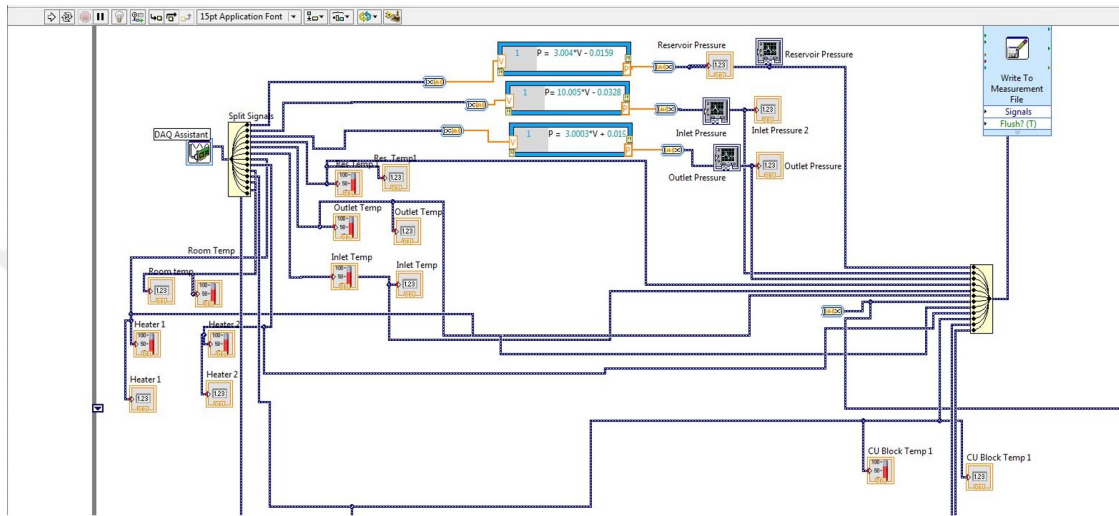


Figure 2.14. Block diagram structure in LabView.

The test data were collected at high frequencies, and a moving average filter was applied during data analysis in MATLAB to reduce noise and smooth the data for more accurate interpretation. Before applying the filter, the raw data were initially examined to check for any anomalies in the system. Figure 2.15 shows the user interface developed in LabView.

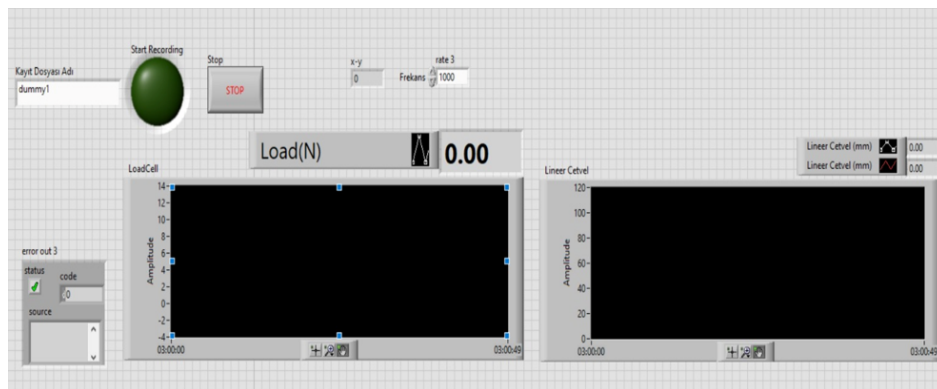


Figure 2.15. User interface developed in LabView.

2.4.3. Test Setup

The characterization tests for the proportional wastegate were conducted at DELTAV's GÖRTEM Test Center. More than ten tests were performed, and upon completion, the discharge coefficient was determined for various valve opening percentages.

DELTAV's GÖRTEM Test Center is located in Teknopark Istanbul, where the company conducts small hybrid rocket firings and pressurized flow tests. There is a 200-liter water tank at the facility, and this tank is pressurized by a manifold consisting of 12 nitrogen cylinders and a regulator. The pressure of the tank is monitored with the pressure sensors on the tank, and the temperature of the water inside is monitored with the temperature sensors. Figure 2.16 illustrates the water tank, along with the nitrogen manifold and regulator used to pressurize the water tank.



Figure 2.16. Water tank and nitrogen manifold used for pressurization at GÖRTEM Test Center.

The piping and instrumentation diagram (P&ID) is an essential tool in process engineering, providing a visual representation of a process's piping, instrumentation,

and control systems, which aids in design, operation, and maintenance. To ensure comprehensive preparation for the water tests to characterize the proportional wastegate, the P&ID of the test area was created. Figure 2.17 shows the P&ID schematic of the systems to be used for valve characterization tests in the GÖRTEM facility.

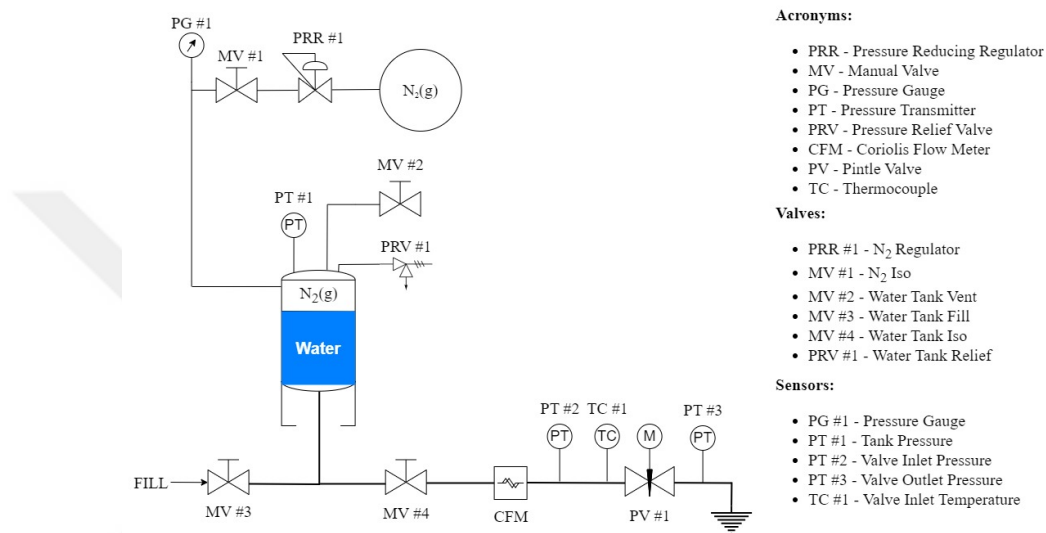


Figure 2.17. P&ID schematic of the test setup used for proportional wastegate's characterization tests.

2.4.4. Test Procedures

In the GÖRTEM test area, the procedures for characterizing the proportional valve in water tests were performed in a sequential manner. First the water tank was filled with water and the proportional valve was mounted on the line. Then, pressure sensors were connected to the inlet and outlet lines of the proportional valve and it was checked from the control room whether these sensors were receiving data. Then, to see if the electric motor in the proportional valve was working properly, commands were given to the electric motor using EPOS Studio. Finally, to bring the proportional valve to the closed position, commands were given to the electric motor to gradually go to the closed position via the EPOS Studio application and the current drawn by the electric motor was checked. The movement was stopped when the current value exceeded a certain value.

After the proportional valve is ready, it is checked whether the water line is loose or not, and if no leakage is observed in the connections, the people are taken to a safe area, the control room, and the water tank is pressurized with the help of a regulator using a nitrogen manifold with one person. In general, the pressure of the water tank is set to approximately 10 bar for safety during the tests and to ensure that the regulator can feed the system more effectively. A pressure gauge with a manual indicator is positioned after the regulator, allowing for manual adjustment of tank pressure. Additionally, the water tank pressure is continuously monitored from the control room. After the pressurization process is completed, the person who performs the pressurization finally opens the manual isolation valve of the water tank and thus the system is ready for testing. The user sends the increment values corresponding to the opening percentages of the proportional valve that they have predetermined according to the purpose of the test to the proportional valve and operate the proportional valve.

The test ends either when the predetermined movements are completed or when the water in the water tank runs out. In both cases, the proportional valve is first turned to the closed position and then someone closes the regulator that pressurizes the water tank. After the pressure in the water tank is released, the manual isolation valve of the water tank is closed and the water tank is drained from the filling line. At this time, the proportional valve is opened once more and the water remaining between the manual isolation valve and the proportional valve is drained. In order to maintain the cleanliness of the systems in the test area, it is checked whether there is any open space in the system and finally the test area is closed. Figure 2.18 shows a picture of the proportional valve during the test.

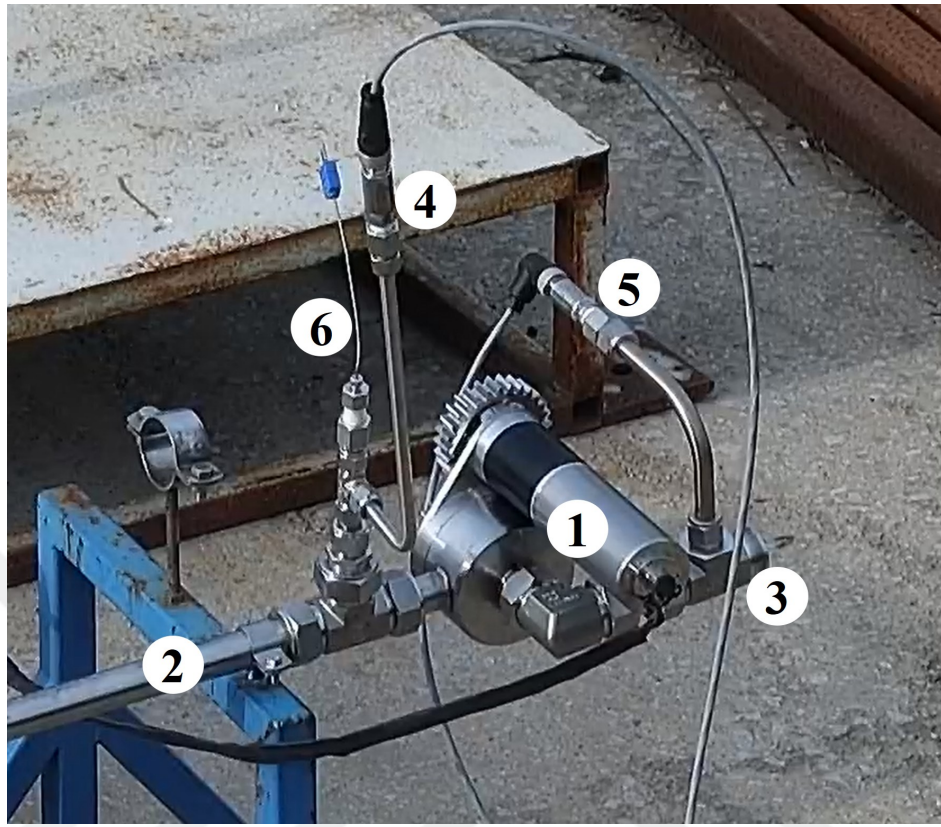


Figure 2.18. Picture from the water test.

In Figure 2.18, number 1 marks the proportional valve, number 2 marks the inlet line of the proportional valve, number 3 marks the outlet of the proportional valve, number 4 marks the pressure measuring sensor at the inlet of the proportional valve, number 5 marks the pressure measuring sensor at the outlet of the proportional valve, and number 6 marks the temperature reading sensor at the inlet of the proportional valve.

2.4.5. Results of the Tests

During the first test, due to leaks observed when the proportional valve was in the closed position, the test was not carried out and the design of valve was changed. After these changes, no leaks were observed in other tests. Since the Coriolis mass flow meter system (CFM) in the test system could not measure values below 60 g/s, no mass flow rate value could be read until the opening of the valve reached 3 percent. The valve

opening ratio was increased by 1 percent and opened up to 10 percent. Afterwards, the valve was closed completely, and it was examined to see whether the system would perform the same behavior again. Then again, the opening was increased by 1 percent starting from 10 percent and advanced to 15 percent, and the valve was closed again. Finally, the opening percentage of the valve was increased from 15 to 20 percent again.

In Figure 2.19, the left axis displays the tank pressure in blue, the wastegate inlet pressure in orange, and the wastegate outlet pressure in yellow. Since the valve is opened to the atmosphere and the pressure sensors give a gauge pressure value, the outlet pressure appears to be 0 bar here. On the right axis, the mass flow rate, measured by the Coriolis mass flow meter, is shown in purple. Although the tank was tried to be fed with the nitrogen regulator during the test, the regulator could not feed the tank sufficiently as the water in the tank decreased. For this reason, at 188 and 307 seconds when the valve is closed, the regulator has ensured that the tank pressure is around 10 bar set for the test.

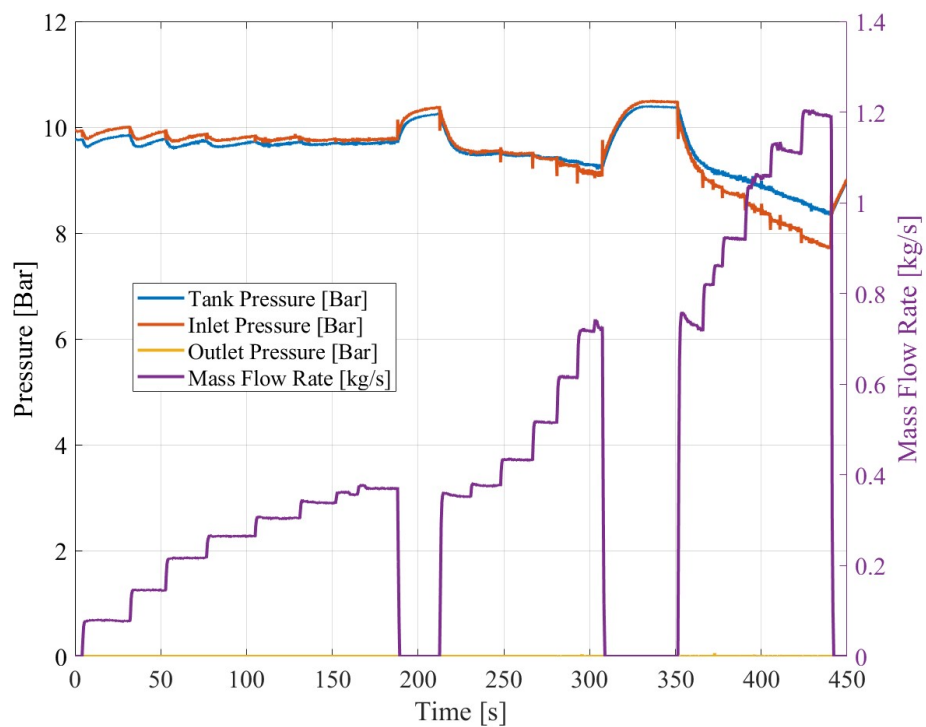


Figure 2.19. Pressures and mass flow rate results of characterization test 3.

After obtaining all the required data during the test, the discharge coefficient of the valve was calculated using a modified form of Equation 2.1 as

$$C_{d,incompressible} = \frac{\dot{m}}{A_{opening} \sqrt{2 \times \rho_{fluid} \times (P_1 - P_2)}}. \quad (2.15)$$

In Figure 2.20, the mass flow rate is shown on the left axis, and the valve opening percentage at that time and the discharge coefficient obtained by using Equation 2.15 are shown on the right axis.

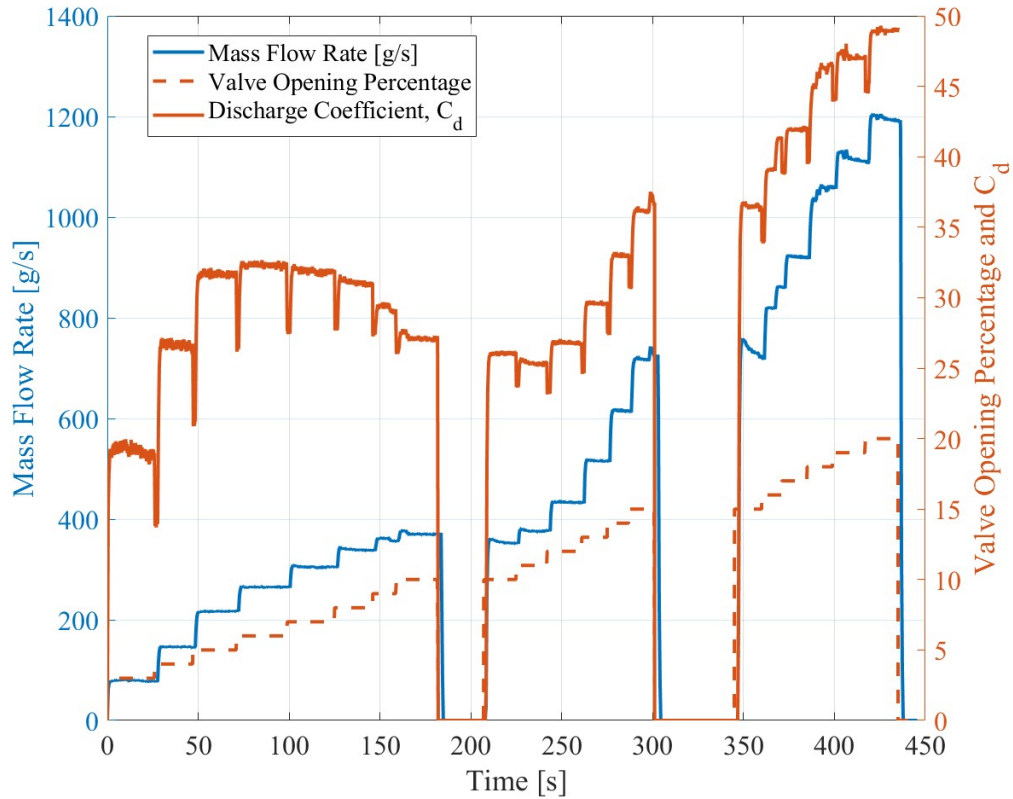


Figure 2.20. Mass flow rate, wastegate opening percentage and discharge coefficients, C_d , results of characterization test 3.

Finally, the opening ratio of the valve and the corresponding average mass flow rate and discharge coefficient values are given in Table 2.1. As can be seen here, when the valve is closed and then reopened to the same position during the test, the discharge

coefficient values calculated are close to each other, and it has been observed that the test results show repeatability.

Table 2.1. Average C_d and $\dot{m}$ corresponding to valve opening percentage from test 3.

Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)	Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)
3	19,4	78	12	26,7	430
4	26,8	144	13	29,5	513
5	31,7	218	14	33,2	620
6	32,3	266	15	36,1	716
7	31,8	304	0	0	0
8	31,2	335	15	36,5	735
9	29,5	354	16	40,1	841
10	27,3	368	17	41,9	918
0	0	0	18	46,3	1057
10	26,1	355	19	46,9	1112
11	25,4	379	20	48,9	1198

Since proportional valves are generally used with openings larger than 20 percent, no more tests were carried out between 0 and 20 percent opening, and tests between 20 percent and 100 percent, which is the actual usage range of the valve, were started.

The pressure and flow results of the fifth test, in which the valve opening between 20 percent and 80 percent is tested by increasing 10 percent at each stage, are given in Figure 2.19. Although the tank pressure was adjusted to 10 bar in this test, it decreased during the test due to the regulator's inability to sufficiently feed the system.

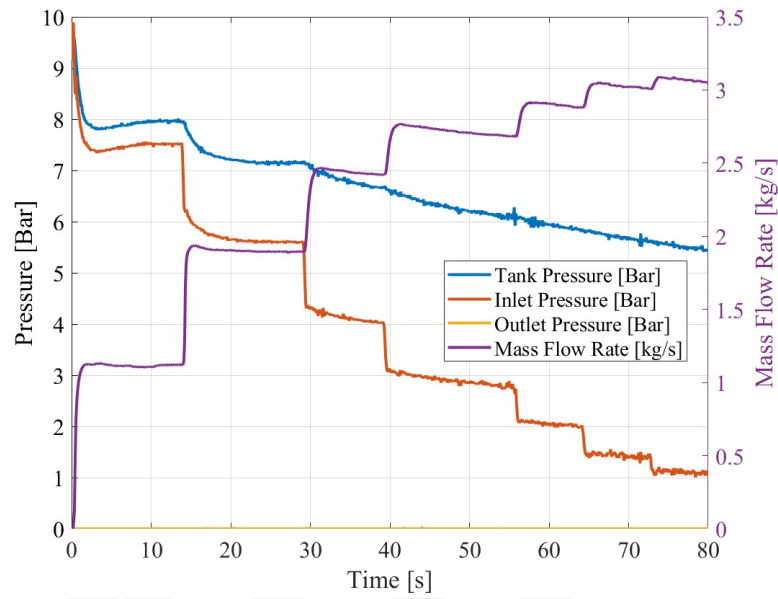


Figure 2.21. Pressures and mass flow rate results of characterization test 5.

In Figure 2.22, the mass flow rate is shown on the left axis, and the valve opening ratio at that moment and the discharge coefficient obtained by using Equation 2.15 are shown on the right axis.

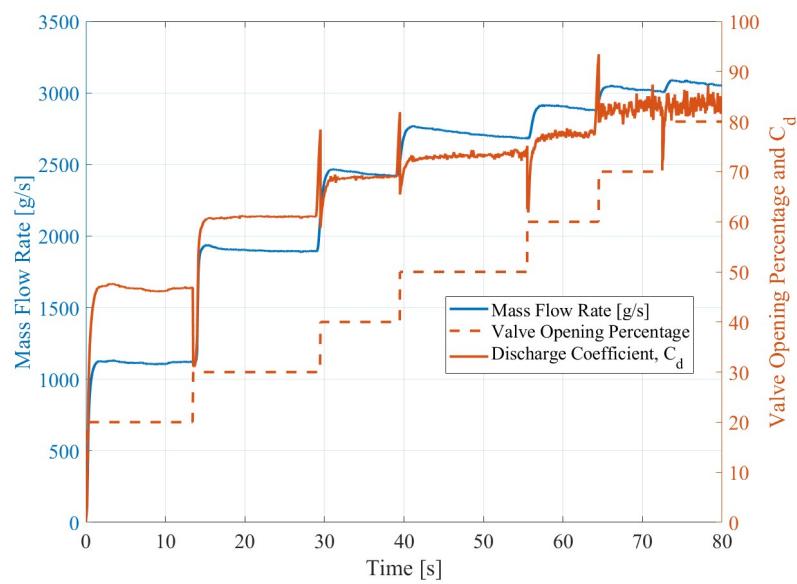


Figure 2.22. Mass flow rate, wastegate opening percentage and discharge coefficients, C_d , results of characterization test 5.

Finally, the opening ratio of the valve and the corresponding average mass flow rate and discharge coefficient values are given in Table 2.2. According to the results obtained here, the discharge coefficient values of the valve increased as the valve was opened, as expected, and reached 80 percent.

Table 2.2. Average C_d and $\dot{m}$ corresponding to valve opening percentage from test 5.

Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)
20	46,7	1115
30	60,9	1894
40	68,8	2441
50	73,2	2707
60	77,4	2899
70	82,8	3020
80	83,2	3073

The pressure and mass flow rate graph of the eleventh test, which was performed on a different day without a design change in the system, in which the opening values in the fifth test were simulated to show that the valve operates in a repeatable manner, is shown in Figure 2.23.

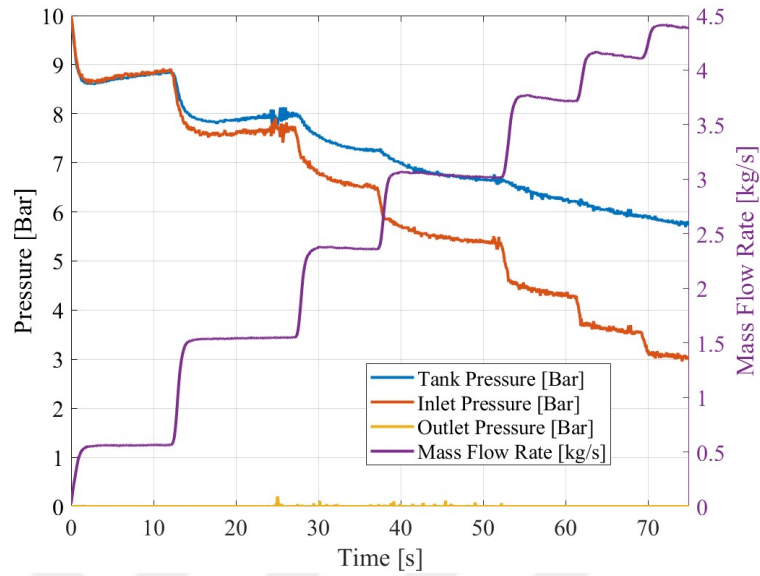


Figure 2.23. Pressures and mass flow rate results of characterization test 11.

In Figure 2.24, the mass flow rate is shown on the left axis, and the valve opening ratio at that time and the discharge coefficient obtained by using Equation 2.15 are shown on the right axis.

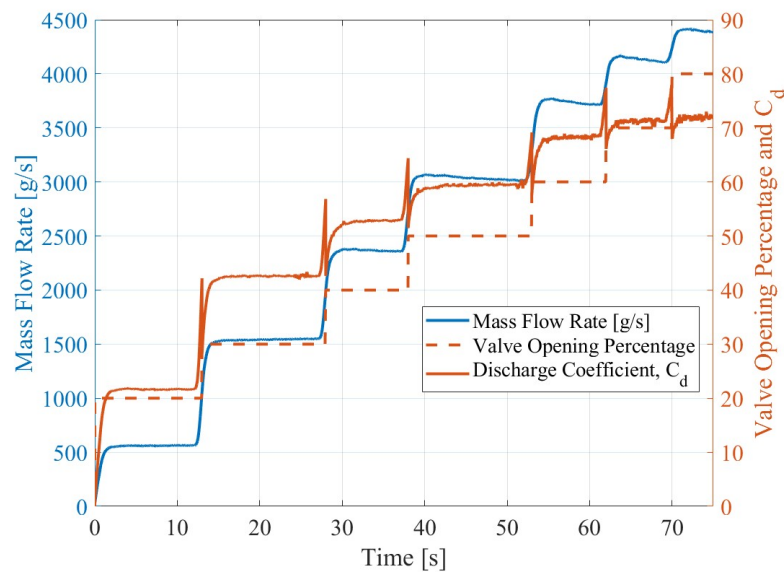


Figure 2.24. Mass flow rate, wastegate opening percentage and discharge coefficients, C_d , results of characterization test 11.

Finally, the opening ratio of the valve and the corresponding average flow rate and discharge coefficient values are given in Table 2.3. When the values in Table 2.3 are compared with the values given in Table 2.2 from the fifth test, an average of 15 percent decrease was observed in the discharge coefficient values of the valve.

Table 2.3. Average C_d and $\dot{m}$ corresponding to valve opening percentage from test 11.

Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)
20	21,6	558
30	42,6	1540
40	52,7	2367
50	59,4	3033
60	68,3	3734
70	71,3	4135
80	71,9	4396

When the two tests were compared, it was observed that the only distinction was the difference in inlet pressures. Later, when research on this subject was conducted in the literature, it was noticed that this situation was due to the phenomenon called cavitation [35].

In areas where the static pressure is low, gas or vapor can be released from the liquid, resulting in bubble formation, a process known as cavitation. Cavitation happens when the local pressure drops to the same level as the vapor pressure of the liquid. This phenomenon can significantly impact the discharge coefficient. Additionally, when these bubbles implode or collapse, they can negatively influence the flow field and potentially cause considerable wear in the passages of the cavitation nozzle. Under cavitating conditions, the behavior of the discharge coefficient (C_d) differs significantly from that observed under non-cavitating conditions [35]. The cavitation number is calculated as

$$\text{Cavitation Number, } C = \frac{P_1 - P_2}{P_1 - P_v} \quad (2.16)$$

where P_1 is the upstream pressure, P_2 is the downstream pressure, and P_v is the vapor pressure of the fluid. In the literature, in the studies examining the relationship between the cavitation number and the discharge coefficient, the reciprocal of the cavitation number is used, and this unknown is called K [35]. The formula of K is given as

$$K = \frac{1}{\text{Cavitation Number, } C} = \frac{P_1 - P_v}{P_1 - P_2}. \quad (2.17)$$

Data obtained from studies conducted by different researchers show the relationship between the discharge coefficient and the K value as seen in Figure 2.25. According to Figure 2.25, the discharge coefficient increases in direct proportion to K up to a critical K value. After the critical K value, the increase in the K value does not change the discharge coefficient, that is, the flow has non-cavitating flow characteristics. Although the critical K value cannot be calculated exactly, it is a value around 2 and the C_d value is around 0.83 [35].

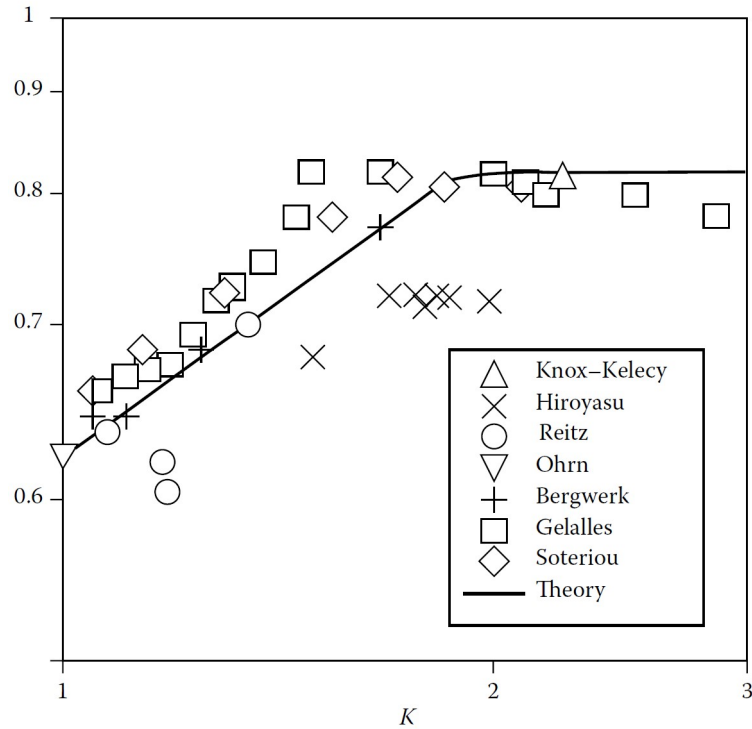


Figure 2.25. Relationship between C_d and K [35].

Considering the significant effect of the K value on the discharge coefficient, while calculating the valve's discharge coefficient value depending on the opening, the K value was also considered as a new parameter. The discharge coefficient values of Test 5 and Test 11, in which the fifth test is simulated, including the cavitation number at different openings, are shown in Table 2.4.

Table 2.4. Average C_d and K corresponding to valve opening percentage comparison between test 5 and test 11.

Test 5			Test 11		
Opening Percentage (%)	K	C_d (%)	Opening Percentage (%)	K	C_d (%)
20	1,13	46,7	20	1,11	21,6
30	1,17	60,9	30	1,13	42,6
40	1,24	68,8	40	1,15	52,7
50	1,34	73,2	50	1,18	59,4
60	1,48	77,4	60	1,23	68,3
70	1,69	82,8	70	1,28	71,3
80	1,89	83,2	80	1,32	71,9

As can be seen from Table 2.4, since there is a difference in the cavitation numbers between the two tests, the discharge coefficient values were different, and the discharge coefficient increased as the cavitation number increased. In Test 5, when the cavitation number was about 1.7, the discharge coefficient, C_d , approached 0.84, which is the maximum value expected in the literature.

Lastly, it was tested how the mass flow rate and the discharge coefficient would change when the proportional valve was opened more than 100%. In the twelfth test, the orifice of the proportional valve was opened from 70% to 120%. Figure 2.26 shows the pressure and mass flow rate results of the twelfth test.

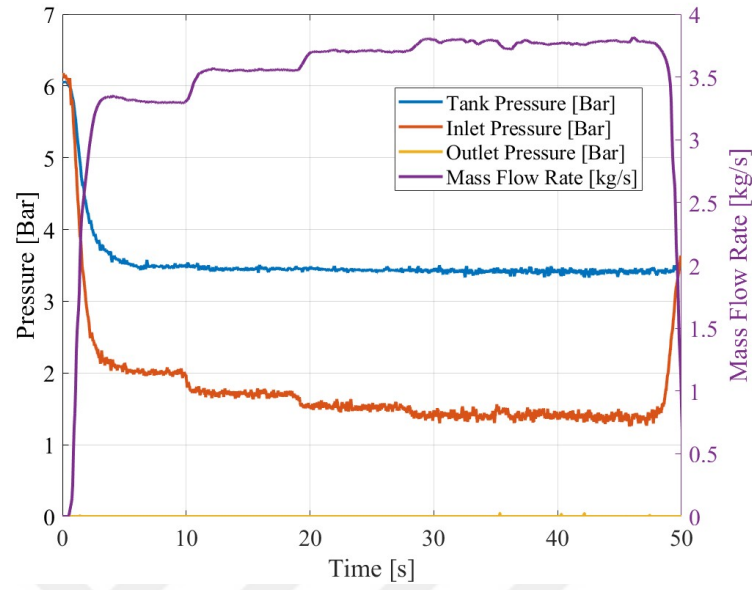


Figure 2.26. Pressures and mass flow rate results of characterization test 12.

In Figure 2.27, the mass flow rate is shown on the left axis, and the valve opening ratio at that time and the discharge coefficient obtained by using Equation 2.15 are shown on the right axis.

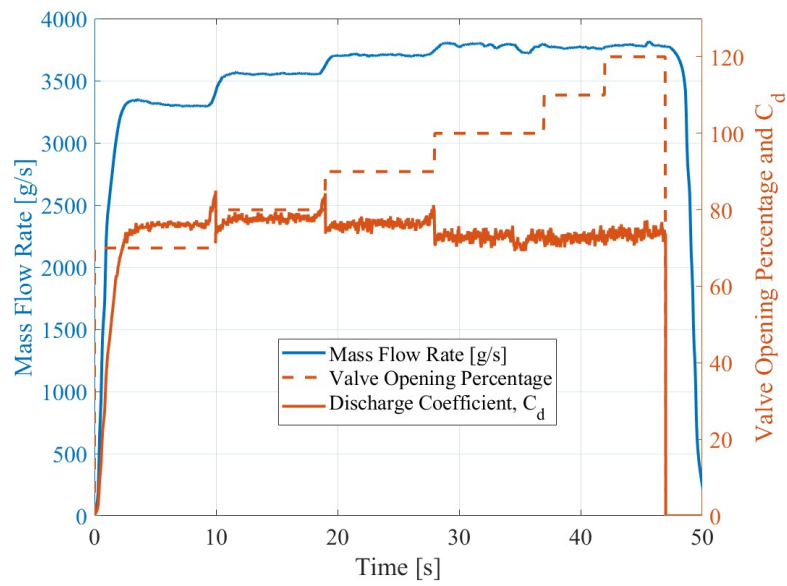


Figure 2.27. Mass flow rate, wastegate opening percentage and discharge coefficients, C_d , results of characterization test 12.

Finally, the opening ratio of the valve and the corresponding average mass flow rate, K , and discharge coefficient values are given in Table 2.5.

Table 2.5. Average K , C_d , and $\dot{m}$ corresponding to valve opening percentage from test 12.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
70	1,48	75,6	3307
80	1,57	77,7	3550
90	1,64	76,3	3706
100	1,69	72,6	3802
110	1,70	72,7	3766
120	1,72	73,9	3782

As can be seen in Table 2.5, when the opening ratio of the proportional valve exceeds 100%, there is no difference in the mass flow rate and discharge coefficient values compared to the case when the valve is 100% open.

The results of other water tests are given in the Appendix C.

2.5. Characterization of Proportional Wastegate

According to the literature, it was seen that two important characterization studies should be carried out regarding the proportional wastegate. Based on the test results, first the C_d characterization of the pintle valve at different valve opening percentages was carried out and then the characterization of C_d according to the cavitation number was carried out.

2.5.1. C_d Characterization

Considering the final design and the general operating ranges of proportional valves, the C_d value of the proportional wastegate was derived for 3 different ranges: 0-20%, 20%-80% and 80%-100%. In determining the C_d values, various curve-fitting techniques were evaluated. The polynomial fitting method was chosen, allowing the calculated C_d values to be accurately defined across different operational ranges [36].

Two water tests were conducted for the valve's 0–20% opening range, as this range is not critical for proportional wastegate operation. As mentioned in the characterization tests of the proportional valve, the mass flow rate of water through the proportional valve could not be measured for openings below 3%. Therefore, the C_d was calculated only within the 3–20% opening range. The behavior of C_d in this range was modeled with a fourth-degree polynomial $y = -0.0047x^4 + 0.2303x^3 - 3.8187x^2 + 25.857x - 31.459$ calculated by the curve fitting method. The data and curve fit used for this range are shown in Figure 2.28.

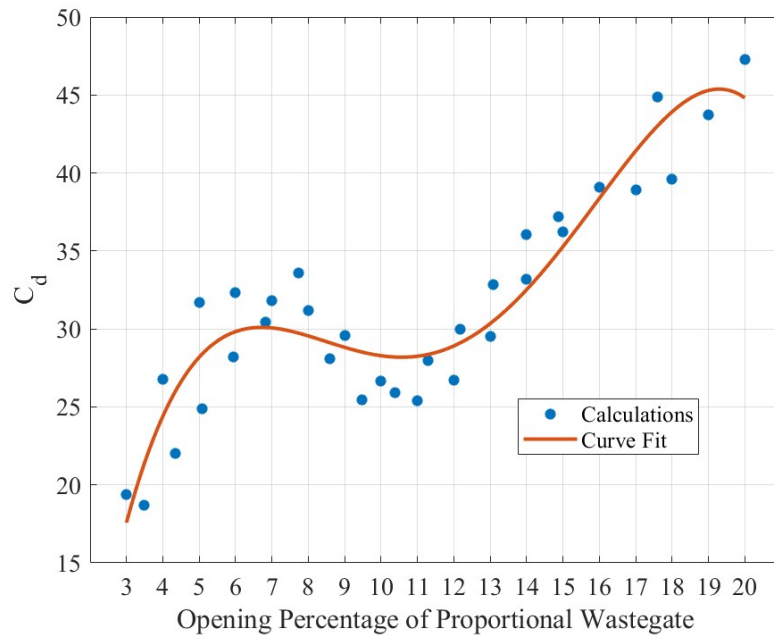


Figure 2.28. C_d corresponding to proportional wastegate opening percentage between 3%-20%.

Since the range of 20%-80% is the main operating range of proportional valves, multiple tests were conducted in this range to ensure accurate characterization. The behavior of C_d in this range was modeled with a second degree polynomial $y = -0.0087x^2 + 1.3842x + 22.509$ calculated by the curve fitting method. Figure 2.29 shows the data and the curve fit used for this range.

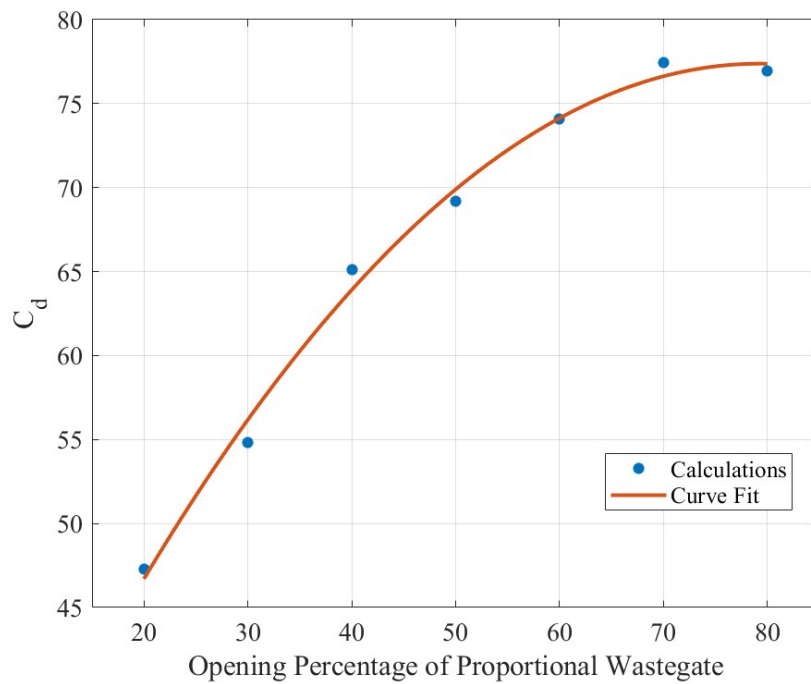


Figure 2.29. C_d corresponding to proportional wastegate opening percentage between 20%-80%.

Finally, to determine the maximum flow rate achievable by the proportional wastegate, numerous tests were conducted in the 80%-100% range. In this range, the behavior of C_d was modeled with a second degree polynomial $y = -0.0054x^2 + 0.7166x + 54.414$ calculated by the curve fitting method. In Figure 2.30, the data and curve fit used for this range are shown.

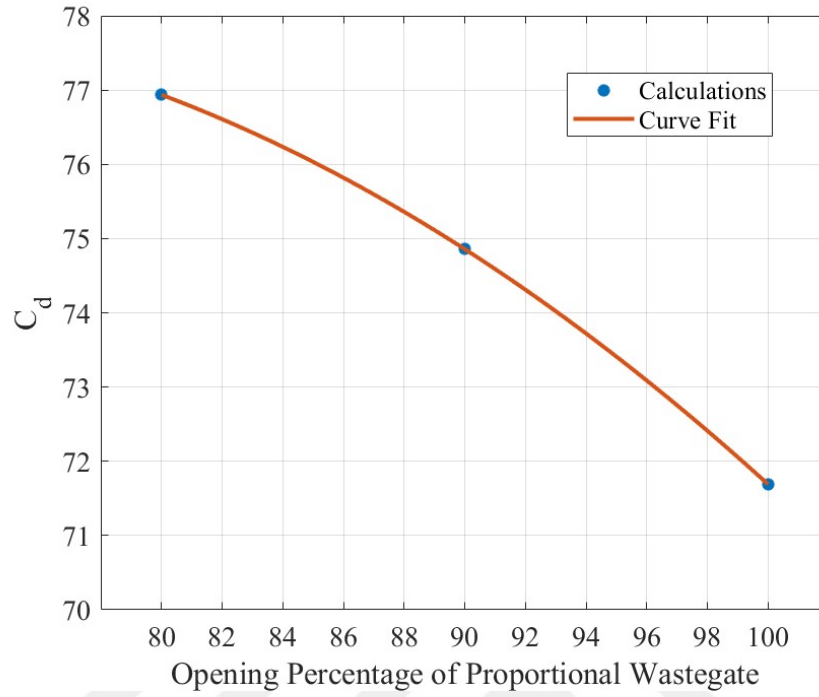


Figure 2.30. C_d corresponding to proportional wastegate opening percentage between 80%-100%.

2.5.2. C_d versus K Characterization

An examination of the test results revealed that the C_d values calculated for the same proportional valve openings varied across different tests. Literature review suggests that this variation is due to changes in the cavitation number, K , which in turn depend on fluctuations in inlet pressure of proportional wastegate for tests conducted on different days. C_d values were examined depending on the K for the proportional wastegate, as used in the studies in the literature and as shown in Figure 2.31. The relationship between C_d and K was determined using a linear regression model, resulting in the equation $y = 0.1223x + 0.5665$.

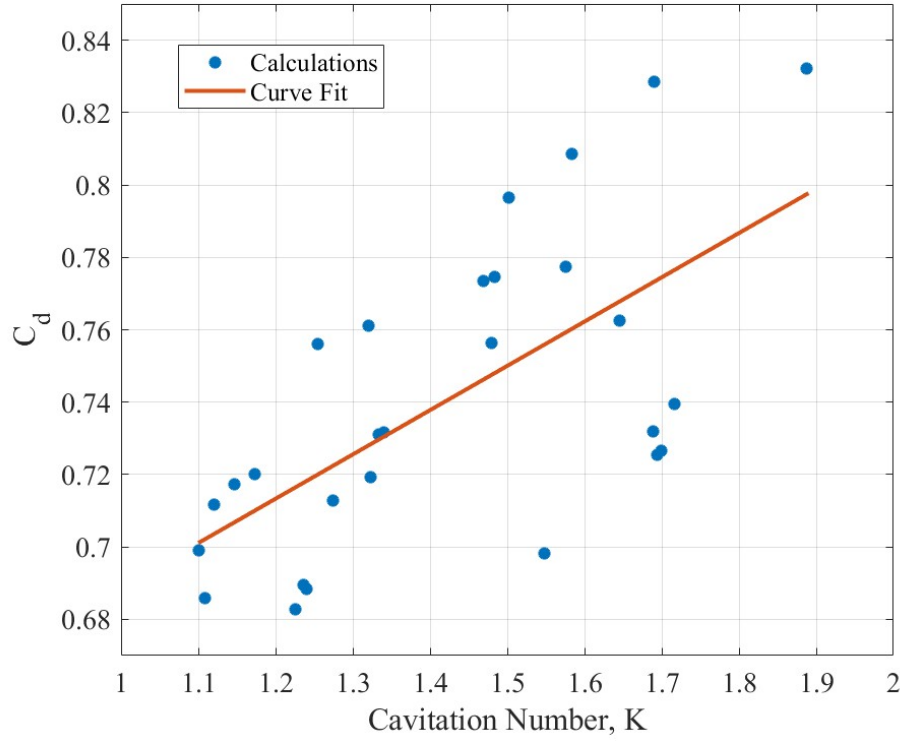


Figure 2.31. C_d versus K relation obtained from characterization tests.

To evaluate the fit quality of the linear regression model describing the relationship between C_d and K , the coefficient of determination (R^2) was calculated. The coefficient of determination for the linear regression model is calculated as [37]

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2} \quad (2.18)$$

where R^2 is the coefficient of determination, y_i are the calculated C_d values, $\hat{y}_i$ are the predicted values, $\bar{y}$ is the mean of calculated C_d values. Curve fit given in Figure 2.31 has R^2 value of 0.42 which indicates that the model has limited predictive accuracy. An R^2 value of 0.42 suggests that only 42% of the variance in C_d is explained by the variation in K , while the remaining 58% of the variance is due to other factors not captured by the model. This relatively low R^2 implies that the linear regression model may not adequately represent the relationship between C_d and K , and it may be beneficial to include additional variables to improve prediction accuracy. Therefore, the characterization of the proportional wastegate should be done not only by considering

C_d or K but also by considering the opening ratio of the proportional valve.

The C_d values obtained from the tests were multiplied by the opening area of the proportional wastegate and then normalized by dividing by the maximum area of the wastegate, yielding the value $C_d A^*$. The resulting equation from the linear regression model is $y = 0.98825x - 0.9504$. Figure 2.32 shows the calculated $C_d A^*$ values along with the fitted curve as a function of the K value.

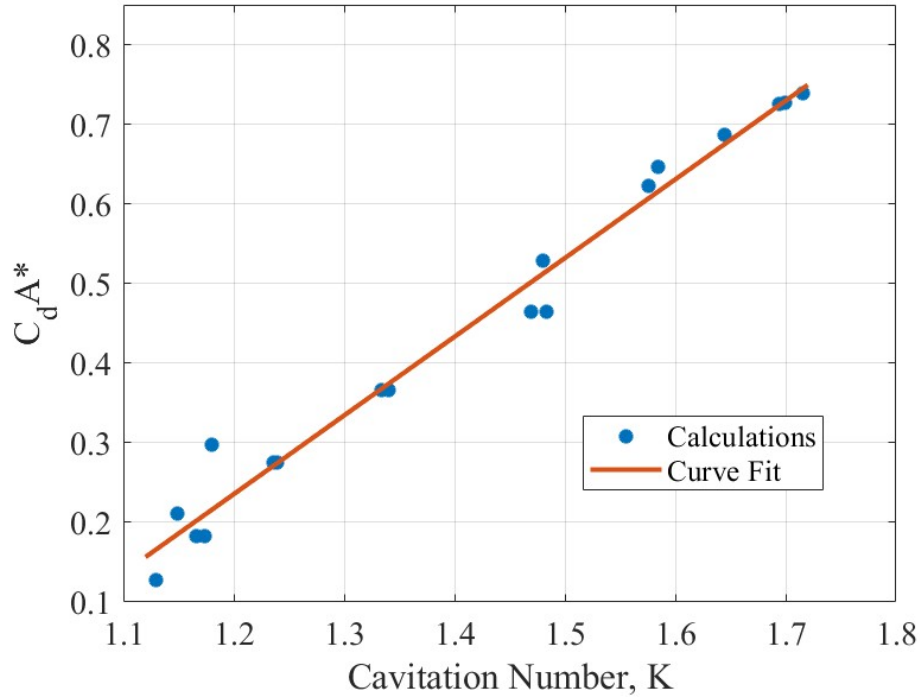


Figure 2.32. $C_d A^*$ versus K relation obtained from characterization tests.

The R^2 value of the linear regression model, which examines the relationship between the $C_d A^*$ value (calculated using the normalized valve area defined as the new variable) and the K , was calculated to be 0.98. This value indicates that the characterization of the proportional wastegate, with the $C_d A^*$ obtained from the newly defined normalized area variable, follows a strong linear trend.

2.5.2.1. Uncertainty Analysis. The accuracy of the testing process directly impacts the reliability of the characterization of the proportional wastegate's discharge coefficient. However, uncertainties in the testing arise due to factors such as the resolution and accuracy of the sensors, as well as external perturbations. The level of uncertainty in the discharge coefficient characterization can be evaluated through linear regression analysis of the measured data. During the tests for characterizing the proportional wastegate, multiple data points were collected under consistent and varying conditions using the same sensors to calculate the discharge coefficient.

In Figure 2.33, blue circles indicate the discharge coefficient normalized by the valve opening ratio based on the cavitation number, while the orange continuous line represents the linear curve fit derived from these data. The dashed purple lines indicate the 95% prediction interval, corresponding to a two sigma standard deviation, with an uncertainty of 0.0656 in discharge coefficient. As mentioned above, the coefficient of determination of this linear fit is 0.98 and this value indicates a strong linear fit with relatively low uncertainty.

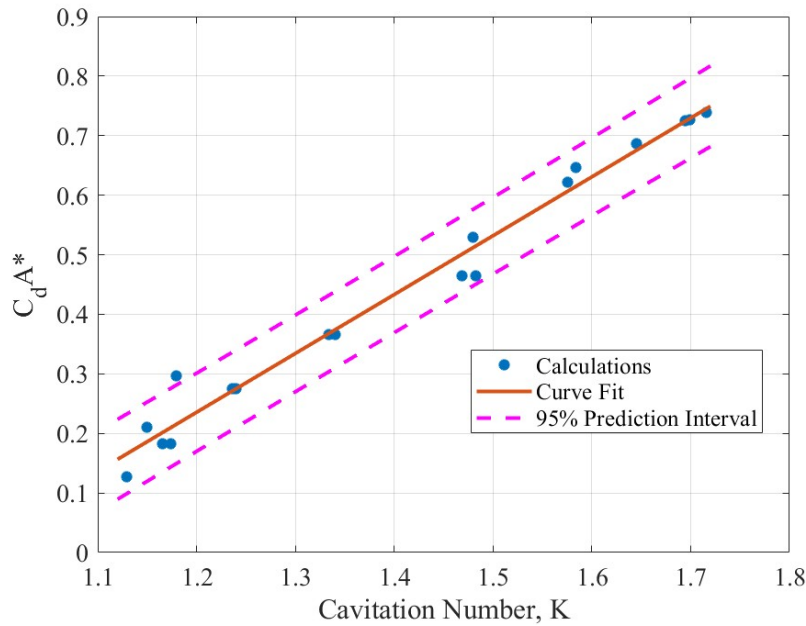


Figure 2.33. $C_d A^*$ versus K relation obtained from characterization tests with linear fit and 95% prediction interval.

3. DEVELOPMENT OF RPM SENSOR FOR PROPORTIONAL WASTEGATE CONTROL

In turbopump systems, the one of the most important indicators of the transfer of the desired power to the pump by the turbine is the rotational speed of the shaft, because if the pump rotates at the speed for which it is designed, it provides the desired pressurization. If the revolutions per minute (RPM) of the turbopump system is less than expected, it means that enough energy is not supplied to the turbine, and if it is more than expected, it means that the turbine is given more energy than necessary. The proportional wastegate will adjust the energy supplied to the turbine by modifying the discharge orifice: when the energy going to the turbine is insufficient, the orifice will be constricted to provide more energy, and when there is excess energy, the orifice will be expanded to reduce the energy supplied. As a result, it is necessary to measure the RPM value of the turbopump system accurately and quickly in order to decide whether the wastegate will operate in the opening or closing direction. In this chapter, the RPM sensor development process using the proximity sensor and the execution of the sensor's functionality tests will be discussed.

3.1. RPM Sensor Types

In turbopump systems, different methods are used to read the RPM value. Since the turbopump system used rotates at 24000 RPM, it is classified as a high-speed application. In the literature, optical encoders, non-contact tachometers, magnetic pickups, and high frequency proximity sensors are used for such high-speed RPM measurement applications [38].

3.1.1. Optical RPM Sensors

These sensors work by detecting interruptions in light, usually achieved by placing a reflective or marked surface on a rotating part. When a light source, such as a laser,

is directed onto the rotating component, the sensor detects the interruptions and calculates the RPM based on the frequency of the interruptions. Optical sensors are often used in environments where high accuracy is required, and they can be non-intrusive, as they don't require physical contact with the object being measured. However, they can be susceptible to dust, dirt, and vibration, which can impact accuracy [39]. Figure 3.1 illustrates a laser tachometer, which is a type of optical RPM sensor.



Figure 3.1. Laser tachometer [40].

3.1.2. Magnetic RPM Sensors

Magnetic sensors measure rotational speed by detecting changes in magnetic fields created by rotating magnets or ferrous teeth on a gear. As the magnetic field fluctuates, the sensor generates a voltage pulse, which is then used to calculate RPM. Magnetic sensors are robust and can function in harsh environments with high levels of dust or temperature variations, making them suitable for automotive and industrial applications. Figure 3.2 illustrates a magnetic RPM sensor, which detects rotational speed by measuring changes in the magnetic field as a gear rotates [41].

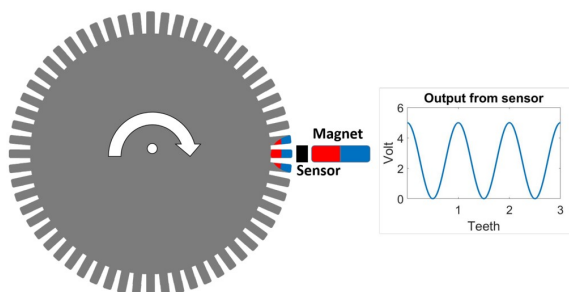


Figure 3.2. Magnetic RPM sensor [41].

3.1.3. Inductive RPM Sensors

These sensors work similarly to magnetic sensors but rely on the inductive effect caused by the proximity of metallic components. Inductive sensors detect changes in the electromagnetic field as a metal object passes in front of them. They are durable and are often used in heavy-duty machinery. However, they are not suitable for non-metallic objects [39].

3.2. RPM Sensor Selection

During the design phase of the turbopump system, an RPM measurement setup using a laser tachometer was envisioned. As a result, holes were drilled in the casing surrounding the shaft that connects the turbine and pump, and reflective tapes were applied to the shaft's surface to facilitate laser tachometer readings. However, tests revealed that the laser tachometer was susceptible to system vibrations and prone to data loss if any leakage occurred from the pump. Figure 3.3 shows a pre-test layout of the turbopump system with the laser tachometer.

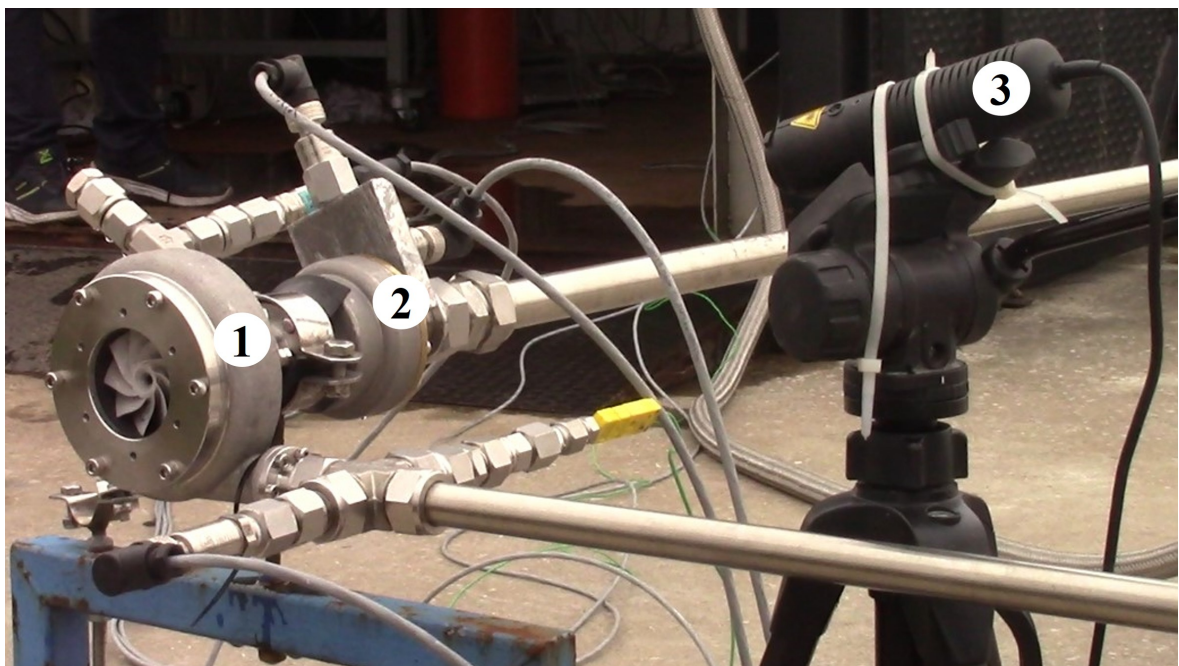


Figure 3.3. Test setup of the turbopump system with the laser tachometer.

In Figure 3.3, number 1 shows the turbine, number 2 shows the pump, and number 3 shows the laser tachometer. The laser tachometer measures the RPM from the shaft connecting the turbine and the pump.

In the test to see the measurement accuracy and speed of the optical RPM sensor, although a proportional wastegate was not used, the energy supplied to the turbine was controlled through a bang-bang system using two solenoid valves. This setup maintained a constant turbine inlet pressure by using feedback solely from the inlet pressure, without any feedback from the RPM values.

Figure 3.4 shows the RPM values measured with a laser tachometer during the turbopump test. Examining the pump outlet pressure, which serves as an additional indicator of the turbopump system's performance, confirmed that the pump was operating as expected. However, it was observed that the laser tachometer, affected by system vibrations and leaks, was unable to provide accurate measurements during the test.

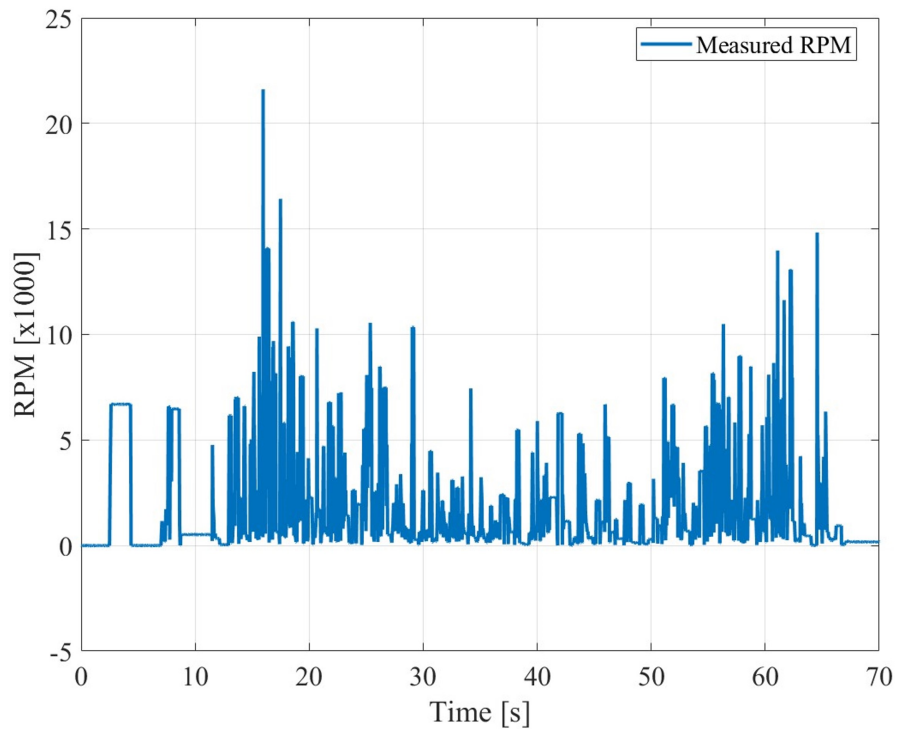


Figure 3.4. RPM measurements of the turbopump system with the laser tachometer.

When the laser tachometer did not perform as expected, an examination of other RPM measurement sensors on the market revealed that the currently available options provided analog outputs and were capable of measuring speeds ranging from 200 to 400,000 RPM, depending on the number of blades. Additionally, these sensors demonstrated durability and would offer convenience in controlling the proportional wastegate. However, the chosen sensor is a magnetic RPM sensor that requires the turbine blades to be made of aluminum or titanium for accurate measurement. Since the material of the turbine blades in used is Inconel, this sensor is not applicable. Furthermore, these sensors are quite expensive. Figure 3.5 shows how these speed sensors are mounted on the turbine [42].

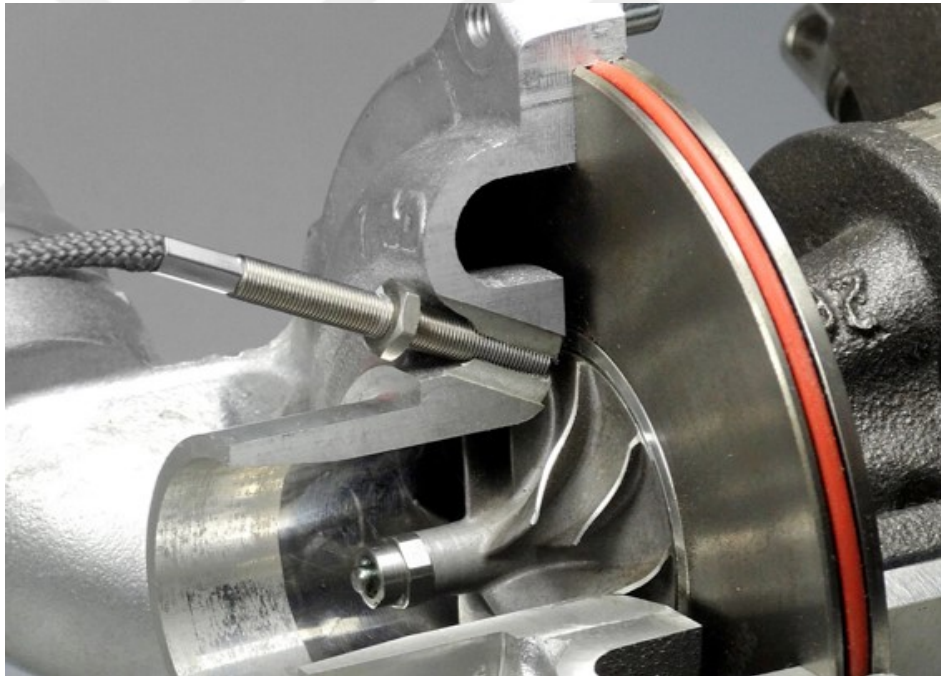


Figure 3.5. Speed sensor mounted on a turbine [42].

A high-frequency inductive proximity sensor was chosen for RPM measurement due to the narrow space available for measurements from the shaft in the turbopump system. This sensor is less impacted by potential leaks in the pump or turbine, ensuring accurate readings. Additionally, it offers a more cost-effective solution compared to other types of sensors.

Upon examination of these sensors, it is observed that their output is digital, providing only binary data (0 and 1) rather than the analog signals typically ranging from 4 to 20 mA that reflect varying values based on rotation speed. This digital output presents a disadvantage, as the avionics systems responsible for controlling the proportional wastegate may experience delays or data loss when processing and interpreting the digital signals. In contrast, analog data directly conveys information about the rotation speed, facilitating more immediate and efficient communication with the avionics systems.

The selected sensor is the DCC 05 V 0.8 PSK-K-TSL coded product of Di-Soric company. A picture of the sensor is given in Figure 3.6. The material of the housing of the sensor is stainless steel and the material of the cable is polyurethane [43].



Figure 3.6. Selected inductive proximity sensor [43].

The selected sensor was chosen from the company's miniature product line to facilitate easier placement in confined spaces. Given that the sensor features M5x0.5 threads, the installation process was straightforward. In addition, a measurement distance of 0.8 mm was selected, as opposed to the standard 1.5 mm or 2 mm products, to minimize the risk of interaction with adjacent metal components in the restricted

area where it is installed, thereby reducing the likelihood of false outputs. The sensor switching output is 'PNP' and this sensor gives 1 digitally when a metal object comes closer than 0.8 mm to the sensor. When the sensor gives 1 digitally, the LED light on it also lights up [43].

Apart from being small and measuring at a very close distance, the most important factor in the selection of this sensor was that it could receive data at high frequency to measure the high rotational speed of the turbopump system. The minimum frequency required for the sensor to make an accurate measurement depends on the shaft speed of the turbopump system and how many teeth there will be on the shaft. Since the shaft in the turbopump system is intended to rotate at 24000 RPM, the rotation speed of the shaft per second is calculated as

$$\text{Revolution per second, rps} = \frac{\text{Revolution}}{1 \text{ minute}} \times \frac{1 \text{ minute}}{60 \text{ seconds}} = \frac{24000}{60} = 400. \quad (3.1)$$

According to the result found in Equation 3.1, a point on the shaft rotates 400 times per second. The more teeth on the shaft, the higher the frequency required for the sensor to read without missing one of these teeth. Since any imbalance of the shaft connecting the turbopump system will cause significant malfunctions in the system at high speeds, an even number of symmetrical teeth has been made on the shaft instead of an odd number of teeth.

After determining that an even number of teeth would be placed on the shaft, the processing speed of the avionics systems—responsible for interpreting the sensor output and subsequently controlling the proportional wastegate—became a critical limitation. Initially, the design intended to incorporate a total of four teeth, spaced at 90-degree intervals around the shaft. However, due to the avionics system's communication frequency, which operates at approximately 1200 Hz, it was ultimately decided to install only two teeth on the shaft, positioned 180 degrees apart. This adjustment ensures that the system can effectively process the RPM measurements without exceeding the communication capabilities of the avionics systems. The reason why the number of

teeth is desired to be increased in the beginning is that when calculating the RPM according to the outputs from the sensor, if a tooth is missed due to noise or any other reason and no data can be obtained from the sensor, the next tooth is seen faster, and the RPM calculation is immediately corrected. The minimum frequency required for the sensor to accurately measure the RPM value is calculated as

$$\text{Minimum frequency required} = 400 \times \text{number of teeth} = 400 \times 2 = 800 \text{ Hz.} \quad (3.2)$$

As a result, since the minimum frequency value required for the measurement of the RPM value of the system is lower than the communication speed of the avionic systems, there is no problem for the control of the valve. In addition, the measurement frequency of the selected sensor is 5000 Hz, and there is no restriction here either.

3.3. RPM Sensor Measurement Algorithm

Before initiating the preparations for testing the sensor, consideration was given to the method of calculating RPM from the digital outputs generated by the sensor. Since the avionics systems responsible for controlling the electric motor of the proportional wastegate can receive data at approximately 1200 Hz, this frequency exceeds the 800 Hz threshold necessary for the sensor to make accurate measurements with two teeth on the shaft at the turbopump system's operational speed of 24,000 RPM. Consequently, the time interval between the detection of one tooth by the RPM sensor and the subsequent detection of the other tooth can be effectively measured and recorded by the avionics systems. Two methods for calculating RPM are considered, with the intention of verifying each other by using the digital data—0 and 1—received from the RPM sensor.

In the first method, RPM is calculated by determining the time difference between the first detection of one tooth by the sensor and the first detection of the next tooth. This time interval represents the period between two teeth and, when multiplied by the total number of teeth on the shaft, gives the time needed for one full rotation. By

inverting this value, the rotational frequency of the shaft can be determined, which is then converted to the number of revolutions per minute (RPM). This approach provides a straightforward calculation based on individual time intervals, facilitating accurate RPM measurements for turbopump control. Figure 3.7 shows the schematic representation of the first method for calculating RPM using a proximity sensor.

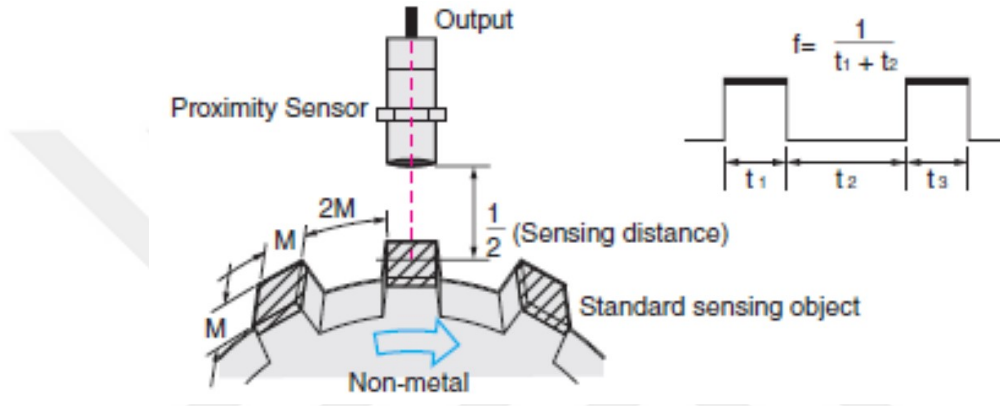


Figure 3.7. Schematic of the first method for measuring rotational speed with an RPM sensor [44].

The steps for RPM calculation in the first method are as

$$\Delta t_{\text{tooth-to-tooth}} = t_1 + t_2$$

Time required for one full rotation in a second = $\Delta t_{\text{tooth-to-tooth}} \times \text{number of tooth}$

$$\text{RPM} = \frac{60 \text{ seconds}}{\text{Time required for one full rotation in a second}}. \quad (3.3)$$

In the second method for calculating RPM, the total number of teeth detected by the proximity sensor over a specified time period, such as 0.2 or 0.5 seconds, is counted using a counter. This count is then divided by the number of teeth on the shaft to determine the shaft's rotation speed during that period. Finally, this rotation speed is converted to RPM. The RPM calculation steps in the second method are as

$$\begin{aligned} \text{Total number of rotations in a specified time period, } n &= \frac{\text{counter}}{\text{number of tooth}} \\ \text{Rotation speed of turbopump in a second} &= \frac{n}{\text{specified time period}} \quad (3.4) \\ \text{RPM} &= 60 \text{ seconds} \times \text{Rotation speed of turbopump in a second.} \end{aligned}$$

Basically, although the same thing has been done in both methods, the RPM value obtained by the first method is more accurate. The reason why the second method gives more inaccurate results is the accumulation of sensor and avionic system errors in the selected time interval.

3.4. RPM Sensor Tests

A high-speed electric motor was used to test the successful operation of the RPM sensor and avionics systems without damaging the turbopump system. A coupling was designed to attach to the shaft of the electric motor, featuring two teeth to enable the RPM sensor to measure the shaft's rotation speed. The designed coupling is shown in Figure 3.8.



Figure 3.8. Coupling used in RPM sensor tests.

The electric motor used in the tests rotates at the RPM value given by the user with the help of its controller. The electric motor rotates at a maximum of 400 Hz, which meets the 24000 RPM required for the turbopump system. The picture of the electric motor controller is shown in Figure 3.9.



Figure 3.9. Electric motor controller.

An L-bracket is made to connect the proximity sensor to the electric motor, and the distance of the sensor to the electric motor shaft is adjusted with the teeth on the sensor and two nuts. Since the proximity sensor has a sensitive measuring range, the shaft of the electric motor is rotated by hand to see if it is at a suitable distance, and it is observed whether the LED light on the sensor is on. If the light is on and the sensor does not touch the shaft, the system is ready for testing, otherwise the sensor's distance is adjusted according to the situation. Figure 3.10 shows the assembly of the electric motor and the proximity sensor for the RPM measurement test.

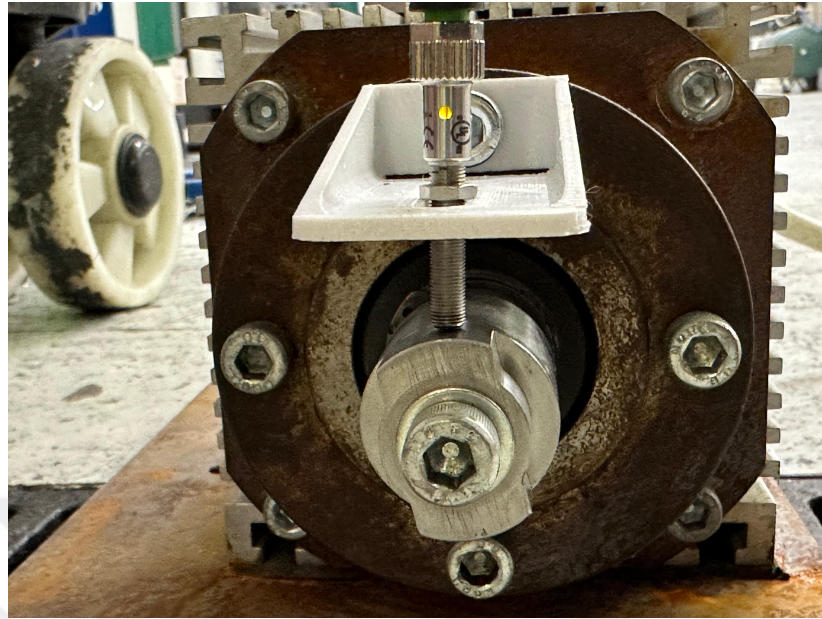


Figure 3.10. Proximity sensor mounted on electric motor.

In the first test, the rotation speed given by the electric motor and the measured RPM values aligned as expected, indicating no apparent issues. However, upon analyzing the data later, it became clear that the avionics systems had only measured certain RPM values of the electric motor. The reason for this was determined to be the measurement interval of the avionics systems, which operated in microseconds, causing the $\Delta t_{\text{tooth-to-tooth}}$ values to change at only 1-microsecond intervals depending on the different rotation speeds.

While solving the problem in avionics systems, in order to observe whether the proximity sensor can successfully receive data at high frequency, the cable connections at the output of the sensor were changed to match the DAQ system, which can take measurements at 5000 Hz. Since the sensor only gives digital output, the sensor did not give current when there was nothing in the measuring distance, and therefore the DAQ system gave 0 as output. When something was in the measuring range of the sensor, it passed a current of 100 mA and gave a value different from 0 and greater than 0 in the DAQ system, according to the resistance of the sensor.

With the help of MATLAB, the time intervals in which this DAQ's output changes were found, and RPM was calculated by using the formulas in Equation 3.3 used by avionic systems. Test results are shown in Figure 3.11. When the noise originating from the DAQ system itself was filtered out, it was observed that the sensor successfully measured the 24000 RPM value targeted by the turbopump system. The reason why the DAQ system, which works successfully in the tests, cannot be used for the control of the proportional wastegate valve is that the DAQ system cannot give real-time data to the system that will control the proportional wastegate.

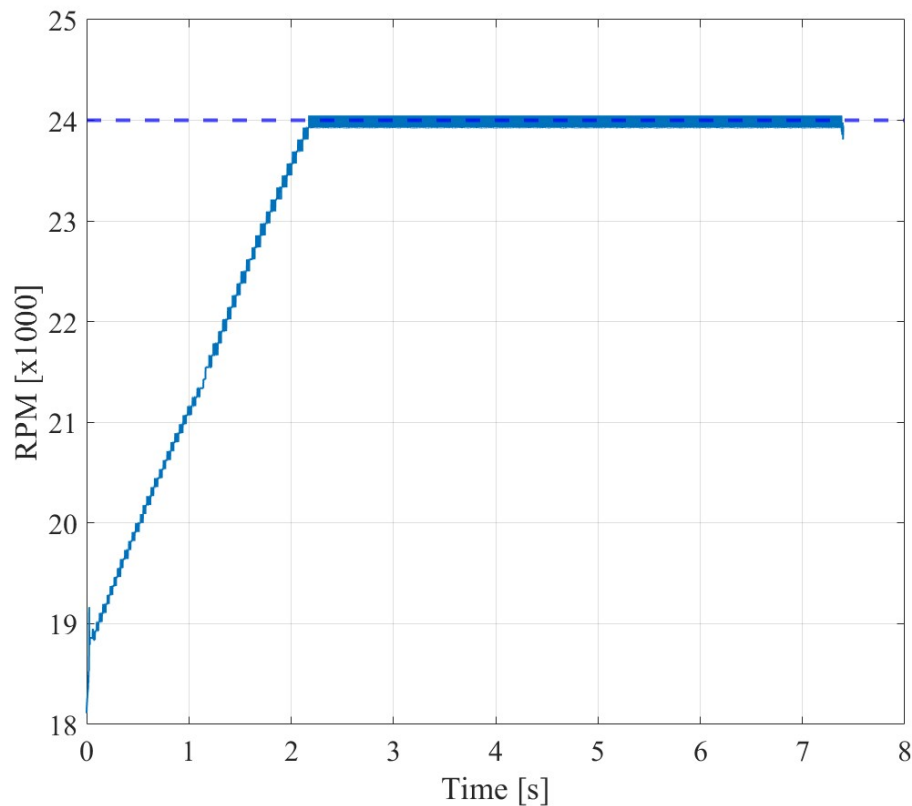


Figure 3.11. Measured RPM versus time from RPM sensor with DAQ system.

After demonstrating that the sensor works successfully, the time interval of the avionics systems was reduced from microseconds to nanoseconds to resolve the error in the avionics systems, thereby increasing the sensitivity of the RPM measurement system. When tests were conducted following these improvements, it was observed

that the system successfully received data and measured RPM even at high speeds. Figure 3.12 shows the RPM values obtained from one of the tests performed after the system was improved. During this test, the system was operated at various rotation speeds using the controller of the high-speed electric motor, allowing for the verification of the RPM measurement sensor and algorithm across different RPM values. As can be seen, the RPM value can be obtained more precisely with avionic systems, and it can also be measured with a less noise.

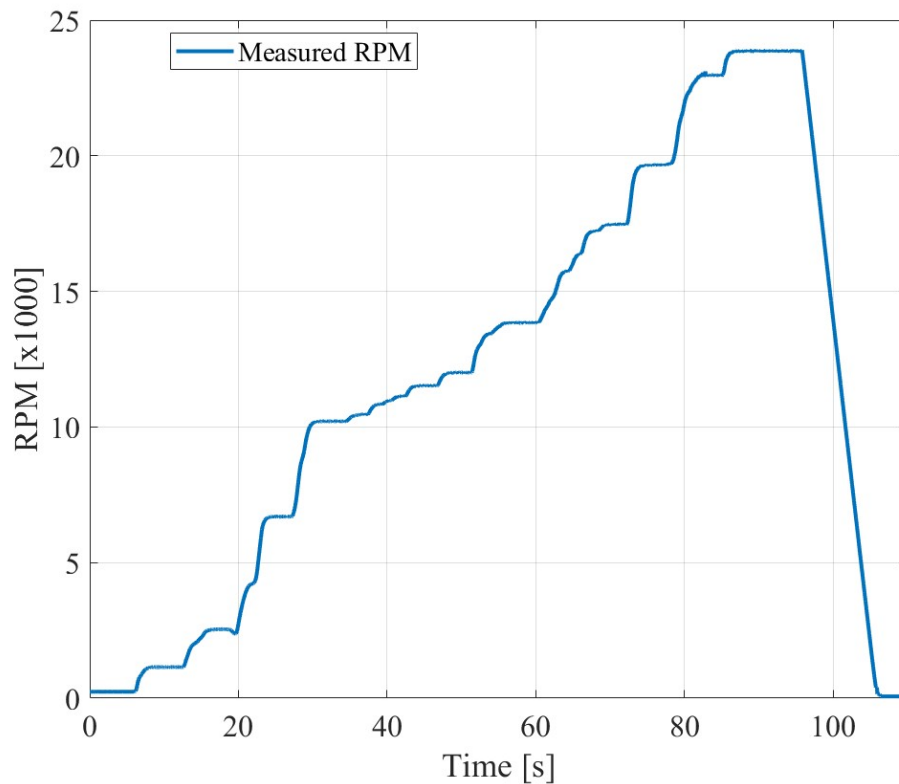


Figure 3.12. Measured RPM versus time from RPM sensor with avionic systems.

Testing the RPM sensor system at a maximum of 24000 RPM can be said to be the only shortcoming of the RPM measurement tests made with the electric motor and the proximity sensor, because it has not been observed how the system will behave at rotational speeds higher than 24000 RPM.

4. PID CONTROLLER DESIGN AND SIMULATION

In this chapter, the design process of the PID controller, which will control the proportional wastegate to ensure that the turbopump system operates at the desired rotational speed, will be discussed. Then, the preparation process of the simulation created in MATLAB Simulink, which mimics the test environment before conducting real-world tests and evaluates the functionality of the controller, will be explained. Finally, the simulation outputs will be evaluated.

4.1. PID Control Theory

The key characteristic of the PID controller is its ability to leverage the three control terms—proportional, integral, and derivative—to achieve precise and optimal control over the output. In Figure 4.1, the block diagram illustrates the principles behind the generation and application of these terms. It depicts a PID controller that continuously computes the error value by comparing the desired setpoint with the measured process variable. The controller then applies corrections derived from the proportional, integral, and derivative terms. To reduce the error over time, it adjusts a control variable—such as the position of a control valve—according to a new value calculated from a weighted combination of these terms [45].

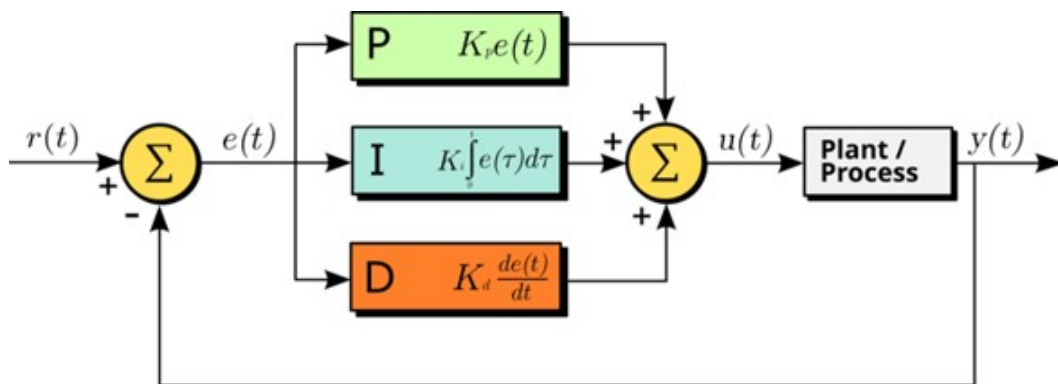


Figure 4.1. Block diagram of PID control loop [46].

In Figure 4.1, $r(t)$ is the desired process value, $y(t)$ is the measured process variable, $e(t)$ is the error value which is the difference between the desired process value and the measured process variable and $u(t)$ is the control variable. The overall control function is given as

$$u(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{de(t)}{dt} \quad (4.1)$$

where K_p is coefficient for proportional term, K_i is coefficient for integral term, and K_d is coefficient for derivative term. These terms can be denoted with P , I , and D respectively as well. In a PID model, each of these terms has unique meaning and effect.

Term P is directly proportional to the current error value. For instance, if the error is significant, the control output will be correspondingly large, based on the gain factor K_p . Relying solely on proportional control will result in a persistent error between the desired process value, set point (SP), and the measured process variable (PV), as the controller needs an error to produce a proportional response. In steady-state conditions, this results in a constant SP-PV “offset” where the system reaches an equilibrium [47].

A high proportional gain leads to a significant change in the output for a given change in the error. However, if the proportional gain is set too high, it can make the system unstable. On the other hand, a low proportional gain results in a smaller output response to a large input error, making the controller less responsive or sensitive. If the proportional gain is too low, the control action may be insufficient when reacting to disturbances in the system. Tuning theory and industrial practice suggest that the proportional term should account for the majority of the output change [47].

The I term addresses the accumulated past values of the error term, $e(t)$, by integrating them over time. For instance, if there is a remaining error after proportional control has been applied, the integral term works to eliminate this residual error by applying a control effect based on the cumulative historical error. Once the error is fully

corrected, the integral term will stop increasing. Consequently, while the proportional effect diminishes as the error decreases, this reduction is offset by the increasing integral effect [47].

The integral term speeds up the process's approach to the set point and addresses the residual steady-state error that a purely proportional controller cannot eliminate. However, because the integral term reacts to accumulated past errors, it can sometimes lead to the current value overshooting the set point.

The D term provides an estimate of the future trend of the error term, $e(t)$, based on its current rate of change. Often referred to as "anticipatory control," this term aims to mitigate the impact of the error by applying a control action proportional to the rate at which the error is changing. The faster the error changes, the stronger the controlling or damping effect will be [47].

Derivative action anticipates the future behavior of the system, which helps to improve settling time and enhance stability. However, an ideal derivative is non-causal, so practical implementations of PID controllers typically incorporate a low-pass filter for the derivative term to reduce high-frequency gain and noise. In practice, derivative action is rarely used—estimated to be part of only 25% of deployed controllers—due to its unpredictable impact on system stability in real-world scenarios [47].

4.2. Simulation Setup

While calculating the coefficient values of the controller that will control the movements of the wastegate proportional valve, the iteration process required during the design was tried to be shortened by using the MATLAB Simulink program. The reason why the controller coefficient values cannot be calculated manually is that the turbopump system cannot be physically modeled. For this reason, the transfer function of the system could not be obtained, and a solution could not be reached by analytical (hand) calculation.

Before calculating the PID coefficients required for the controller, the physical systems used during the test were started to be modeled using the Simscape module of MATLAB Simulink. The turbine and centrifugal pump blocks from the Simscape library were used to model the turbine and pump in the turbopump system. In these two blocks, the data tabulated from the data sheet given by the manufacturer of the turbine and the pump under normal conditions, such as the values of flow and efficiency versus pressure ratio, are used. However, since these data are not available for the system used, the closest system to the current system was obtained in the simulation by using the previous test results.

Figure 4.2 shows the block parameters used in the simulated turbine, where the isentropic and mechanical efficiency values of the modeled system have been modified until it mimics the real system.

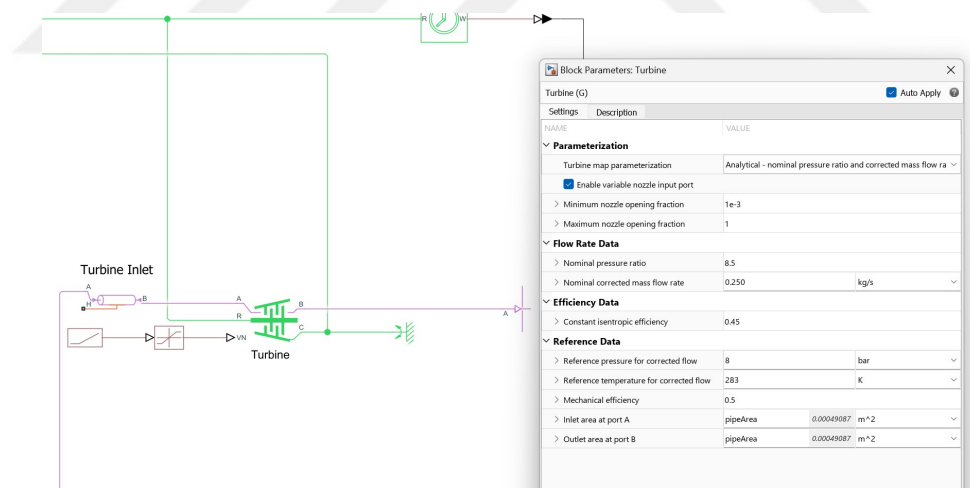


Figure 4.2. Turbine block used in MATLAB Simulink.

The centrifugal pump and block parameters used in the simulation are also shown in Figure 4.3. Values given by the pump's designer are used in the pump block, but tabulated data could not be used since these data are given only according to the design point, not according to different shaft speeds.

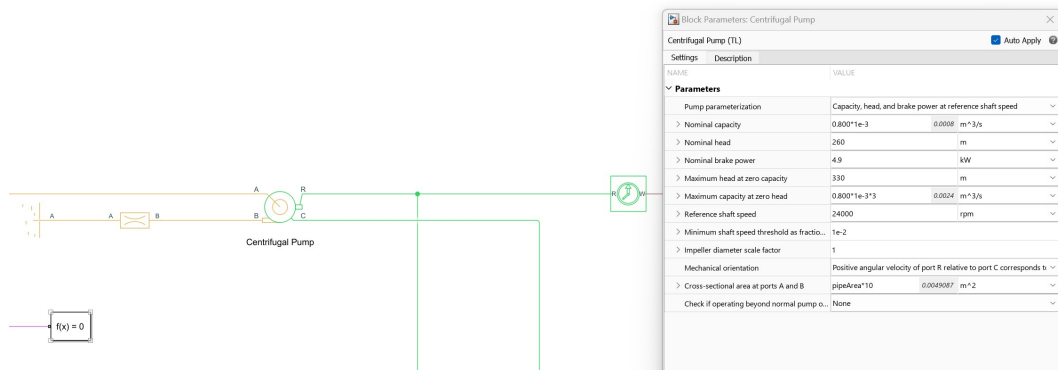


Figure 4.3. Centrifugal pump block used in MATLAB Simulink.

For both systems, port A shows the inputs of the systems, and port B shows the outputs of the systems. The rotational speed of the shaft is measured from the difference between Port C, which is connected to the mechanical rotation reference, and Port R, which is rotating depending on the power produced by the system. The rotation speed of the shaft was measured with a sensor block and then the noise in the system was modeled with a random noise generating block and added to the model.

For the proportional wastegate valve, the pintle valve block found in Simulink was used. In this block, the opening area data was used according to the increment value given to the electric motor of the valve obtained in the theoretical calculation section. The pintle valve and block parameters used in the simulation are shown in Figure 4.4.

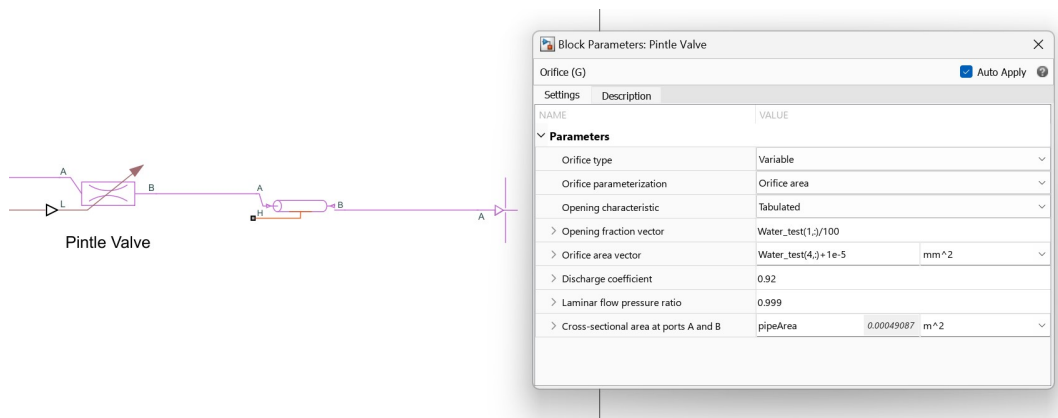


Figure 4.4. Pintle valve block used in MATLAB Simulink.

In the simulation, the turbine rotated with cold gas was modeled, because the chemical properties of the gas supplied to the turbine rotated with hot gas generator slightly change during the firing, it will be more difficult to model and the priority is to determine the PID coefficients for the controller, so the cold gas system with similar characteristics to the hot gas generator is modeled. The reason why the cold gas generator is similar to the hot gas generator system is that the pressure and temperature of the cold gas generator decrease over time, and its energy decreases, similar to the decrease in pressure and temperature of the hot gas system over time.

A high pressure nitrogen manifold is used as a cold gas generator. The properties of nitrogen gas were found using CoolProp and given to Simscape block as tabular data. Since the flow rate of the gas that must pass through the turbine to produce the energy required to keep the pump rotating at the desired RPM is approximately known and also modeled in Simscape, an attempt was made to calculate the pressure and flow rate to be given to the system by considering that some of the gas will also pass through the proportional wastegate valve. For this, the pressure at which the high pressure nitrogen manifold will be adjusted with the regulator before the test was determined, and then the orifice diameter at which the desired flow would pass was calculated using the choked flow formula given in Equation 2.3, and an orifice was placed in front of the turbine and proportional wastegate valve.

As a result of the computational fluid dynamics studies and tests conducted for the turbine, the narrowest area where the mass flow rate is determined in the turbine was calculated as 132.73 mm^2 . This value corresponds to a throat with a diameter of approximately 13 mm. In order for the turbine to rotate at the desired speed of 24000 RPM, 250 gr/s N_2 gas with a pressure of 7.5-8 bar must enter the turbine.

In order to see how the PID controller works, it was planned to pass some gas through the proportional wastegate valve at the same time as the turbine. Therefore, calculations were made so that the nitrogen flow rate passing through the proportional wastegate valve, like the turbine, would be approximately 250 gr/s.

As will be discussed in the test section later, the physical properties of the water tank in the test area were used to model how the water tank would be pressurized with another nitrogen manifold in the test area via Simulink.

The pressure and orifice diameter required to provide a total nitrogen flow rate of 500 gr/s was calculated via Simulink, with 250 gr/s passing through the turbine and 250 gr/s passing through the proportional wastegate valve. Accordingly, the nitrogen manifold with a volume of 600 liters will have a pressure of 120 bar and a mass flow rate of 500 gr/s nitrogen will pass through the system with the help of an orifice with a diameter of 5 mm.

In the upper part of Figure 4.5, the water tank that feeds the pump with water and the nitrogen manifold that pressurizes the water tank are shown. In the lower part of Figure 4.5, the nitrogen manifold that produces cold gas for the turbine that drives the pump and the proportional wastegate valve that adjusts the turbopump speed are shown.

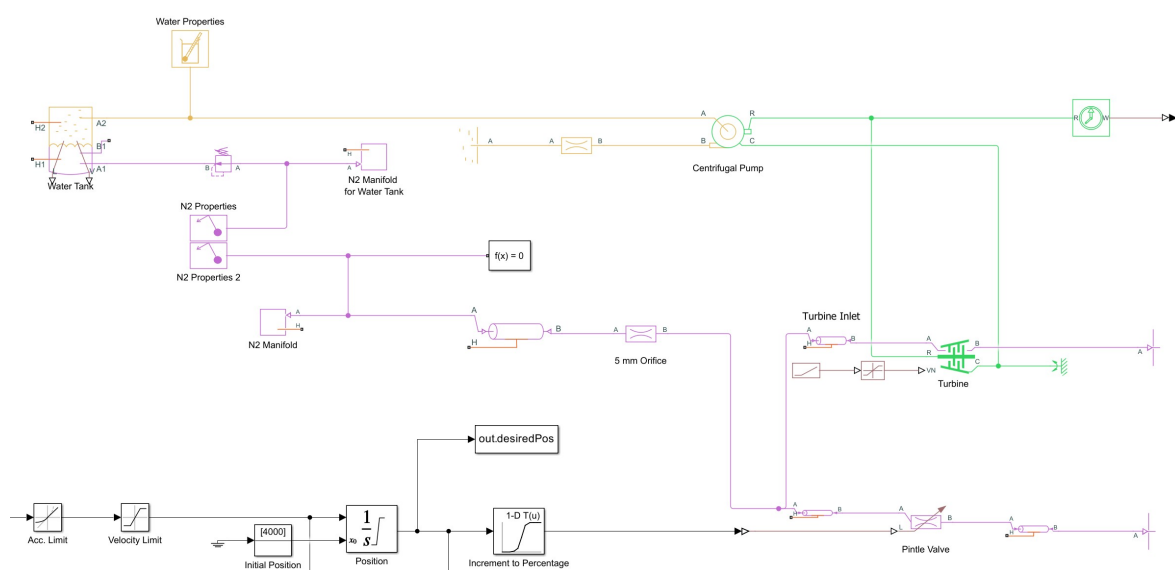


Figure 4.5. Pressurization and feeding line of turbopump system in MATLAB Simulink.

After the physical system to be used in the tests was established via Simulink, the control coefficients of the controller that would control the proportional wastegate valve and thus keep the rotation speed of the turbopump system at the desired values were found and implemented into the system. The physical system of the turbopump system to be fed with the cold gas generator was established via MATLAB Simulink and the design of the controller that will move the proportional wastegate valve that will ensure the turbine rotates at the desired RPM was started. According to Equation 4.1, a PID controller containing proportional, integrator and derivative terms was designed via Simulink. The PID controller designed in Simulink is shown in Figure 4.6.

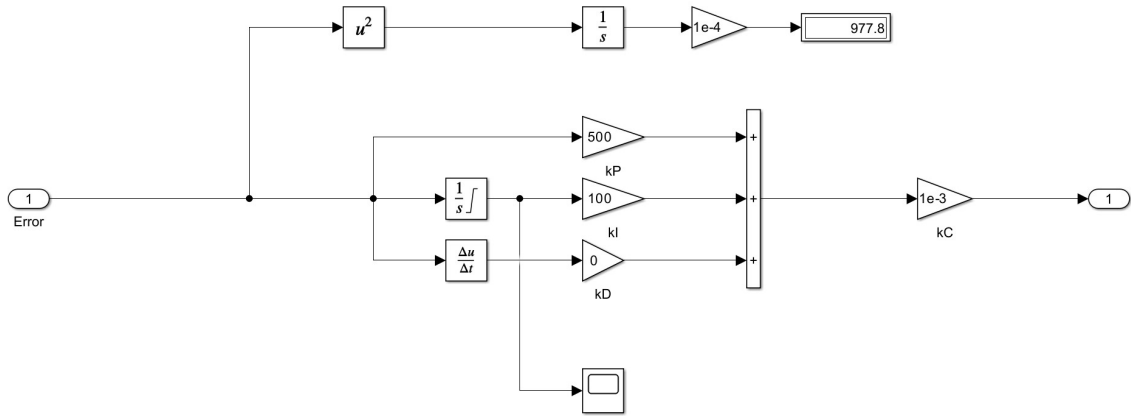


Figure 4.6. PID controller used in MATLAB Simulink.

In Figure 4.6, the error term represents the difference between the desired process value of 24000 RPM from the controller and the value obtained from the rotational motion sensor in the turbopump system. Although the derivative term was also present in the first trials, the coefficient of the derivative term was set to 0 due to the errors caused by this term in the tests performed. A saturation limit was added to the integrator term to prevent error accumulation, and this limit is 1500. The outputs produced by the controller using the control terms are then multiplied by the controller gain, K_c , and used as the control variable to move the electric motor of the proportional wastegate valve.

In addition, in the upper part of Figure 4.6, the total error amount in the system is examined in the simulation to examine the behavior of the controller with different control terms. In this way, many different options were examined quickly with the help of simulation, away from the trouble of doing real tests.

The PID controller block in MATLAB Simulink was also used as the first step in finding the coefficients of the control terms. This block contains various control parameters and also allows the control terms to be tuned via Simulink. The parameters of the PID controller block are shown in Figure 4.7. In real-life tests and simulations, the controller shown in Figure 4.6 was used.

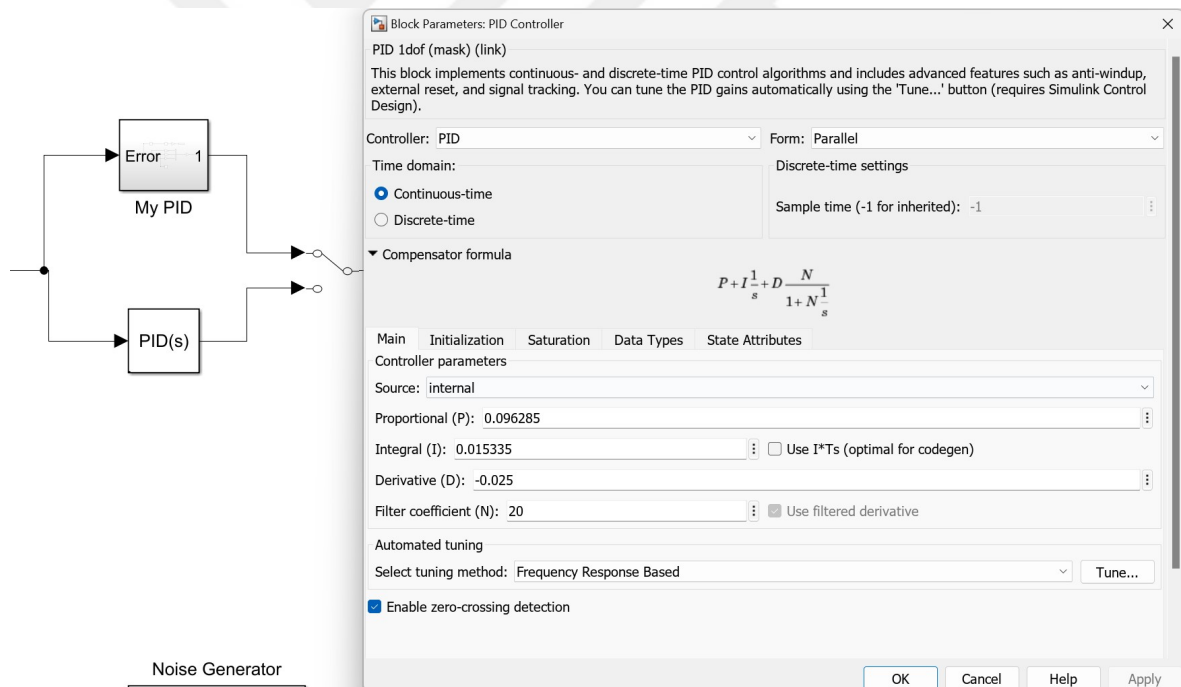


Figure 4.7. PID controller block in MATLAB Simulink.

In Figure 4.8, it is shown that the controller output reaches the proportional wastegate valve and then completes the control loop and returns to the controller. The speed and acceleration limits of the electric motor used in the valve were taken from the datasheet and technical specifications and used in the simulation.

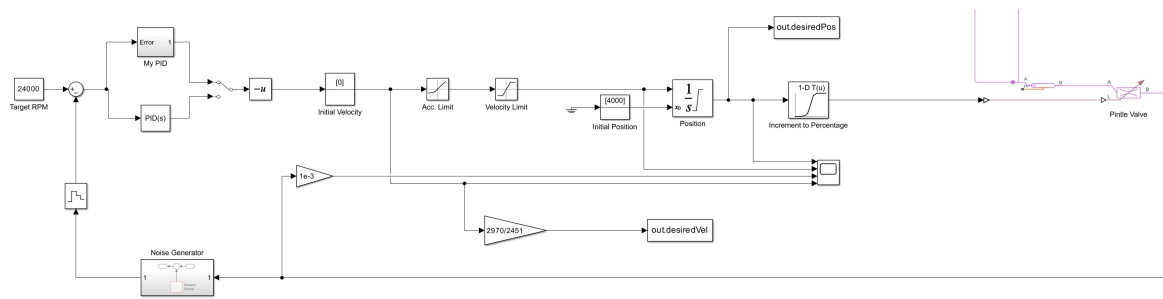


Figure 4.8. PID controller and proportional wastegate used in MATLAB Simulink.

In the tests and simulations, the valve was started at 4000 increments, which corresponds to 80% open rate. The reason for this is to prevent the turbopump system from rotating at high speed in case of an error in the controller in real-life tests and to prevent damage to the system.

When designing the controller for the system, two parameters of the electric motor were tested as control variables. In the first of these, the controller gave the electric motor the position it should go as output, and in the second, it gave the speed value it should return to instead of the position. Although both variables gave the same outputs as a result, the reason for this change was that the control loops within the electric motor were different against position and speed inputs. In the model shown in Figure 4.8, the controlled variable is the position of the valve. This mode was also used in real-life tests.

In Figure 4.9, a model where the controlling variable is speed is shown. In this model, the PID controller is created in a similar way and the coefficients of the control terms are different. It has been observed in technical documents that as a result of the controller giving the speed as output, the calculation cycle of the electric motor will decrease by one step and the response time will be shortened. Although the tests on the simulation environment were completed, since this mode could not be used in real-life tests, the difference with the other mode could not be observed.

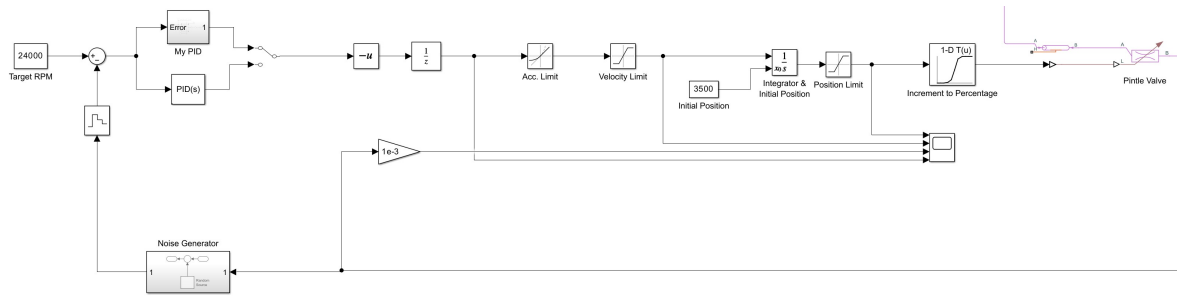


Figure 4.9. PID controller with speed as controlling variable in MATLAB Simulink.

In Simulink simulations, the noise generator shown in Figure 4.8 and Figure 4.9 was used to simulate the sensor noises that would occur in the tests.

4.3. Simulation Results

After determining the control coefficients of the PID controller and integrating the test environment into the simulation, the simulation results were examined. Simulations were made for 30 seconds, and MATLAB Simulink's own variable time step solver was used.

Before looking at how the PID controller works, it is necessary to examine the behavior of the cold gas generator during the test. The nitrogen manifold used as the cold gas generator will experience a pressure and temperature drop over time after the flow starts. In Figure 4.10, the pressure and temperature of the nitrogen manifold used are shown. Accordingly, the initial pressure of the system is 120 bar and the temperature is 300 Kelvin, and later the pressure of the system drops to 90 bar and the temperature drops to 275 Kelvin.

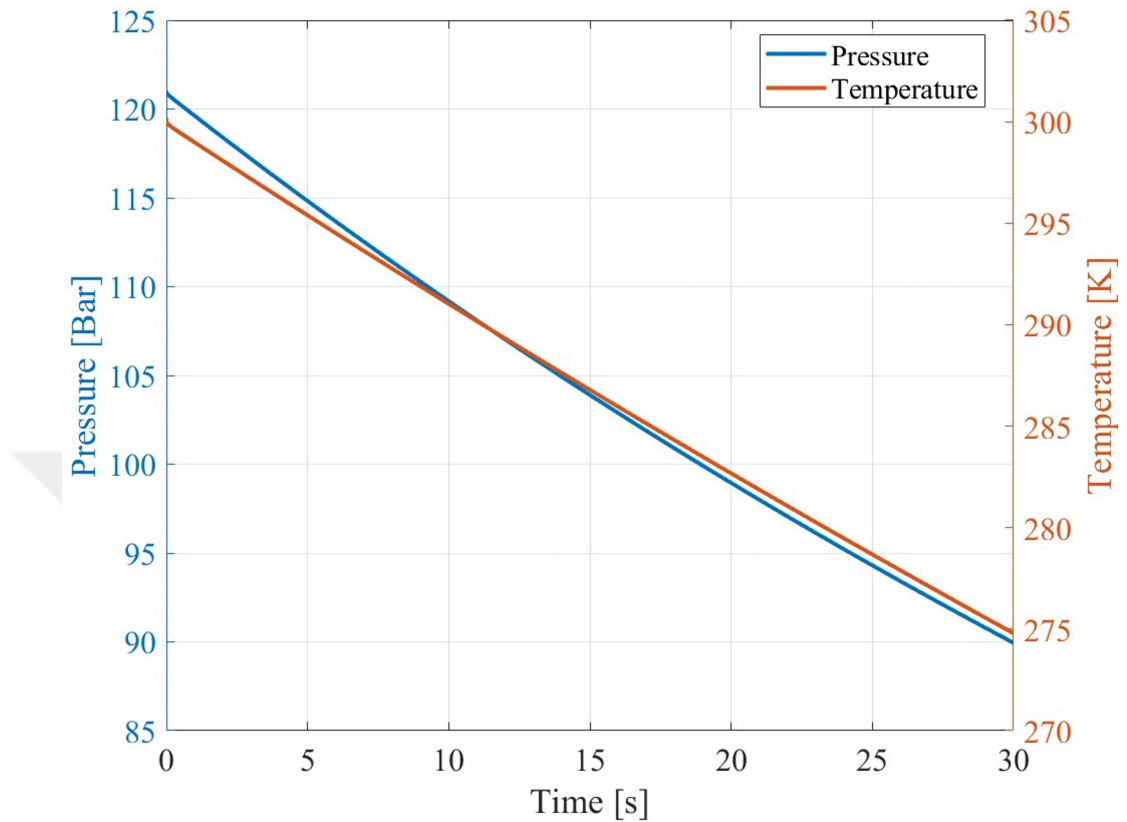


Figure 4.10. Pressure and temperature of N_2 manifold from simulation.

Despite the significant pressure drop in the nitrogen manifold, since the flow is choked by the orifice placed before the turbine and proportional wastegate valve, no change is observed in the pressure of the gas reaching the turbine and proportional wastegate valve. In Figure 4.11, the pressure and temperature of the nitrogen at the orifice outlet in the system are given. Accordingly, the nitrogen pressure at the orifice outlet is approximately 7.50 bar throughout the simulation, but its temperature decreases from 276 Kelvin to 255 Kelvin over time. This causes the energy given by the cold gas generator to decrease over time, and it is expected that this decrease will be corrected by changing the opening ratio of the proportional wastegate valve controlled by the PID controller so that the rotation speed of the turbopump does not change.

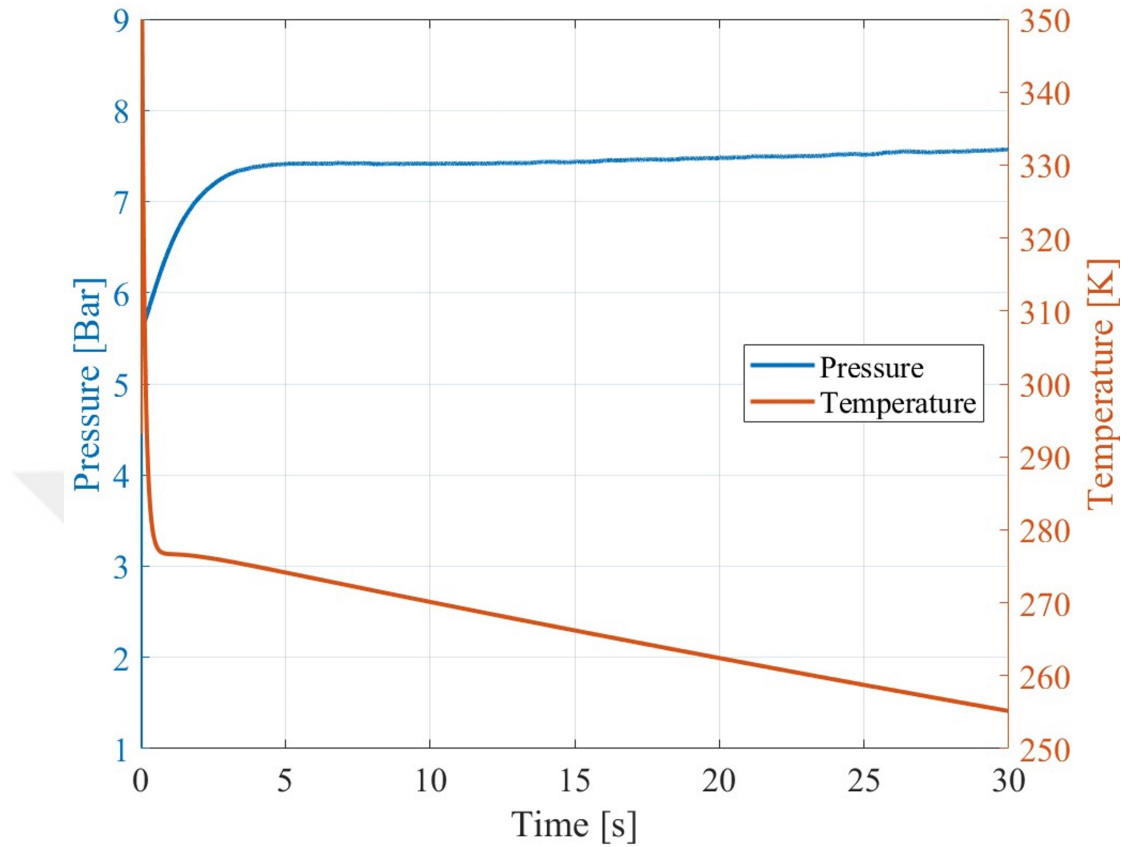


Figure 4.11. Pressure and temperature at the outlet of orifice from simulation.

Once it was known how the cold gas generator would behave during the simulation, the response of the rest of the system and the controller to this situation was examined. Figure 4.12 shows the RPM value obtained as a result of the simulation and the change in position of the proportional wastegate valve during this time. According to the simulation, the system reaches the desired RPM value of 24000 in approximately 2.5 seconds. Since the energy in the cold gas generator decreases over time during the simulation, the proportional wastegate valve reduces the opening rate to keep the RPM value of the turbopump system at the target, thus reducing the energy it exhausts.

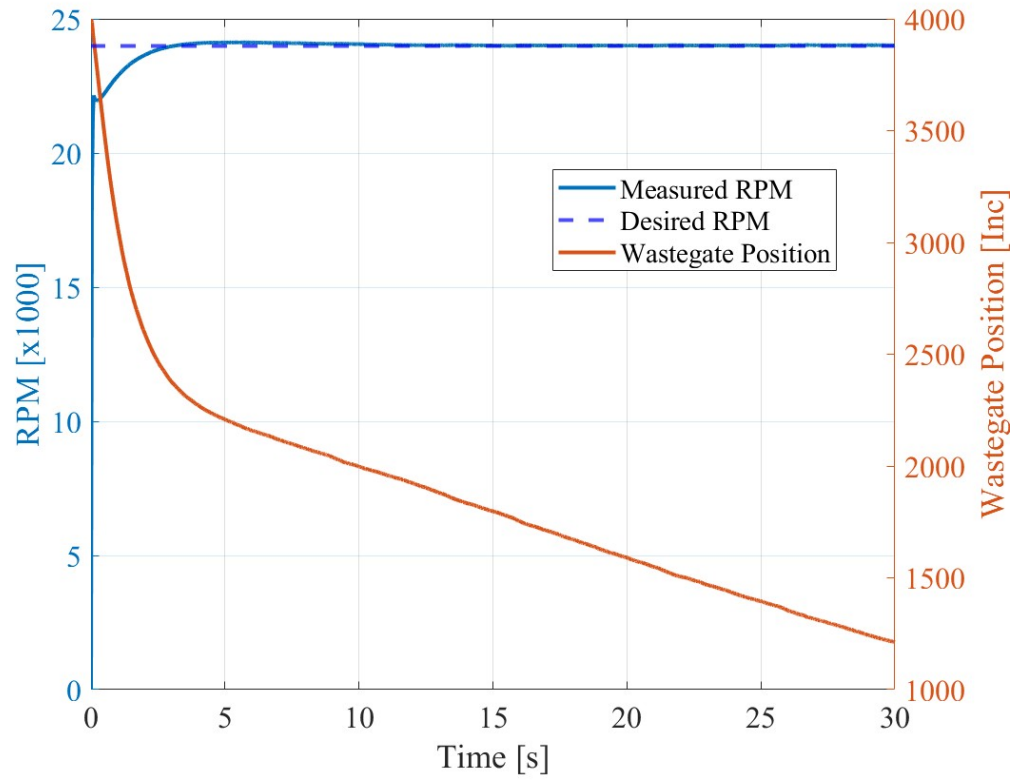


Figure 4.12. RPM and position of proportional wastegate from simulation.

In Figure 4.13, the mass flow rate in the turbine, the wastegate valve and the system are given respectively. According to this graph, the cold gas generator initially has a nitrogen mass flow rate of 500 g/s, but it decreases over time throughout the simulation due to the decrease in manifold pressure, and finally gives a nitrogen mass flow rate of 395 g/s. It is seen that the nitrogen mass flow rate passing through the turbine is initially 230 g/s for the turbopump to have the planned rotation speed, and then it becomes 250 g/s to compensate for the energy decrease due to the cooling of the nitrogen. The proportional wastegate valve starts with a nitrogen mass flow rate of approximately 270 g/s and then decreases to 145 g/s.

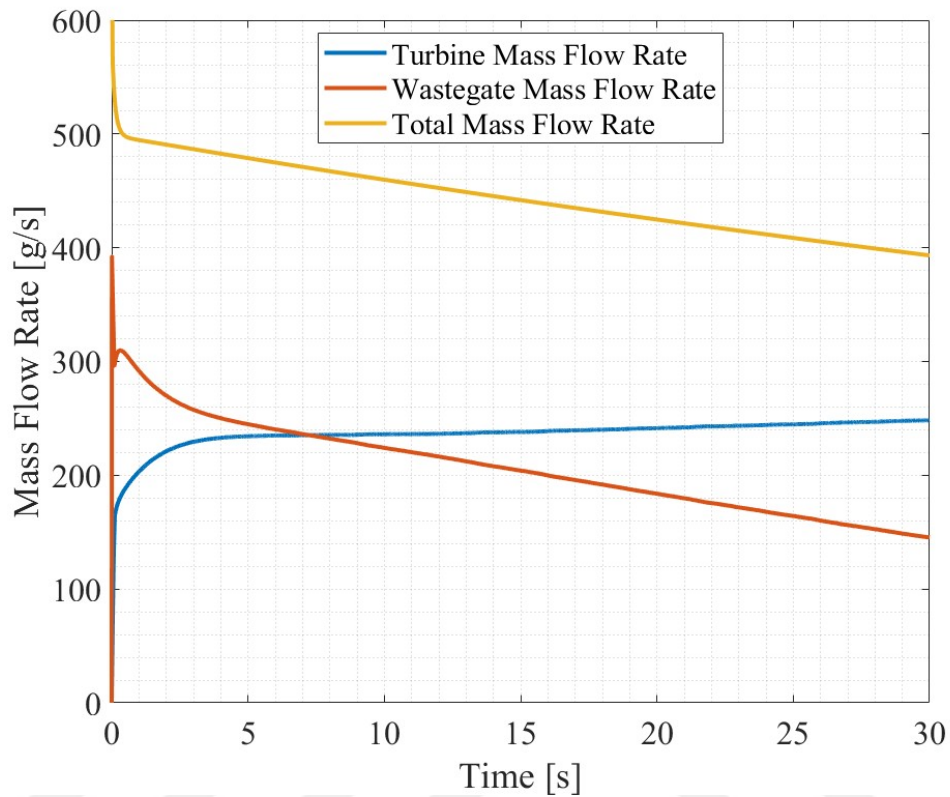


Figure 4.13. Mass flow rate in the system from simulation.

In order to model the sensor noises, present in the tests, the error term reaching the controller is distorted by the noise generator. The effect of this situation on the output of the PID controller is shown in Figure 4.14. Although the controller outputs are noisy, their effect on the rotation of the turbopump system at the desired rotation speed is insignificant.

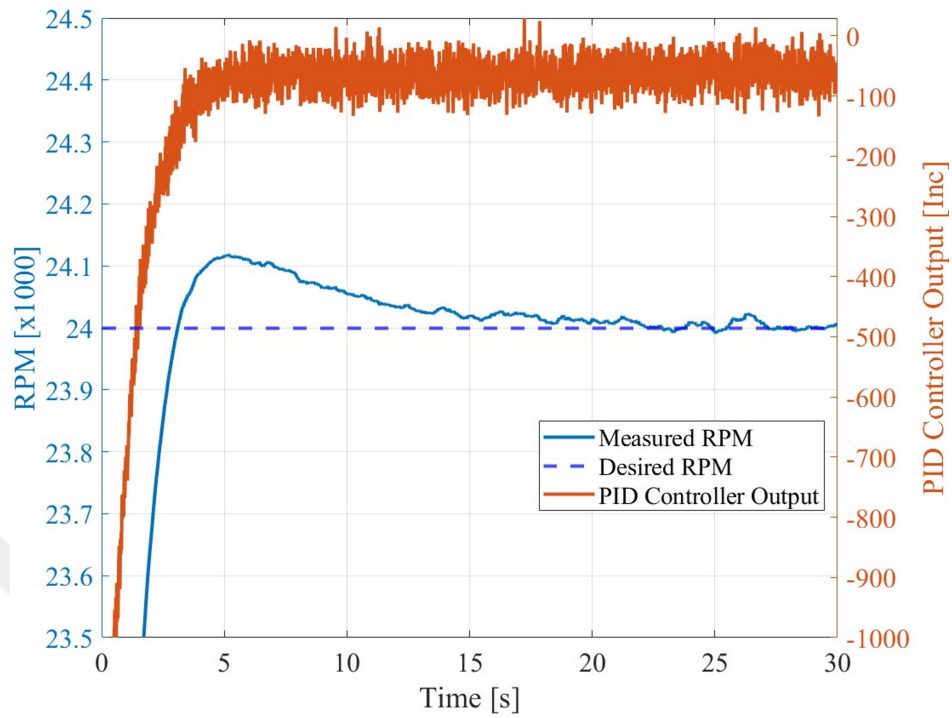


Figure 4.14. RPM and PID controller output from simulation.

When the turbopump system has the desired rotation speed, the output pressure and mass flow rate of the pump, which are one of the most important outputs that the turbine produces the necessary power and feeds the pump as desired, are shown in Figure 4.15. The inlet pressure of the pump is 10 bar and the outlet pressure of the pump is approximately 33.4 bar, which is stable throughout the simulation, and the mass flow rate of the water passing through the pump at this time is approximately 930 g/s.

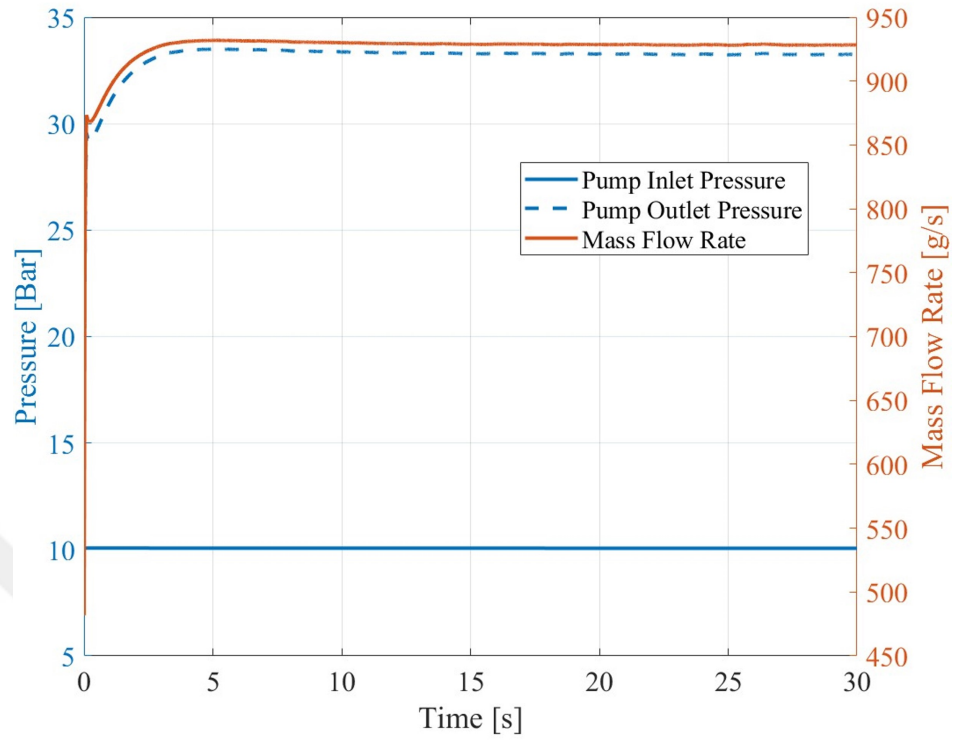


Figure 4.15. Pump inlet and outlet pressure and mass flow rate from simulation.

The full version of the model prepared in MATLAB Simulink, which is used to determine the control coefficients in the PID controller, to examine the response of the controller by simulating the test environment and to save time and resources in the tests to be performed, is given in Figure 4.16

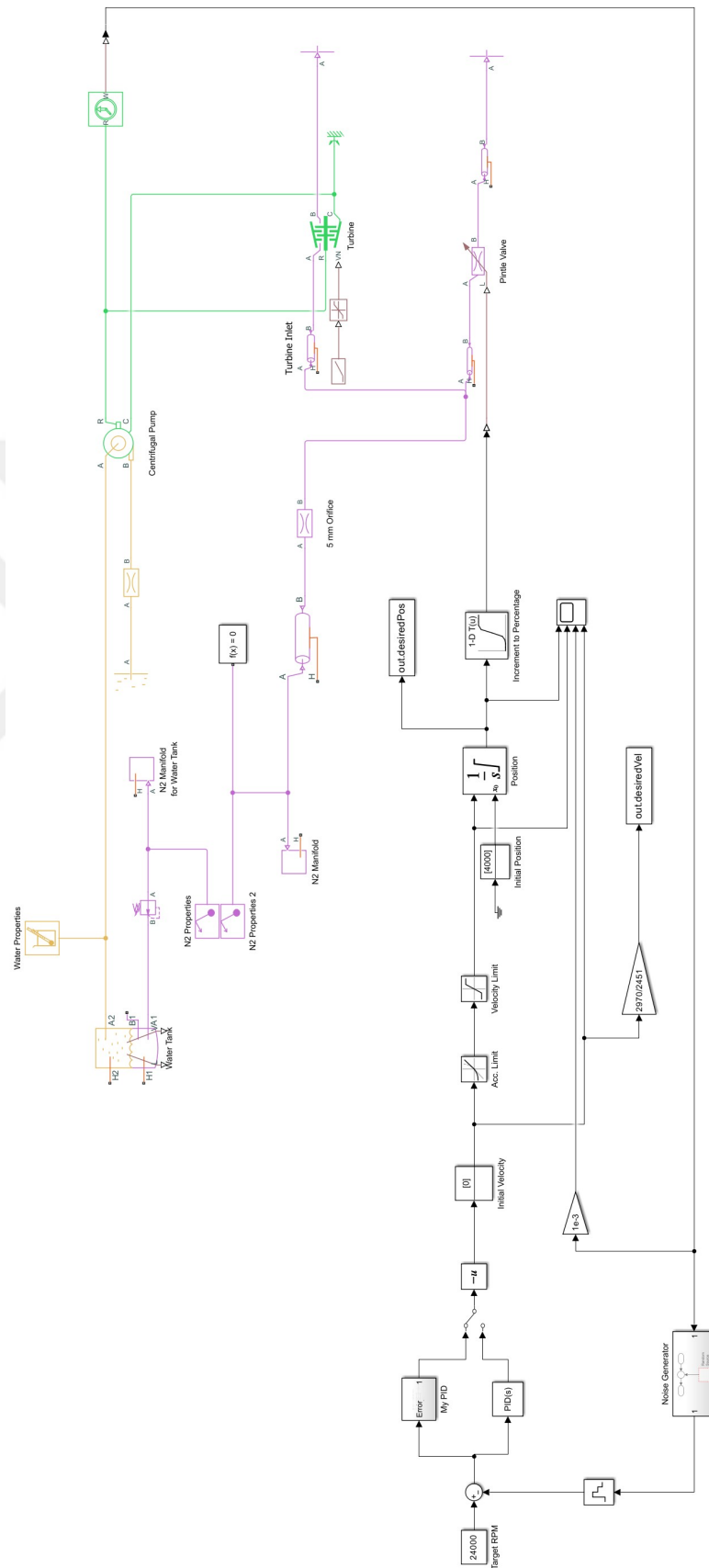


Figure 4.16. MATLAB Simulink model used for PID controller design.

5. INTEGRATED SYSTEM TESTS AND RESULTS

After completing the development processes and testing of the subsystems, integrated tests involving the proportional wastegate, RPM sensor, and PID controller were initiated. In these tests, the system was initially tested with the hot gas generator, followed by testing with the cold gas generator. The test results were carefully analyzed, with attention given to the problems encountered and their underlying causes. Additionally, the actions implemented to address these issues were clearly explained.

5.1. Test Setup with Hot Gas Generator

During the preparation for the integrated tests, the assembly of the turbopump system was completed, making the system ready for testing. Subsequently, the hybrid rocket engine, designated as the hot gas generator, was assembled. Finally, the functionality of the proportional wastegate and avionic systems was verified.

Platforms were constructed at the GÖRTEM test site to enable proper assembly of the systems for testing. The pump in the turbopump system was connected to the water tank in GÖRTEM with a flexible pipe. The outlet of the hot gas generator was connected to the inlet of the turbine and the inlet of the proportional wastegate with a T-shaped connection piece. In this way, the amount of energy going to the turbine would be adjusted with a proportional wastegate when necessary. In order to better examine and understand the test results, pressure and temperature sensors were placed at the points where data needed to be taken. The sensors and DAQ system used in the integrated tests are identical to those employed in the characterization tests of the proportional wastegate. Figure 5.1 presents the P&ID diagram prepared for the integrated test with the hot gas generator.

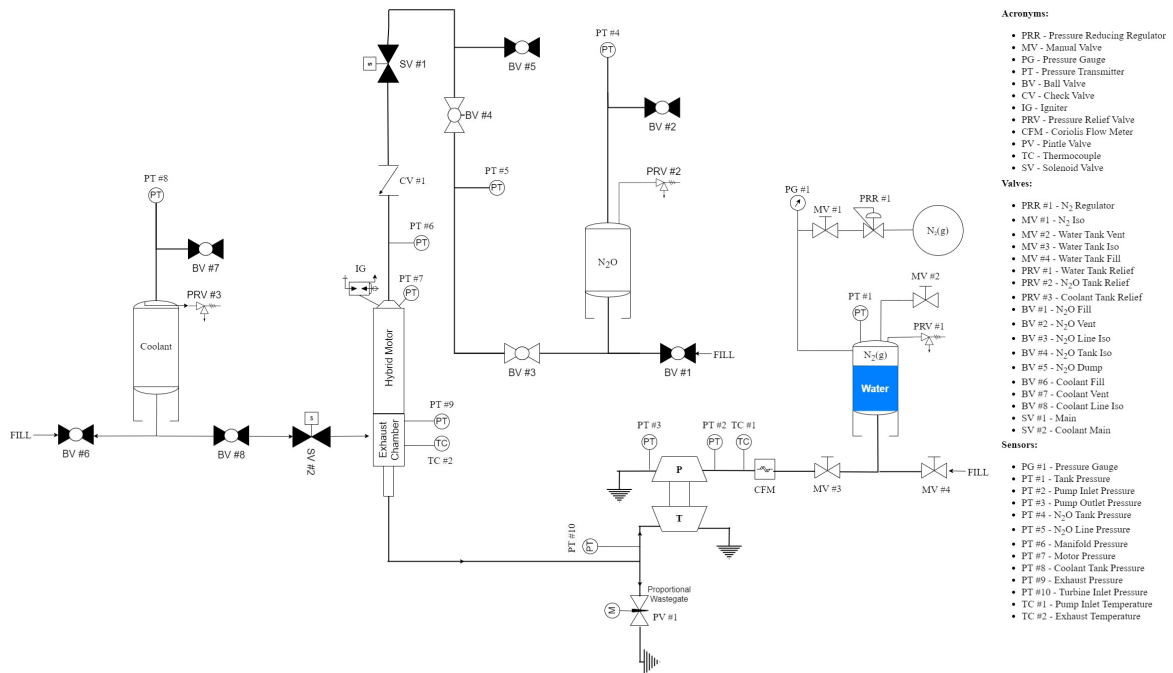


Figure 5.1. P&ID schematic of the test setup used for integrated test with hot gas generator.

Before the test, RPM sensor was connected close to the coupling on the turbopump system and the shaft of the turbopump was rotated by hand to see if the RPM sensor was able to take data. After the final checks of the avionics systems and DAQ system were made before the firing, the people were sent to the control room as a safety precaution. Finally, one person adjusted the pressure of the water tank with the help of the regulator and opened the manual isolation valve of the water tank and water was sent to the pump at the previously determined pressure. When everything was determined to be nominal, the firing sequence was started and the test started. Figure 5.2 shows a picture of the integrated test system.

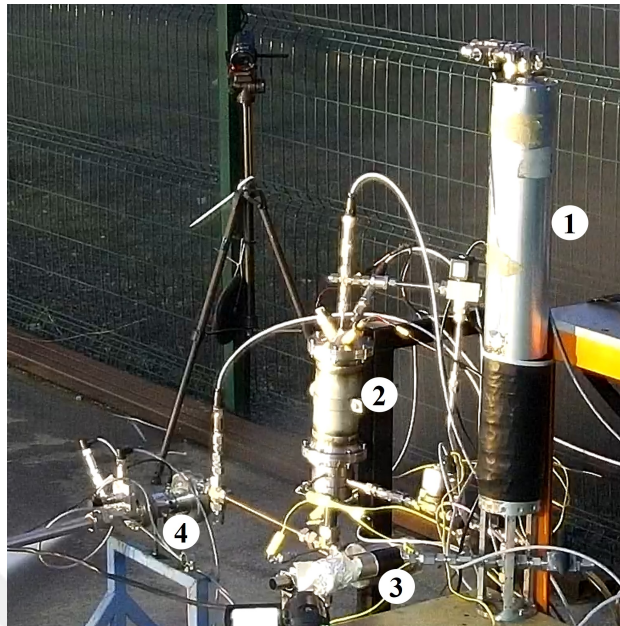


Figure 5.2. Integrated test setup.

In Figure 5.2, number 1 marks the oxidizer tank used in the hybrid rocket motor firing, number 2 marks the hybrid rocket motor, number 3 marks the proportional wastegate valve, and number 4 marks the turbopump system. The sub-components of the turbopump system are shown in Figure 5.3. In Figure 5.3, *a* marks the turbine, *b* marks the pump, and *c* marks the RPM sensor that measures the rotational speed of the turbopump.

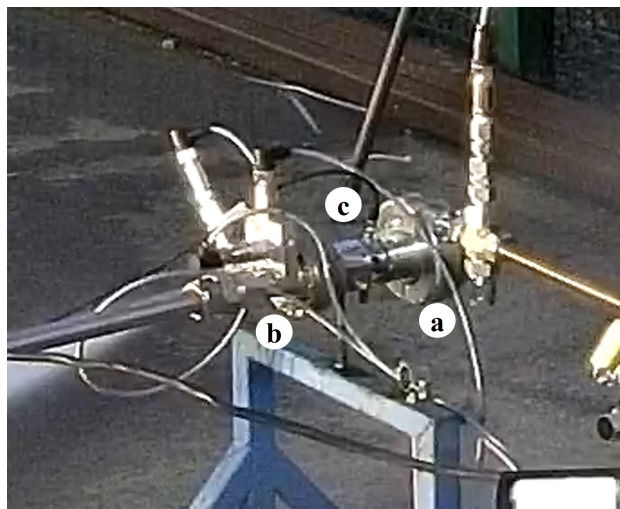


Figure 5.3. Turbopump system used in integrated test.

5.2. Test Results with Hot Gas Generator

Although the duration of the first integrated test was planned to be 60 seconds, the firing was stopped at about 30 seconds of the test because large amplitude fluctuations were observed in the RPM of the turbopump system. Although the hot gas generator was stable and working as planned the test was stopped to avoid any damage to the system. In Figure 5.4, pump inlet and outlet pressures, turbine inlet pressure and hybrid rocket motor chamber pressure are shown on the left axis, and the measured RPM value of the turbopump system is shown on the right axis. The data in Figure 5.4 are shown by taking their moving averages to be more understandable, so the noise in the data is eliminated.

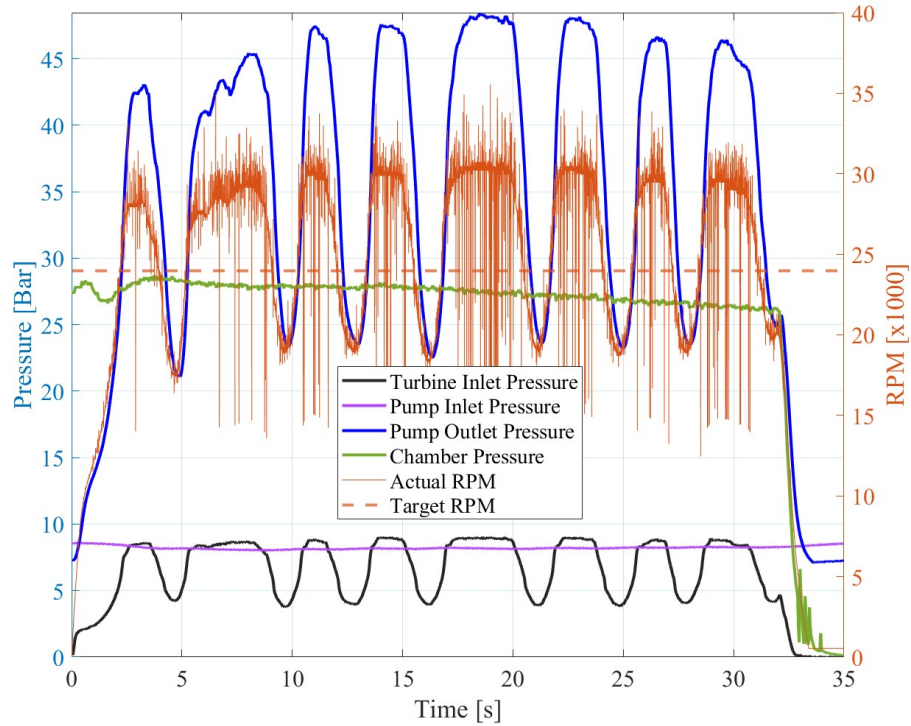


Figure 5.4. Pressures and measured RPM obtained from integrated test 1.

When Figure 5.4 is examined, it is seen that the proportional wastegate works in the closing direction and the turbine inlet pressure increases, since the RPM value was

lower than 24000 RPM, which is the target of the PID controller, at the beginning of the test. As the inlet pressure of the turbine increases, the energy entering the turbine increases and therefore the RPM value of the system has exceeded the target value. Meanwhile, it was observed that the pump outlet pressure increased in parallel with the increase in RPM. Then, the PID controller operated the valve in the opening direction, reducing the turbine inlet pressure and lowering the RPM value from 30000 to 18000. Since the RPM value remained below the target of the PID controller, the controller operated the proportional wastegate in the closing direction once again, and this time the system worked above the target value. Afterwards, wastegate opening and closing events started to repeat and the test was stopped because there was no improvement in the system over time.

After the test, when the collected data were examined, a few errors and omissions were noticed, and they were started to be corrected. First of all, the deficiencies are not caused by the mechanical systems, but by the avionics systems and the PID controller, so no change has been made in the mechanical system.

Due to two main problems during testing, the turbopump system could not be kept at the planned RPM. The first is that the derivative term D in the PID controller dominates the output of the controller during rapid RPM changes. The second problem is that although the avionic systems can communicate at high frequency, approximately at 1200 Hz, for RPM measurement, they cannot send commands as fast to the motor controlling the proportional wastegate. When the test data were examined, it was seen that the position of the proportional wastegate was sent to the avionics systems only at 3 Hz, and therefore it was seen that the PID controller was incompletely informed about the current position of the wastegate and therefore gave incorrect outputs to the electric motor driver controlling the valve.

Figure 5.5 shows the RPM measurements reaching the PID controller on the left axis and the position information that the proportional wastegate sends to the PID controller during the test on the right axis. As can be seen from Figure 5.5, the

proportional wastegate sends its position slowly and it closes and opens periodically because the PID outputs are incorrect due to the derivative term. When the RPM measurement was examined, it was observed that there was more noise than the desktop tests. Efforts have been made to prevent these noises from affecting the PID output.

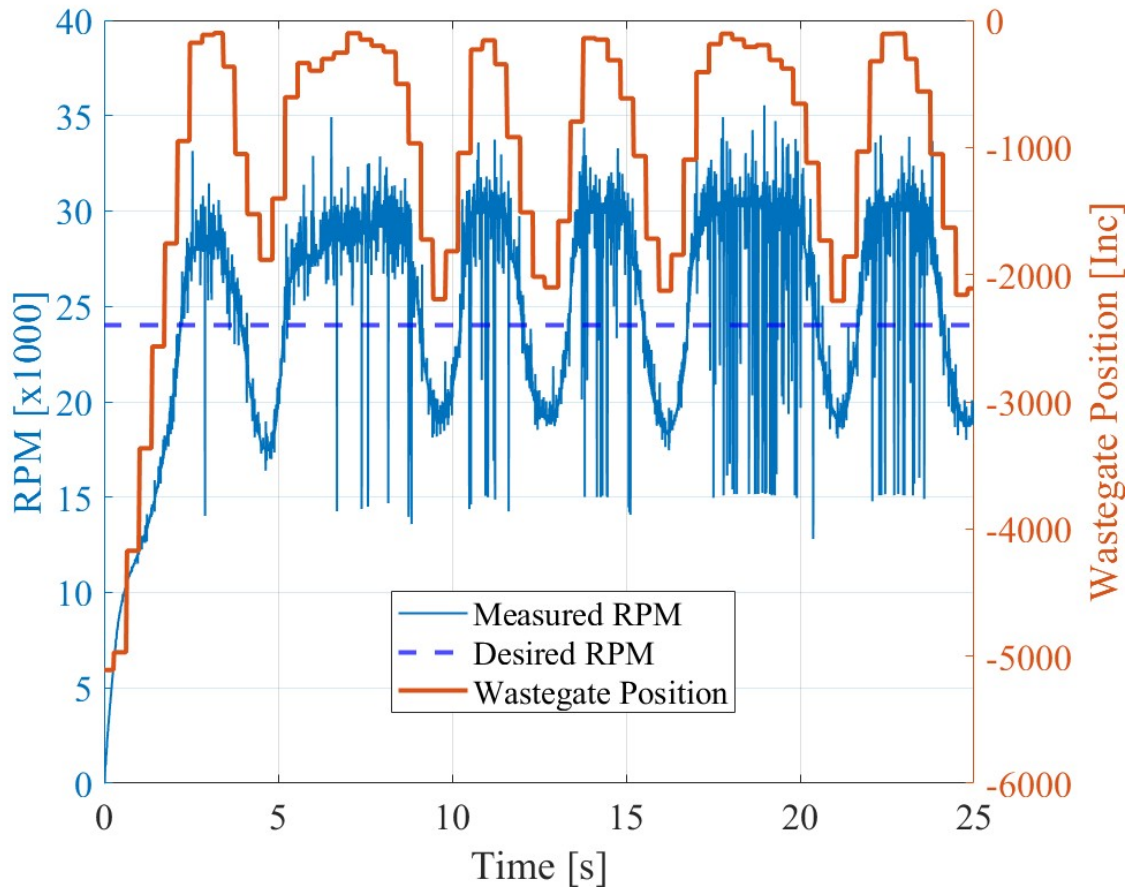


Figure 5.5. RPM and wastegate position results from integrated test 1.

In order to solve the first problem, the simulation was updated in the MATLAB Simulink, and the derivative term was removed from the PID controller, and the control was made only with PI. In order to solve the second problem, the speed of communication with the electric motor driver was tried to be increased. Here, the communication protocol has been changed to a protocol that provides faster information flow. While the RPM sensor was not used for measurement during desktop tests, successful communication was achieved with the driver of the electric motor up to 250 Hz. However,

since it was observed that the controller used in avionic systems could not transmit data fast enough when the RPM sensor was used for measurement, communication with the electric motor began at 30 Hz, where the avionics system could both measure RPM using the proximity sensor and successfully communicate with the electric motor driver.

In addition, filters have been used to reduce the noise in RPM measurement. For the filter used in the RPM measurement, the RPM measurement data taken at high frequency was averaged to be equal to the communication frequency with the electric motor driver. In order to avoid instantaneous fluctuations, the previous measurement was compared with the current measurement, and if the current measurement is less than 80 percent or greater than 120 percent of the previous one, the current measurement is not included in the average because such a change in RPM is mechanically not possible at high speeds. If there is such a change due to a mechanical problem, it will be evident with the next measurement, because since the measurements are made at high frequency, a change that is not caused by noise will be immediately noticeable. The filter used in RPM measurement is shown in Figure 5.6.

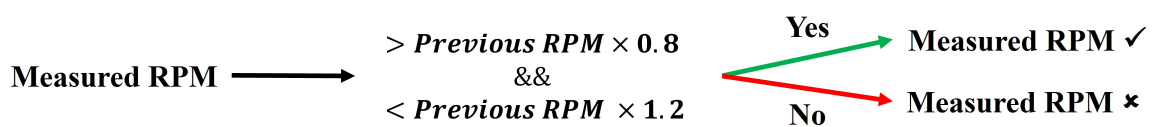


Figure 5.6. RPM measurement verification logic.

Once the issues with the avionics systems are resolved, a new test is planned to see how the system will work with the new changes. Since testing with a hot gas generator is more complex, it is planned to provide energy to the turbine by using nitrogen gas as cold gas.

5.3. Test Setup with Cold Gas Generator

The high-pressure nitrogen intended for use as a cold gas generator is sourced from the nitrogen manifold at the GÖRTEM test center. This manifold consists of 12 interconnected nitrogen cylinders, each with a pressure of 240 bar. The pressure of the nitrogen manifold is adjusted using a regulator. An orifice was placed before the line to be connected to the turbine and proportional wastegate and the flow was choked. In this way, the mass flow rate that will enter the system is made independent from the outlet of the orifice, making the system more stable. The diameter of the orifice depends on the pressure at which the regulator connected to the nitrogen manifold is set and the mass flow rate to be passed through the system. Although approximately 500 grams of nitrogen is consumed per second during the test, there will be a slight drop in manifold pressure throughout the test and therefore the nitrogen flow rate will decrease over time. Since there is a similar situation in the hot gas generator, the tests using cold gas are also similar to the real situation. The decrease in the mass flow rate will be solved by closing the proportional wastegate and keeping the energy entering the turbine constant. Figure 5.7 presents the P&ID diagram prepared for the integrated test with the cold gas generator.

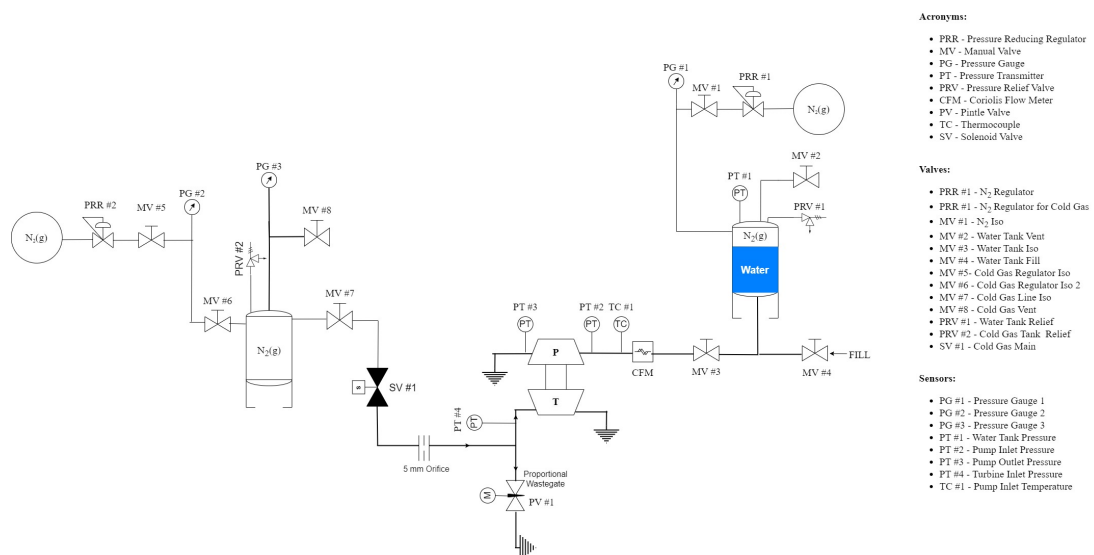


Figure 5.7. P&ID schematic of the test setup used for integrated test with cold gas generator.

The test of the turbopump system with cold gas is shown in Figure 5.8. During the test, the regulator was set to 130 bar and the orifice was chosen as 5 mm to provide the desired mass flow at this pressure. Since there was no mechanical change in the systems, the system was ready for testing using old fixtures.

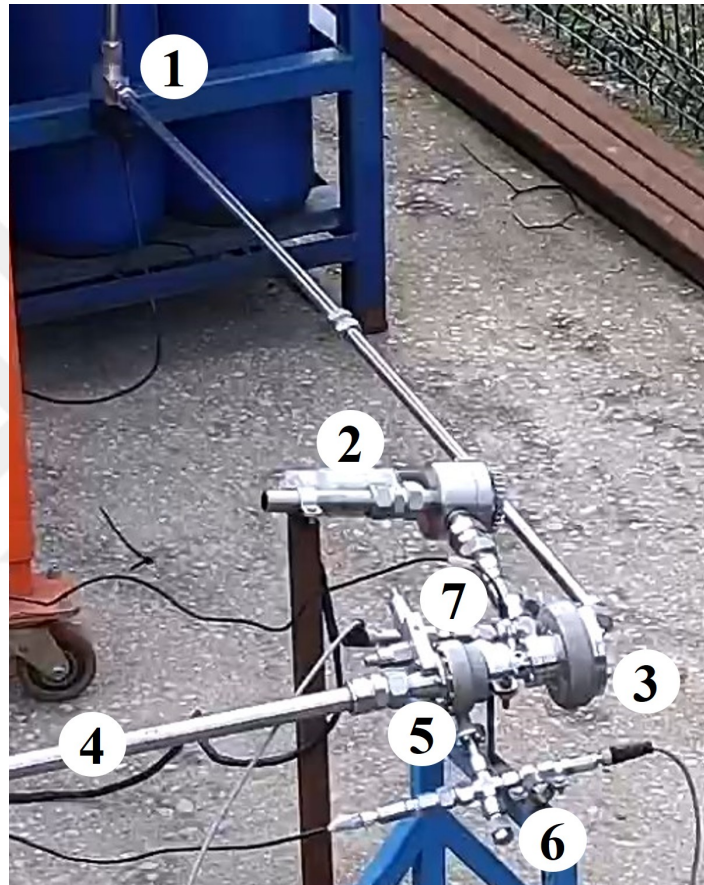


Figure 5.8. Integrated test with cold gas.

In Figure 5.8, number 1 marks the solenoid isolation valve of the cold gas generator, number 2 marks the proportional wastegate, number 3 marks the turbine, number 4 marks the pump inlet (coming from the water tank), number 5 marks the pump, number 6 marks the pump outlet, and number 7 marks the RPM sensor that measures the rotation speed of the turbopump.

5.4. Test Results with Cold Gas Generator

Although the duration of the second integrated test was planned as 60 seconds, the test was stopped by closing the solenoid valve, which provides the gas flow to the system, after about 30 seconds due to the anomalies during the test. Figure 5.9 shows the measured RPM on the left and the position of the proportional wastegate on the right. As can be seen from Figure 5.9, just before the test, the position of the proportional wastegate has reached the 100 increment value, which is the limit position in the closing direction, from its determined position of 4000 increment. Afterwards the sensor measured the rotational speed of the shaft at more than 30000 RPM for a while and immediately dropped to less than 20000 RPM and the PI controller therefore continued to command the wastegate to close the orifice area to increase the energy entering the turbine. After the turbopump worked like this for a while, when the RPM value started to be measured as 35000 RPM, the controller managed to reduce the rotation speed of the system to the targeted 24000 RPM by commanding the proportional wastegate to open quickly. After the system worked stably for a while, the test was stopped because a fluctuation started in the RPM measurements.

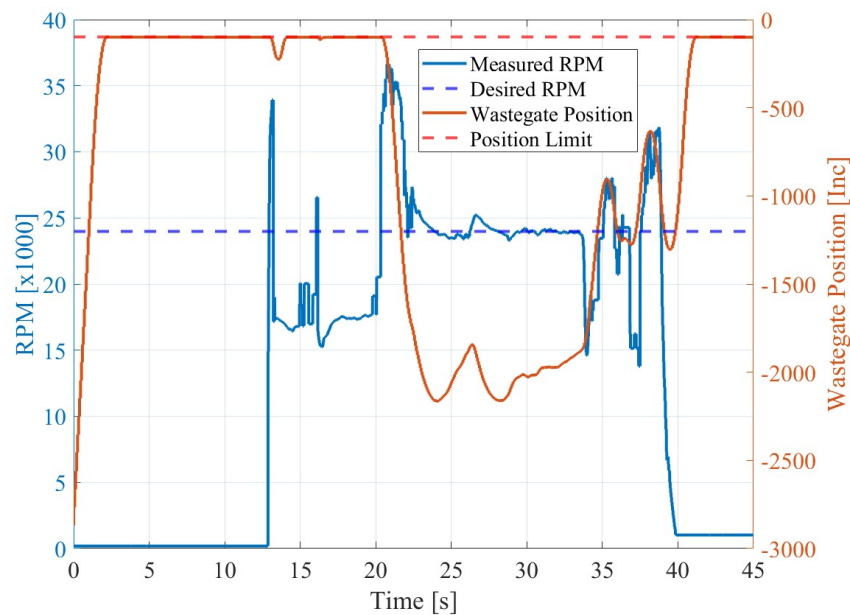


Figure 5.9. RPM and wastegate position results from integrated test 2.

The pressure data collected by the DAQ system is shown in Figure 5.10. When Figure 5.10 is examined, it is seen that the turbine inlet pressure is above the expected value due to the proportional wastegate being closed since the beginning of the test, and the outlet pressure of the pump is very high due to this excess energy given to the turbine. Pressures in Figure 5.10 was analyzed, and it was seen that 35000 RPM measured by the sensor at the beginning of the test was correct and then 1200 Hz, which is the maximum frequency at which the avionic systems could measure, was exceeded, so the avionics systems could not catch one of the teeth on the shaft and could not measure the RPM. When the avionics systems miss one of the teeth, the RPM value is halved according to Equation 3.3 used for the RPM calculation. For this reason, the PI controller misjudges the situation it is in during the test and directs the proportional wastegate in the wrong direction.

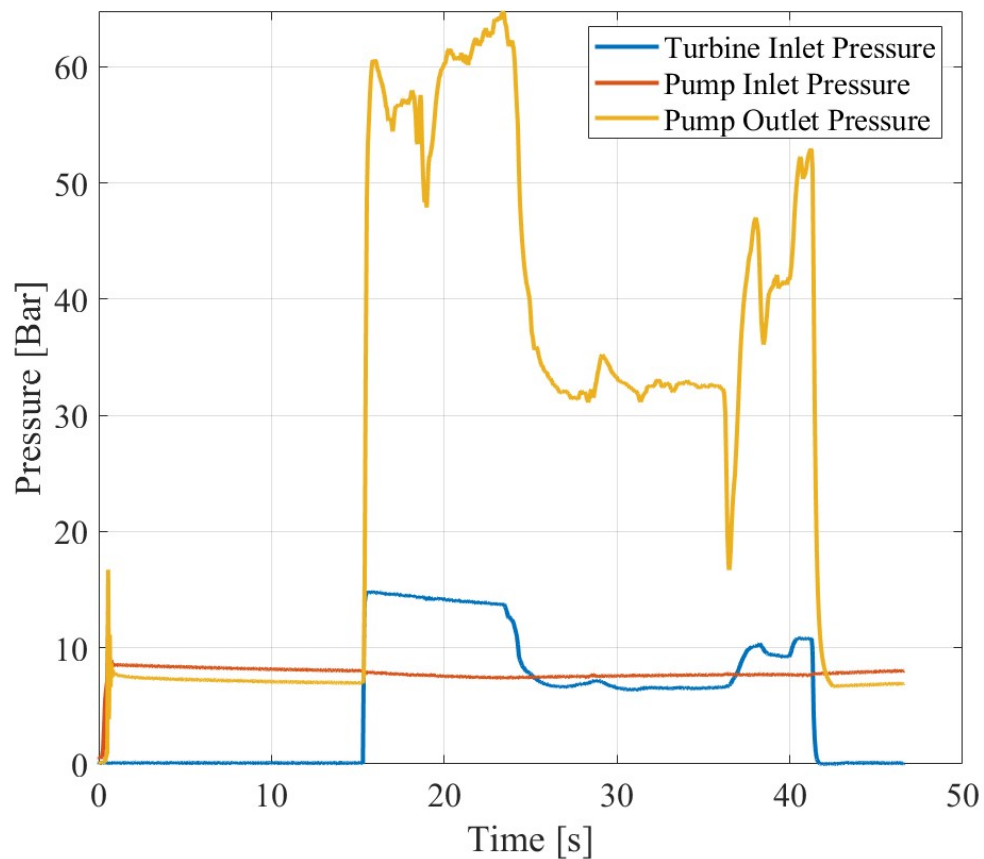


Figure 5.10. Turbine and pump pressures versus time from integrated test 2.

The reason for the first problem here is that the shaft moves momentarily while pressurized water is supplied to the pump before the test and therefore the controller starts to work. If the predetermined RPM measurement for the PI controller to start, 2000 RPM, reaches the controller due to a noise or an error before the turbopump system starts to work, the proportional wastegate will operate in the closing direction as a result of its action according to this data before the test. Therefore, when the test starts, the turbopump system will rotate at a higher than expected speed, as a large amount of energy from the cold gas generator enters the turbine. In order to solve this situation, for the PI controller to start, measurements must be received from the RPM sensor for a certain number of times, such as 5 times, above 2000 RPM. Considering that the avionics systems operate at 30 Hz, after the final arrangement, the turbopump system must rotate for approximately 160 milliseconds for the controller to start, and this time is sufficient to prevent the PI controller from working before the test.

Secondly, avionics systems started to receive pressure sensor data from the pump outlet and a limit value was set here. In this way, if the turbopump system starts to rotate at a higher frequency than the avionics systems can measure, and the measured pump outlet pressure is above the predetermined pump outlet pressure, the PI controller will act according to the pump outlet pressure instead of the RPM value it measures, since the pressure sensor is more robust. The PI controller will operate the proportional wastegate in the opening direction, the energy entering the turbine will be reduced and the RPM will be expected to decrease. In addition, the position limit of the proportional wastegate, which was 100 increments in the past, was increased to 1000 in order to prevent damage to the turbopump system in the next tests, thus preventing too much energy being supplied to the turbine due to any error. Because during this test, the bearings on the pump side became inoperable due to the high rotational speed.

The changes made to improve the controller system regarding the errors seen from the previous test are briefly explained in Figure 5.11.

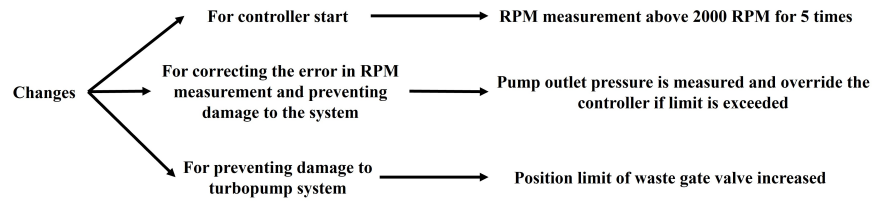


Figure 5.11. Changes made in the system for improvements.

After the turbopump system was examined and assembled after the second integrated test, a new test was conducted to see the effect of the last changes. Test parameters such as orifice diameter and tank pressure were left the same as the previous test. The test time was set at 30 seconds to run two tests without exhausting the gas in the nitrogen manifold. Figure 5.12 shows the measured RPM on the left and the position of the proportional wastegate on the right. The system worked successfully and the turbopump system was kept around 24000 RPM, but at the beginning of the test, it was seen that the system took approximately 7 seconds to reach the desired RPM value. Since such a delay in the turbopump system reaching the desired RPM value during rocket ignition would result in the oxidizer not being pressurized sufficiently and the desired oxidizer flow rate not being achieved, studies have been initiated to solve the problem.

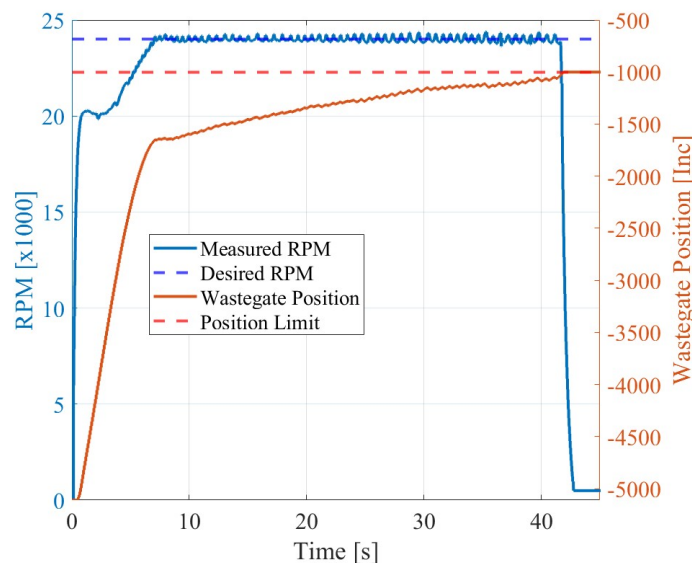


Figure 5.12. RPM and wastegate position results from integrated test 3.

When the position of the proportional wastegate is examined in Figure 5.12, the proportional wastegate moves in the direction of closing rapidly from its initial position, and after the turbopump system starts to rotate at the desired level, it continues to close slowly. Since the pressure could not be read from the nitrogen manifold, the pressure before the orifice could not be displayed. The expectation is that the energy entering the turbine decreases over time because the pressure of the nitrogen manifold decreases and the temperature of the nitrogen decreases due to expansion. Therefore, the controller closed the valve slowly throughout the test to keep the RPM of the system at the desired level.

Figure 5.13 shows the turbine inlet pressure, pump inlet pressure and pump outlet pressure collected by the DAQ system.

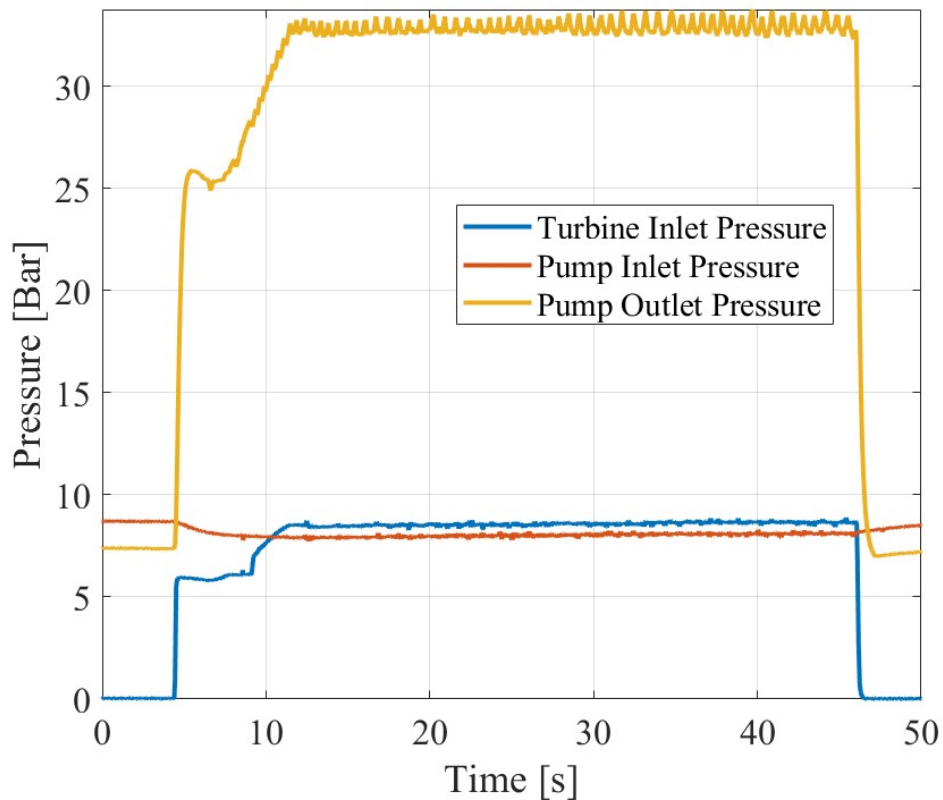


Figure 5.13. Turbine and pump pressures versus time from integrated test 3.

Figure 5.14 shows the pump outlet pressure on the right and the RPM measurement on the left. According to Figure 5.14, as a result of fluctuations in RPM, there are fluctuations in the pump outlet pressure with an amplitude of about 0.7-0.8 bar. Although the oscillations are only about 3% of the pump outlet pressure, high-frequency oscillations in the pump outlet pressure can destabilize the rocket engine because they cause high-frequency oscillations in the oxidizer flow rate.

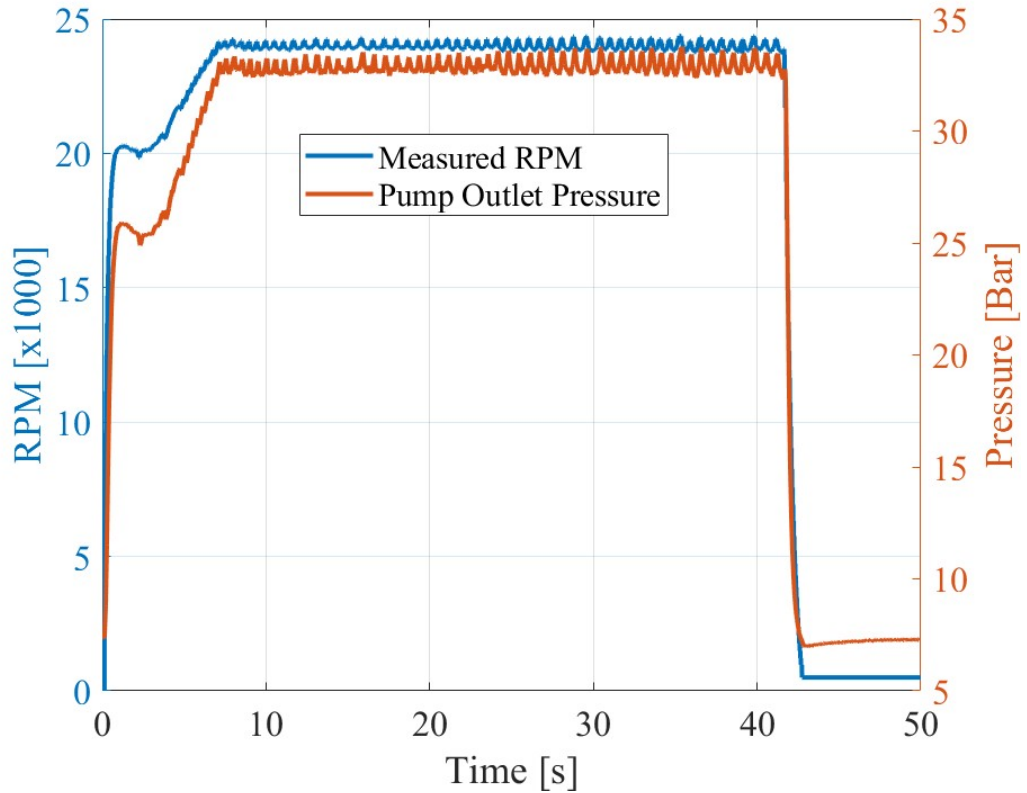


Figure 5.14. Measured RPM and pump outlet pressure versus time in test 3.

Immediately after the third integrated test, another test was performed to check the repeatability of the system operation since there was gas left in the nitrogen manifold, which is the cold gas generator, for one more test. In the fourth integrated test, since the confidence in the system increased due to the third test, the position limit of the proportional wastegate, which was 1000 increments, was reduced to 700, unlike the third test. Figure 5.15 shows the measured RPM on the left and the position of the

proportional wastegate on the right. The system worked successfully as in the third test and the turbopump system was kept at approximately 24000 RPM. As in the third test, it was observed that it took approximately 7 seconds for the system to reach the desired RPM at the beginning of the test. Unlike the third test, an increase in the amplitude of the oscillations and a slight decrease in their frequency were observed in the fourth test.

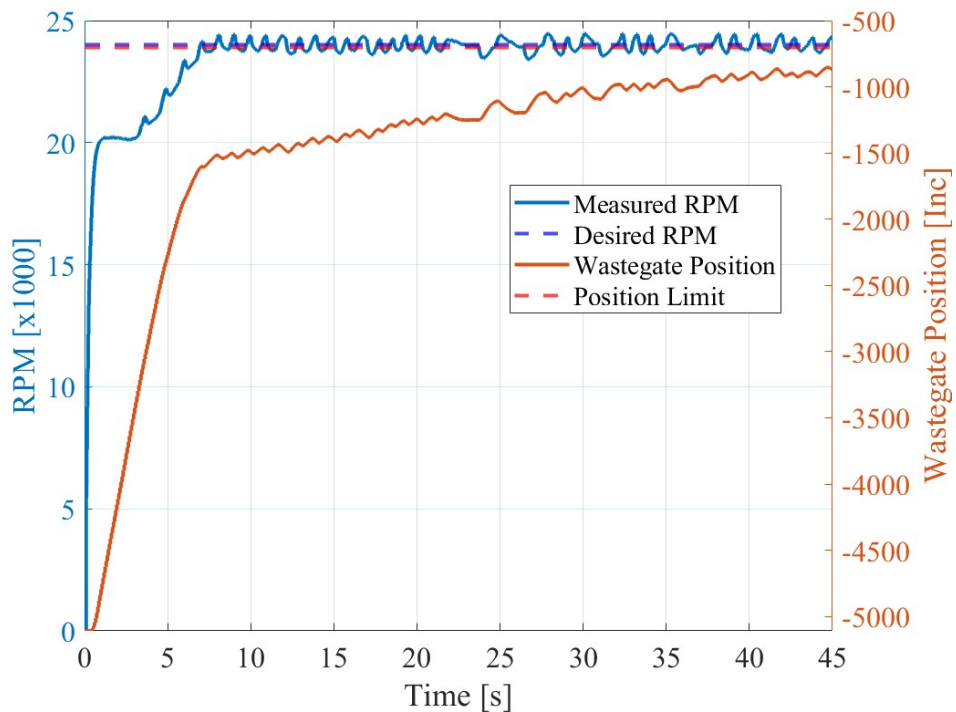


Figure 5.15. RPM and wastegate position results from integrated test 4.

Figure 5.16 shows the turbine inlet pressure, pump inlet pressure and pump outlet pressure collected by the DAQ system.

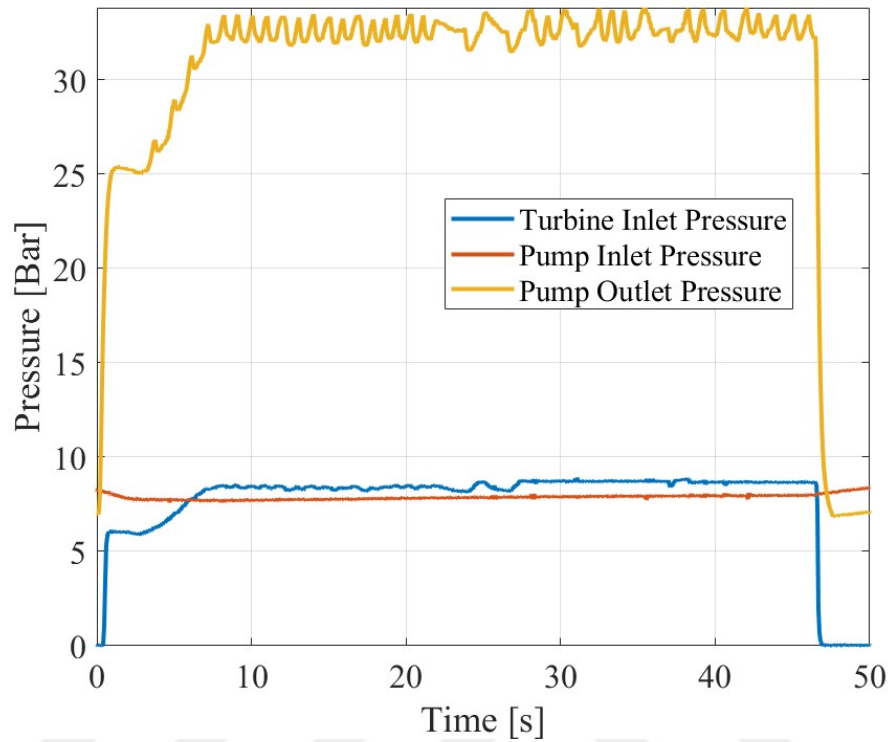


Figure 5.16. Turbine and pump pressures versus time from integrated test 4.

Figure 5.17 shows the pump outlet pressure on the right and the RPM measurement on the left. When Figure 5.15, Figure 5.16, and Figure 5.17 are examined together, it is seen that the proportional wastegate moves slowly in the opening direction from the twenty-first to the twenty-fourth seconds and therefore the turbine inlet pressure drops by approximately 0.5 bar. It is seen that the drop in turbine inlet pressure does not affect the rotation speed of the turbopump for a long time, only a sharp drop in RPM in the last 400 ms. The pump outlet pressure also lost 1 bar in this sharp drop. It is thought that such problems will be prevented with the new PID controller that will reduce the oscillations.

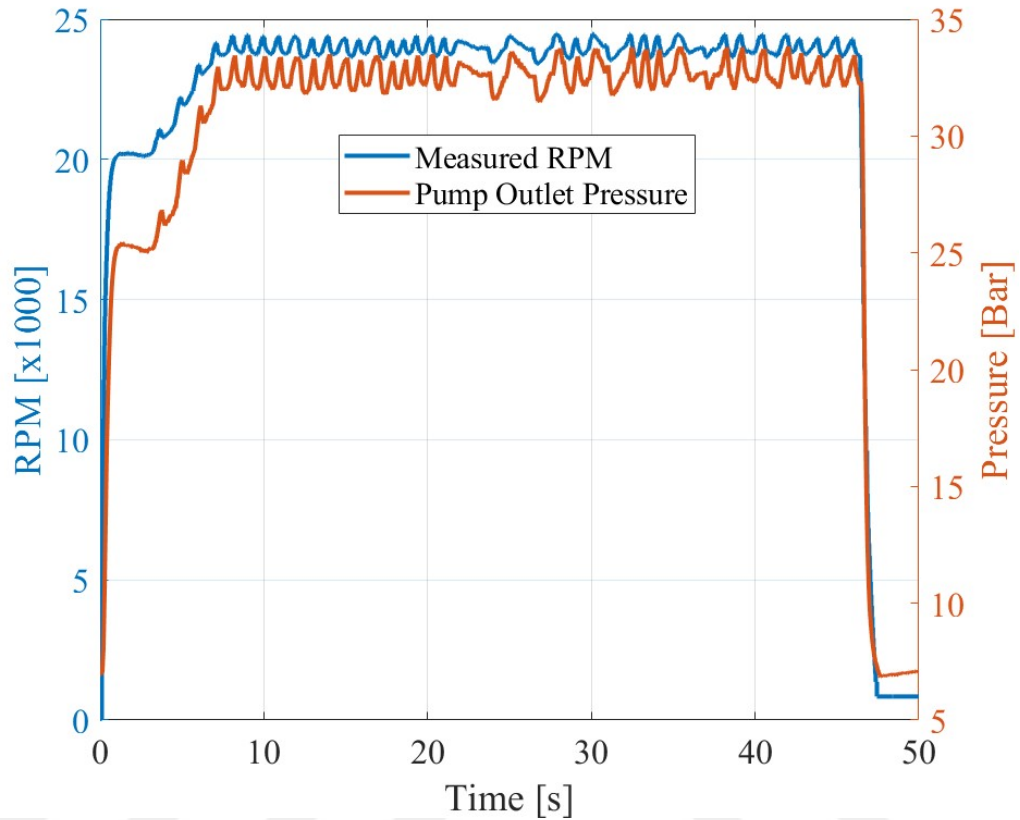


Figure 5.17. Measured RPM and pump outlet pressure versus time in test 4.

After the third and fourth integrated tests, the error values going to the controller were changed to ensure that the turbopump system reaches the desired RPM level more quickly and to have less oscillation while keeping it at the desired RPM level. The PI controller is configured to produce varying outputs by applying different coefficients to error term across three distinct intervals, based on the deviation between the measured RPM and the target RPM value. The purpose of this change is to allow the controller to control the proportional wastegate faster with larger outputs up to a certain level according to the measured RPM value, and to reduce the oscillations by controlling it more slowly by giving smaller outputs when it approaches the desired RPM value.

The operations applied by the controller to the error term are shown in Figure 5.18. According to Figure 5.18, the first range is valid when the error term is greater than 2000 and the controller will square the error term and divide it by 1000.

In this way, the error term will appear larger than it is and the proportional wastegate will move faster. The second range is valid when the error term is greater than 350 and less than 2000 and the controller will take the error term as it is. In this way, the proportional wastegate will slow down when the desired RPM value is approached. In the third range, when the error term is equal to or less than 350, the controller will take the root of the error term and multiply it by 5. In this way, the system will move much slower around the desired RPM and will not make the oscillations seen in the third and fourth integrated tests.

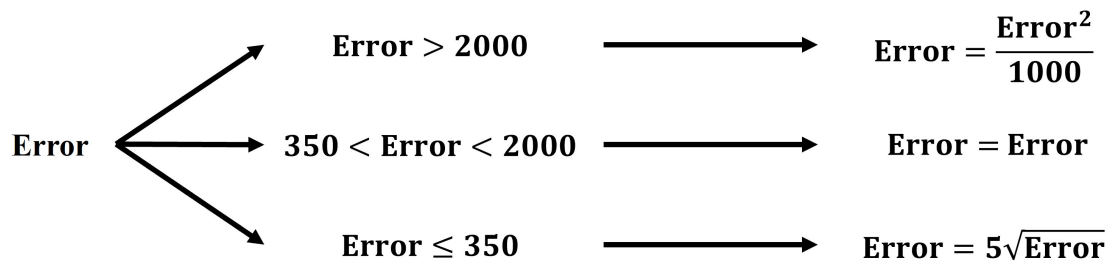


Figure 5.18. New error term calculation logic.

A fifth integrated test, planned to last 30 seconds, was conducted to examine the changes made to the controller. Figure 5.19 shows the measured RPM on the left and the position of the proportional wastegate on the right. The test was terminated early due to oscillations occurring after the sixteenth second. When the beginning of the test is examined, it is seen that the turbopump system reaches 22000 RPM in approximately 1 second and slowly reaches the desired 24000 RPM due to the change in the coefficients of the error term according to the newly designed controller. When the old tests are examined, it is seen that the system takes 5 seconds to reach 22000 RPM, while the new system reaches the desired 24000 RPM in approximately 4 seconds. The turbopump system successfully rotated at the desired RPM for approximately 10 seconds and no oscillations were observed. Towards the end of the test, an oscillation was observed as a result of the sudden response of the controller due to the change in the coefficients of the error term and the test was stopped in order to prevent any damage to the turbopump system.

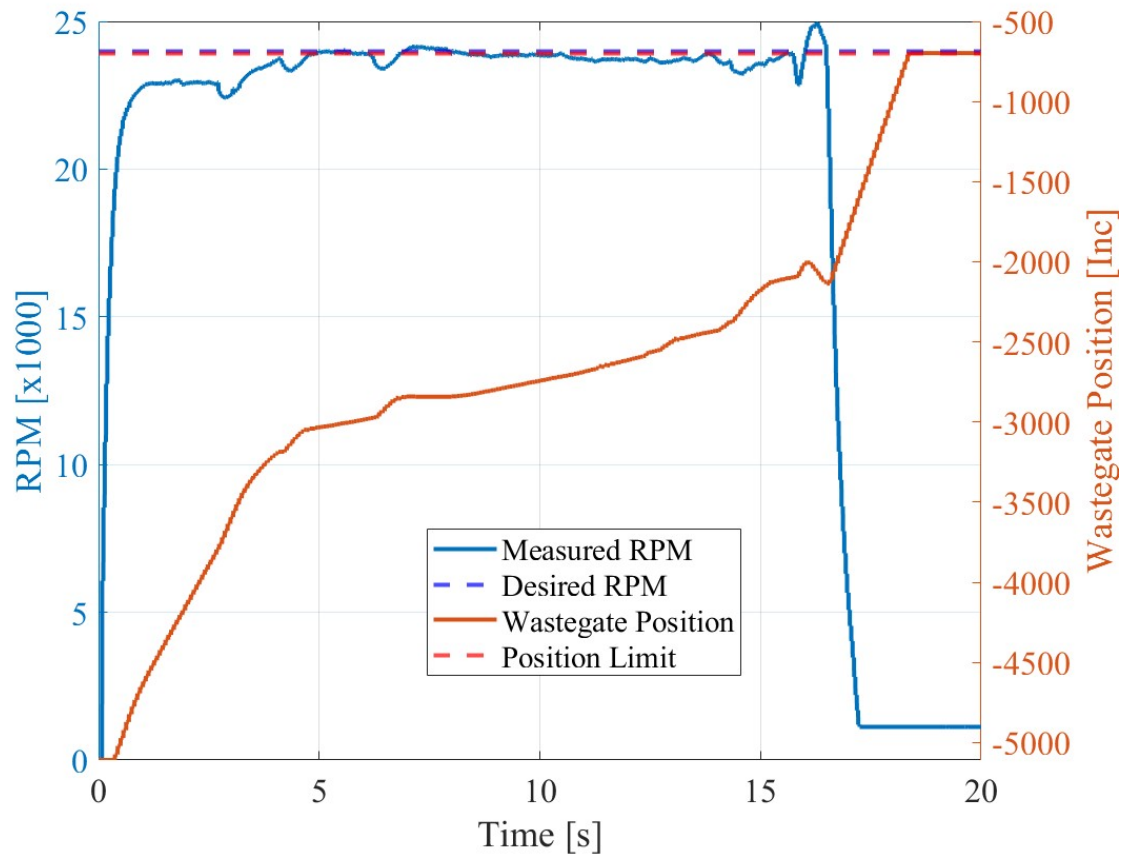


Figure 5.19. RPM and wastegate position results from integrated test 5.

Figure 5.20 shows the turbine inlet pressure, pump inlet pressure and pump outlet pressure collected by the DAQ system. When the turbine inlet pressure was examined, it was seen that the pressure value measured at the part where it remained constant at the beginning of the test was slightly low. Since the current RPM value and pump pressure cannot be obtained with this pressure value, it is thought that there is a blockage in the pressure sensor.

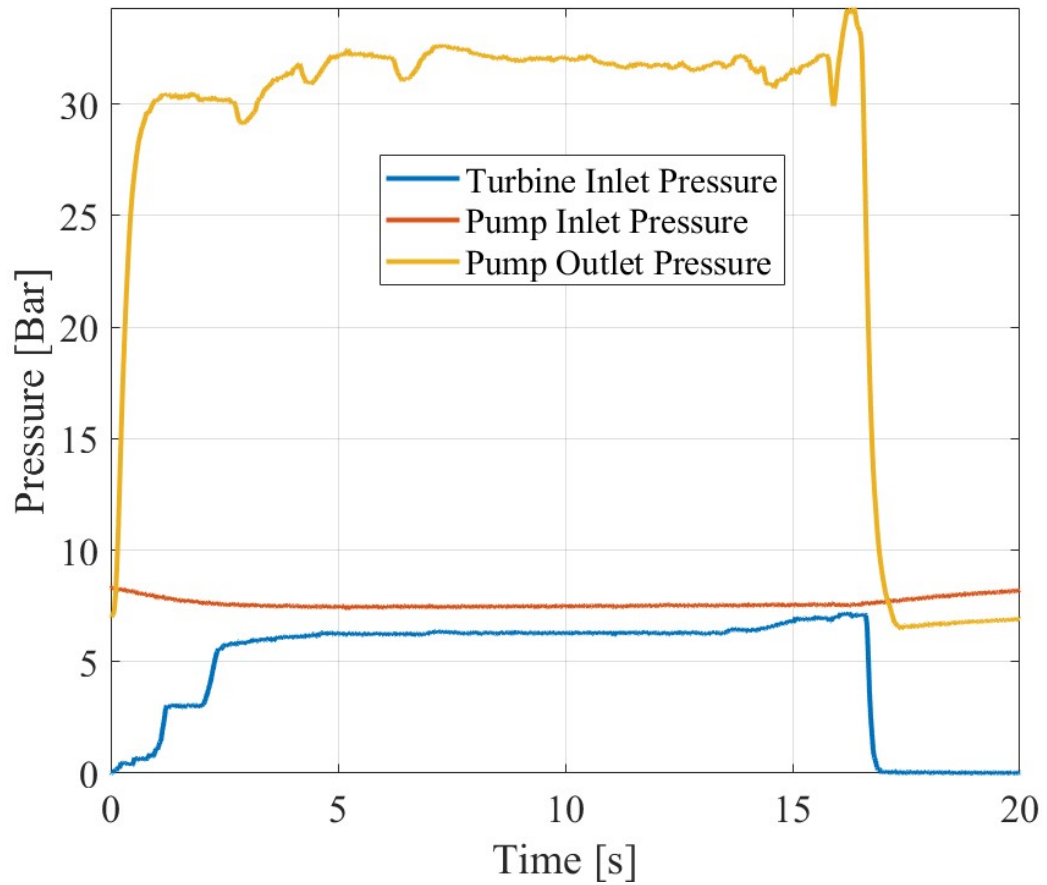


Figure 5.20. Turbine and pump pressures versus time from integrated test 5.

Figure 5.21 shows the pump outlet pressure on the right and the RPM measurement on the left. As a result, the time it takes for the turbopump to reach the desired RPM at the beginning of the test has been shortened with the new controller. In addition, the oscillations seen in RPM and pump outlet pressure have almost been eliminated. Studies have been started on the oscillations that occurred after the test.

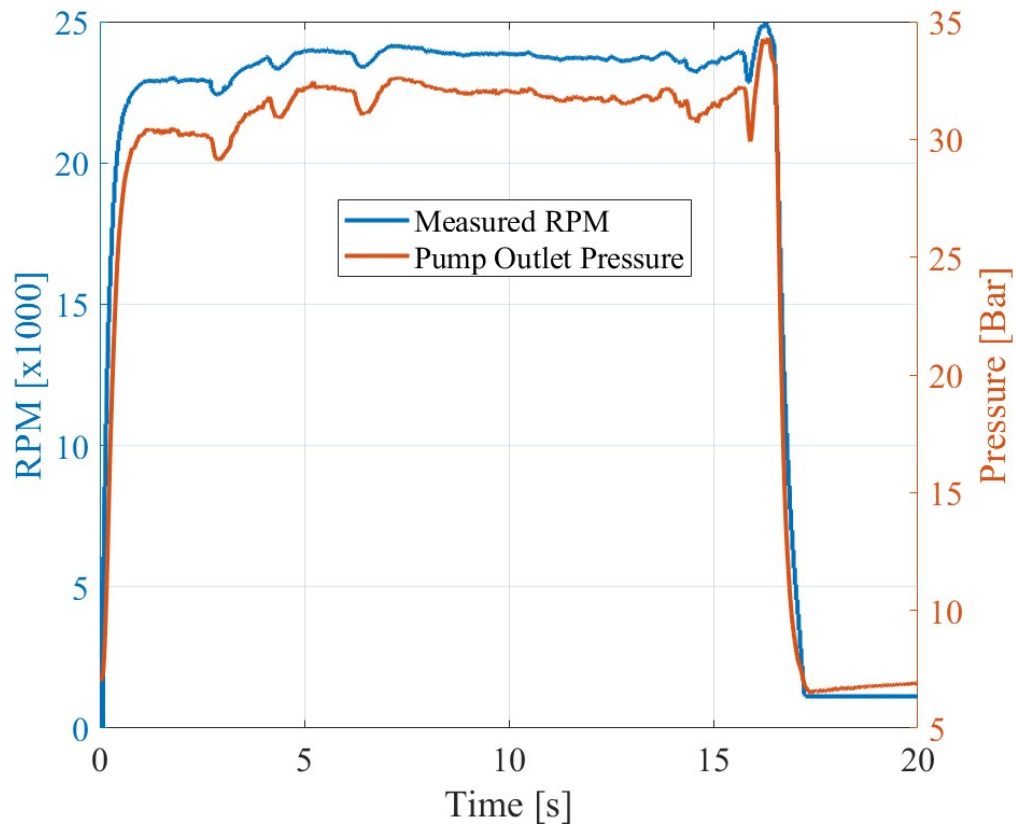


Figure 5.21. Measured RPM and pump outlet pressure versus time in test 5.

Since there was still enough nitrogen to do one more test, another test was conducted to see if the fluctuation that occurred at the end of the fifth test would occur again. Figure 5.22 shows the measured RPM on the left and the position of the proportional wastegate on the right. When Figure 5.22 is examined, it is seen that the proportional wastegate started the test in the closed position, which was previously encountered in the second test. Due to this situation that was overlooked before the test started, the RPM value came to 32000 RPMs and then the sensor started to measure around half of the real value since it was saturated. Since the proportional wastegate valve was already moving in the opening direction to reduce the energy in the turbine, the RPM value came to the range that the sensor could measure by itself. Even if this situation did not happen, the controller would activate the proportional wastegate to reduce the energy in the turbine due to the pump pressure exceeding the previously determined limit with the second precaution mentioned in Figure 5.11.

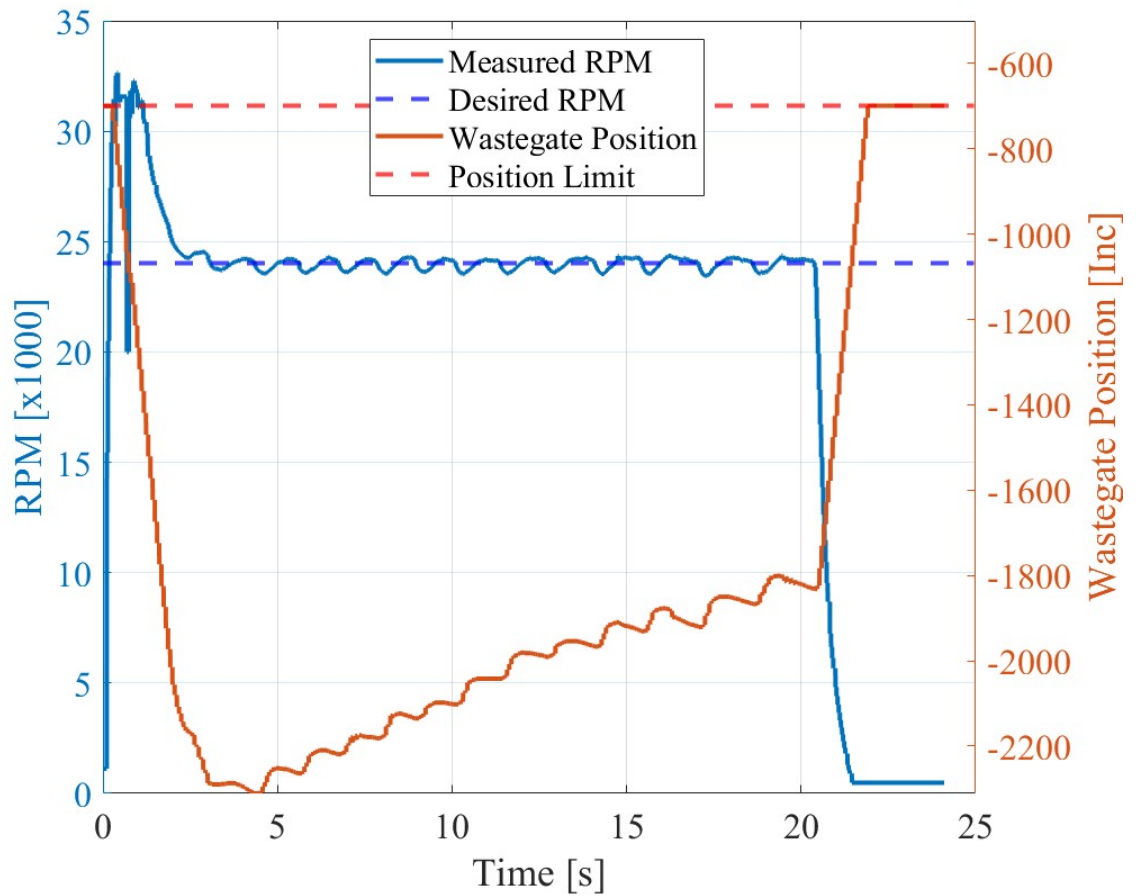


Figure 5.22. RPM and wastegate position results from integrated test 6.

Figure 5.23 shows the turbine inlet pressure, pump inlet pressure and pump outlet pressure collected by the DAQ system. When Figure 5.23 is examined, it is seen that the pump outlet pressure, which is set as 50 bar pressure limit for the controller, has reached 57 bars. If the RPM sensor had not started to take proper measurements, the proportional wastegate would have worked in the direction of opening since the pump outlet pressure exceeded the limit.

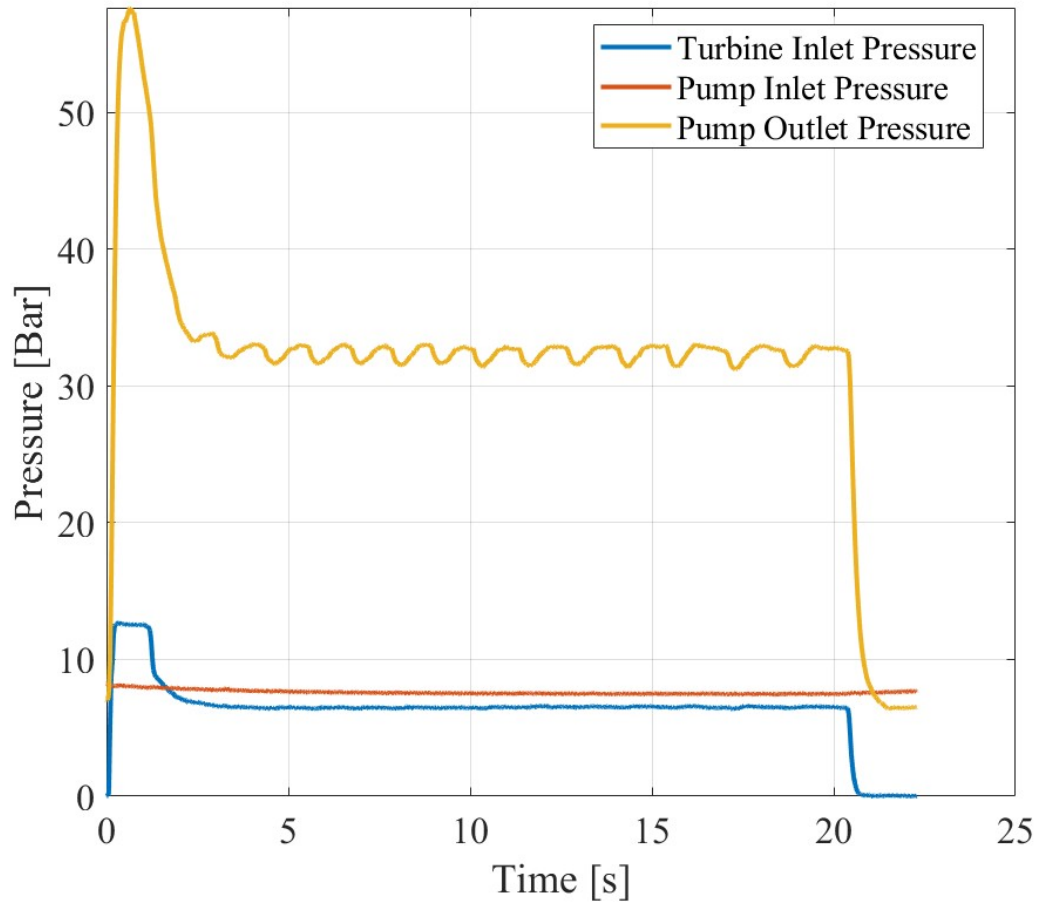


Figure 5.23. Turbine and pump pressures versus time from integrated test 6.

Figure 5.24 shows the pump outlet pressure on the right and the RPM measurement on the left. The controller intervenes very quickly in the error at the beginning of the test and then keeps the RPM value of the turbopump at the desired level. Contrary to the fifth test, although oscillations were seen again in the sixth test, when compared with the third and fourth tests, it is seen that the frequency of oscillations decreased.

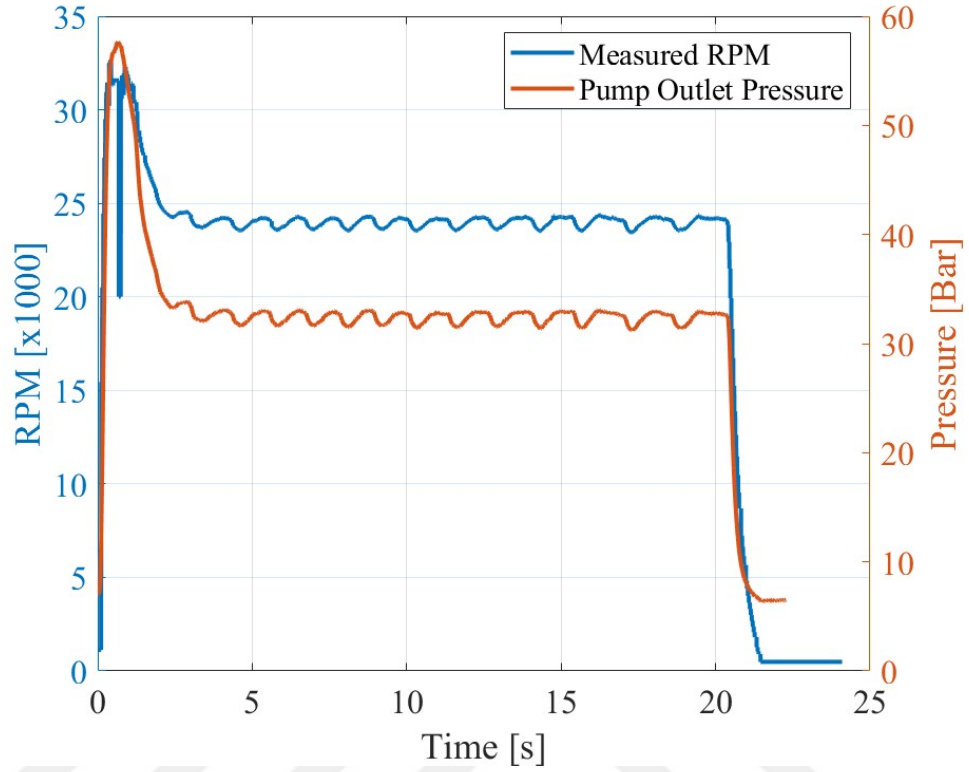


Figure 5.24. Measured RPM and pump outlet pressure versus time in test 6.

The sixth integrated test is the last integrated test and the PID coefficients used in the integrated tests are given in Table 5.1. In Test 5 and Test 6, the error term processed by the controller is changed as given in Figure 5.18.

Table 5.1. PID coefficients used in controller during integrated tests.

	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
K_p	1000	250	500	500	500	500
K_i	250	250	250	250	100	100
K_d	28	0	0	0	0	0

As a result of the integrated tests, solutions were implemented for the problems encountered during the process, and it was observed that the proposed system successfully kept the turbopump system within the desired RPM range.

6. CONCLUSION

This thesis presents the modeling, control, and testing of a proportional wastegate designed to regulate the energy supplied to the turbine that drives the pump in the pressurization systems of liquid rocket engines. The study begins by introducing chemical, electrical, and nuclear propulsion systems, followed by an explanation of the feed systems used in liquid rocket engines. The different cycles utilized in turbopump systems for liquid rockets are reviewed, with a focus on the gas generator cycle, which is employed in this thesis. Relevant examples from the literature are reviewed to provide context. Additionally, wastegate types found in automotive turbochargers, which are analogous to the proposed wastegate system, are examined alongside the control systems for these wastegates, and relevant studies in the literature are discussed. Finally, the significance and scope of this study are outlined.

The development process of the proportional wastegate is explained in detail, covering the design, constraints, and requirements. To understand the behavior of the wastegate within the control algorithm, modeling and characterization studies were performed. These studies included analytical calculations of the mass flow rates that the valve would allow at different opening ratios. To validate the analytical results and further characterize the valve, more than ten water tests were conducted under varying conditions and at different valve openings. The characterization focused on two primary variables, with the first being the examination of the discharge coefficient, C_d , at different opening ratios. The discharge coefficients were studied at three different operating ranges of the proportional wastegate, and curves defining the system were developed using curve fitting methods.

Subsequently, the relationship between the discharge coefficient and the cavitation number was examined, drawing on similar studies from the literature. However, since these studies typically focused on valves with fixed areas, no significant results were observed in the initial analyses. To address this, the discharge coefficient was

normalized using the valve opening ratio. In the refined analysis, a strong linear trend was identified, with an R^2 value of 0.98, achieved through the linear regression model. This successful characterization of the proportional wastegate, which involved both the discharge coefficient and cavitation number, provided the necessary data to complete the valve's performance profile. This characterization information was then used in the simulations to further refine and test the control system.

The RPM sensor, developed to measure the rotation speed of the turbopump system for controlling the proportional wastegate, was based on a proximity sensor. The development process began with a review of RPM sensors in the literature, followed by the selection of a suitable sensor according to design constraints and requirements. Since the inductive proximity sensor used could not directly measure rotation speed, an algorithm was created to calculate the RPM from its output. The functionality of the RPM sensor was tested using an electric motor operating at high speeds, and the results demonstrated that the sensor provided fast, accurate, and precise measurements.

The general control theory was studied in depth during the design phase of the controller for the proportional wastegate. This included understanding control strategies, such as proportional, integral, and derivative control (PID), to ensure the effective regulation of the wastegate system. Subsequently, all subsystems were modeled in detail using the MATLAB Simulink environment. The model comprises several key subsystems: the proportional wastegate, cold gas generator system, turbopump system, electric motor driving the proportional wastegate, and the controller system. Through simulations conducted in MATLAB Simulink, the interactions between these components were modeled and analyzed. These simulations allowed for the derivation of the necessary PID controller coefficients, which were essential for tuning the controller to achieve the desired performance. In this way, both time and money were saved before the real-world tests and damage to the systems was prevented in the event of a possible error. Finally, the simulation results were examined in detail, and all necessary preparations for integrated testing were completed.

To assess the functionality of the proportional wastegate and the designed controller, two integrated test environments were established using hot and cold gas generators. In the first test, with the hot gas generator, the turbopump system's RPM could not be maintained within the desired range. The test results revealed errors that were not detectable during the previous subsystem tests, and these issues were addressed and corrected. Given the time and cost associated with testing with the hot gas generator, five additional tests were conducted using a cold gas generator with similar characteristics. The results from these tests led to further refinements of the controller system. New PID coefficients were calculated based on the difference between the measured and desired RPMs, allowing for more accurate control of the system. It has been observed that the controller system, which receives feedback from the RPM sensor and controls the proportional wastegate, effectively regulates the speed of the turbopump system, even in the presence of disturbances.

6.1. Future Work

Thus far, successful tests have demonstrated the proposed system's ability to maintain the turbopump's rotation speed within the desired range. However, some issues were observed, such as delays in reaching the target RPM and minor RPM fluctuations throughout operation. These issues will be addressed by adjusting the coefficients of the PID controller to enhance response time and stability.

To reduce the number of real-life tests required, updates can be applied to the simulation model based on the obtained test results, allowing it to more accurately reflect real-life conditions. This approach enables the simulation to better match actual test outcomes, thereby reducing the need for future physical tests.

The electric motor that controls the proportional wastegate currently operates in position mode, with the PID controller setting the target position. However, upon further examination of the motor's internal controller, it appears that operating the motor in speed mode rather than position mode could reduce internal feedback cycles,

allowing for smoother control of the proportional wastegate. Studies have been conducted on this approach, and real-life testing will be required to evaluate the system's performance with this adjustment.

An examination of the tests conducted for characterizing the proportional wastegate indicates that, despite extensive data for a variety of scenarios, there is limited data available for high cavitation numbers. Additionally, because the valve is rarely operated in the 0-20% opening range, data in this range is also rare. To enhance the characterization of the proportional wastegate, additional water tests could be conducted in future studies to collect data specifically for these conditions. This additional data would enable a more comprehensive and precise characterization of the valve's performance across its entire operating range.

In the initial test with the hot gas generator, although the generator itself operated smoothly, the turbopump rotation speed could not be controlled as desired due to system errors. The system was subsequently improved with many updates and corrections. Once the remaining issues awaiting updates are addressed, a subsequent test with the hot gas generator is planned.

REFERENCES

1. Sutton, G. P. and O. Biblarz, *Rocket Propulsion Elements*, John Wiley & Sons, New Jersey, 9 edn., 2016.
2. Kokal, U., *Development of a Mili-Newton Level Thrust Stand for Thrust Measurements of Electric Propulsion Systems and Uk90 Hall Effect Thruster*, Master's Thesis, Boğaziçi University, 2019.
3. Macdonald, M. and V. Badescu, *The International Handbook of Space Technology*, Springer Berlin, Heidelberg, 2014.
4. Cantwell, B., A. Karabeyoglu and D. Altman, "Recent Advances in Hybrid Propulsion", *International Journal of Energetic Materials and Chemical Propulsion*, Vol. 9, pp. 305–326, 2010.
5. Zolla, P. M., M. Migliorino, D. Bianchi, F. Nasuti, R. Pellegrini and E. Cavallini, "A Computational Tool for the Design of Hybrid Rockets", *Aerotecnica Missili & Spazio*, Vol. 100, pp. 253–262, 2021.
6. Story, G., T. Abel, S. Claffin, O. Park, J. Arves and D. Kearney, "Hybrid Propulsion Demonstration Program 250K Hybrid Motor", *39th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit*, 2003.
7. "DELTAV-About Us", <https://deltav.com.tr/en/index.php/about/>, accessed on September 30, 2024.
8. Jahn, R. G., "Electric Propulsion", *American Scientist*, Vol. 52, No. 2, pp. 207–217, 1964.
9. Coote, D. J., K. P. Power, H. P. Gerrish and G. Doughty, "Review of Nuclear Thermal Propulsion Ground Test Options", *51st AIAA/SAE/ASEE Joint Propulsion*

Conference, Vol. 1, pp. 890–897, 2015.

10. Atasoy, M., *Design & Analysis and Development of Turbine System for Hybrid Rocket Applications*, Master's Thesis, Koç University, 2022.
11. Lee, J., T.-S. Roh, H. Huh and H. Lee, "Performance Analysis and Mass Estimation of a Small-Sized Liquid Rocket Engine with Electric-Pump Cycle", *International Journal of Aeronautical and Space Sciences*, Vol. 22, pp. 94–107, 2020.
12. Rachov, P. A. P., H. Tacca and D. Lentini, "Electric Feed Systems for Liquid-Propellant Rockets", *Journal of Propulsion and Power*, Vol. 29, No. 5, pp. 1171–1180, 2013.
13. Sutton, G., *History of Liquid Propellant Rocket Engines*, American Institute of Aeronautics and Astronautics, Virginia, 2006.
14. RocketLab, "Rutherford", <https://web.archive.org/web/20160919222251/https://rocketlabusa.com/about-us/propulsion/>, accessed on September 30, 2024.
15. Atkinson, I., "Astra Scrubs DARPA Launch Challenge Attempt", <https://www.nasaspaceflight.com/2020/03/astra-darpa-launch-challenge/>, accessed on September 30, 2024.
16. Fatuev, I. Y. and A. A. Ganin, "New Propellants Ignition System in LV "Soyuz" Rocket Engine Chambers", *55th International Astronautical Congress*, Vol. 11, pp. 6980–6986, 2004.
17. NASA, "This Week in NASA History: 1st Full-Thrust, Long-Duration F-1 Engine Test — May 26, 1962", <https://www.nasa.gov/image-article/this-week-nasa-history-1st-full-thrust-long-duration-f-1-engine-test-may-26-1962/>, accessed on September 31, 2024.

18. SpaceX, “Falcon 9”, <https://www.spacex.com/vehicles/falcon-9/>, accessed on November 3, 2024.
19. SpaceX, “Merlin 1D Test Fire”, <http://www.spacex.com/media-gallery/detail/1/1661/172>, accessed on September 31, 2024.
20. Brito, I., *Modeling and Control of Wastegate Equipped Turbocharged Engines*, Master’s Thesis, University of New Mexico, 2010.
21. Prasad, N., N. Ganesh and A. Kumarasamy, “Technologies for High Power Density Diesel Engines”, *Defence Science Journal*, Vol. 67, p. 370, 2017.
22. Holmbom, R. and B. Liang, *Boost Control with Turbo Speed Sensor and Electric Wastegate*, Master’s Thesis, Linköping University, 2016.
23. Motorsport, W., “External Wastegate – Whats it all about?”, <https://www.waltonmotorsport.com/external-wastegate/>, accessed on September 30, 2024.
24. Andersson, H., *Model Based Control of Throttle, EGR and Wastegate : A System Analysis of the Gas Flows in an SI-Engine*, Master’s Thesis, Linköping University, 2017.
25. Vavra, J., J. Macek, O. Vítek and M. Takats, “Investigation of Radial Turbocharger Turbine Characteristics under Real Conditions”, *SAE Technical Paper*, 2009.
26. Ruffin, A. *et al.*, *Study and Development of Throttleable Hybrid Rocket Motors*, Master’s Thesis, Università Degli Studi Di Padova, 2018.
27. Bell, I. H., J. Wronski, S. Quoilin and V. Lemort, “Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp”, *Industrial & Engineering Chemistry Research*, Vol. 53, No. 6, pp. 2498–2508, 2014.

28. Vennard, J. and R. Street, *Elementary Fluid Mechanics*, Wiley, New Jersey, 1982.
29. Kayser, J. and R. Shambaugh, “Discharge Coefficients for Compressible Flow through Small-Diameter Orifices and Convergent Nozzles”, *Chemical Engineering Science - CHEM ENG SCI*, Vol. 46, pp. 1697–1711, 1991.
30. Antunes, J. M. G., *The Use of Hydrogen as a Fuel for Compression Ignition Engines*, Master’s Thesis, Newcastle University, 2011.
31. Honeywell, “VersaFlow Coriolis Flowmeter”, <https://process.honeywell.com/us/en/products/field-instruments/flow-meters/versaflow-coriolis-flowmeter>, accessed on September 30, 2024.
32. ifm, “Pressure Transmitter-PT2402”, <https://www.ifm.com/tr/en/product/PT2402>, accessed on September 30, 2024.
33. Omega, “Thermocouple Probes”, <https://www.omega.com/en-us/resources/thermocouple-hub>, accessed on September 30, 2024.
34. Group, M., “Positioning Controllers”, <https://www.maxongroup.com/en/drives-and-systems/controls/positioning-controllers>, accessed on November 3, 2024.
35. Lefebvre, A. and V. McDonell, *Atomization and Sprays*, CRC Press, Florida, 2017.
36. Mathews, J. and K. Fink, *Numerical Methods Using MATLAB*, Featured Titles for Numerical Analysis Series, Pearson Prentice Hall, New Jersey, 2004.
37. Moore, L. M., “The Basic Practice of Statistics”, *Technometrics*, Vol. 38, No. 4, pp. 404–405, 1996.
38. Johansson, M., *Evaluation of Sensor Solutions a Motor Speed Control Methods for BLDCM/PMSM in Aerospace Applications*, Master’s Thesis, Lulea University of

Technology, 2017.

39. Bestas, “Types of Speed Sensors: Advantages and Disadvantages”, <https://www.bestas.com.tr/en/blog/faq/types-of-speed-sensors-advantages-and-disadvantages>, accessed on November 3, 2024.
40. Elkome, “Monarch Instrument 6180-030 Optical Laser RPM Sensor”, <https://shop.elkome.com/en/6180-030-optical-laser-rpm-sensor>, accessed on November 3, 2024.
41. Östling, J., *High Accuracy Speed and Angular Position Detection by Dual Sensor*, Master’s Thesis, Uppsala University, 2018.
42. Micro-Epsilon, “Turbocharger Speed Sensors”, <https://www.micro-epsilon.com/inclination-acceleration-sensors/rotation-speed-sensors/turbocharger-speed-dz140/>, accessed on September 30, 2024.
43. Di-Soric, “DCC 05 V 0.8 PSK-K-TSL”, https://www.di-soric.com/PM/Sensors/Inductive-sensors/Inductive-proximity-sensors/INM-100-Miniature/DCC-05-V-0.8-PSK-K-TSL__210436, accessed on September 30, 2024.
44. Omron, “Proximity Sensors”, https://www.ia.omron.com/support/guide/41/explanation_of_terms.html, accessed on September 30, 2024.
45. Franklin, G., A. Emami-Naeini and J. Powell, *Feedback Control of Dynamic Systems*, Pearson, New Jersey, 2010.
46. Brito, T., C. Lira and W. Vasconcelos, “Development of a PID-Fuzzy Controller in the Water Level Control of a Pressurizer of a Nuclear Reactor”, *Brazilian Journal of Radiation Sciences*, Vol. 8, No. 3B, 2021.
47. Araki, M., “PID Control”, *Control Systems, Robotics and Automation*, Vol. 2, pp. 58–79, 2009.

48. Öznlbant, B., *Fuel Grain Modeling and Optimization for Hybrid Propulsion Systems using Additive Manufacturing*, Master's Thesis, Koç University, 2023.
49. Terzi, B., *Design, Analysis and Testing of Cryogenic Centrifugal Pump for Hybrid Rocket Applications*, Master's Thesis, Koç University, 2023.



APPENDIX A: VARIOUS GEOMETRIES FROM CAD DRAWINGS USED TO CALCULATE PINTLE VALVE OPENING AREA

In order to check the accuracy of the formula used to find the narrowest area in the proportional wastegate, the geometries created for 1.5 mm stroke are given below. The areas of the given geometries are calculated with the truncated cone lateral surface area formula given as

$$Area = \frac{(R + r)}{2} \times s \times \pi \quad (A.1)$$

where R is the base radius, r is the top radius, and s is the slant height.

The lateral surface area of the cone calculated for the geometry used in modeling the proportional wastegate given in Figure A.1 is 69.16 mm².

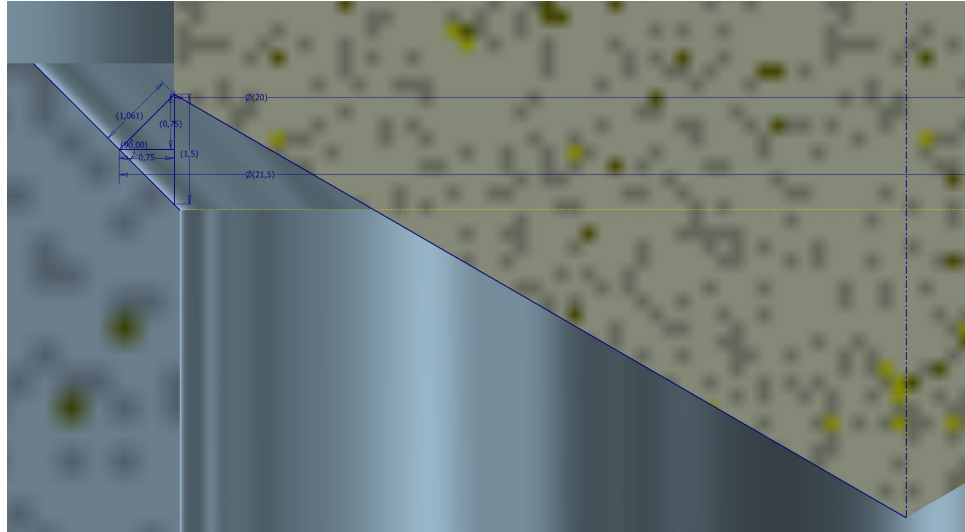


Figure A.1. Geometry used to calculate the smallest area when proportional wastegate has 1.5 mm stroke from CAD drawing.

The lateral surface area of the cone calculated for the geometry given in Figure A.2 is 70.88 mm^2 .

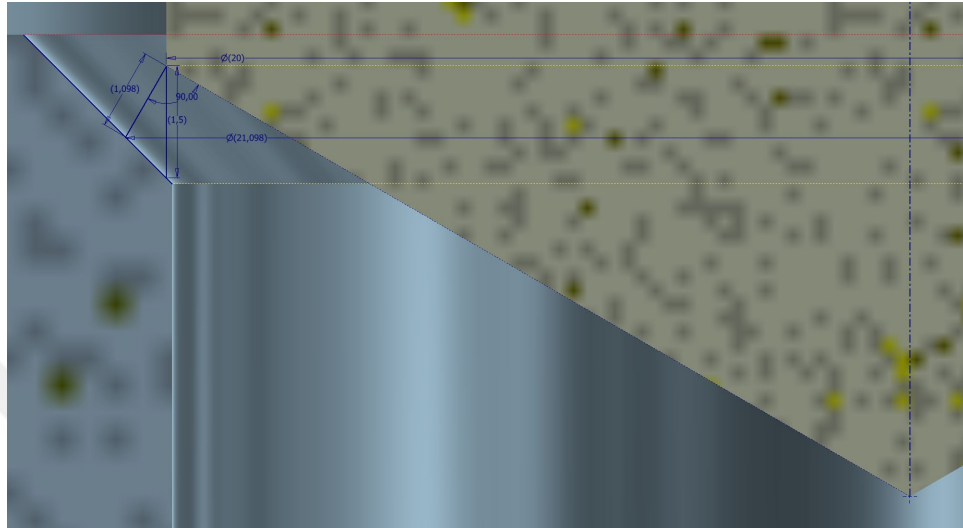


Figure A.2. Geometry used to calculate the second smallest area when proportional wastegate has 1.5 mm stroke from CAD drawing.

The lateral surface area of the cone calculated for the geometry given in Figure A.3 is 78.97 mm^2 .

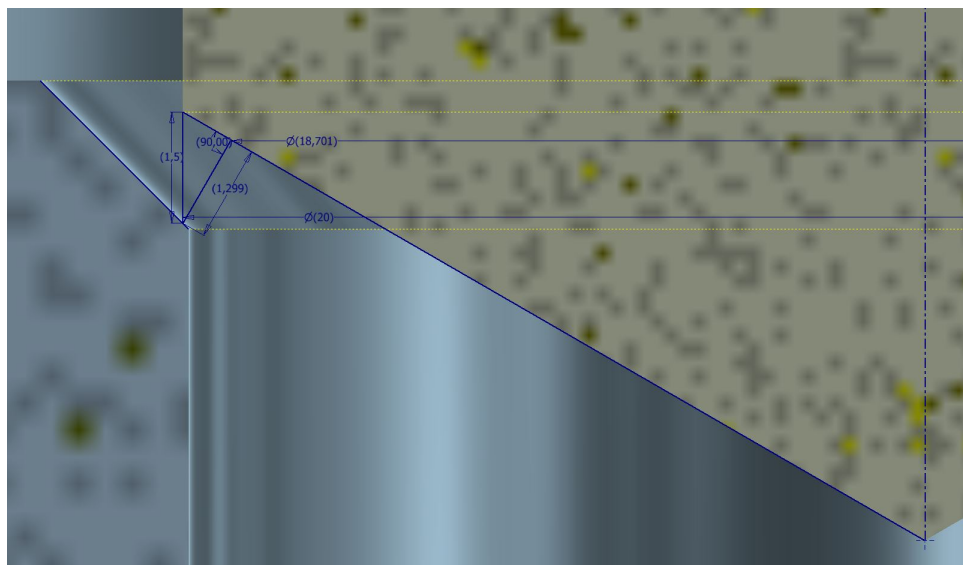


Figure A.3. Geometry used to calculate the third smallest area when proportional wastegate has 1.5 mm stroke from CAD drawing.

The lateral surface area of the cone calculated for the geometry given in Figure A.4 is 80.00 mm^2 .

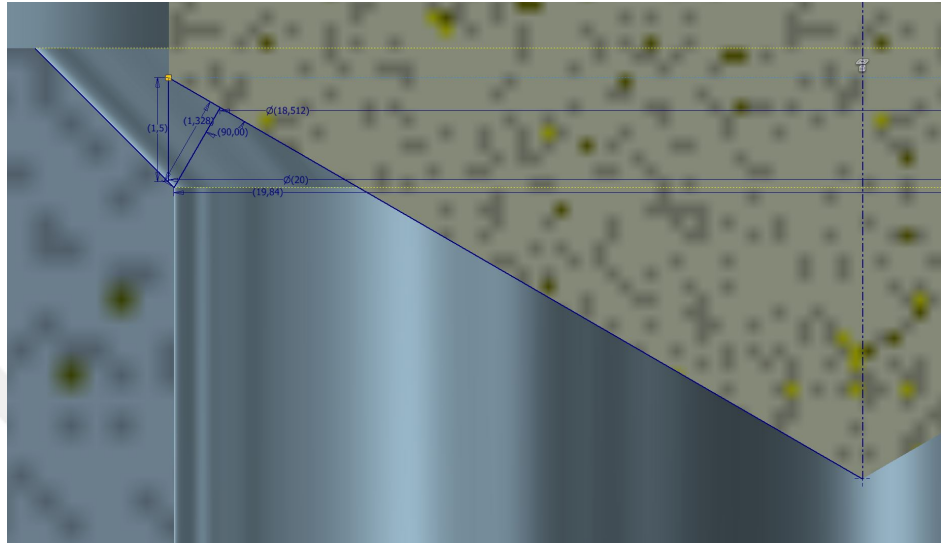


Figure A.4. Geometry used to calculate the fourth smallest area when proportional wastegate has 1.5 mm stroke from CAD drawing.

The lateral surface area of the cone calculated for the geometry given in Figure A.5 is 80.49 mm^2 .

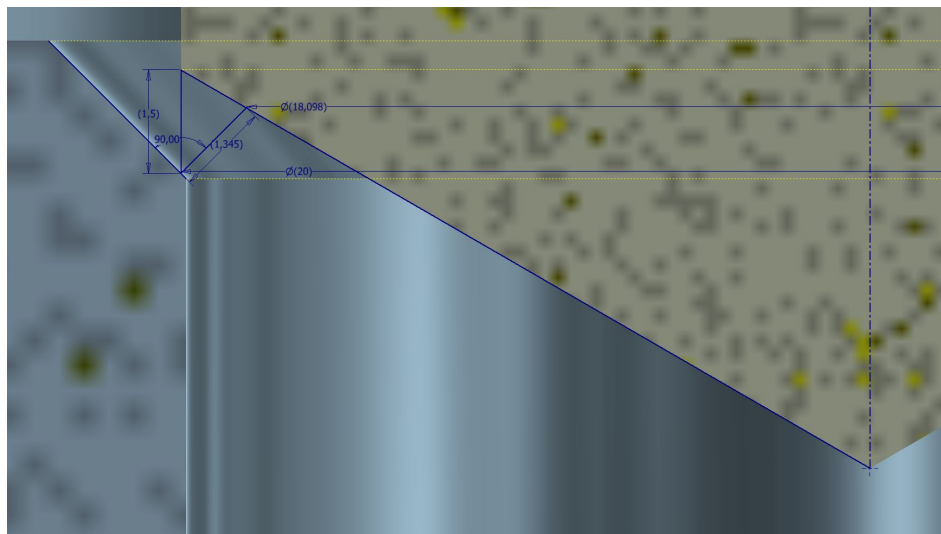


Figure A.5. Geometry used to calculate the fifth smallest area when proportional wastegate has 1.5 mm stroke from CAD drawing.

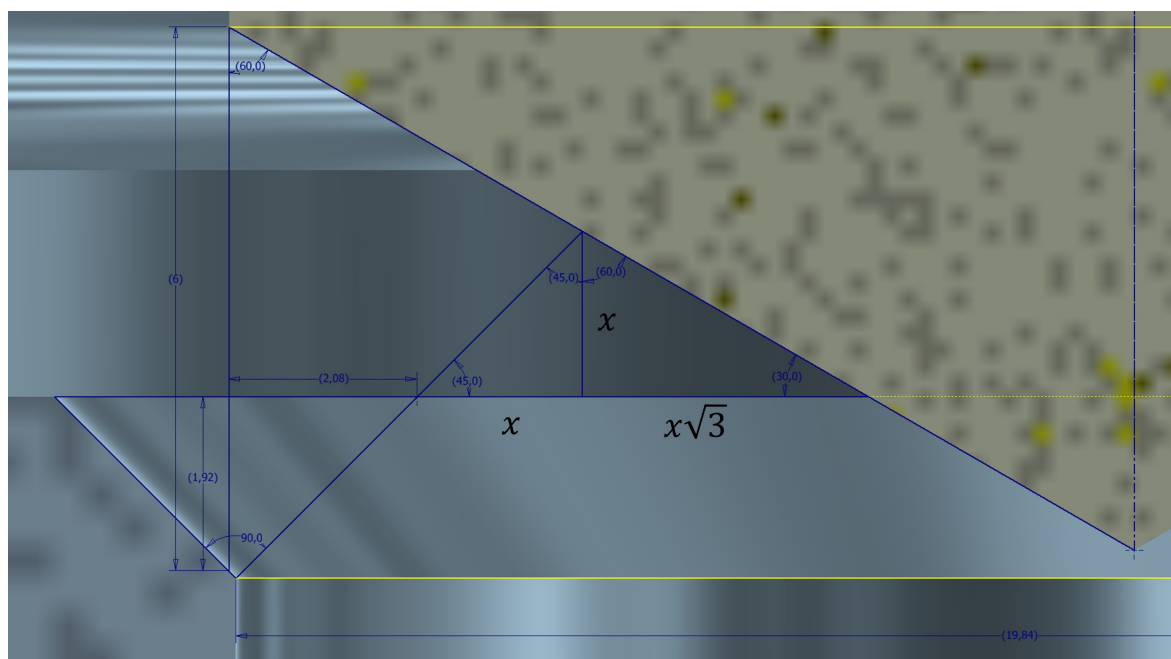


Figure B.2. Orifice area calculation of proportional wastegate with 6 mm stroke using CAD drawing, step 2.

$$x = \frac{\text{Side 1}}{1 + \sqrt{3}}. \quad (\text{B.2})$$

In Figure B.3, the slant height value s is shown on the CAD.

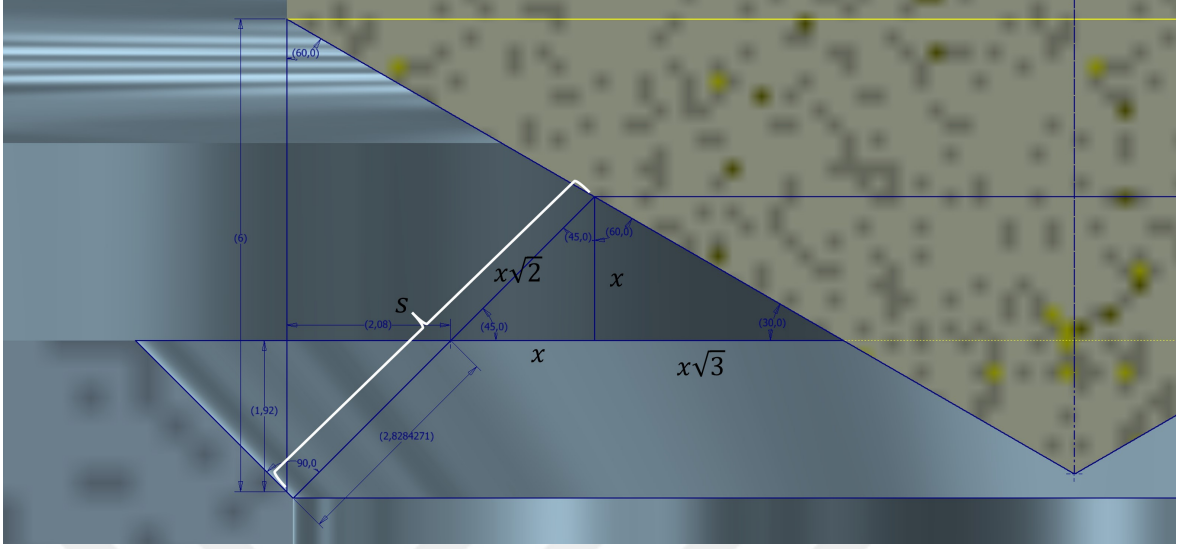


Figure B.3. Orifice area calculation of proportional wastegate with 6 mm stroke using CAD drawing, step 3.

s is calculated as

$$s = 2\sqrt{2} + x\sqrt{2}. \quad (\text{B.3})$$

In Figure B.4, the boundaries of the truncated cone that gives the narrowest area, the orifice, after the 4.141 mm stroke, which is the point where the area formula used for the proportional wastegate changes, are shown. In Figure B.4, R is the base radius, and r is the top radius.

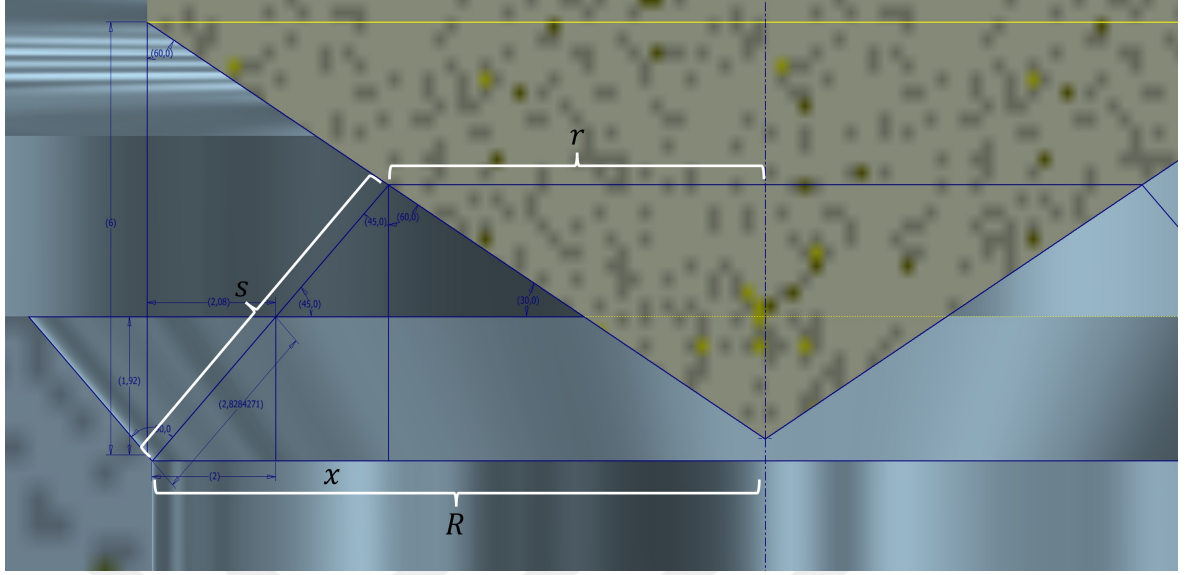


Figure B.4. Orifice area calculation of proportional wastegate with 6 mm stroke using CAD drawing, step 4.

Finally, using the variables given in Figure B.4, Equation 2.14 can be rewritten as

$$\text{Area} = \pi \times s(i) \times \left(\frac{R+r}{2}\right). \quad (\text{B.4})$$

APPENDIX C: WATER TEST RESULTS

The results of the water tests performed to characterize the proportional waste-gate are presented below. Since the results of Tests 3, 5, 11, and 12 were previously shared, they are not included again here.

In the first test, the test was not continued due to leakage in the proportional waste gate during the test. For Test 2, opening ratio of the proportional wastegate and the corresponding average mass flow rate and discharge coefficient values are given in Table C.1.

Table C.1. Average C_d and $\dot{m}$ corresponding to valve opening area from test 2.

Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)	Opening Percentage (%)	C_d (%)	$\dot{m}$ (g/s)
3,48	18,67	81	10,38	25,93	314
4,34	21,98	110	11,28	28	368
5,07	24,89	151	12,18	30	423
5,95	28,22	200	13,09	32,13	478
6,82	30,45	248	0	0	0
7,72	33,58	303	13,09	33,57	515
8,6	33,48	334	13,98	36,08	570
0	0	0	14,88	37,18	622
8,6	22,67	232	17,61	44,89	869
9,48	25,46	286	0	0	0

For Test 4, opening ratio of the proportional wastegate and the corresponding average mass flow rate, K and discharge coefficient values are given in Table C.2.

Table C.2. Average K , C_d , and $\dot{m}$ corresponding to valve opening area from test 4.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
20	1,12	46,23	1128
30	1,17	60,88	1936
40	1,24	68,96	2455
50	1,33	73,11	2740
60	1,47	77,35	2936

In Test 6, the proportional waste gate was opened at low speed within the range of 20%-80%, and the discharge coefficient of the system was examined. However, there is no data available to share from this test. For Test 7, opening ratio of the proportional wastegate and the corresponding average mass flow rate, K and discharge coefficient values are given in Table C.3.

Table C.3. Average K , C_d , and $\dot{m}$ corresponding to valve opening area from test 7.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
60	1,25	75,6	3896
70	1,32	76,11	4075

For Test 8, opening ratio of the proportional wastegate and the corresponding average mass flow rate, K and discharge coefficient values are given in Table C.4. In this test, the goal was to observe any changes in the discharge coefficient once the proportional waste gate was fully opened to 100%. However, no change in the discharge coefficient was observed during this phase of the test.

Table C.4. Average K , C_d , and $\dot{m}$ corresponding to valve opening area from test 8.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
70	1,50	79,7	3435
80	1,58	80,8	3672
100	1,69	73,19	3796
110			3839
120			3805

For Test 9, opening ratio of the proportional wastegate and the corresponding average mass flow rate, K and discharge coefficient values are given in Table C.5.

Table C.5. Average K , C_d , and $\dot{m}$ corresponding to valve opening area from test 9.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
40	1,10	69,9	3856
50	1,12	71,16	4438
60	1,15	71,73	4852
70	1,17	72,01	5251

For Test 10, opening ratio of the proportional wastegate and the corresponding average mass flow rate, K and discharge coefficient values are given in Table C.6. In this test, the pressure of the water tank was increased, and the test was conducted at a high mass flow rate. Throughout the test, the discharge coefficient of the proportional valve remained constant.

Table C.6. Average K , C_d , and $\dot{m}$ corresponding to valve opening area from test 10.

Opening Percentage (%)	K	C_d (%)	$\dot{m}$ (g/s)
80	1,11	67,3	7508