ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL

PRESSURE REGULATOR DESIGN FOR LIQUID PROPELLANT ROCKET ENGINES

M.Sc. THESIS

Çağrı İNANÇ

Department of Aeronautical and Astronautical Engineering

Aeronautical and Astronautical Engineering Programme

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MAY 2023

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ

SIVI YAKITLI ROKET MOTORLARI İÇİN BASINÇ REGÜLATÖRÜ TASARIMI

YÜKSEK LİSANS TEZİ

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Uçak ve Uzay Mühendisliği Anabilim Dalı

Uçak ve Uzay Mühendisliği Programı

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Çağrı İNANÇ, a M.Sc. student of ITU Graduate School student ID 511191191, successfully defended the thesis entitled “PRESSURE REGULATOR DESIGN FOR LIQUID PROPELLANT ROCKET ENGINES”, which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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To my family and my lovely partner,

FOREWORD

It is a great pleasure and a sense of accomplishment that I present this master thesis. I would like to express my heartfelt gratitude to my advisor, Prof. Dr. Alim Rüstem ASLAN for the invaluable support.

My deepest appreciation also goes to my family, whose unconditional love and encouragement have been my pillars of strength, and my friends, whose unwavering support and friendship have kept me motivated throughout this journey.

May 2023 Çağrı İNANÇ

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ABBREVIATIONS

CV	**:** Control Volume
DOF	**:** Degree of Freedom
LPRE	**:** Liquid Propellant Rocket Engine
MATLAB	**:** Matrix Laboratory
ODE	**:** Ordinary Differential Equation
RK	**:** Runge-Kutta Method

SYMBOLS

$\boldsymbol{\eta}$	: Kinematic Viscosity [m^2/s]
$\mathbf{A}$	: Orifice Area [m^2]
$\mathbf{c_d}$	: Discharge coefficient
$\mathbf{T}$	: Temperature, [K]
$\mathbf{T_{out}}$	: CV_{out} Temperature, [K]
$\mathbf{P_{out}}$	: Outlet Pressure, [Pa]
$\mathbf{V_{out}}$	: CV_{out} Volume, [m^3]
$\mathbf{T_{in}}$	: Inlet Gas Temperature, [K]
$\mathbf{P_{in}}$	: Inlet Pressure, [Pa]
$\mathbf{V_{in}}$	: CV_{in} Volume, [m^3]
$\mathbf{T_{sen}}$	: CV_{sen} Temperature, [K]
$\mathbf{P_{sen}}$	: CV_{sen} Pressure, [Pa]
$\mathbf{V_{sen}}$	: CV_{sen} Volume, [m^3]
$\boldsymbol{\theta}$	: Poppet's angle, [rad]
$\mathbf{P_{cr}}$	: Critical Pressure Ratio
γ	: Specific Heat Ratio
$\mathbf{c_p}$	: Specific Heat, [J/kg-K]
$\mathbf{R}$	: Gas Constant, [J/kg-K]]
$\dot{\mathbf{m}}$	: Mass Flow Rate, [kg/s]
$\dot{\mathbf{m}}_{out}$	: Mass Flow Rate through outlet, [kg/s]
$\dot{\mathbf{m}}_{sen}$	: Mass Flow Rate through sensing, [kg/s]
$\mathbf{d}_{or}$	: Diameter of orifice, [m]
$\mathbf{x}$	: Displacement of sliding parts, [m]
$\dot{\mathbf{x}}$	: Speed of sliding parts, [m/s]
$\ddot{\mathbf{x}}$	: Acceleration of sliding parts, [m/s^2]
$\mathbf{m_{tot}}$	: Mass of Sliding Parts, [kg]
$\mathbf{m_{sp}}$	: Effective Mass of Springs, [kg]
$\mathbf{m_{spring1}}$	: Mass of Piston Spring, [kg]
$\mathbf{m_{spring2}}$	: Mass of Poppet Spring, [kg]

$\mathbf{m_{poppet}}$: Mass of Poppet, [kg]

$\mathbf{m_{piston}}$: Mass of Piston, [kg]

$\mathbf{F_{net}}$: Net Force acting on sliding parts, [N]

$\mathbf{F_{springs}}$: Total Spring Force, [N]

$\mathbf{F_{spring1}}$: Piston Spring Force, [N]

$\mathbf{F_{spring2}}$: Poppet Spring Force, [N]

$\mathbf{F_{preload}}$: Preload Force from Springs, [N]

$\mathbf{F_{atm}}$: Force from Atmospheric Pressure, [N]

$\mathbf{F_f}$: Friction Force, [N]

$\mathbf{P_{atm}}$: Atmospheric Pressure, [Pa]

$\mathbf{A_{atm}}$: Area under effect of Atmospheric Pressure, [m^2]

$\mathbf{A_{sen}}$: Area of piston under effect of Outlet pressure, [m^2]

$\mathbf{A_{pis}}$: Area of piston under effect of Atmospheric Pressure, [m^2]

$\mathbf{A_{out}}$: Area of poppet under effect of Outlet Pressure, [m^2]

$\mathbf{A_{pop}}$: Area of poppet under effect of Atmospheric Pressure, [m^2]

$\mathbf{x_{max}}$: Maximum Stroke of Poppet, [m]

$\mathbf{k_b}$: Spring Constant of Piston Spring, [N/m]

$\mathbf{k_s}$: Spring Constant of Poppet Spring, [N/m]

$\mathbf{x_b}$: Initial Compression of Piston Spring, [m]

$\mathbf{x_s}$: Initial Compression of Poppet Spring, [m]

$\boldsymbol{\rho}$: Density, [kg/m^3]

$\mathbf{V_0}$: Beginning Volume, [m^3]

$\mathbf{d_{pop}}$: Diameter of poppet, [m]

$\mathbf{d_{pis}}$: Diameter of piston, [m]

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PRESSURE REGULATOR DESIGN FOR LIQUID PROPELLANT ROCKET ENGINES

SUMMARY

Liquid propellant rocket engines (LPRE) are powerful and complex systems that play an important role in space exploration and flight technologies. These engines use a propellant pair, usually an oxidizer and a combustible propellant. The thrust obtained from the combustion reaction of these propellants, propels the rocket. Liquid propellant rocket engines have the advantages of high thrust, high efficiency and controllability. These engines are used in many application areas such as space exploration, satellite launches, deep space exploration and unmanned space missions. The design of these high-tech engines involves a series of complex engineering problems such as fuel supply, combustion control and cooling. Liquid propellant rocket engines are of great importance for the future of manned and unmanned space travel, and are becoming more efficient, reliable and powerful with continuously developed technologies.

Flow control components in liquid propellant rocket engines are an important part that ensures correct and regular feeding of liquid propellants into the combustion chamber of the engine. These components ensure that the propellants are filled into the tanks, directed to the combustion chamber at an appropriate pressure and that the combustion reaction takes place in a stable manner. In addition, flow control components must be flexible and able to function safely to adapt to pressure and temperature changes in the combustion chamber. The design of these components requires precise engineering calculations and material selections to optimize the system, minimize energy losses and increase engine performance. Flow control components are a critical element for the reliability, efficiency and successful operation of liquid propellant rocket engines, and are made more effective and durable with constantly improved technologies.

Pressure regulators are an essential component in liquid propellant rocket engines as they play a crucial role in maintaining a constant pressure of gas in the propellant lines. This, in turn, allows for the smooth operation of engine control valves, main propellant valves, gas generator valves, start systems, and most importantly, the pressurization of propellants for combustion.

Pressure regulators in liquid propellant rocket engines can have different types. In this study, four different types of pressure regulators are mentioned. The differences, advantages and disadvantages of these regulator types are listed.

The pressure regulator can be described as a small system consisting of many elements. In the study, the most important elements of the pressure regulator are listed one by one and the types of the elements are mentioned. The advantages and disadvantages of the types of these elements are listed.

The design and manufacturing of the pressure regulators in space applications are always special to meet rigid requirements. The first step of designing an effective

pressure regulator is mathematical model that considers the specific requirements and limitations of the application.

In this thesis, the design, mathematical and dynamic modeling of the pressure regulator is carried out. The developed mathematical and dynamic model uses isentropic equations and force balance equations of the mechanical parts to create a more accurate representation of the regulator's behavior under real-world conditions. It also reveals variable models that may arise for variable conditions such as variable outlet volume. For example, pressure regulators with variable outlet volume use a different mathematical model, while pressure regulators with fixed outlet volume use a different set of equations. Using these developed mathematical and dynamic models, simulations are made for three different regulators with MATLAB Simulink.

In this study, it is assumed that the pressure regulator has three control volumes. These control volumes are; inlet control volume, outlet control volume and sensing control volume.

As the second step of the study, the requirements for designing the pressure regulator are defined. These requirements include important parameters such as inlet pressure, set pressure and mass flow rate. In addition to these parameters, the parameters of the working fluid also must be defined. Nitrogen and helium gases are used as working fluids in this study.

In the third step of the study, the mechanical parameters of the pressure regulator should be determined. In order to determine the mechanical parameters, static calculations are made by using the mathematical model and the parameters are roughly calculated. Then the dynamic model on MATLAB Simulink is used with the optimization tool to obtain the optimum parameters. Optimum parameters are the inputs of the pressure regulators for the simulations.

In this study, calculations of mechanical parameters were made only for the first regulator and other regulators were used to validate the dynamic model by comparing simulation and test results.

Tests of three different pressure regulators are carried out in accordance with the requirements of each of them in the established pneumatic test setup. The data of these tests are collected from the pressure sensors on the line just before and after the pressure regulators and from the mass flow sensor in the line.

In order to validate the accuracy of these developed mathematical and dynamic models, the simulations are made in MATLAB Simulink using the inputs in the test. And the results of the simulations obtained from this model is compared with the test results obtained from there different pressure regulators.

After validation of the developed model, the simulations are made for three different regulators. The requirements and design inputs for each regulator are specified separately. These regulators differ in terms of dimensions, set pressure and mass flow rate variations, and the fluid they regulate. Simulations are performed with four different scenarios which include variations in inlet pressure, mass flow rate, and outlet volume for each regulator. All of the results of scenarios were compared among themselves for each regulator. In the results obtained, the effects of varying inlet pressure, mass flow rate and outlet volume on the outlet pressure of the regulator are observed.

The model developed in this study provides valuable insight into how pressure regulators behave under different conditions and could help improve the design and performance of this critical component in future space applications.

SIVI YAKITLI ROKET MOTORLARI İÇİN BASINÇ REGÜLATÖRÜ TASARIMI

ÖZET

Sıvı yakıtlı roket motorları (SYRM), uzay keşfi ve uçuş teknolojilerinde önemli bir rol oynayan güçlü ve karmaşık sistemlerdir. Bu motorlar, genellikle oksitleyici ve yanıcı yakıt olmak üzere bir yakıt çifti kullanırlar. Bu yakıtların yanma reaksiyonundan elde edilen itki roketi hareket ettirir. Sıvı yakıtlı roket motorları, yüksek itme kuvveti, yüksek verimlilik ve kontrol edilebilirlik avantajlarına sahiptir. Bu motorlar, uzay araştırmaları, uydu fırlatmaları, derin uzay keşifleri ve insansız araçlar gibi birçok uygulama alanında kullanılmaktadır. Yüksek teknoloji gerektiren bu motorların tasarımı, yakıt beslemesi, yanma kontrolü ve soğutma gibi bir dizi karmaşık mühendislik problemlerini içerir. Sıvı yakıtlı roket motorları, insanlı ve insansız uzay seyahatlerinin geleceği açısından büyük öneme sahiptir ve sürekli olarak geliştirilen teknolojilerle daha verimli, güvenilir ve güçlü hale gelmektedir.

Sıvı yakıtlı roket motorlarında akış kontrol bileşenleri, sıvı yakıtların doğru ve düzenli bir şekilde motorun yanma odasına beslenmesini sağlayan önemli bir parçalardır. Bu akış control bileşenleri; yakıtın tanklara doldurulmasını, uygun bir basınçta yanma odasına yönlendirilmesini ve yanma reaksiyonunun istikrarlı bir şekilde gerçekleşmesini sağlar. Ayrıca, akış kontrol bileşenleri, yakıtın yanma odasındaki basınç ve sıcaklık değişikliklerine uyum sağlamak için esneklik göstermeli ve güvenli bir şekilde işlevini sürdürebilmelidir. Bu bileşenlerin tasarımı, sistemi optimize etmek, enerji kayıplarını en aza indirmek ve motorun performansını artırmak için hassas mühendislik hesaplamaları ve malzeme seçimlerini gerektirir. Akış kontrol bileşenleri, sıvı yakıtlı roket motorlarının güvenilirliği, verimliliği ve başarılı bir şekilde çalışabilmesi için kritik bir unsur olup, sürekli geliştirilen teknolojilerle daha etkin ve dayanıklı hale getirilmektedir.

Basınç regülatörleri, yakıtların sabit bir basınçta yanma odasına ulaşmasında çok önemli bir rol oynadıklarından, sıvı yakıtlı roket motorlarında önemli bir akış kontrol bileşenidir. Bu da motor vanalarının, ana yakıt valflerinin, gaz jeneratoru valflerinin, başlatma sistemlerinin düzgün çalışmasına ve en önemlisi yanma için yakıtların basınçlandırılmasına olanak tanır.

Sıvı yakıtlı roket motorlarındaki basınç regülatörleri farklı tiplere sahip olabilirler. Bu çalışmada dört farklı basınç regülatörü tipinden bahsedilmektedir. Bu regülatör tiplerinin birbirlerinden farkları, avantajları ve dezavantajları listelenmektedir.

Basınç regülatörü bir çok parçadan oluşan ufak bir sistem olarak nitelendirilebilirler. Gerçekleştirilen çalışmada basınç regülatörüne ait en önemli parçalar, tek tek listelenmekte ve parçanın türlerinden bahsedilmektedir. Belirtilen bu parçaların tiplerine ait avantajlar ve dezavantajlar listelenmektedir.

Uzay uygulamalarındaki basınç regülatörlerinin tasarımı ve üretimi, zorlu gereksinimleri karşılamak için daha özeldir. Etkili bir basınç regülatörü tasarlamanın ilk adımı, uygulamanın özel gereksinimlerini ve sınırlamalarını dikkate alan matematiksel modeldir.

Bu tezde SYRM'de kullanılan akış kontrol bileşenlerinden basınç regülatörünün ilk olarak basınç regülatörünün matematiksel ve dinamik modellemesi gerçekleştirilmektedir. Matematiksel ve dinamik modeli oluşturulan regülatörün gerçek dünya koşulları altındaki davranışının daha doğru bir temsilini oluşturmak için izentropik denklemleri ve mekanik parçaların kuvvet eşitliği denklemlerini kullanılmaktadır. Ayrıca değişken koşullar için ortaya çıkabilecek değişken modelleri de ortaya koymaktadır. Örneğin, değişken çıkış hacmine sahip basınç regülatörlerinde farklı bir matematiksel model kullanılırken, sabit çıkış hacmine sahip basınç regülatörlerinde farklı bir denklem seti kullanılmaktadır. Geliştirilen bu matematiksel ve dinamik modelleri kullanarak MATLAB Simulink ile üç farklı regülatör için simülasyonlar yapılmaktadır.

Bu çalışmada basınç regülatörünün üç adet control hacmine sahip olduğu kabul edilmiştir. Bu control hacimleri; giriş kontrol hacmi, çıkış kontrol hacmi ve algılama kontrol hacmi olarak sıralanabilir.

Çalışmanın ikinci adımı olarak basınç regülatörünün tasarlanabilmesi için gerekli olan gereksinimler tanımlanmaktadır. Bu gereksinimler giriş basıncı, ayar basıncı ve kütlesel debi gibi önemli girdiler içermektedir. Bu girdilere ek olarak çalışılan akışkana ait parametrelerin de tanımlanması gerekmektedir. Bu çalışmada akışkan olarak azot ve helyum gazları kullanılmaktadır.

Çalışmanın üçüncü adımında ise basınç regülatörüne ait mekanik parametreler belirlenmelidir. Mekanik parametreleri belirleyebilmek için matematiksel model üzerinden statik hesaplamalar yapılarak parametreler kabaca hesaplanır. Daha sonra optimum parametreleri elde etmek için MATLAB Simulink üzerindeki dinamik model, optimizasyon aracı ile birlikte kullanılır. Bu çalışmada, hesaplamalar sadece birinci regülatör için yapılmış ve diğer regülatörler, simülasyon ve test sonuçları karşılaştırılarak dinamik modeli doğrulamak için kullanılmıştır.

Üç farklı basınç regülatörünün, kurulan pnömatik test düzeneğinde her birinin kendi gereksinimleri doğrultusunda testleri gerçekleştirilmektedir. Bu testlere ait veriler regülatörlerin giriş ve çıkışlarındaki basınç sensörelrinden ve hattaki kütlesel debi sensöründen toplanmaktadır.

Geliştirilen matematiksel ve dinamik modellerin doğruluğunu gözlemlemek için testteki girdileri kullanarak MATLAB Simulink ile simülasyonlar yapılmaktadır. Ve bu simülasyonlardan elde edilen sonuçlar ile üç farklı basınç regülatörlerinden elde edilen test sonuçları karşılaştırılmaktadır.

Geliştirilen modelin doğrulaması yapıldıktan sonra üç farklı regülatör için farklı simülasyonlar yapılmaktadır. Her regülatör için gereksinimler ve tasarım girdileri ayrı ayrı belirtilmektedir. Bu regülatörler boyutlar, ayar basıncı, kütlesel debi değişimleri ve çalıştıkları akışkan açısından farklılık göstermektedir. Simülasyonlar, her bir regülatör için giriş basıncı, kütlesel debi ve çıkış hacmindeki değişimleri içeren dört farklı senaryo ile gerçekleştirilmiştir. Tüm senaryoların sonuçları her bir regülatör için kendi aralarında karşılaştırılmıştır. Elde edilen sonuçlarda değişken giriş basıncı, kütlesel debi ve çıkış hacmi altında regülatörün çıkış basıncı gözlenmiştir.

Bu çalışmada geliştrilen model, sıvı yakıtlı roket motorlarında kullanılan basınç regülatörlerinin farklı koşullar altında nasıl davrandığına dair değerli bilgiler sağlar ve bu gelecekteki uzay uygulamalarında bu kritik bileşenin tasarımını ve performansını iyileştirmeye yardımcı olabilir.

1. INTRODUCTION

1.1 Purpose of Thesis

The purpose of this thesis is to design a pressure regulator and develop a valid mathematical and dynamic model for pressure regulators in liquid propellant rocket engines. After developing the mathematical and dynamic model, the validation of it is another important part of the study. The model whose validation is completed is used for different pressure regulators in different conditions to see the performance of pressure regulators on variable inlet pressure, mass flow rate and outlet volume.

1.2 Thesis Overview

First a mathematical model for the pressure regulator is obtained by using ideal gas equations, isentropic equations, and force balance equations. After developing a mathematical model, a pressure regulator is designed for specific requirements. Then, the dynamic model is used in MATLAB Simulink to simulate the regulators performance. The simulations are compared with test results and the mathematical and dynamic model is validated. By using these valid models, the simulations for different regulators are made. For each regulator, different conditions which is variable inlet pressure, mass flow rate and outlet volume are applied to simulations to compare the effects of each condition on the performance of the regulator.

1.3 Literature Review

Liquid Propellant Rocket Engines (LPREs) are very complex. However, they are the key element of reaching the orbit because of their high ISP, thrust control and flexibility in burning time. This gives launch vehicle designers a lot of flexibility to build a cheap and reliable launch system.

Flow control valves are critical components of liquid propellant rocket engines that regulate the flow of propellants to control the engine's thrust and performance. These valves are typically located in the engine's propellant feed lines and are used to adjust the flow rate of the propellants to achieve the desired thrust level and engine efficiency. There are several components which control the motion of fluids through them. Such as;

- shut-off valves,
- control valves,
- pressure regulators,
- relief valves,
- check valves,
- servo valves,
- explosive valves

They are used extensively in liquid propellant rocket systems. All of these fluid control components have an important role in pressurizing systems for propellant feed of reaction controls, in main-engine control systems, and in vehicle tank pressurization systems. For instance, in the Saturn V vehicle, there are 34 pressure regulators, 62 relief valves, 286 check valves, and 64 burst disks, including explosive units. [1]

The task of the pressure regulator is to regulate and maintain the outlet pressure against variable inlet pressure, temperature, and outlet mass flow rate. They are generally involved in reducing the high inlet pressure in a system to the desired outlet pressure. For example, the ullage of the propellant tank is pressurized and kept at a constant value for steady state engine performance by using a pressure regulator. [2]

The pressure regulators in space applications are almost always special units designed specifically to meet unique requirements. The rigid requirements, with particular emphasis on size, weight, and efficient operation of the total system, have limited the use of commercial of the shelf (COTS) products. [1]

Changes in the regulated pressure have an impact on the valve timing and sequencing during the start and shutdown operations of engines in pneumatic control systems. The malfunctioning of a valve in a Saturn engine, which caused high-pressure gas to leak into the helium control system, resulted in an abnormally high control pressure that altered the shutdown timing to the extent that the engine suffered severe damage. [1]

In the past, there was not enough information about the mathematical modelling of pressure regulators. Therefore, the developing of a pressure regulator was a trial-and-error process for a time. In 1961, Tsai and Cassidy published a journal describes the experimental and theoretical investigation of the dynamic behavior of a simple pressure regulator. They aimed to determine the system's response to changes in the inlet pressure, outlet pressure, and flow rate. [3,4] The system includes a diaphragm type pressure regulator connected to air compressor. The experiments are used to combine with theoretical analysis in order to develop a mathematical model of pressure regulator. Then, the experimental results are compared to the model's predictions for validation of its accuracy. The one of the results of this journal is the behavior of pressure regulator is highly dependent on the diaphragm's properties, such as mass and stiffness. They identified lots of key parameters which affects the behavior of the pressure regulator such as inlet and outlet pressure and flow rate. The whole work of them provides very important insights for the design and control of pressure regulators.

In 1984, Krigman, write about the selection criteria and design of the pressure regulators for the required applications. [5] The paper covers the principles of pressure regulation, in addition to that, it includes the various types of pressure regulators such as direct acting, pilot operated and dome loaded regulators. The advantages and disadvantages of each type of the regulators are discussed in this paper. Krigman provides the important factors to select a pressure regulator as inlet, outlet pressure rang, flow rate and the type of gas. Also, the paper has the information about how to set up a pressure regulator system for a safe and reliable operation.

In 1989, the overview of pressure regulators is written by Brasilow [6]. The paper explains how pressure regulator works and its components. Different types of pressure regulators are discussed in the paper such as single stage and two stage regulators and their advantages and disadvantages.

In 1999, Delenne, Mode and Blaudez published a paper about modelling and simulation of pressure regulators.[7] The importance of accurate modeling and simulation of pressure regulator is discussed by them. They created a mathematical model based on energy conservation and the ideal gas law. After that, the behavior of pressure regulator is simulated under different conditions such as varying inlet pressure, outlet pressures and flow rates by using this mathematical model.

The paper aims that the model and simulation can be used for optimizing the design of pressure regulators and developing more efficient control algorithms for pressure regulators.

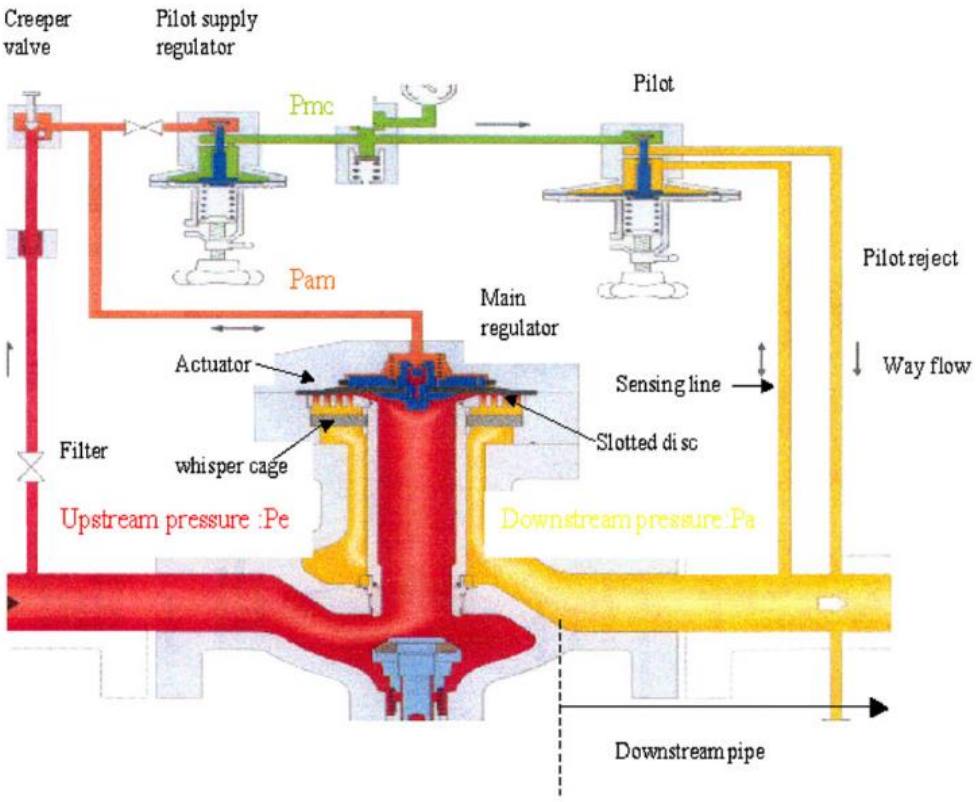

Figure 1.1: Schematic of pilot-controlled diaphragm type regulator [7]

After one year, Delenne and Mode added the simulation of pressure oscillations to his work. [8] They pointed out that the oscillations on the pressure regulator can cause instability and poor performance in the system. Because of that, they develop a model that predict oscillations on pressure regulators. The mathematical model on this paper includes the dynamics of the diaphragm. And again, they simulated the pressure regulator performance under different conditions to predict and prevent accurately the oscillations in the pressure regulator.

Vujic, in 2001, created a dynamic simulation process for pressure regulators and derived the mathematical equations. [10] Vujic notes that the most of the models are developed for pressure regulators have focused on steady state behavior and not capturing the dynamic of the system. The Laplace transforms are used to obtain the transfer functions of the regulator which is used to analyze the stability and transient response of the pressure regulator.

Then, the validation of the dynamic model is made by using experimental results and it demonstrated that the model accurately predicts the transient behavior.

In 2008, Zafer and Luecke write a mathematical analysis of the stability of pressure regulators.[11] The paper gives information about developing a dynamical model to analyze its performance and identify the factors that affect it of a pressure regulator. They used the experimental results of a regulator to determine the values of the parameters. The main aim of the paper is creating a scientific instrument to determine the most important parameters on the stability of the pressure regulator. Some of the effective parameters can be listed as opening area, spring constant and discharging coefficient.

2. BASICS OF PRESSURE REGULATORS

For a pressure regulator with gas medium, the main purpose is to be able to provide the flow which is demanded by the rest of the system. While providing the flow demanded, the pressure regulator needs to maintain the outlet pressure stable for a safe and reliable operation. A pressure regulator provides the demanded flow automatically. When pressure regulators were not in use, to have a safe system, the pressure of the downstream needs to be checked during the operation. And if the pressure is increasing, the inlet valve is adjusted to close manually until the pressure is decreasing.

In the Figure 2.1, a basic system which includes the pressure regulator and the valve is showed.

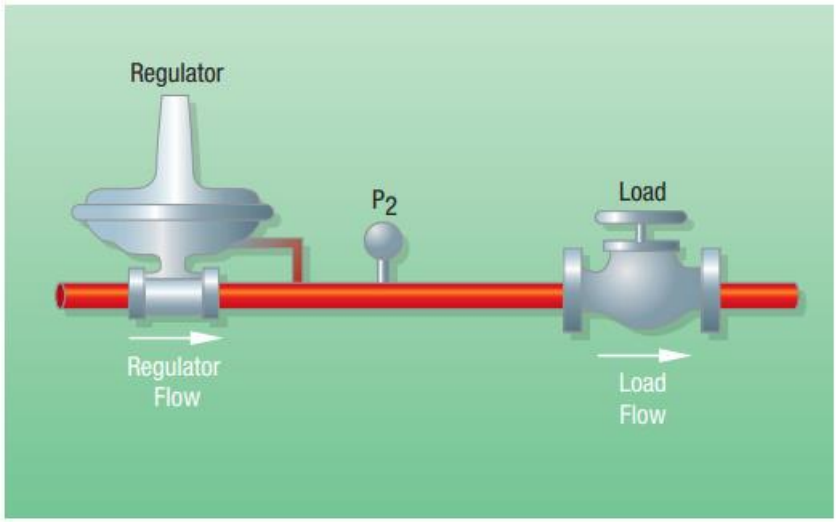

Figure 2.1: Tank pressurization system

The valve creates the flow demand with its opening area and the pressure regulator must fulfill this demand. If the flow is less than the demand, then the system pressure is getting lower, and if the flow is higher than the demand, then the system pressure is getting higher. This can be summed up as the pressure of the system is maintained when the pressure regulator ensures the flow demanded.

For an ideal case, if the pressure regulators can provide the flow immediately without any fluctuations, then there is no need to understand the dynamics of the pressure regulators. However, in real world, this perfect situation cannot be experienced. There are always some instabilities and fluctuations occur in the system. These instabilities can arise due to several reasons. Therefore, understanding the dynamics of the pressure regulator is important to control the behavior of it, and to predict a safe and reliable operation.

There is a restricting element to control the flow in the pressure regulator as shown Figure 2.2. The restricting element works with the loads on it, and change the area of the orifice to provide the flow by the effects of the forces. Mostly, for a pressure regulator, the downstream pressure getting higher means the restricting element tends to close to ensure the pressure getting reducing to the nominal value of it.

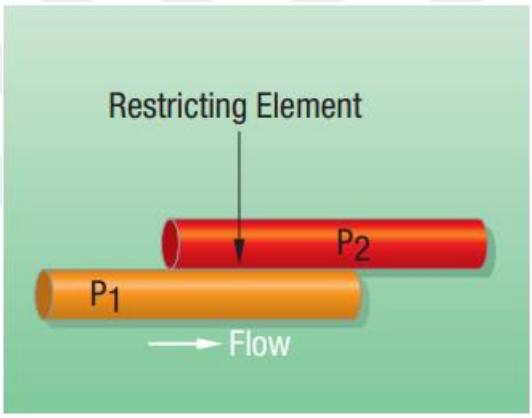

Figure 2.2: Schematic of restricting element's function [11]

With the reference force inside the regulator, the sensing element and the restricting element regulate the flow. The outlet pressure can be adjusted by preloading to the reference force source, and this preload can have an adjustable mechanism as shown in Figure 2.3. This mechanism allows to control the downstream pressure by loading up or down.

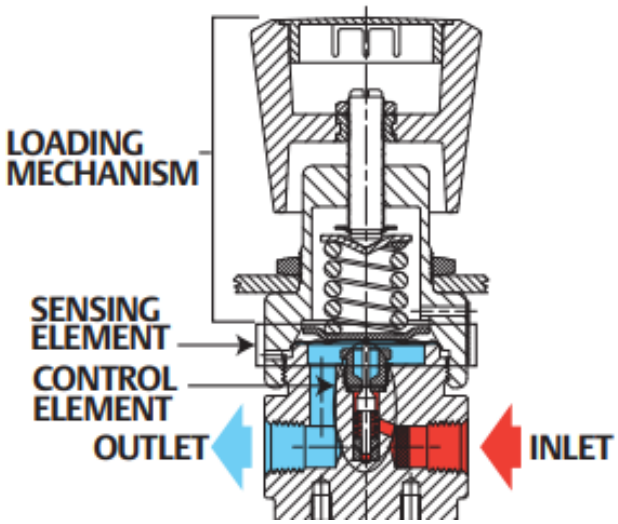

Figure 2.3: Basic regulator schematic

2.1 Regulator Types

The beginning to design of the pressure regulator is choosing the type of the pressure regulator by the designer. The design requirements should be determined well before deciding on the design of the pressure regulator. These requirements may include environmental conditions, performance and physical requirements. The knowledge of flight and transportation vibration, mechanical shock, acceleration values and thermal loads provides important details for design. A product designed without using this information could cause loss of time and money when it cannot adapt to flight conditions even if it works on the ground. [1]

The selection of the pressure regulator type is a complex operation. While designing the pressure regulator, the designer should start with the simplest solution that meets the requirements. If this design has problems, for example a heavy product or a product non resisting to vibration, then one step more complex design may be preferred by the designer.

In this study, 4 different the most commonly used pressure regulators in LPREs are examined:

- Direct acting
- Dome loaded
- Pilot operated
- Bang Bang type

2.1.1 Direct acting regulators

The direct acting regulator is the simplest type of all. It has three essential components: a sensing element (such as a diaphragm), an adjusting valve unit (consisting of a poppet and sealing element), and a reference load (a spring). Direct-acting regulators reduce pressure with a spring-loaded pre-load applied to the valve. The pressure acting on the poppet is multiplied by the poppet's surface area and pushes the poppet to overcome the spring force. The poppet moves and gas begins to flow through the open channel. The flowing gas flows through a sensing hole into the volume of the diaphragm, exerting an effect on the diaphragm. The diaphragm pushes the poppet again, and the poppet reaches equilibrium to reach the desired pressure value. The process described in Figure 2.4 can be seen in more detail:

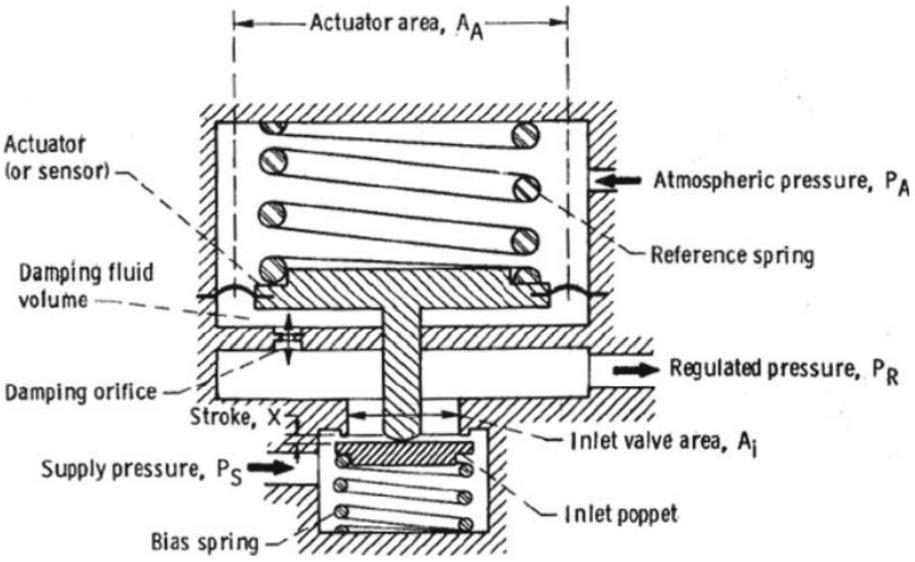

Figure 2.4: Direct acting pressure regulator [1]

2.1.2 Dome loaded regulators

A dome-loaded type regulator does not rely on a mechanical component for its load source. Instead, it uses a constant reference pressure exerted on the dome above the sensing element by a pilot gas. The pilot gas, which can be the same as the gas whose pressure is being regulated, is fed to the dome through a small loader valve. This valve lowers the supply pressure to the desired reference pressure. The process described in Figure 2.5 can be seen in more detail:

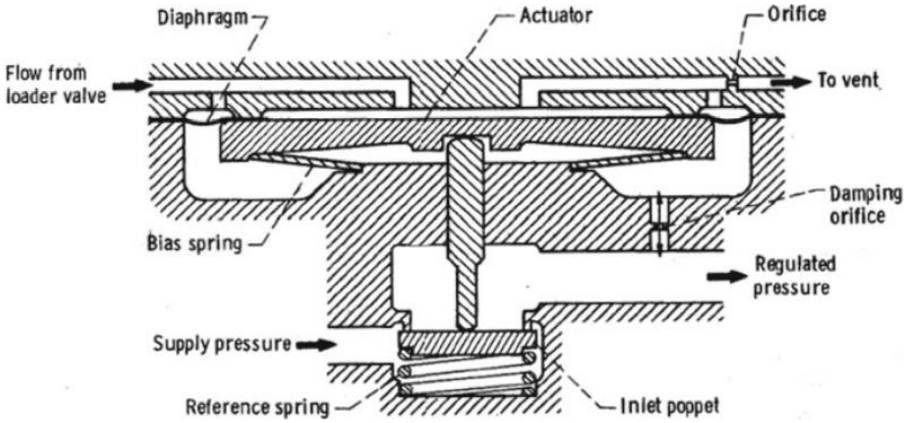

Figure 2.5: Dome Loaded Regulator [1]

2.1.3 Pilot operated regulators

In pilot-operated regulators, a steady stream of control-pressure gas (known as bleed flow) is provided to the restricting valve actuator. The pilot valve or controller detects the regulated pressure and adjusts the bleed flow from the actuator. The change in control pressure that occurs as a result is proportionate to alterations in the regulated pressure.

While designing pressure regulators, sensing the outlet pressure is very important. Because of that, pilot operated regulators are invented. We can see the pilot operated regulators as two different regulators which are combined. In general, the main regulator is the bigger, while other is the pilot one. The pilot one control the main one with sensing the outlet pressure and adjusting the load pressure. [12]

The main advantage of the pilot operated regulator is maintaining more stable outlet volume by applying the important design points to the pilot regulators. However, as a disadvantage, we can say the pilot operated pressure regulator may oscillate and show unstable performance. The process described in Figure 2.6 can be seen in more detail:

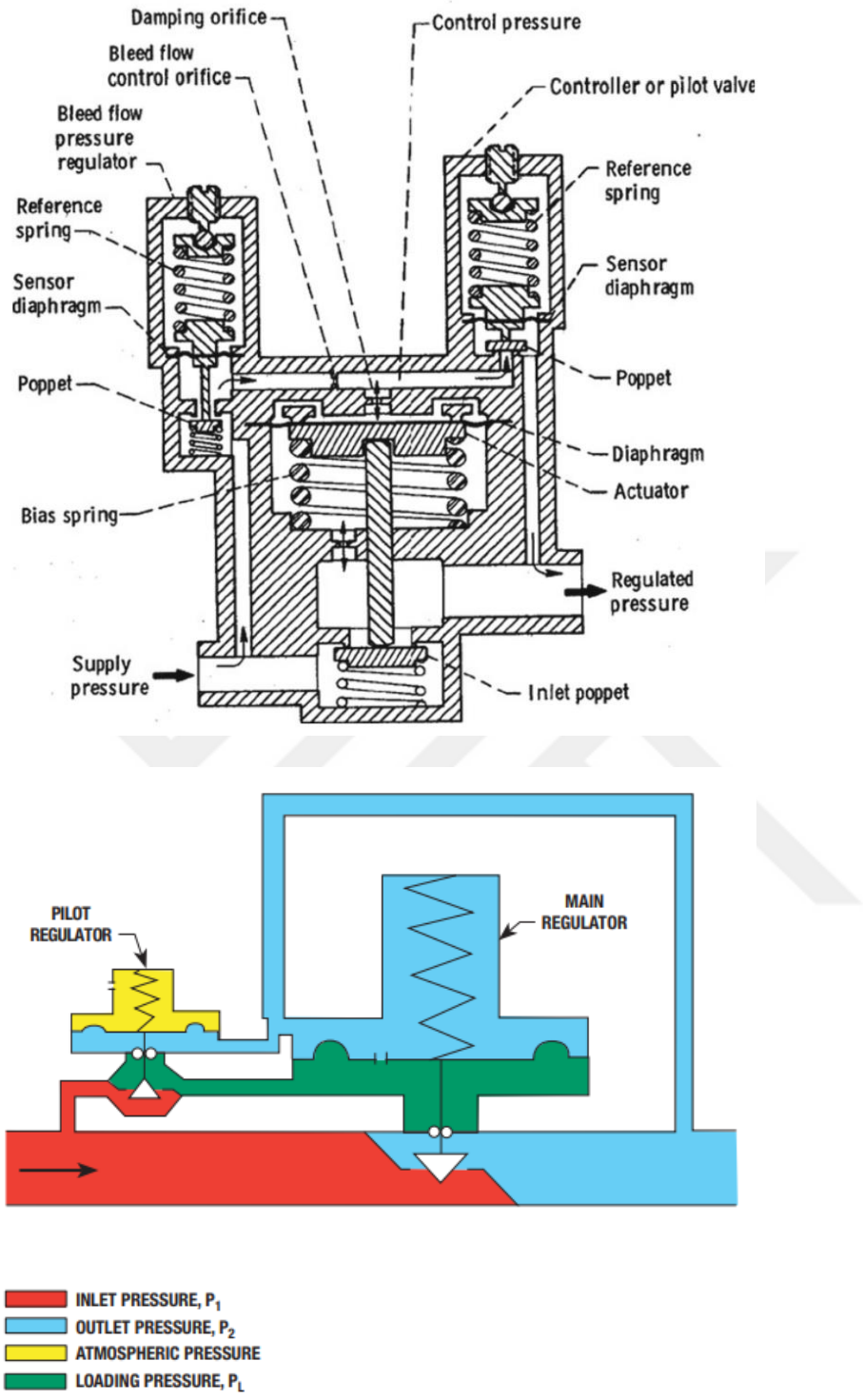

Figure 2.6: Pilot operated regulator [1, 12]

2.1.4 Bang bang regulators

Bang bang regulators are not actually a regulator that known. They are combined solenoid valves. They have two positions as open and close. The solenoids are getting signal to open or close from the pressure sensors on the downstream.

For example, when the pressure on the propellant tanks dropped, they are opened until reaching the set pressure. Disadvantages of this type of regulators is having a non-stable downstream pressure, since there is not a continuous flow into downstream. A graph of the downstream pressure of a bang bang regulator is shown in Figure 2.7.

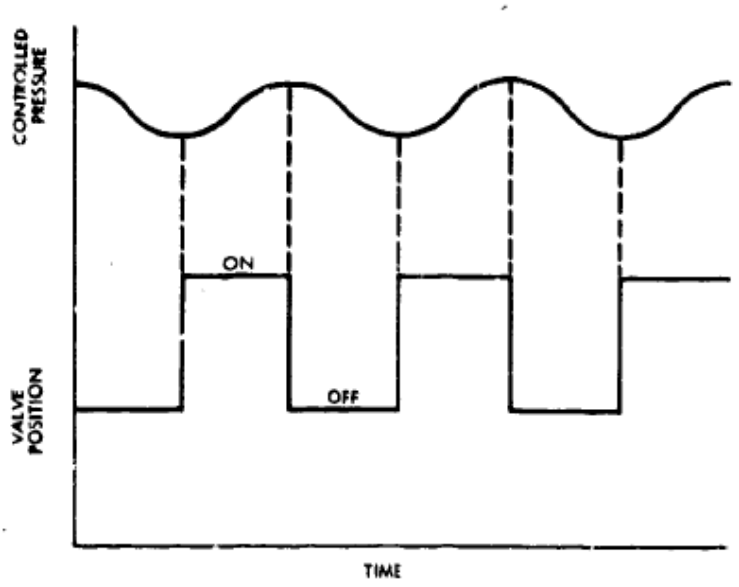

Figure 2.7: Downstream pressure of bang bang regulators

2.2 Regulator Elements

The pressure regulators have lots of elements inside of it. However, there are three main elements which affect valve design and performance extremely. These three elements are restricting element, reference loading element and sensing element.

2.2.1 Restricting element

The main function of the regulator's restricting element is to manage the fluid flow according to the system's demand and maintain the desired regulated pressure. To determine the necessary flow area, an equivalent orifice size is calculated using the equation for isentropic compressible flow based on the rated flow at the most challenging operating conditions. The restricting element’s seat diameter and stroke are then adjusted to meet the required flow area. If the restricting element is closed fully, then the flow is stopped.

Regulator restricting element design is determined by flow demand, pressure drop, leakage requirements, type of fluid, temperature extremes, and the preference of the designer.

The restricting element is mostly named as poppet. Some of the poppet designs are showed in Figure 2.8.

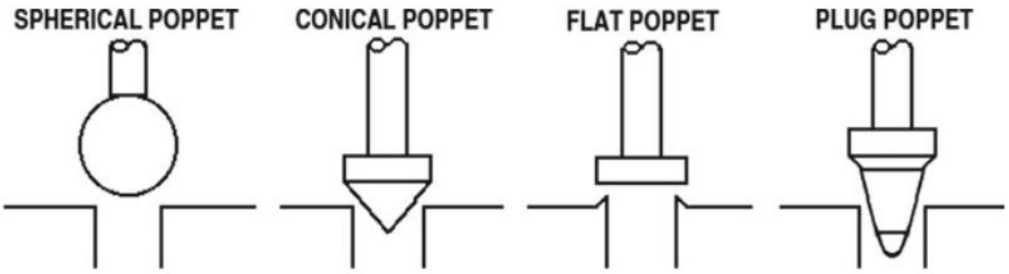

Figure 2.8: Poppet types [1]

There are two different type of restricting element which affect mechanic equations:

- Balanced poppet
- Unbalanced poppet

The balanced poppet shown in Figure 2.9 is the design which is not affected from the inlet pressure. It is sealed from the inlet pressure not to get backside of it. Also, the body of poppet has same diameter with poppet seat. Therefore, inlet pressure affects the same area of the poppet from top and bottom and there is not any net force by inlet pressure on the poppet. Main advantage of it is not affected from varying inlet pressure. However, it's more expensive to manufacture.

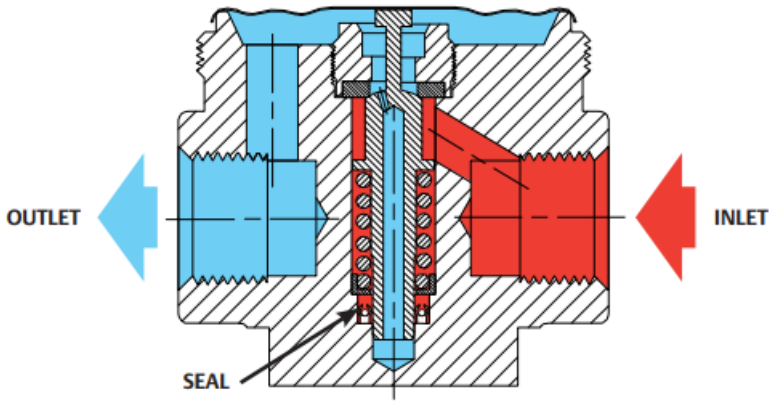

Figure 2.9: Balanced Poppet [12]

The unbalanced poppet shown in Figure 2.10 is the simplest design which inlet pressure affects. Since the net force on the poppet by the inlet pressure is not zero, inlet pressure decides the characteristic of the pressure regulator. As the inlet pressure increases, the load on the poppet increases. And the load on the poppet decreases when the inlet pressure decreases. Therefore, the outlet pressure cannot have a stable value with varying inlet pressure.

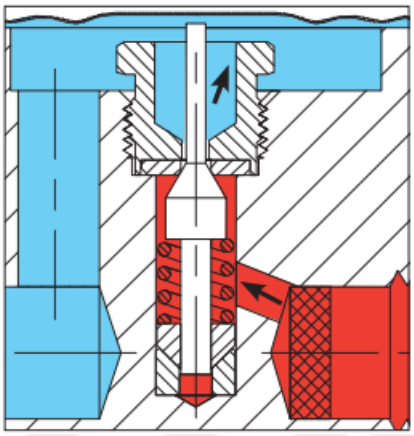

Figure 2.10: Unbalanced poppet [12]

Poppet provides impermeability through the flexible elastomer part on it or on the surface it contacts. The geometrical alignment of the contacting surfaces of the poppet and the elastomer part and keeping the surface roughness in the area where they are seated at low levels are important in minimizing the amount of internal leakage. The contact geometry directly affects the cross-sectional area and thus the arrangement characteristics as the poppet is opened.

2.2.2 Reference loading element

This element determines what the outlet pressure of the pressure regulator will be. In pressure regulators, there is the outlet pressure on one side of the sensing element and the reference loading element on the other. The reference loading element is responsible for applying a force in the opposite direction to the force formed by the output pressure on the sensing element. The net force consisting of these two force differences provides the movement of the poppet. Spring or dome is used as reference force element in pressure regulators as shown Figure 2.11. When the spring is used, the outlet pressure is adjusted by preloading the spring, while in the use of the dome,

the outlet pressure is adjusted by adjusting the pressure of the gas on the other side of the dome.

By increasing the amount of preload in the spring or by increasing the pressure of the gas in the dome, the outlet pressure of the regulator is increased. The use of springs is more preferable, especially in designs in space applications, in terms of safety, sealing, production and assembly convenience. In electronic pressure regulators, it has been observed that the load is created by using a solenoid or piezo.

Figure 2.11: Reference loading elements (left – spring, right – dome) [1]

2.2.3 Sensing element

The sensing element is responsible for detecting any variations in the pressure downstream or at the outlet side of the regulator. There are three common types of sensing elements: Diaphragm, Piston, Bellows.

Diaphragm is a better choice for low outlet pressure and tight tolerances because it is more sensitive. It is also cheaper and simpler, but it is less preferred because of non-linearity. It does not provide a fixed sensing surface area, and it has lower pressure limits. It is shown in Figure 2.12

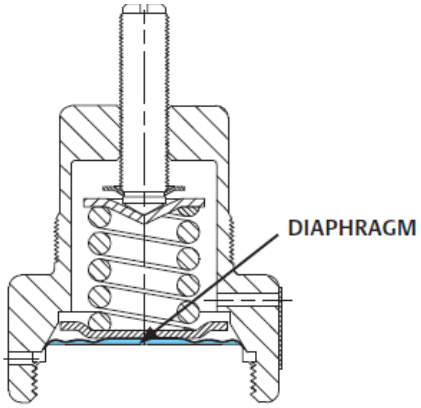

Figure 2.12: Diaphragm type sensing element [12]

In Figure 2.13, piston is shown. Piston is less sensitive than the diaphragm. This difference in sensitivity is due to the fact that the sealing elements required in the use of pistons increase friction. As friction increases in the moving parts of the pressure regulator, the response speed and sensitivity of the system decreases. However, having higher pressure limits and a fixed sensing surface area make it advantageous.

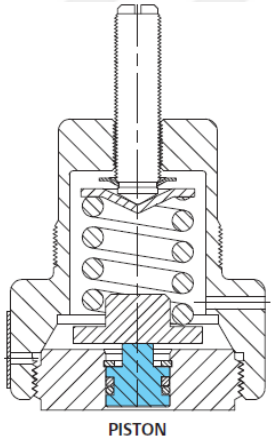

Figure 2.13: Piston type sensing element [12]

Below is successful in sealing and high sensitivity. However, it is not preferred very often because it is expensive, difficult to manufacture and unstable. The process described in Figure 2.14 can be seen in more detail:

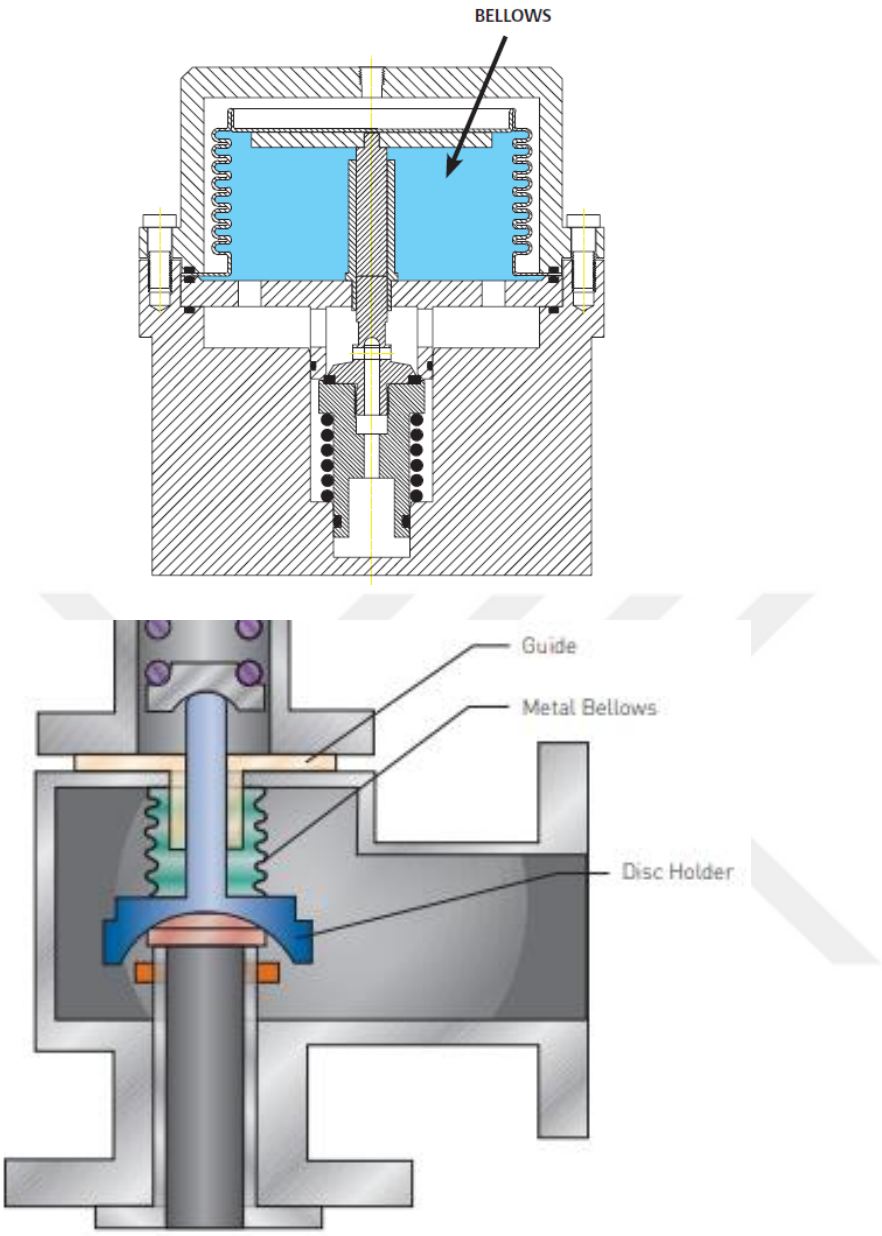

Figure 2.14: Bellow type sensing element [11,12]

3. MATHEMATICAL MODELING

The approach to develop a pressure reducer device is not creating a model for a long time. In order to reduce pressure, engineers have been using trial and error method before. In 1961, Tsai and Cassidy study to create a mathematical model for both the nonlinear and the linearized problems, and analyzed the both results of the mathematical model with using experimental results they get from a working pressure regulator. [3]

The mathematical models for pressure regulators are based on the physical behaviors of compressible fluid flow. The model includes gas dynamics equations and mechanical system equations (equation of motion). In literature, it can be seen that while governing gas dynamic equations, the outlet volume assumed as fixed. There are lots of papers, such as [10, 13, 14, 15], includes a mathematical model for a mechanical pressure regulator with fixed outlet volume. In addition to that, all the performance tests after the manufacturing the designed pressure regulator are made based on fixed outlet volume. However, in rocket engines, the outlet volume can vary significantly due to decreasing the number of propellants on the tanks while consuming them during combustion. This study provides a mathematical model derived for both the fixed outlet volume and the variable outlet volume.

In a typical pressure regulator, there exists a difference in pressure between the inlet and outlet volumes of the system since the inlet pressure is higher than the outlet. This difference leads to the flow of mass from the inlet to outlet direction through the orifice, resulting in a net force on the regulator's sliding parts. As the outlet pressure is getting higher the net force dictates that the poppet moves towards the closing direction, which means reducing the orifice area. The movement of the poppet stops once the net force on the regulator's moving parts reaches zero. When the equilibrium is reached, the downstream pressure stays constant. In this state, the mass flow rate which is passing through the orifice and the required flow rate are equal and balanced.

For the purpose of creating a mathematical model, it is necessary to consider the mass flow rate through the orifices between control volumes, the mechanical equations for the sliding components of regulator, the pressure and temperature derivation of the control volumes. The assumptions, which are made to derive the equations, are listed below:

I. The thermodynamic processes of the medium are assumed to be adiabatic since gas process occurs faster and hence no heat transfer can take place.
II. The fluid behaves as an ideal gas.
III. The pressure forces on the poppet and bellow sensing element are calculated assuming uniform pressure distribution over the entire area.
IV. Sensing element, poppet and springs are considered to always move together as a single part.
V. Coulomb's law of friction is considered between sliding parts and their contact on the regulator's body.

In this study, it is assumed that the pressure regulator has three different control volumes. These volumes are listed below and shown in Figure 3.1.

1) Inlet volume
2) Outlet volume
3) Sensing volume

Inlet volume is the volume where upstream pressure is, outlet volume is the volume where the regulated pressure of downstream is, and sensing volume is the volume which provide damping to the system.

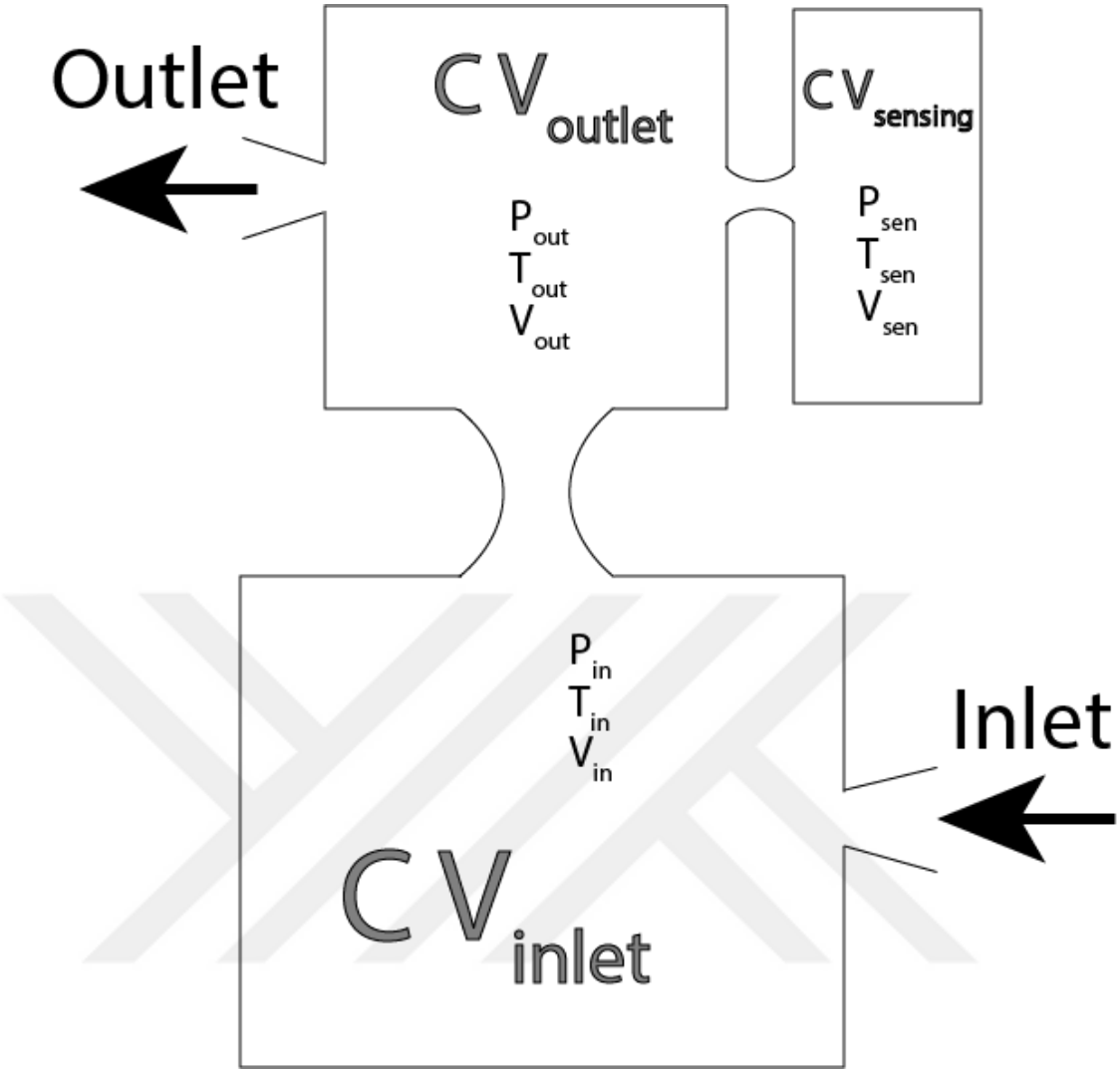

Figure 3.1: Control volumes of pressure regulators

3.1 Mass Flow Rate Equations

All of the control volumes have mass flow through them. The mass flow rate through an orifice for any gas can be calculated using the equation provided by White (2011) and Babu (2003), which takes into account the geometry of the orifice, gas properties, and pressure ratio. The flow can be categorized as either choked or not choked, depending on the pressure ratio, and the mass flow rate behaves differently in each case. [16, 17] In choked flow, the downstream pressure is extremely lower than the upstream pressure which cause the gas to reach the speed of sound at the orifice. In non-choked flow, the mass flow rate is proportional to the pressure drop and is not limited by the maximum velocity of the gas. [18]

The flow is choked when the pressure ratio (outlet pressure divided by inlet pressure) is less than the critical pressure ratio, which is a function of the specific heat ratio of the gas. The critical pressure ratio can be calculated using equation 3.1:

$$P_{cr} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma+1}} \tag{3.1}$$

Mass flow rate of the choked flow can be calculated using equation 3.2 when the pressure ratio is less than the critical pressure ratio:

$$\dot{m} = c_d\, A\, \frac{P_{in}}{\sqrt{R\, T_{in}}} \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma-1}} \sqrt{\frac{2\gamma}{\gamma + 1}} \quad when \quad \frac{P_{out}}{P_{in}} < P_{cr} \tag{3.2}$$

Mass flow rate of the non-choked flow can be calculated using equation 3.3 when the pressure ratio is higher than the critical pressure ratio:

$$\dot{m} = c_d\, A\, \frac{P_{in}}{\sqrt{R\, T_{in}}} \sqrt{\frac{2\gamma}{\gamma - 1}} \sqrt{\left(\frac{P_{out}}{P_{in}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma+1}{\gamma}}} \quad when \quad \frac{P_{out}}{P_{in}} > P_{cr} \tag{3.3}$$

where $A = \pi \left(d - x \sin(\theta) \cos(\theta)\right) x \sin(\theta)\right)$

The orifice area between inlet control volume and outlet control volume can be theorized as a lateral surface area of a truncated cone and the area is variable with the displacement of sliding parts of regulator. The lateral surface area of the truncated cone between poppet and its seat is shown in Figure 3.2:

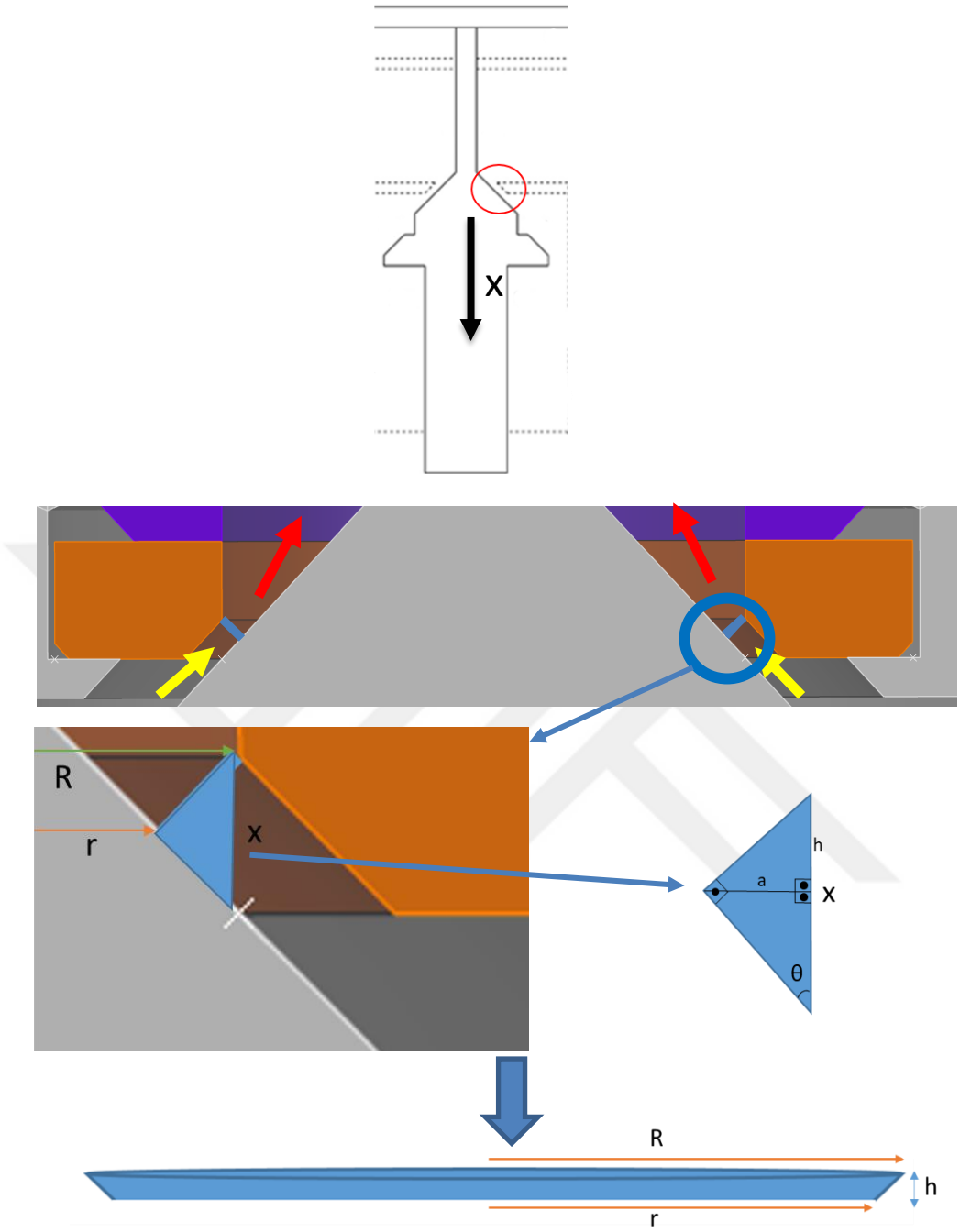

Figure 3.2: Orifice area (zoomed in)

The lateral surface area of a truncated cone can be calculated using equation 3.4 where R is the radius of the big base, r is the radius of the little base and the h is the slant height. [19]

$$Lateral\ Surface\ Area\ of\ Cone\ = \pi\ (r + R)\ \sqrt{(R - r)^2 + h^2} \tag{3.4}$$

For the poppet-seat system, R can be used as the radius of the seat, and r can be calculated as shown on Figure 3.2 by using equations 3.5, 3.6, 3.7 and 3.8.

$$R = \frac{d}{2} \tag{3.5}$$

$$r = R - a = \frac{d}{2} - a \tag{3.6}$$

$$a = x\ sin\ (\theta) cos(\theta) \tag{3.7}$$

$$h = x\ sin^2(\theta) \tag{3.8}$$

After substituting them to the equation of surface area, the orifice area equation is obtained as equations 3.9 and 3.10:

$$A = \pi\ (d - x \sin(\theta) \cos\ (\theta))\ \sqrt{(x \sin(\theta) \cos(\theta))^2 + (x \sin^2\ (\theta))^2} \tag{3.9}$$

$$A = \pi\ (d - x \sin(\theta) \cos\ (\theta))\ x \sin\ (\theta) \tag{3.10}$$

3.2 Force Balance Equations

The equation of motion is a fundamental concept in mechanics. This concept, which relates an object's position, velocity and acceleration as a function of time, is derived from Newton's second law of motion. Newton's second law of motion states that the acceleration of an object is directly proportional to the net force acting on it, and inversely proportional to its mass. This is expressed as the equation 3.11 [20]

$$\vec{F} = m\vec{a} \tag{3.11}$$

Where m is the mass, "a" is acceleration. Also, acceleration can be calculated as the second derivative of the position of the sliding parts which is "x" in equation 3.12.

$$\mathrm{F} = m\ddot{x} \tag{3.12}$$

The masses of the regulator system can be separated into three parts which are poppet's, piston's and the springs' mass. In a dynamic system which include springs in it, the one third of the mass of springs will be added to the total mass of the system.

Therefore, the effective mass of two springs which is used on the regulator system is shown in equation 3.13:

$$m_{sp} = \frac{m_{spring1}}{3} + \frac{m_{spring2}}{3} = \frac{m_{spring1} + m_{spring2}}{3} \tag{3.13}$$

Then, the total mass of the system can be used as shown in equation 3.14:

$$m_{tot} = m_{sp} + m_{poppet} + m_{piston} \tag{3.14}$$

The free body diagram is shown in Figure 3.3.

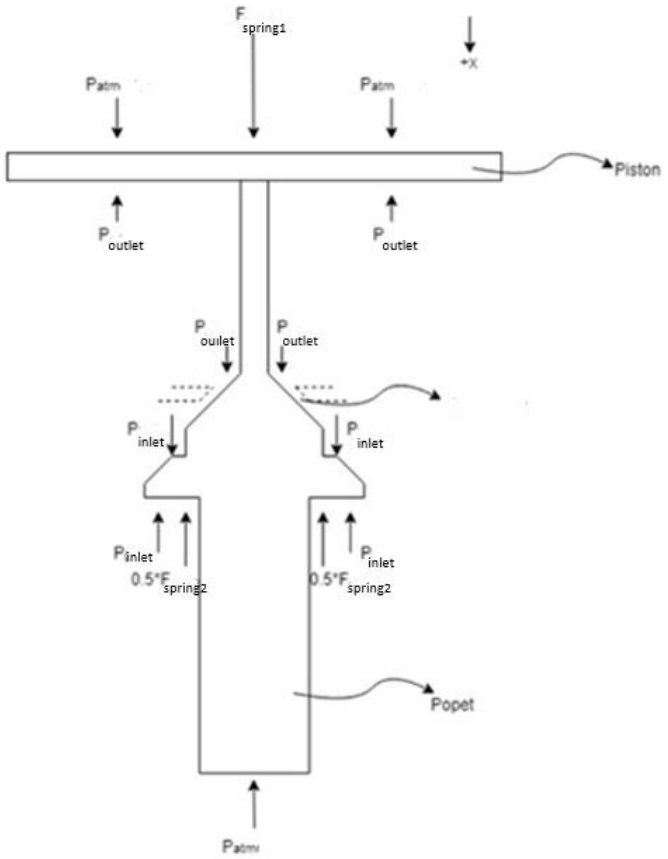

Figure 3.3: Free body diagram

As it is seen from the free body diagram, net force on the sliding parts can be written as equation 3.15:

$$\sum F = m_{tot}\ddot{x} = F_{springs} + P_{out}A_{outlet} + P_{atm}A_{atm} - P_{sen}A_{sen} - P_{in}A_{inlet} - F_f \tag{3.15}$$

The spring force includes preload and the effect of poppet position in 3.16, 3.17, 3.18:

$$F_{springs} = F_{springbig} - F_{springsmall} \tag{3.16}$$

$$F_{springbig} = k_b(x_b - x) \tag{3.17}$$

$$F_{springsmall} = k_s(x_s + x) \tag{3.18}$$

In these formulas, x_b and x_s stands for initial compressions of springs, so total preload force can be written in equation 3.19:

$$F_{preload} = k_b x_b - k_s x_s \tag{3.19}$$

By substituting the equations 3.16 and 3.19 into the 3.15 the equation for net force can be written in 3.20:

$$F_{net} = m_{tot}\ddot{x} = F_{preload} - x(k_1 + k_2) + P_{out}A_{outlet} + P_{atm}A_{atm} - P_{sen}A_{sen} - P_{in}A_{inlet} - F_f \tag{3.20}$$

Area under effect of atmospheric pressure can be extended in equations 3.21 and 3.22:

$$F_{atm} = P_{atm}A_{atm} = P_{atm}A_{pis} - P_{atm}A_{pop} \tag{3.21}$$

$$A_{atm} = A_{pis} - A_{pop} \tag{3.22}$$

The friction force of the sliding parts can be estimated by the coulomb friction between dynamic seals of sliding parts and the surfaces they slide in equation 3.23.

$$F_f = sign(\dot{x})(F_{fpiston} + F_{fpoppet}) \tag{3.23}$$

For the piston and the poppet, the dynamic seals have different sizes and pressures. Therefore, the equations can be estimated as equations 3.24 and 3.25.

$$F_{fpiston} = \mu\, A_{fpiston}\, P_{sen} \tag{3.24}$$

$$F_{fpoppet} = \mu\, A_{fpoppet}\, P_{in} \tag{3.25}$$

3.3 Gas Equations

The pressure regulator's operation on gas is modeled by considering the behavior of compressible fluid flow. As it's mentioned above in the assumptions, the fluid is assumed as an ideal gas in this system. For the control volumes which is mentioned after assumptions, the ideal gas equation can be expressed in 3.26.

$$P\,V = m\,R\,T \tag{3.26}$$

Also, the thermodynamic processes of the gas are assumed to be adiabatic and reversible. According to the second law of thermodynamic, the relation of pressure and density raised to the specific heat ratio power (ɣ) is shown in equation 3.27:

$$\frac{P}{\rho^{\gamma}} = constant \tag{3.27}$$

Also, this equation can be written as equation 3.28:

$$\frac{P}{(\frac{m}{V})^{\gamma}} = constant \tag{3.28}$$

Substituting 3.26 into 3.28 gives equation 3.29 for temperature:

$$\frac{T}{(\frac{m}{V})^{\gamma-1}} = constant \tag{3.29}$$

Deriving the equations 3.28 and 3.29 provides the rate of change of the pressure and temperature between control volumes. However, the derivation includes volume itself. Therefore, the result of the derivations is depended to the volume. The volume (V0) of the control volume outlet could be fixed or variable.

Fixed Outlet Volume

If the volume (V_0) of the control volume outlet is fixed as equation 3.30:

$$V = V_0 \tag{3.30}$$

Pressure equation can be written as equation 3.31:

$$\dot{P} = \frac{\gamma R}{V} T \dot{m} \tag{3.31}$$

Temperature equation can be written as equation 3.32:

$$\dot{T} = \frac{RT^2}{VP}(\gamma - 1)\dot{m} \tag{3.32}$$

Variable Outlet Volume

If the volume (V_0) of the control volume outlet is fixed as equation 3.33:

$$V = V_0 + Ax \tag{3.33}$$

Pressure equation can be written as equation 3.34:

$$\dot{P} = \frac{\gamma R}{V} T \dot{m} - \frac{\gamma AP}{V} \dot{x} \tag{3.34}$$

Temperature equation can be written as equation 3.35:

$$\dot{T} = \frac{RT^2}{VP}(\gamma - 1)\dot{m} - \frac{(\gamma - 1)AT}{V}\dot{x} \tag{3.35}$$

3.4 Equations Applied for Control Volumes

The equations are derived and written for a general form. In the pressure regulator system, the equations will differ from the control volume to another control volume.

Sensing Control Volume

Sensing control volume has only one interface where mass flow transfer happens. The interface is between the sensing control volume and the outlet control volume.

Total mass flow for CV_{sen} as equation 3.36:

$$\sum \dot{m} = \dot{m}_{sen} \tag{3.36}$$

Since P_{out} and P_{sen} are close to the balance, the sensing mass flow is non-chocked:

For $P_{out} > P_{sen}$, it is written as equation 3.37:

$$\dot{m}_{sen} = c_d\, A_{or2}\, \frac{P_{out}}{\sqrt{R\, T_{out}}} \sqrt{\frac{2\gamma}{\gamma - 1}} \sqrt{\left(\frac{P_{sen}}{P_{out}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{sen}}{P_{out}}\right)^{\frac{\gamma+1}{\gamma}}} \tag{3.37}$$

For $P_{sen} > P_{out}$, it is written as equation 3.38:

$$\dot{m}_{sen} = -c_d\, A_{or2}\, \frac{P_{sen}}{\sqrt{R\, T_{sen}}} \sqrt{\frac{2\gamma}{\gamma - 1}} \sqrt{\left(\frac{P_{out}}{P_{sen}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{out}}{P_{sen}}\right)^{\frac{\gamma+1}{\gamma}}} \tag{3.38}$$

If the flow goes into the sensing control volume, the sensing mass flow is assumed positive and if the flow goes out from the sensing control volume to the outlet control volume, then the sensing mass flow is assumed negative.

The volume of the sensing control volume is varying with the movement of the piston. Therefore, the volume of it includes the position of the sliding parts of the regulator. The effect of the varying volume on the pressure and temperature equations is given as follow.

Volume of CV_{sen} is given in equation 3.39:

$$V_{sen} = V_{0/sen} + A_{sen} x \tag{3.39}$$

Pressure equation can be written as equation 3.40:

$$\frac{dP_{sen}}{dt} = \frac{\gamma R}{V_{sen}}\, T_{sen}\, \dot{m}_{sen} - \frac{\gamma A_{sen} P_{sen}}{V_{sen}}\, \dot{x} \tag{3.40}$$

Temperature equation can be written as equation 3.40:

$$\frac{dT_{sen}}{dt} = \frac{R T_{sen}^2}{V_{sen} P_{sen}} (\gamma - 1) \dot{m}_{sen} - \frac{(\gamma - 1) A_{sen} T_{sen}}{V_{sen}} \dot{x} \tag{3.41}$$

Outlet Control Volume

Outlet control volume has three interfaces for mass flow going in or going out. The interface between inlet control volume and outlet control volume is the feeder interface of the outlet control volume. The interface between outlet control volume and the sensing control volume is always trying to get in balance. The mass flow which goes

to outlet from the control volume is the outlet mass flow. In most of the cases, the outlet mass flow is a design parameter for the pressure regulator and it is controlled by a needle valve after the regulator. Therefore, the outlet mass flow is taken as input.

Total mass flow for CV_{out} as equation 3.42:

$$\sum \dot{m} = \dot{m}_{in} - \dot{m}_{sen} - \dot{m}_{out} \tag{3.42}$$

The sensing mass flow has minus sign before it since the sensing mass flow has the positive value if it goes into the sensing control volume, and it has negative value if the sensing mass flow goes out from the sensing control volume to the outlet control volume.

Mostly, the flow between CV_{inlet} and CV_{outlet} is choked flow since the pressure inlet pressure is extremely higher than outlet pressure. Therefore, the mass equation for CV_{outlet} is shown equation 3.43:

$$\dot{m}_{out} = c_d\, A\, \frac{P_{in}}{\sqrt{R\, T_{in}}} \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \sqrt{\frac{2\gamma}{\gamma + 1}} \quad when \quad \frac{P_{out}}{P_{in}} < P_{cr} \tag{3.43}$$

However, in some cases, inlet pressure could be closer to outlet pressure. Therefore, pressure ratio is more than the critical pressure ratio and the flow is non-chocked flow. The mass flow equation for this situation is given as equation 3.44:

$$\dot{m}_{out} = c_d\, A\, \frac{P_{in}}{\sqrt{R\, T_{in}}} \sqrt{\frac{2\gamma}{\gamma - 1}} \sqrt{\left(\frac{P_{out}}{P_{in}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma+1}{\gamma}}} \quad when \quad \frac{P_{out}}{P_{in}} > P_{cr} \tag{3.44}$$

Volume of CV_{out} is given in equation 3.45:

$$V_{out} = V_0 + A_{outlet} x \tag{3.45}$$

Pressure equation can be written as equation 3.46:

$$\frac{dP_{out}}{dt} = \frac{\gamma R}{V_{out}} \left(T_{in}\, \dot{m}_{in} - T_{sen}\, \dot{m}_{sen} - T_{out}\, \dot{m}_{out}\right) - \frac{\gamma A_{out} P_{out}}{V_{out}}\, \dot{x} \tag{3.46}$$

Temperature equation can be written as equation 3.47:

$$\frac{dT_{out}}{dt} = \frac{RT_{out}^2}{V_{out}P_{out}}\left(\left(\gamma\frac{T_{in}}{T_{out}} - 1\right)\dot{m}_{in} - (\gamma - 1)\dot{m}_{out} - (\gamma - 1)\dot{m}_{sen}\right) - \frac{(\gamma - 1)A_{out}T_{out}}{V_{out}}\dot{x} \tag{3.47}$$

4. DESIGNING

4.1 Requirements for Design

Firstly, the purpose of the regulator needs to be defined. In this study, pressurization propellant tank is aimed by the regulators designed as shown in Figure 4.1.

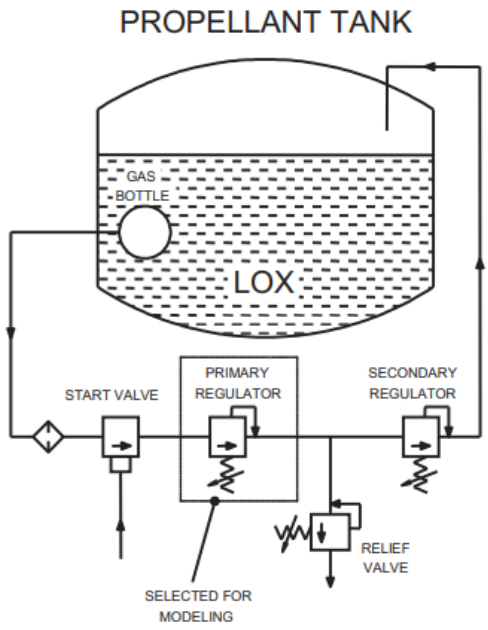

Figure 4.1: Tank pressurization system

For a pressure regulator which pressurized the propellant tank, the parameters below need to be determined:

- Inlet pressure
- Set pressure
- Nominal mass flow rate
- Maximum mass flow rate
- Working fluid
- Temperature of fluid
- Pipe size
- Outlet volume

In this study, three different requirement cases are simulated and tested. Three requirements will be showed in Table 4.1.

Table 4.1: Requirements of all cases

Parameter	Case 1	Case 2	Case 3
Inlet pressure [Bar]	300-50	300-50	300-50
Set pressure [Bar]	8.5	18.5	23.5
Atmospheric or Absolute	Atmospheric	Absolute	Absolute
Nominal mass flow rate [kg/s]	0.010	0.045	0.055
Maximum mass flow rate [kg/s]	0.040	0.120	0.150
Working Fluid	Nitrogen	Nitrogen	Helium
Temperature of fluid [K]	300	300	300
Pipe size [mm]	22	22	20
Outlet volume [m^3]	270e-6	270e-6	1200e-6

Besides these requirements, the variable outlet volume for all three cases is simulated in order to see the difference between fixed and variable outlet volume.

4.2 Design Inputs

After the requirements are defined, there are some other parameters to finish the design of the pressure regulator. These parameters are fluid parameters and mechanical parameters.

The fluid parameters are depended to the working fluid. In this study, Helium and Nitrogen are used as the working fluid. The gas parameters of them are listed in Table 4.2.

Table 4.2: Gas Parameters

Gas Parameter (@300 K)	Symbol	Nitrogen	Helium
Specific Heat Ratio	ɣ	1.4	1.667
Gas Constant [J/kg.K]	R	296.8	2077.1
Cp [J/kg.K]	cp	1040	5190
Cv [J/kg.K]	cv	743	3120
Kinematic Viscosity [m^2/s]	η	15.25e-06	116.9e-06
Dynamic Viscosity [Pa.s]	μ	17.58e-06	19.6e-06
Critical pressure ratio	P_{cr}	0.5283	0.4871

In order to select the mechanical parameters, the requirements such as inlet pressure, set pressure, maximum mass flow rate, nominal mass flow rate, temperature, pipe size have an important role.

Firstly, the static calculations from the mathematical model are made to determine the rough mechanical parameters. After having the rough parameters, the optimum values for the mechanical parameters are calculated by using the optimization tool on MATLAB Simulink.

In this study, the calculations and optimization of the mechanical parameters are made for only Case 1. The other cases are that to compare test results and the simulation results. To calculate the orifice diameter which is called poppet seat diameter, the worst case is assumed. The worst case has maximum mass flow rate, minimum discharge coefficient, minimum inlet pressure. It is shown in Table 4.3. The calculations are made on equations 4.1, 4.2, 4.3, 4.4, 4.5, 4.6 and 4.7.

Table 4.3: Worst Case Scenario for Case 1

Parameter	Value
Inlet Pressure [Bar]	300 (decreasing)
Set Pressure [Bar]	8.5
Working Fluid	Nitrogen
Maximum Mass Flow Rate [kg/s]	0.04
Max. Stroke of poppet [mm]	2

For staying the safe side, discharge coefficient (cd) is assumed as 0.5, stroke is used as three quarters of its maximum value and inlet pressure as on its last regulating value.

$$cd = 0.5 \tag{4.1}$$

$$x = 2 * \frac{3}{4} = 1.5\ [mm] \tag{4.2}$$

$$P_{in-last} = \frac{P_{out}}{P_{cr}} = \frac{8.5}{0.5283} = 16.1\ [Bar] \tag{4.3}$$

Firstly, the area needed for worst case is calculated by rearranging equation 3.2

$$A = \frac{\dot{m}}{cd \frac{P_{in}}{\sqrt{R\,T_{in}}} \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \sqrt{\frac{2\gamma}{\gamma + 1}}} \tag{4.4}$$

$$A = 2.17x10^{-5}\ m^2 \tag{4.5}$$

By arranging (3.10), orifice diameter equation is found

$$d_{or} = \frac{A}{\pi\ x\ \sin(\theta)} + x \sin(\theta) \cos(\theta) \tag{4.6}$$

$$d_{or} = 7.2\ mm \tag{4.7}$$

To be have a balanced poppet, the poppet diameter is chosen same with orifice diameter. After the calculating the orifice diameter roughly, values of the other mechanical parameters are roughly decided either. They are shown in Table 4.4.

Table 4.4: Rough mechanical values of piston and poppet

Parameter	Symbol	Value
Piston diameter [mm]	d_{pis}	35
Poppet Diameter [mm]	d_{pop}	7.2
Area of piston under effect of outlet pressure [mm^2]	A_{sen}	955
Area of piston under effect of Atmospheric Pressure [mm^2]	A_{pis}	962
Area of poppet under effect of Outlet Pressure [mm^2]	A_{out}	31.4
Area of poppet under effect of Atmospheric Pressure [mm^2]	A_{pop}	44.2

Preload is calculated roughly from the total force on piston and poppet by the outlet pressure in equations 4.8 and 4.9.

$$Preload = P_{out}(A_{sen} - A_{out}) - P_{atm}(A_{pis} - A_{pop}) \quad (4.8)$$

$$Preload = 693.3\ N \quad (4.9)$$

After all rough calculations, the mechanical parameters of the Case 1 are optimized by using optimization tool on MATLAB Simulink. The last values of the mechanical parameters for simulating and manufacturing the pressure regulator are shown in Table 4.5:

Table 4.5: Applied mechanical parameters for Case 1

Parameter	Symbol	Value
Orifice diameter [mm]	d_{or}	7.5
Poppet diameter [mm]	d_{pop}	7.5
Piston diameter [mm]	d_{pis}	35
Poppet rod diameter [mm]	d_{rod}	3
Orifice through sensing area [mm]	d_{or2}	5
Poppet surface angle [deg]	θ	45
Spring constant of piston side [N/mm]	k_b	147
Spring constant of poppet side [N/mm]	k_s	49.4
Preload [N]	F_{pre}	700

The design inputs of all cases for simulations are shown in Table 4.6.

Table 4.6: Design inputs of all cases

Parameter	Symbol	Case 1	Case 2	Case 3
Atmospheric or Absolute	-	Atmospheric	Absolute	Absolute
Mass of sliding parts [kg]	m_{tot}	0.08	0.125	0.25
Orifice diameter [mm]	d_{or}	7.5	9.3	12
Poppet diameter [mm]	d_{pop}	7.5	9.3	12.5
Piston diameter [mm]	d_{pis}	35	50	40
Poppet rod diameter [mm]	d_{rod}	3	3	3
Orifice through sensing volume [mm]	d_{or2}	5	5	4.5
Poppet surface angle [deg]	θ	45	45	45
Spring constant of piston side [N/mm]	k_b	147	270	212
Spring constant of poppet side [N/mm]	k_s	49.4	90	64
Spring displacement of piston side [mm]	x_b	7.1	14	15.5
Spring displacement of poppet side [mm]	x_s	6.4	6	3.1
Max Stroke [mm]	x_{max}	2	2.1	2
Preload [N]	$F_{preload}$	730	3590	3080
Outlet volume [m^3]	V_0	270e-6	270e-6	1200e-6

5. RESULTS AND DISCUSSION

In this study, a dynamic model of pressure regulator has been prepared on MATLAB Simulink by using the mathematical model. The model is able to calculate outlet pressure by using regulator's design inputs, inlet pressure and outlet mass flow as inputs.

On Simulink, the solver is selected as the Runge-Kutta. The Runge-Kutta method is a widely used numerical integration technique for solving ODEs. It approximates the solution by evaluating the derivatives at different points within a time step and using a weighted average of these evaluations. The method's accuracy can be improved by increasing the number of evaluation points and adjusting the weights.

By selecting the Runge-Kutta solver in Simulink, it can be taken advantage of its numerical integration capabilities to simulate and analyze dynamic systems described by ODEs. Simulink handles the implementation of the Runge-Kutta method.

The design inputs are listed on the Table 4.6. All simulations are made with the inputs provided on this table. The scenarios of the simulations are mentioned on the results section for each.

5.1 Test Results

In space applications, the performance of pneumatic systems is needed to be tested as an essential process. These tests help the engineers to ensure the safety and reliability of the system.

In this study, the performance tests of the pressure regulators are done on the pneumatic test system which Figure 5.1 shows.

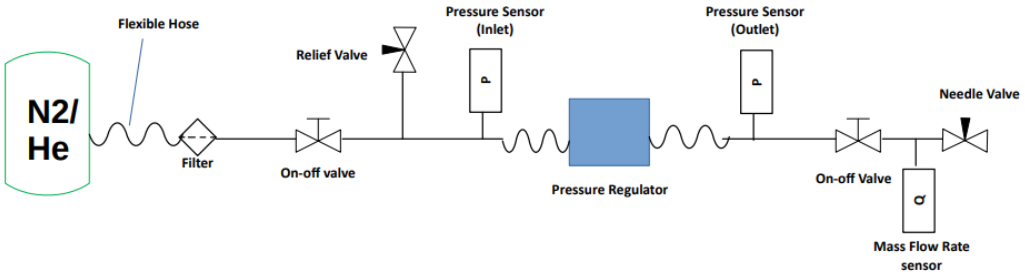

Figure 5.1: Test setup

The test system includes a gas tank can be filled with pressurized Nitrogen or Helium. A flexible hose connects the tank to a filter to ensure the particulars cannot reach to the pressure regulator and affect its performance. After filter, there is an on-off valve to give the pressure to the line. Between on-off valve and pressure regulator, there is a relief valve for safety. On the inlet and the outlet of the pressure regulator, there are pressure sensors to get the performance data from the test. In the end of the line, there is a needle valve to adjust the mass flow rate. Just before the needle valve, there is a mass flow rate sensor.

In the tests, the pressure regulators are tested with different mass flow rates and different inlet pressures. The data for both pressure sensors and mass flow rate sensor are collected as 1 kHz.

5.1.1 Case 1

Case 1 design inputs are mentioned on Table 4.6. This regulator adjusted for 8 bar on this test. The parameters and its values which are set for this test are listed in Table 5.1. Test lasted for 200 seconds. The test results of inlet/outlet pressure and mass flow rate are shown in Figure 5.2, Figure 5.3.

Table 5.1: Test parameters for Case 1

Parameter	Value
Inlet pressure [Bar]	53-14 (decreasing)
Set Pressure [Bar]	8
Total Time [s]	200
Mass flow rate [g/s]	12
Outlet volume [l]	0.4

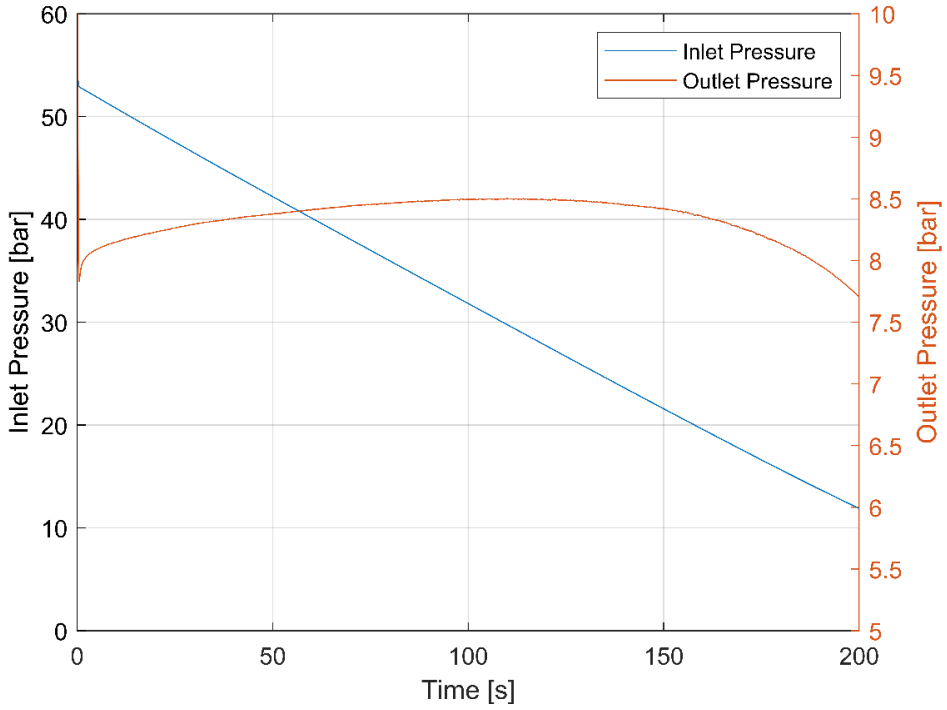

Figure 5.2: Inlet(left axis)/Outlet(right axis) pressure for tests of case 1

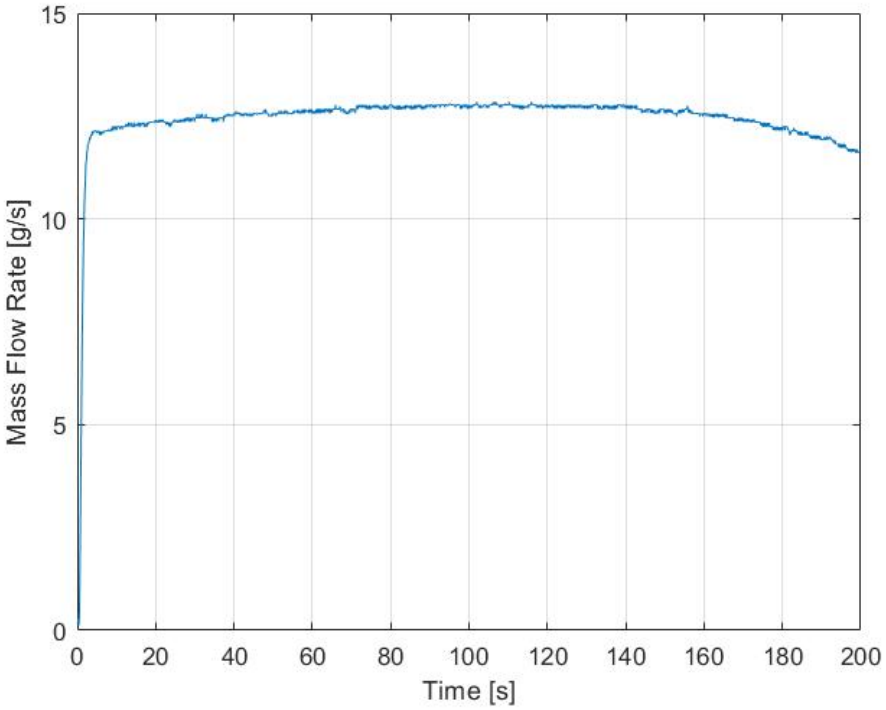

Figure 5.3: Mass flow rate [g/s] vs Time for tests of case 1

Figure 5.2 shows that, the outlet pressure of the pressure regulator starts 7.9 bar and peaks at 8.5 bar and decreases again. Figure 5.3 shows that, the mass flow rate stays still.

5.1.2 Case 2

Case 2 design inputs are mentioned on Table 4.6. This regulator adjusted for 17.5 bar on this test. The parameters and its values which are set for this test are listed in Table 5.2. Test lasted for 40 seconds. The test results of inlet/outlet pressure and mass flow rate are shown in Figure 5.4, Figure 5.5.

Table 5.2: Test parameters for Case 2

Parameter	Value
Inlet pressure [Bar]	83-48 (decreasing)
Set Pressure [Bar]	17.5
Total Time [s]	40
Mass flow rate [g/s]	51
Outlet volume [l]	0.4

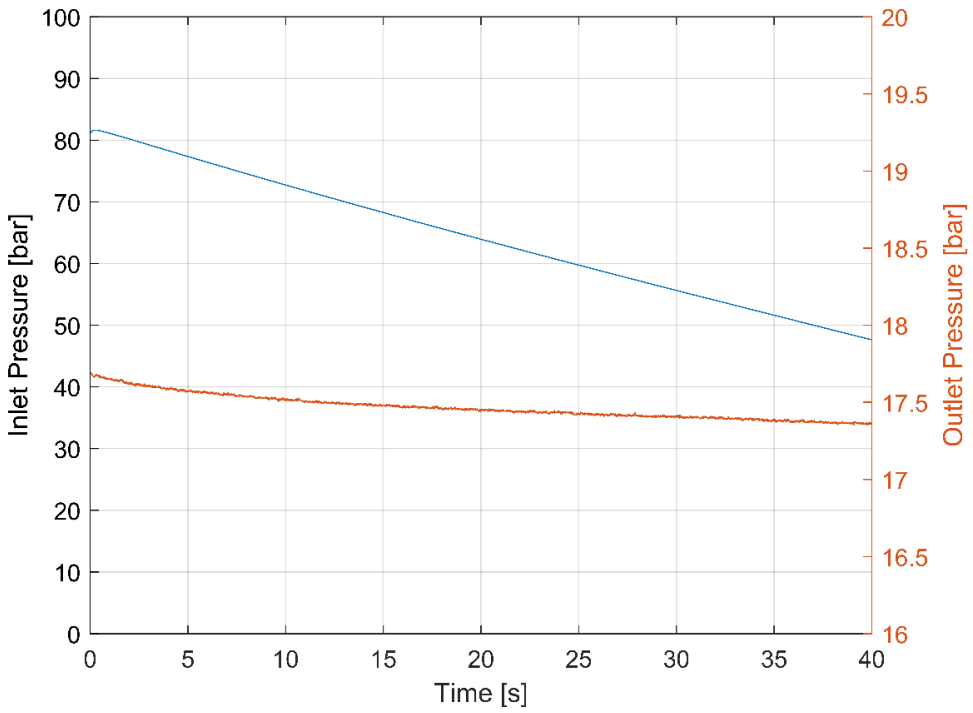

Figure 5.4: Inlet(left axis)/Outlet(right axis) pressure for tests of case 2

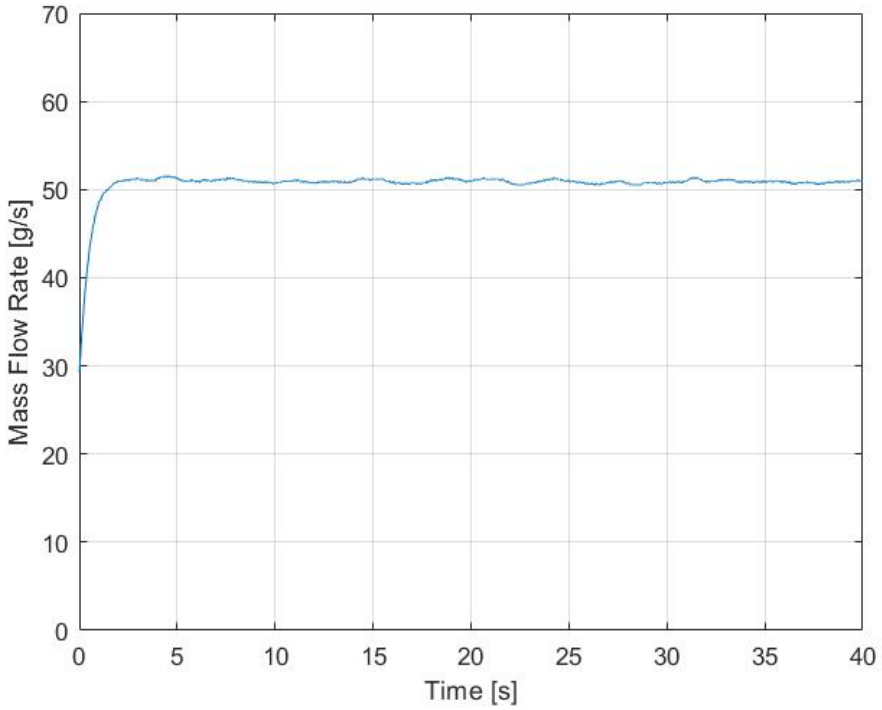

Figure 5.5: Mass flow rate [g/s] vs Time for tests of case 2

Figure 5.4 shows that, the outlet pressure of the pressure regulator starts 17.7 bar and decreases until 17.4 bar. Figure 5.5 shows that, the mass flow rate stays still.

5.1.3 Case 3

Case 3 design inputs are mentioned on Table 4.6. This regulator adjusted for 22.5 bar on this test. The parameters and its values which are set for this test are listed in Table 5.3. Test lasted for 10 seconds because the helium capacity is insufficient for long tests. The test results of inlet/outlet pressure and mass flow rate are shown in Figure 5.6, Figure 5.7.

Table 5.3: Test parameters for Case 3

Parameter	Value
Inlet pressure [Bar]	110-70 (decreasing)
Set Pressure [Bar]	22.5
Total Time [s]	10
Mass flow rate [g/s]	24
Outlet volume [l]	0.4

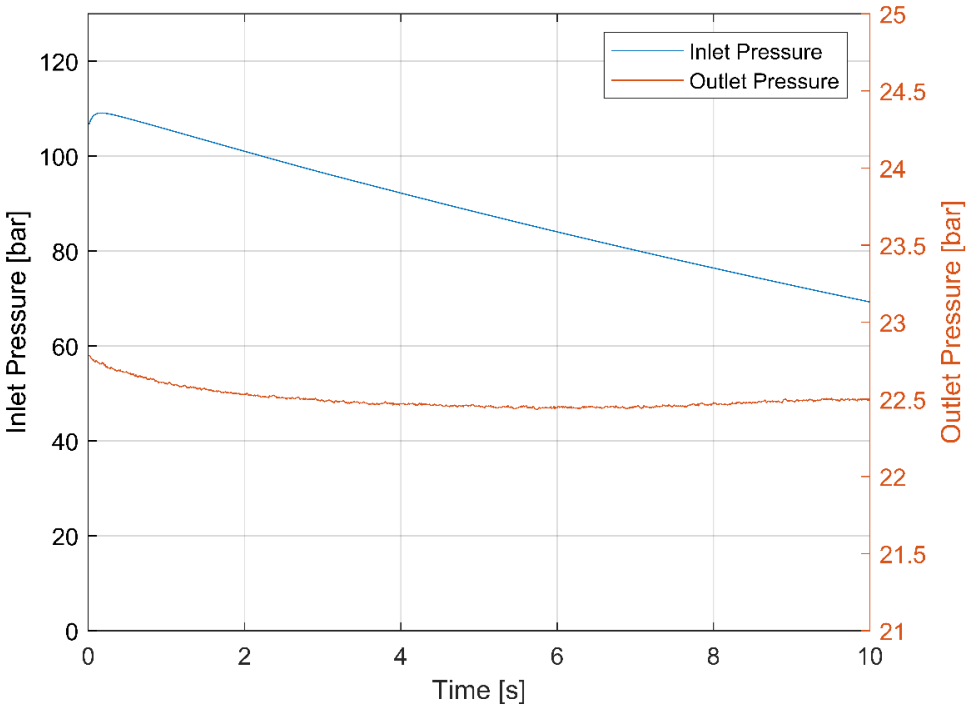

Figure 5.6: Inlet(left axis)/Outlet(right axis) pressure for tests of case 3

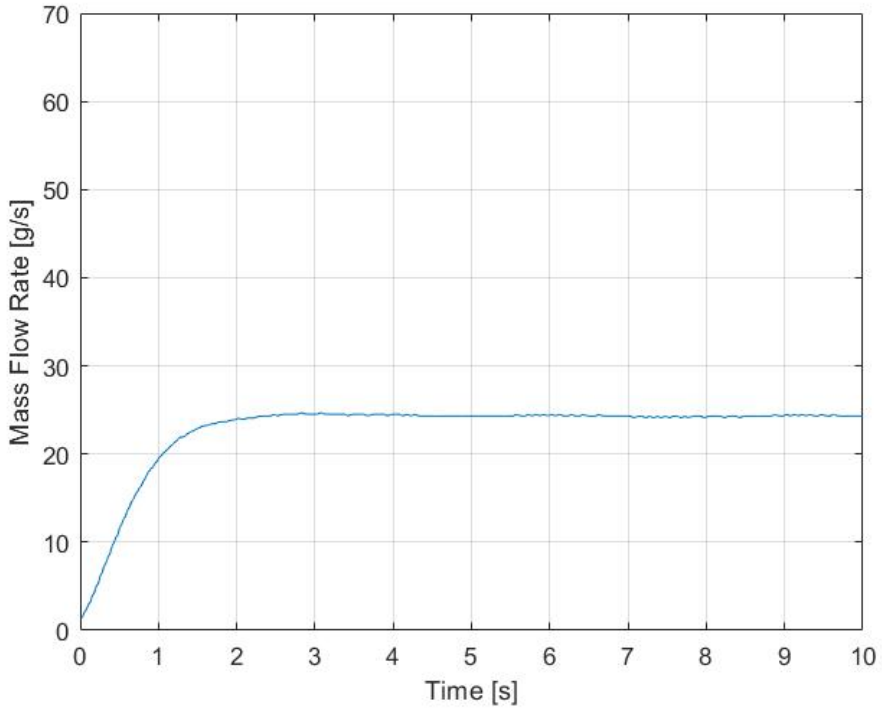

Figure 5.7: Mass flow rate [g/s] vs Time for tests of case 3

Figure 5.6 shows that, the outlet pressure of the pressure regulator starts 22.8 bar and decreases until 22.5 bar. On Figure 5.7, the mass flow rate starts from 0 g/s, reaches 24 g/s and stays still.

5.2 Simulation Validation

In this section, the test results of all cases are compared with the simulation results. The simulations are made for the same mass flow rate and set pressure from the tests. In order to have same scenario with tests; time, mass flow rate and preload inputs are adjusted on the simulations.

The results are compared for only outlet pressure. An error calculation for all cases is made through the process by using equation 5.1.

$$error\% = \frac{|P_{test} - P_{sim}|}{P_{test}} x100 \tag{5.1}$$

5.2.1 Case 1

On a zoomed view between 7-9 bar for outlet pressure in Figure 5.8, the test results for case 1 seems slightly different from the simulation. It is predicted that the reason for this may be due to tolerance errors, manufacturing errors and dimensional inconsistencies during the production of the regulator. However, when looked at in general in Figure 5.9, it is seen that the error rate remains around 2%. This shows that the simulation has a deviation in the 2-3% band.

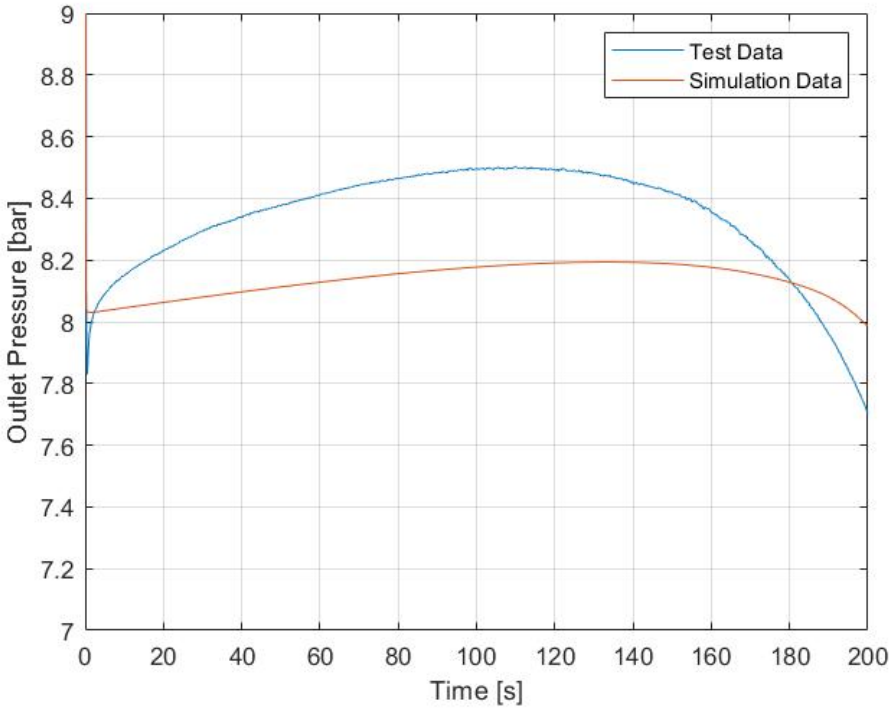

Figure 5.8: Test vs Simulation Outlet Pressure Comparison for case 1

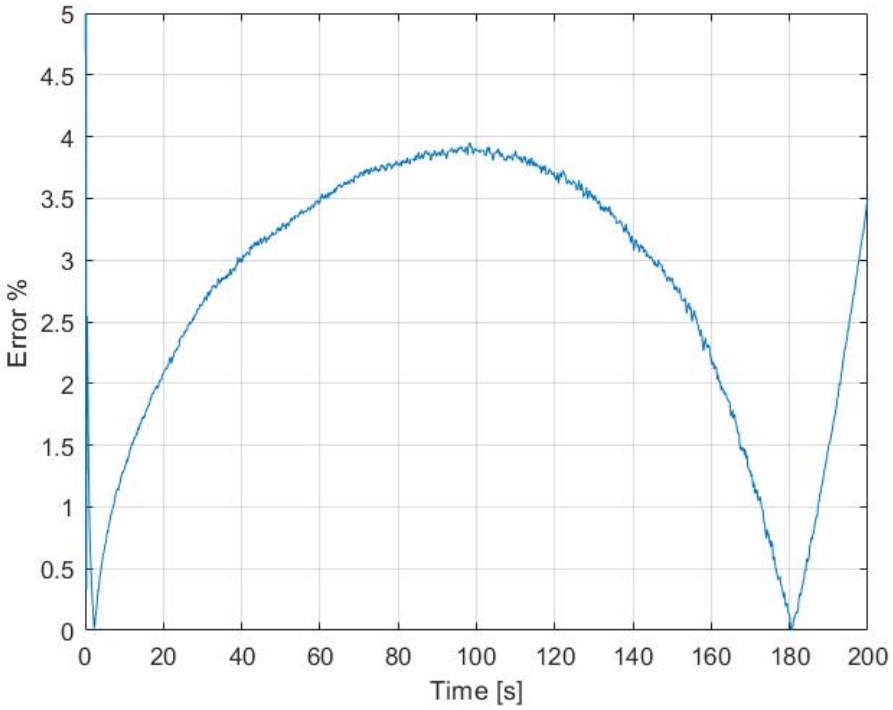

Figure 5.9: Simulation error percent for case 1

5.2.2 Case 2

On a zoomed view between 17-18 bar for outlet pressure in Figure 5.10, the test results for case 2 seems more linear than simulation data. The simulation data has an early down while test result is reaching that much down later. However, both results have nearly same start and same end. Also, in Figure 5.11, the error line reached maximum at 0.5% and remains around 0.25% averagely which seems very promising for simulation. This shows that the simulation has a deviation in the 0.25% band.

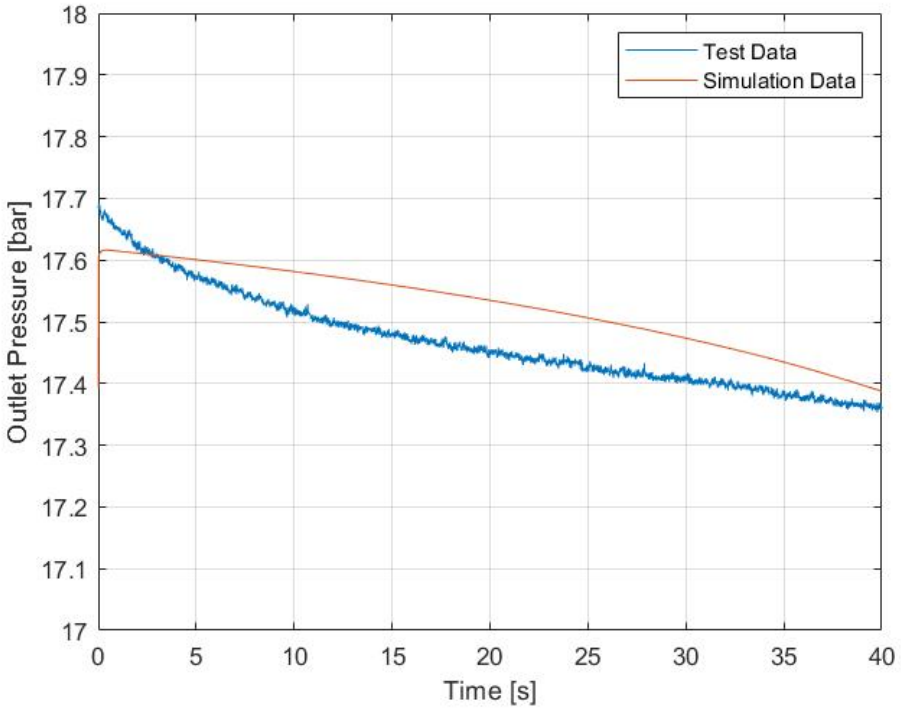

Figure 5.10: Test vs Simulation Outlet Pressure Comparison for case 2

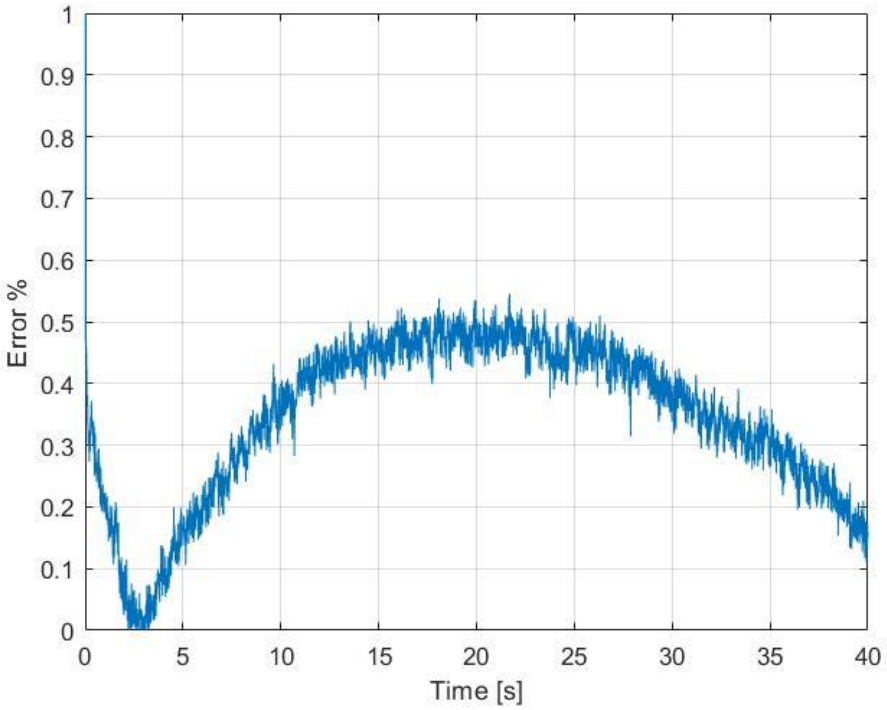

Figure 5.11: Simulation error percent for case 2

5.2.3 Case 3

On a zoomed view between 22-24 bar for outlet pressure in Figure 5.12, the test results for case 3 seems more converging than simulation data. However, both results have nearly same start and same end. Also, in Figure 5.13, the error line reached maximum at 0.38% and remains around %0.15 which seems the simulation results are very realistic. This shows that the simulation has a deviation in the 0.15% band.

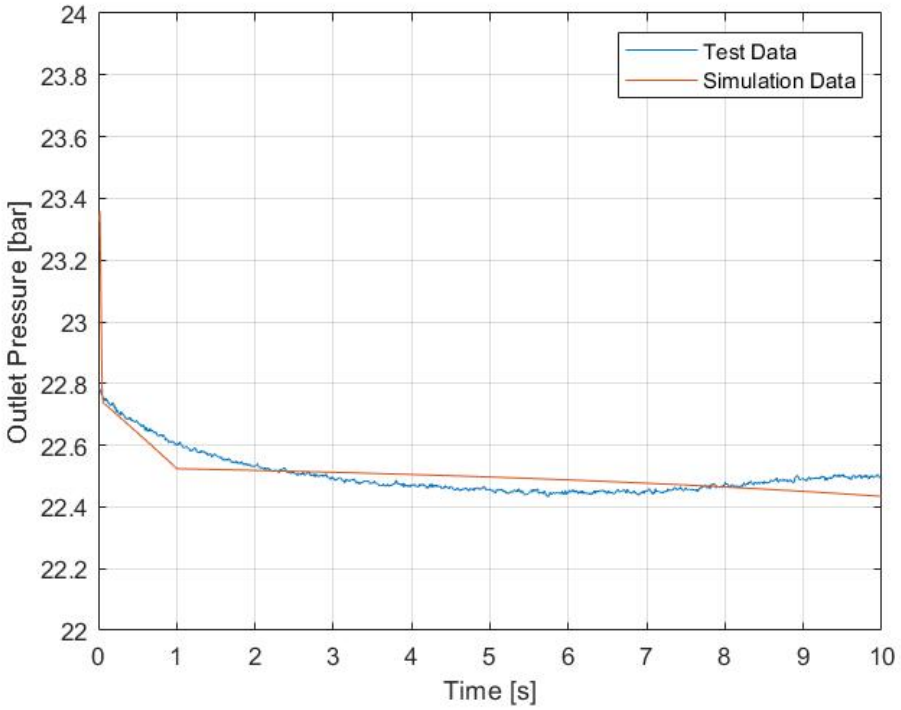

Figure 5.12: Test vs Simulation Outlet Pressure Comparison for case 3

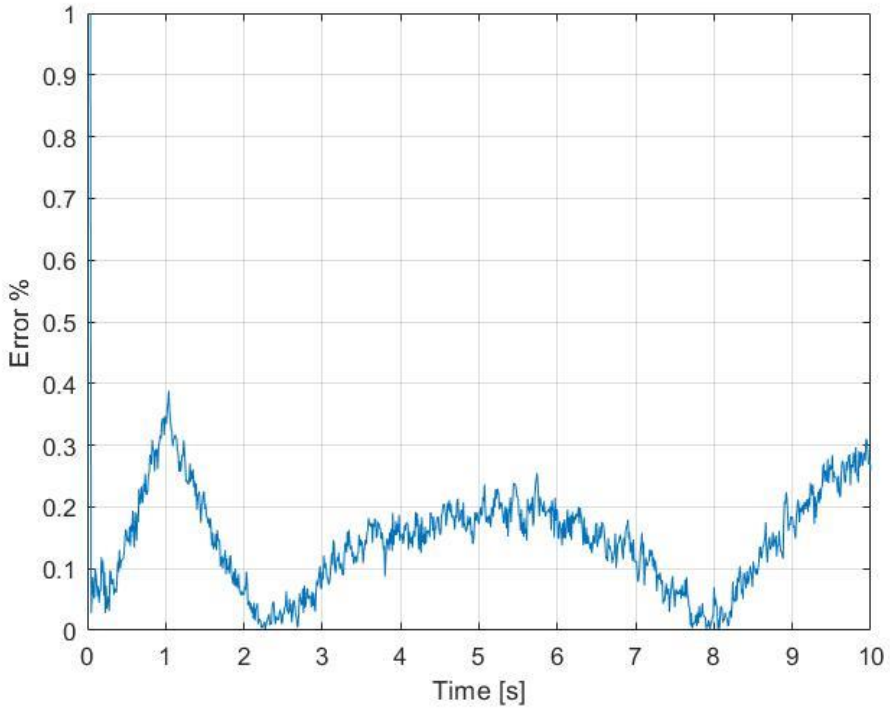

Figure 5.13: Simulation error percent for case 3

5.3 Simulations for Different Scenarios

After validation of the mathematical and dynamic model, the simulations are made for all three cases. There are four different scenarios are simulated for all cases. The scenarios have different from each other such as inlet pressure variation, mass flow rates and outlet volume variation. All scenarios' results show as graphs including inlet and outlet pressure vs time, displacements vs time and mass flow rate vs time. Also, a comparison for all scenarios is shown in the end of each case section.

5.3.1 Case 1

Case 1 requirements and design inputs are mentioned on Table 4.1 and Table 4.6. The main purpose of this case to regulate the pressure to 8 Bar.

Scenario 1:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. On the other hand, mass flow rate and outlet volume are taken as fixed. The values of these parameters are given in Table 5.4. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.14, Figure 5.15, Figure 5.16.

Table 5.4: Input Parameters - Case 1 Scenario 1

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	10
Outlet volume [l]	0.270

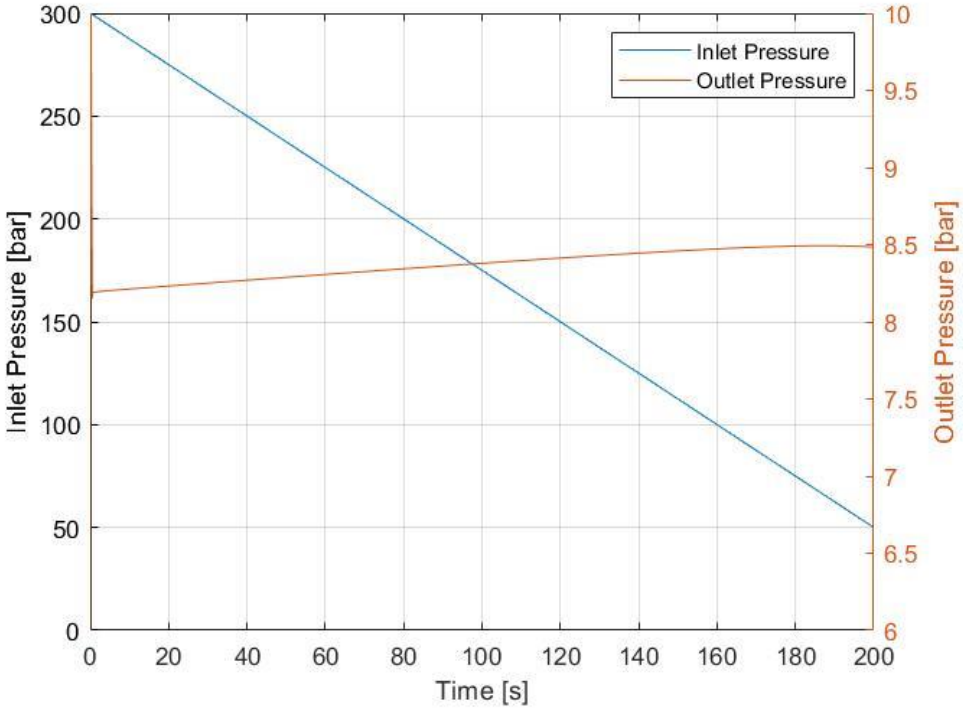

Figure 5.14: Inlet/Outlet Pressure for scenario 1 of case 1

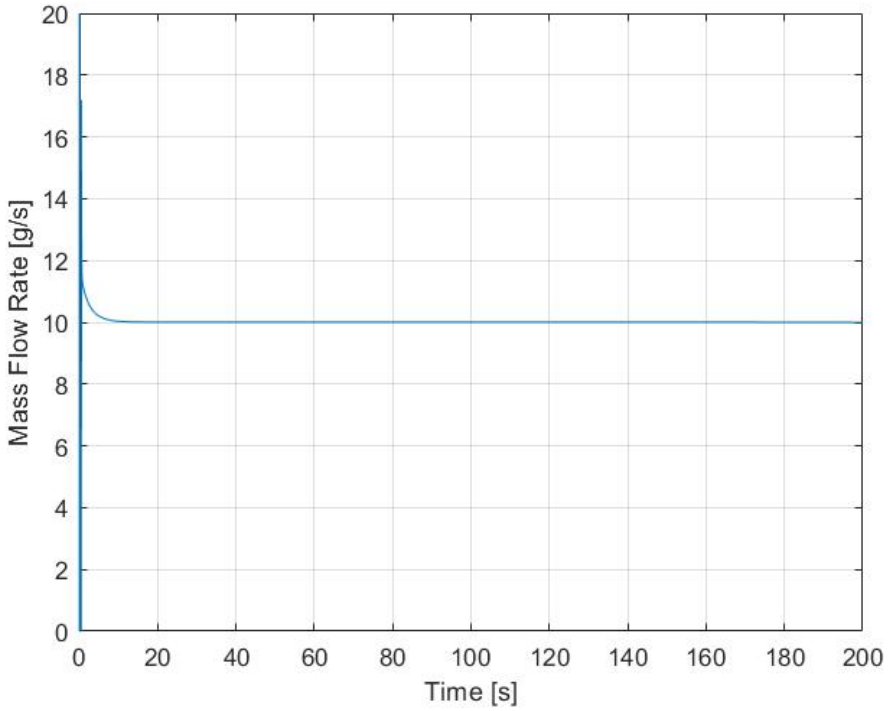

Figure 5.15: Mass flow rate for scenario 1 of case 1

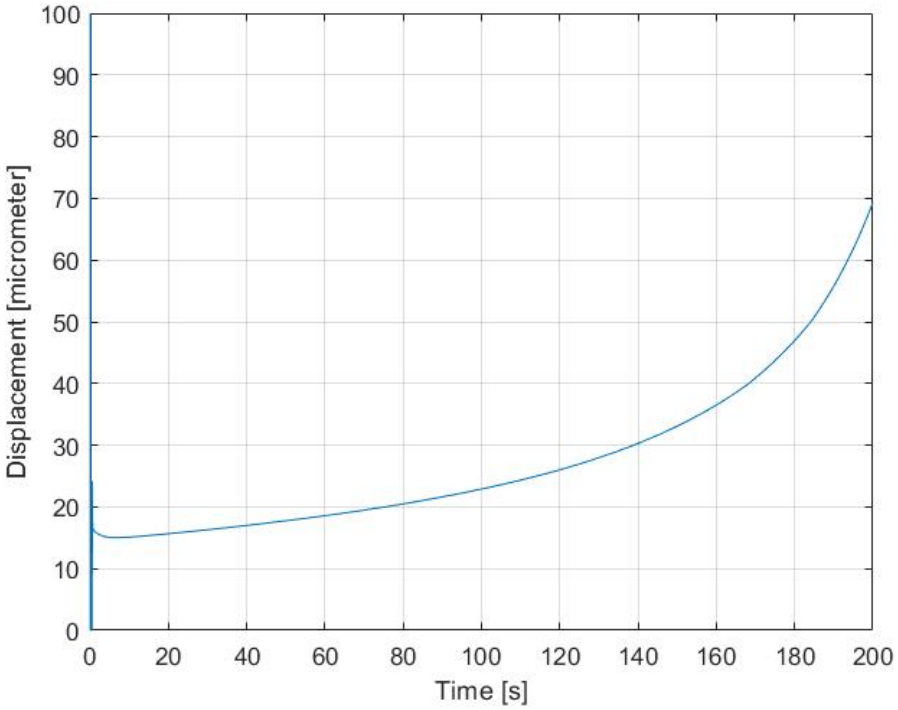

Figure 5.16: Poppet displacement for scenario 1 of case 1

Scenario 2:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. In addition to that, mass flow rate is varying and it changes its value at every 50 s. The outlet volume is taken as fixed. The values of these parameters are given in Table 5.5. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.17, Figure 5.18, Figure 5.19.

Table 5.5: Input Parameters - Case 1 Scenario 2

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	[10 – 100 – 50 – 5 g/s] (varying)
Outlet volume [l]	0.270

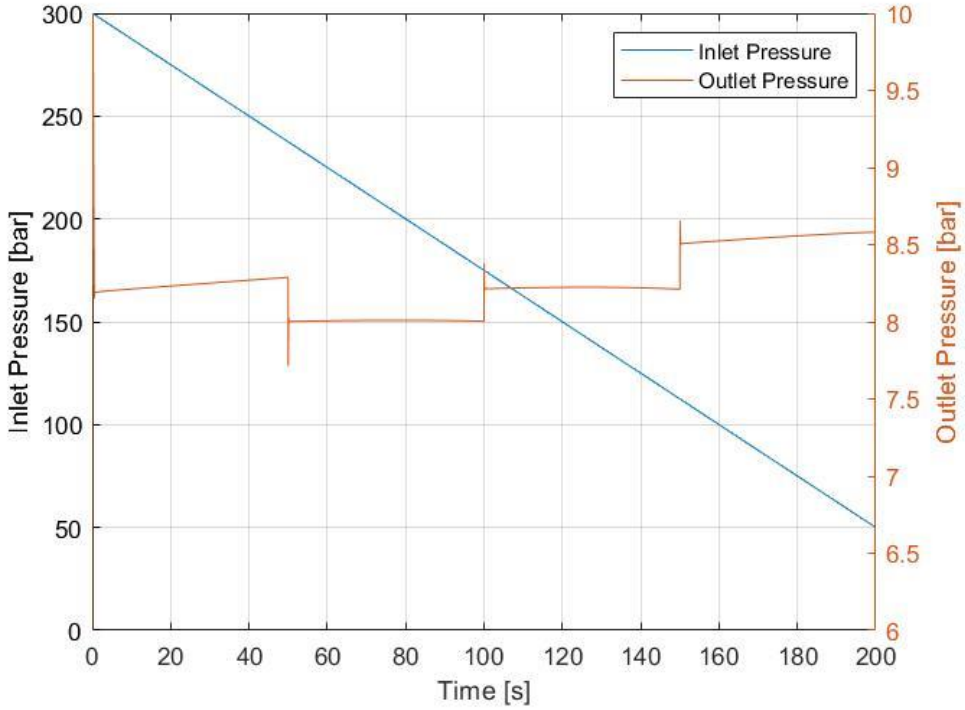

Figure 5.17: Inlet/Outlet Pressure for scenario 2 of case 1

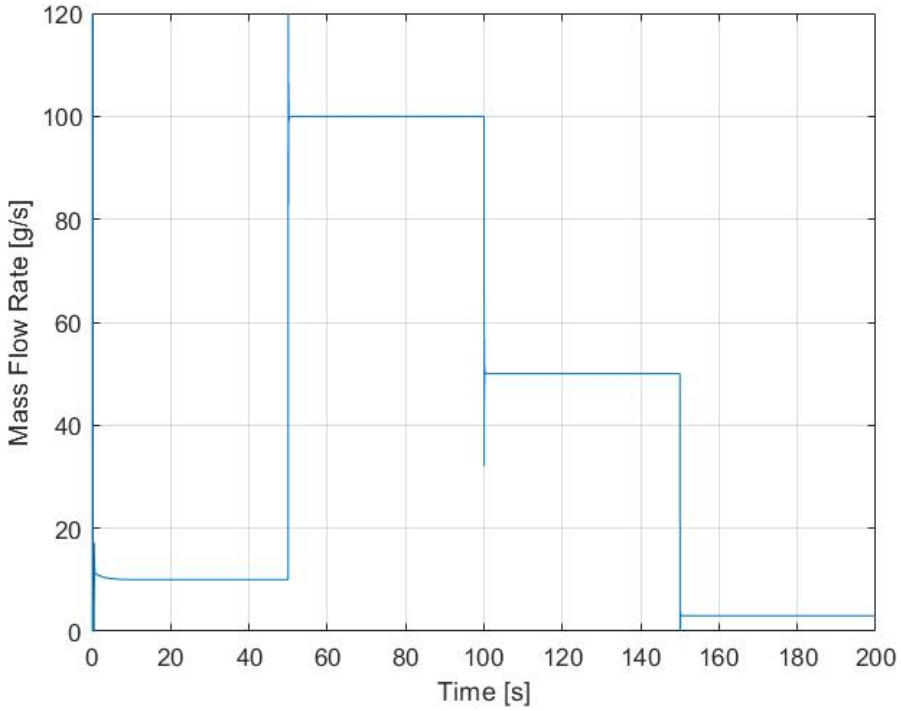

Figure 5.18: Mass flow rate for scenario 2 of case 1

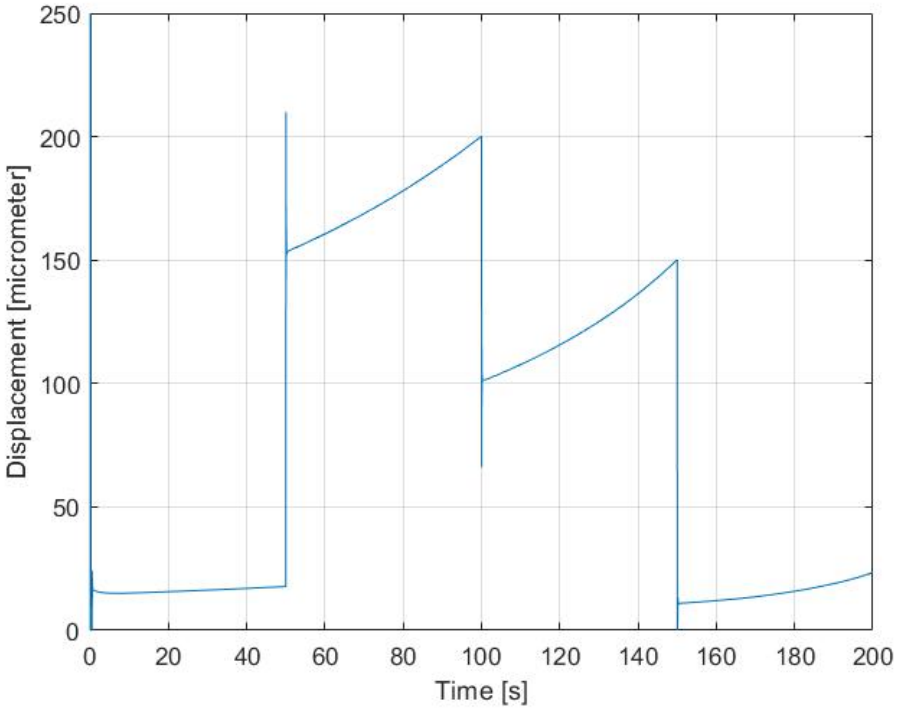

Figure 5.19: Poppet displacement for scenario 2 of case 1

Figure 5.17 shows that, the outlet pressure of the pressure regulator starts 8.2 bar and, at 50 s, it is down to 8 bar due to increase of mass flow rate to 100 g/s. When the mass flow increases, the outlet pressure is decreases. Figure 5.18 shows that, the mass flow rate varies. On Figure 5.19, it can be seen that, the poppet displacement increases as mass flow rate increases. That's why the outlet pressure decreases when mass flow increases. The force on the poppet decreases when x displacement increases, so the pressure required to reach stabilize the poppet is also reduced.

Scenario 3:

In this scenario, the inlet pressure, mass flow rate and outlet volume are fixed for all simulation time. The values of these parameters are given in Table 5.6. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.20, Figure 5.21, Figure 5.22.

Table 5.6: Input Parameters - Case 1 Scenario 3

Parameter	Value
Inlet pressure [Bar]	300
Mass flow rate [g/s]	10
Outlet volume [l]	0.270

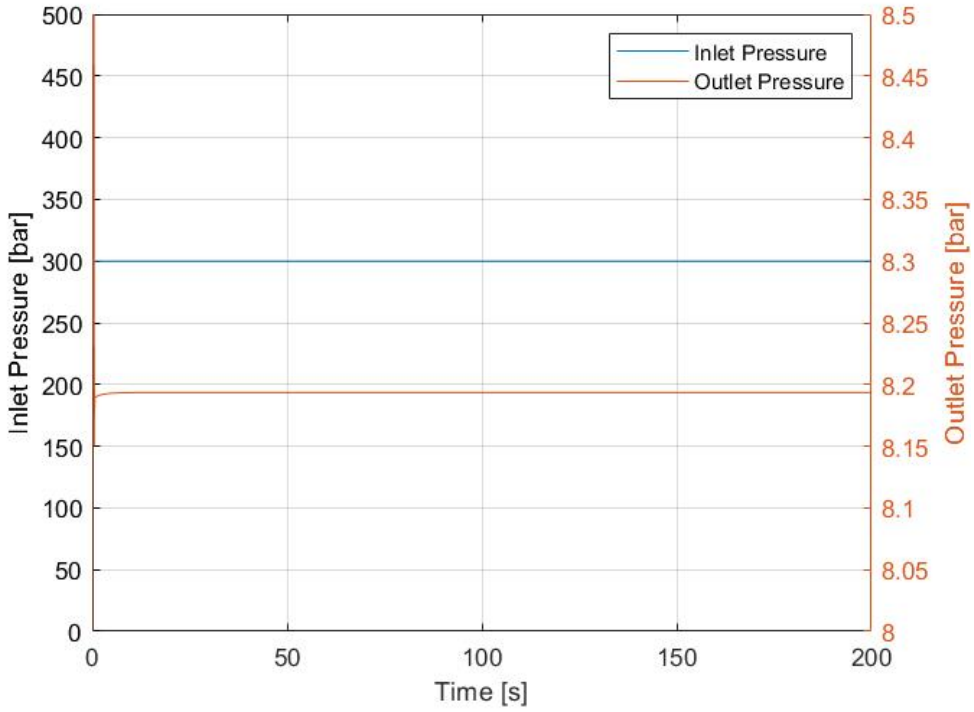

Figure 5.20: Inlet/Outlet Pressure for scenario 3 of case 1

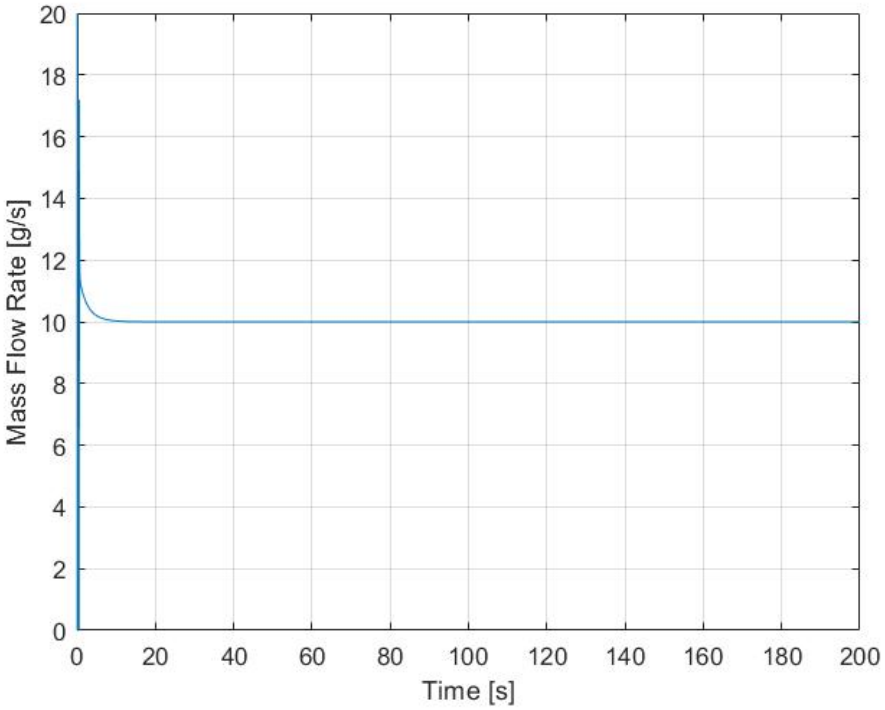

Figure 5.21: Mass flow rate for scenario 3 of case 1

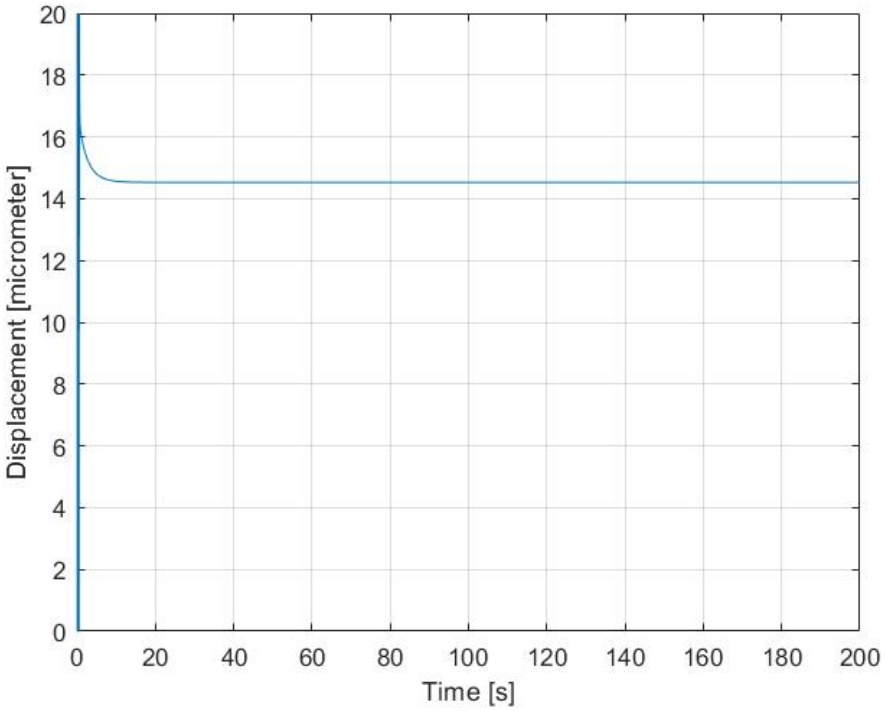

Figure 5.22: Poppet displacement for scenario 3 of case 1

Figure 5.20, Figure 5.21 and Figure 5.22 shows that, the outlet pressure and x displacement of the pressure regulator stays still when there is no change in inlet pressure and mass flow rate.

Scenario 4:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. However, the outlet volume is increasing. On the other hand, mass flow rate is taken as fixed. The values of these parameters are given in Table 5.7. The simulation results of inlet/outlet pressure, mass flow rate, x displacement and outlet volume are shown in Figure 5.23, Figure 5.24, Figure 5.25, Figure 5.26.

Table 5.7: Input Parameters - Case 1 Scenario 4

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	10
Outlet volume [l]	0.270-60 (increasing)

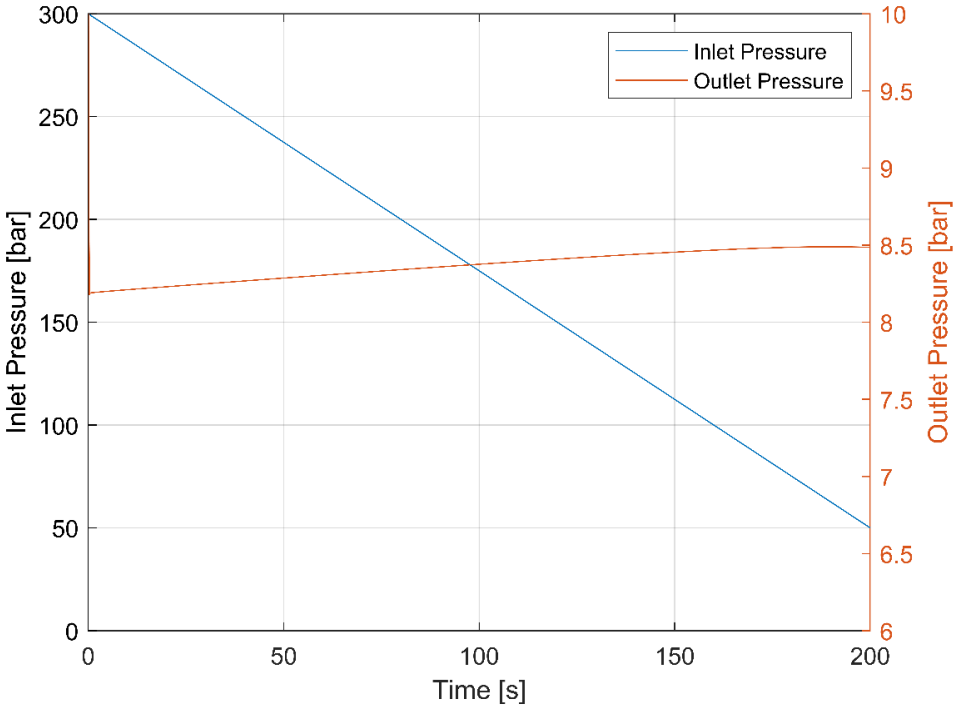

Figure 5.23: Inlet/Outlet Pressure for scenario 4 of case 1

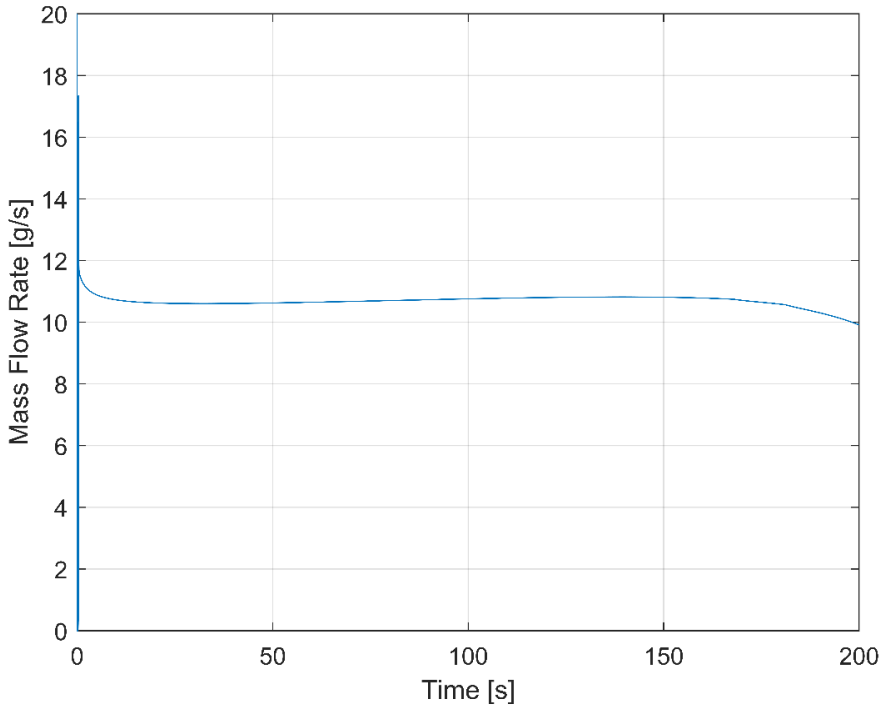

Figure 5.24: Mass flow rate for scenario 4 of case 1

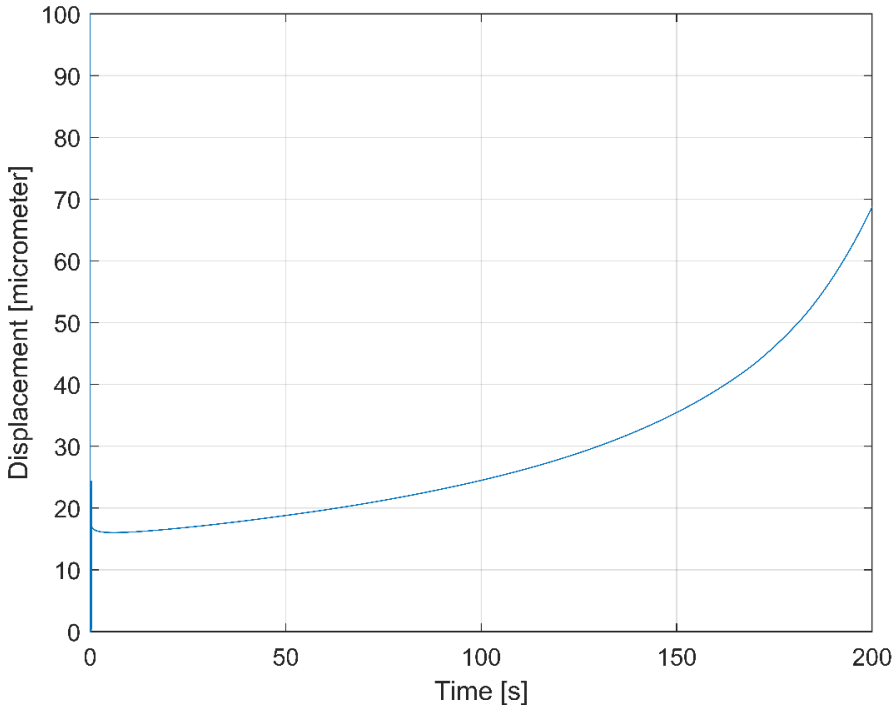

Figure 5.25: Poppet displacement for scenario 4 of case 1

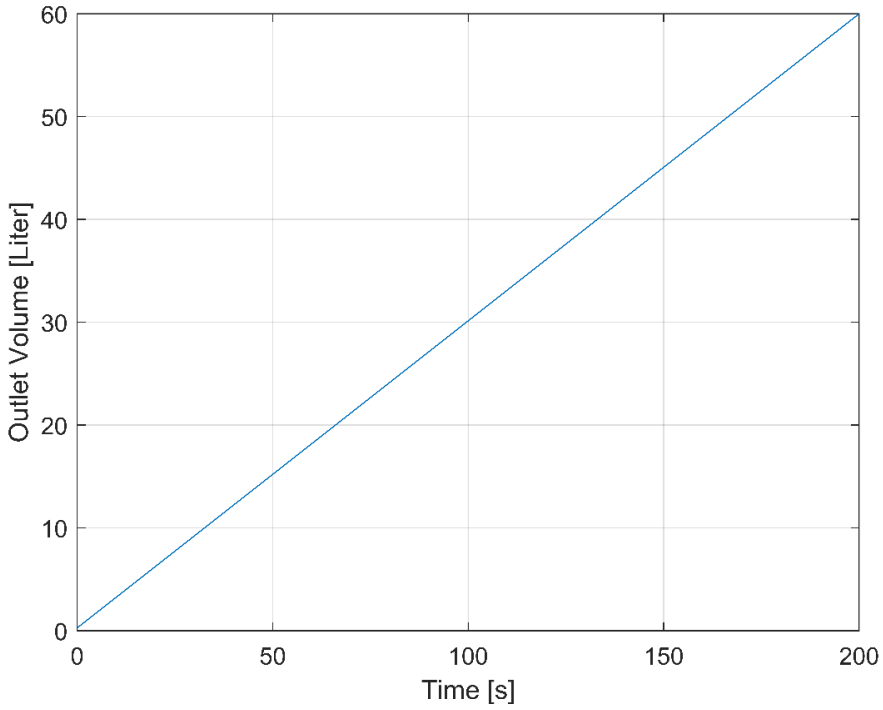

Figure 5.26: Outlet volume for scenario 4 of case 1

The results of scenario 1 and scenario 4 is very close to each other. However, in a close look the minor differences on outlet pressure can be seen.

The outlet pressure is compared for all scenarios in Figure 5.27. Scenario 1 and 4 shows the closest results to each other. Although all the scenarios start at the same point, none of them has the same value in the end.

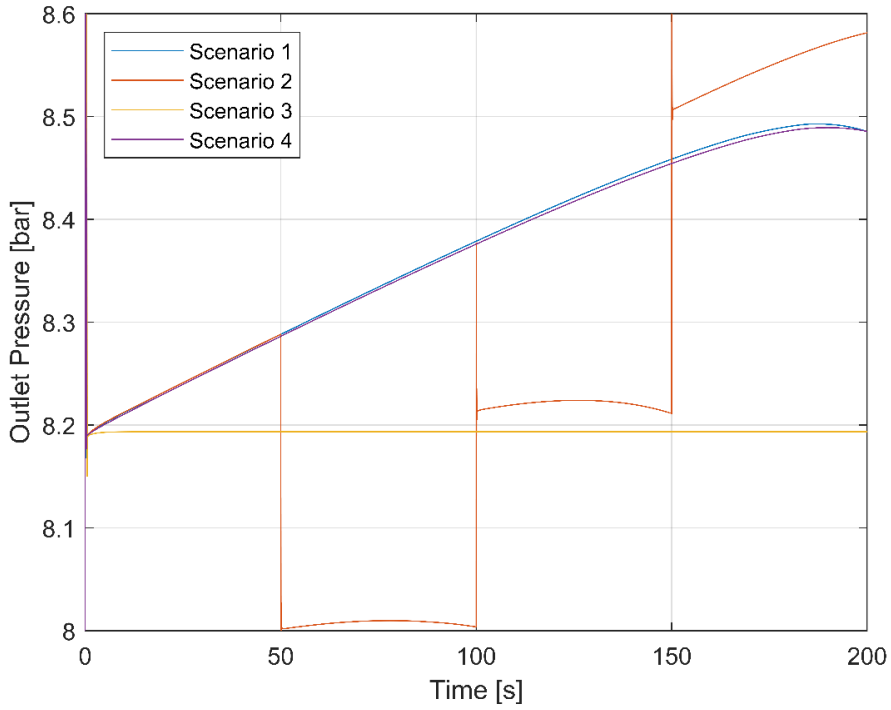

Figure 5.27: Outlet pressure comparison of all scenarios for case 1

5.3.2 Case 2

Case 2 requirements and design inputs are mentioned on Table 4.1 and Table 4.6. The main difference of the regulator of case 2 and case 1 is that case 2 has absolute regulator which means the atmospheric pressure does not affect the performance of this regulator. The purpose of this case to regulate the pressure to 18.5 Bar.

Scenario 1:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. On the other hand, mass flow rate and outlet volume are taken as fixed. The values of these parameters are given in Table 5.8. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.28, Figure 5.29, Figure 5.30.

Table 5.8: Input Parameters - Case 2 Scenario 1

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	45
Outlet volume [l]	0.270

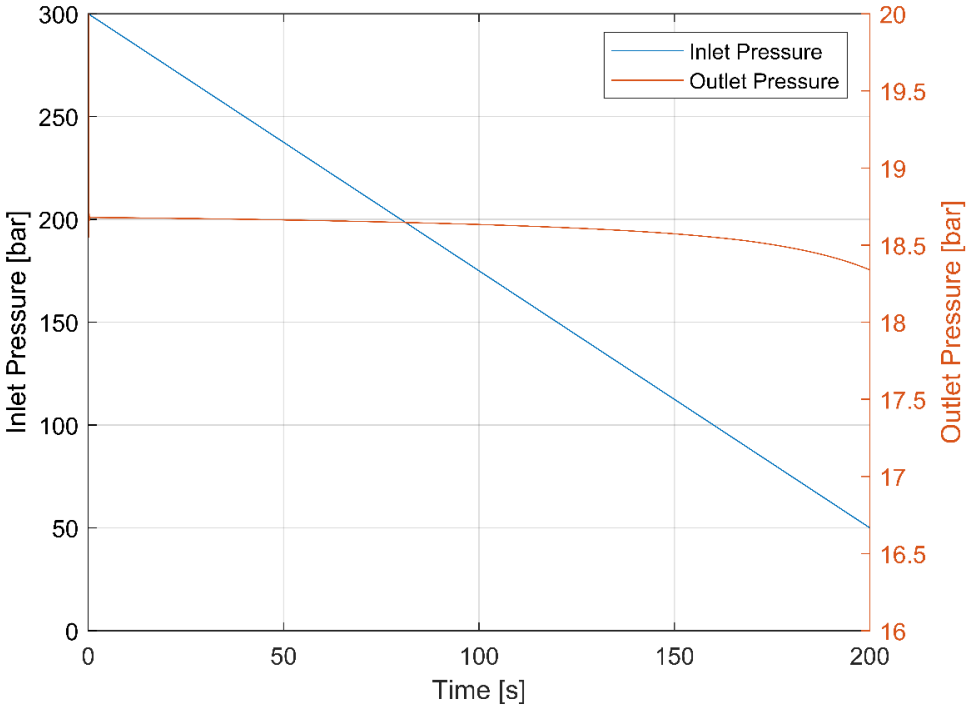

Figure 5.28: Inlet/Outlet Pressure for scenario 1 of case 2

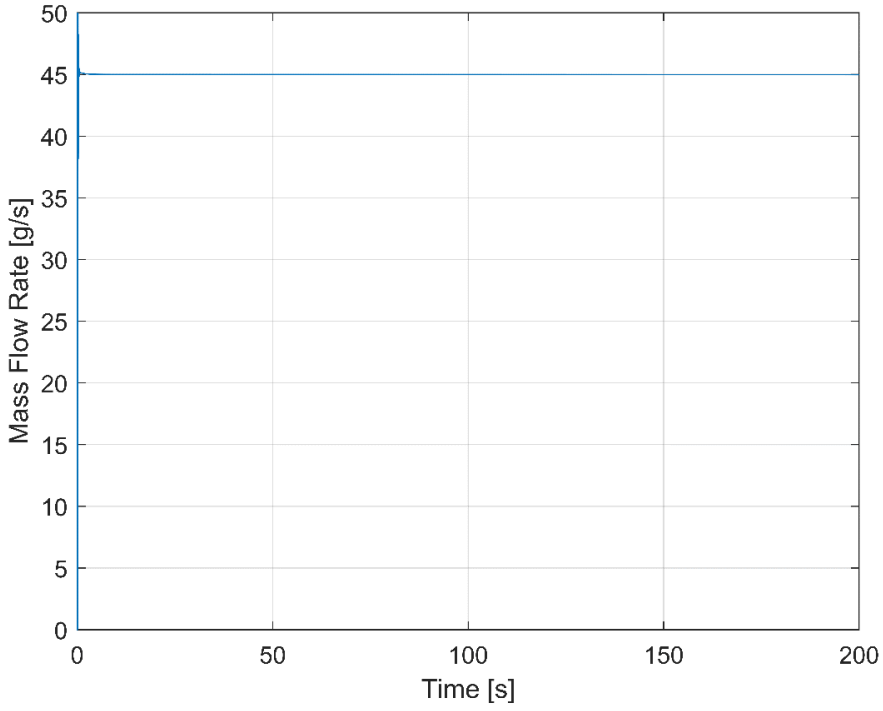

Figure 5.29: Mass flow rate for scenario 1 of case 2

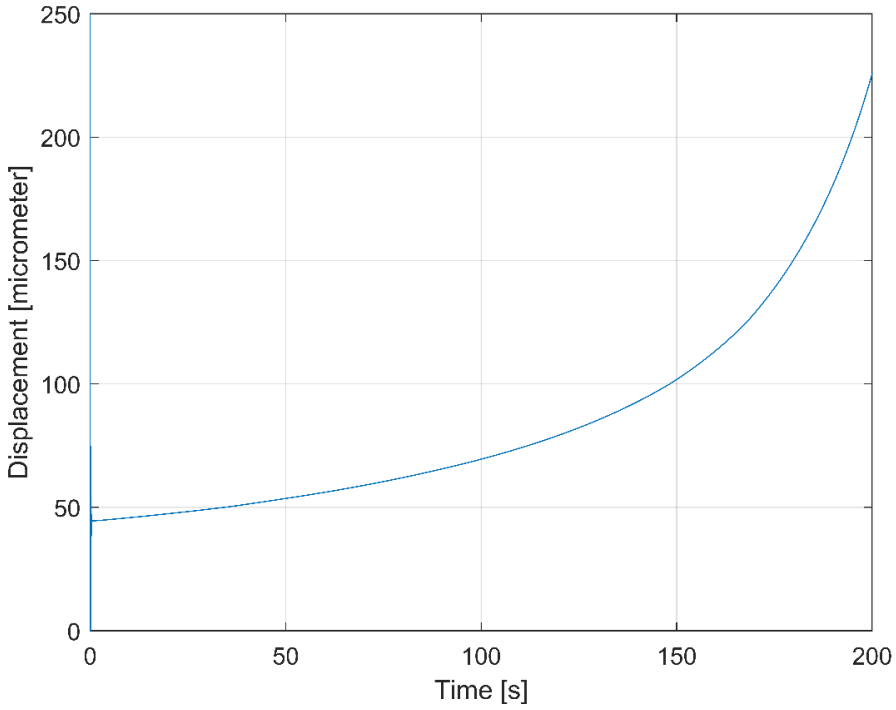

Figure 5.30: Poppet displacement for scenario 1 of case 2

Figure 5.28 shows that, the outlet pressure of the pressure regulator starts 18.65 bar and decreases as inlet pressure decreases. However, decreasing rate of the outlet pressure gradually increases. Figure 5.29 shows that, the mass flow rate stays still. On Figure 5.30, it can be seen that, the poppet displacement increases by time.

Scenario 2:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. In addition to that, mass flow rate is varying and it changes its value at every 50 s. The outlet volume is taken as fixed. The values of these parameters are given in Table 5.9. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.31, Figure 5.32, Figure 5.33.

Table 5.9: Input Parameters - Case 2 Scenario 2

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	[45 – 150 – 75 – 10 g/s] (varying)
Outlet volume [l]	0.270

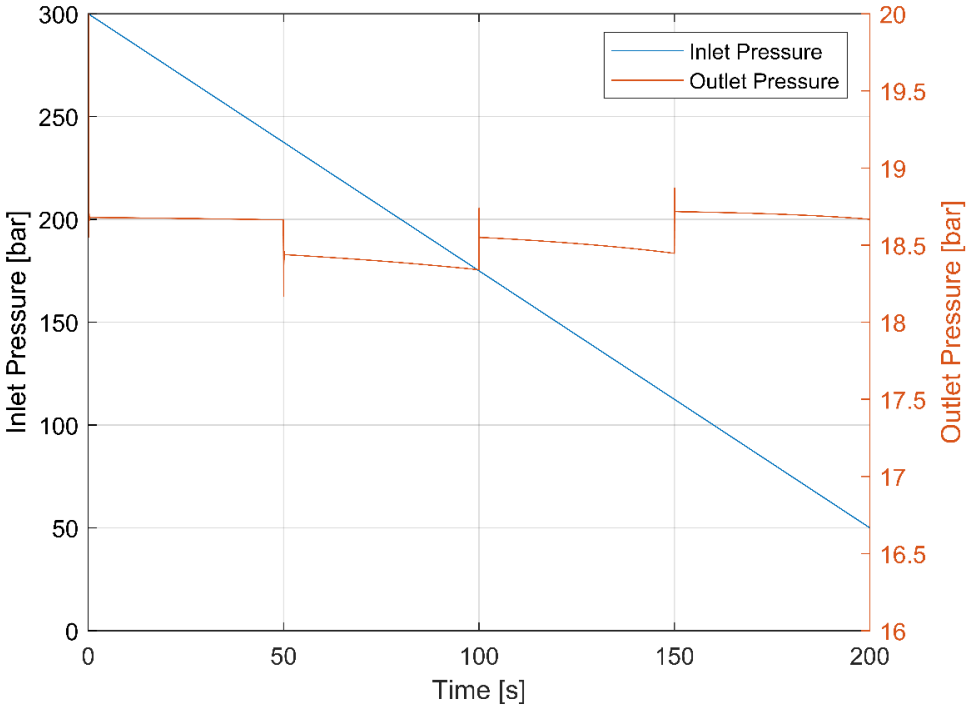

Figure 5.31: Inlet/Outlet Pressure for scenario 2 of case 2

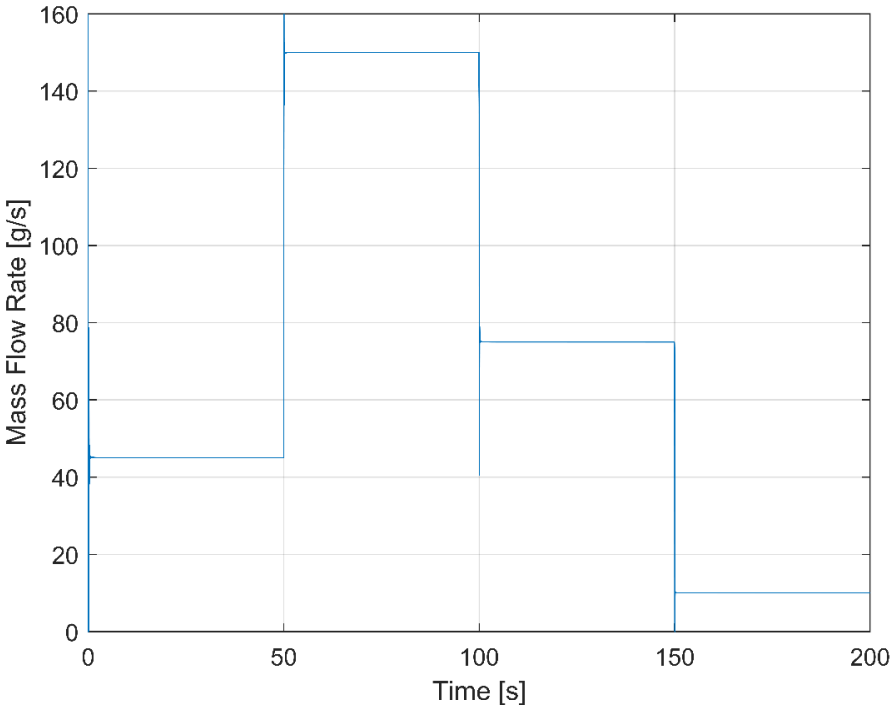

Figure 5.32: Mass flow rate for scenario 2 of case 2

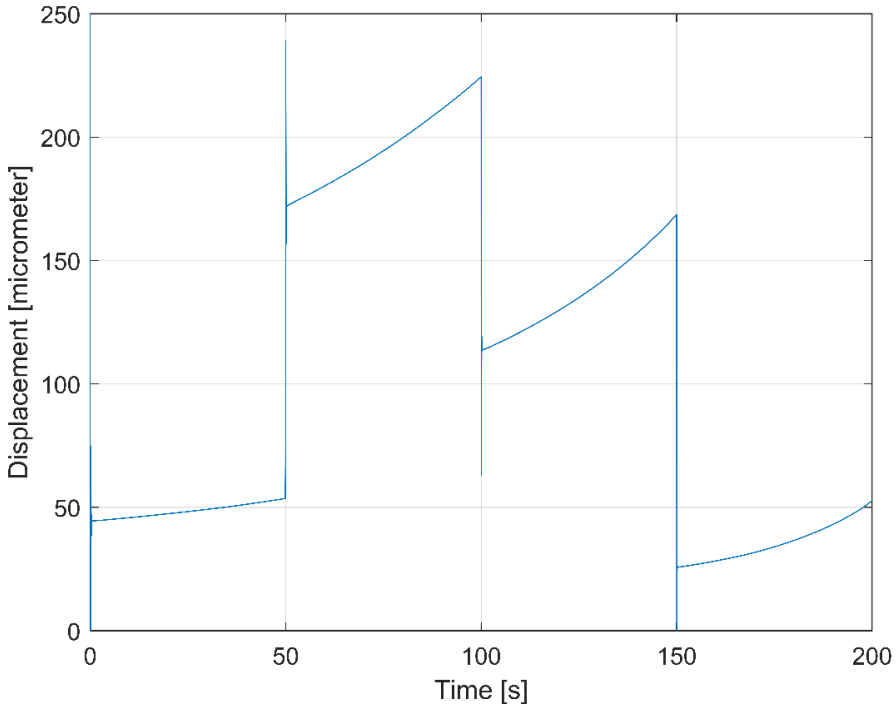

Figure 5.33: Poppet displacement for scenario 2 of case 2

Figure 5.31 shows that, the outlet pressure of the pressure regulator starts 18.65 bar and, at 50 s, it is down to 18.45 bar due to increase of mass flow rate to 145 g/s. When the mass flow increases, the outlet pressure is decreases. Figure 5.32 shows that, the mass flow rate varies. On Figure 5.33, it can be seen that, the poppet displacement increases as mass flow rate increases.

Scenario 3:

In this scenario, the inlet pressure, mass flow rate and outlet volume are fixed for all simulation time. The values of these parameters are given in Table 5.10. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.34, Figure 5.35, Figure 5.36.

Table 5.10: Input Parameters - Case 2 Scenario 3

Parameter	Value
Inlet pressure [Bar]	300
Mass flow rate [g/s]	45
Outlet volume [l]	0.270

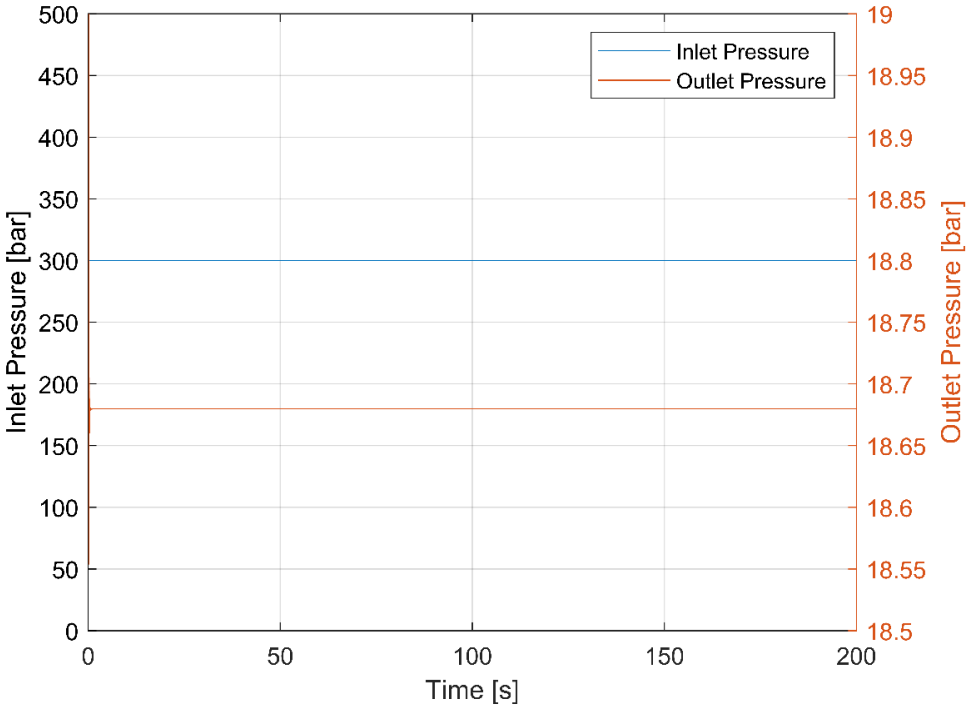

Figure 5.34: Inlet/Outlet Pressure for scenario 3 of case 2

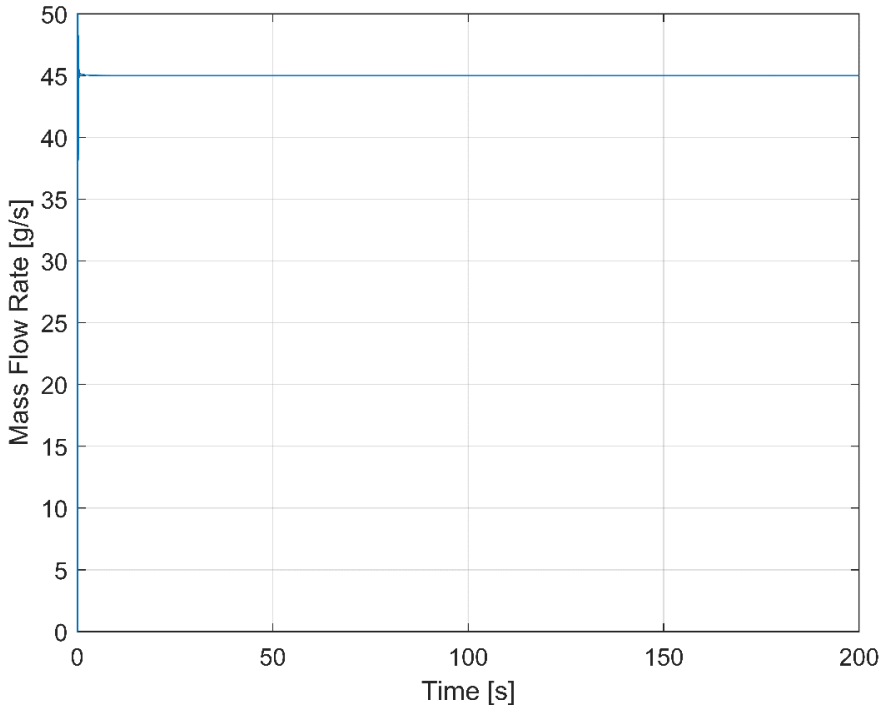

Figure 5.35: Mass flow rate for scenario 3 of case 2

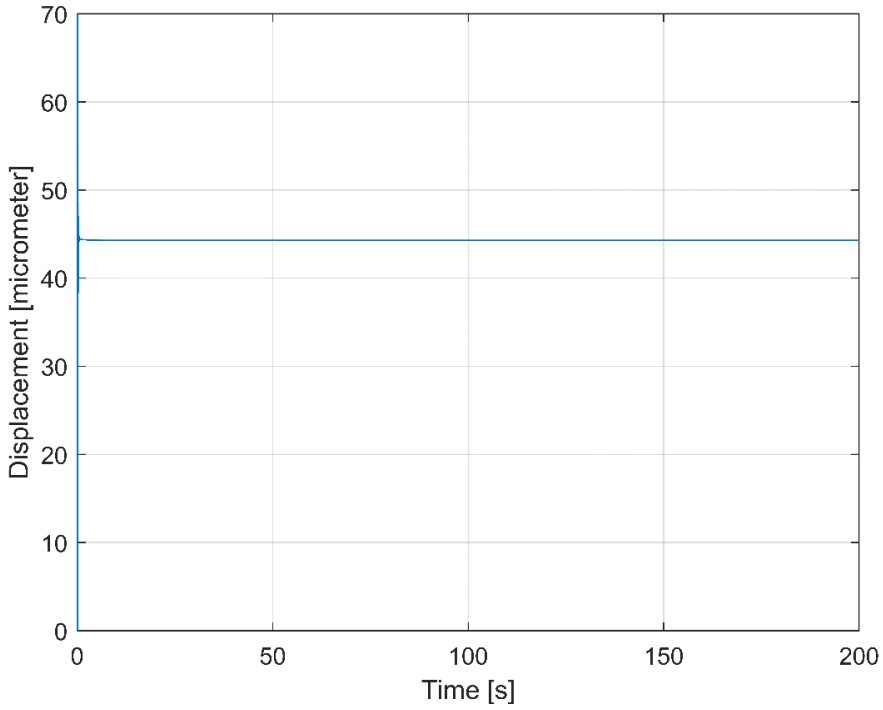

Figure 5.36: Poppet displacement for scenario 3 of case 2

Figure 5.34, Figure 5.35 and Figure 5.36 shows that, the outlet pressure and x displacement of the pressure regulator stays still when there is no change in inlet pressure and mass flow rate.

Scenario 4:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. However, the outlet volume is increasing. On the other hand, mass flow rate is taken as fixed. The values of these parameters are given in Table 5.11. The simulation results of inlet/outlet pressure, mass flow rate, x displacement and outlet volume are shown in Figure 5.37, Figure 5.38, Figure 5.39, Figure 5.40.

Table 5.11: Input Parameters- Case 2 Scenario 4

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	45
Outlet volume [l]	0.270-60 (increasing)

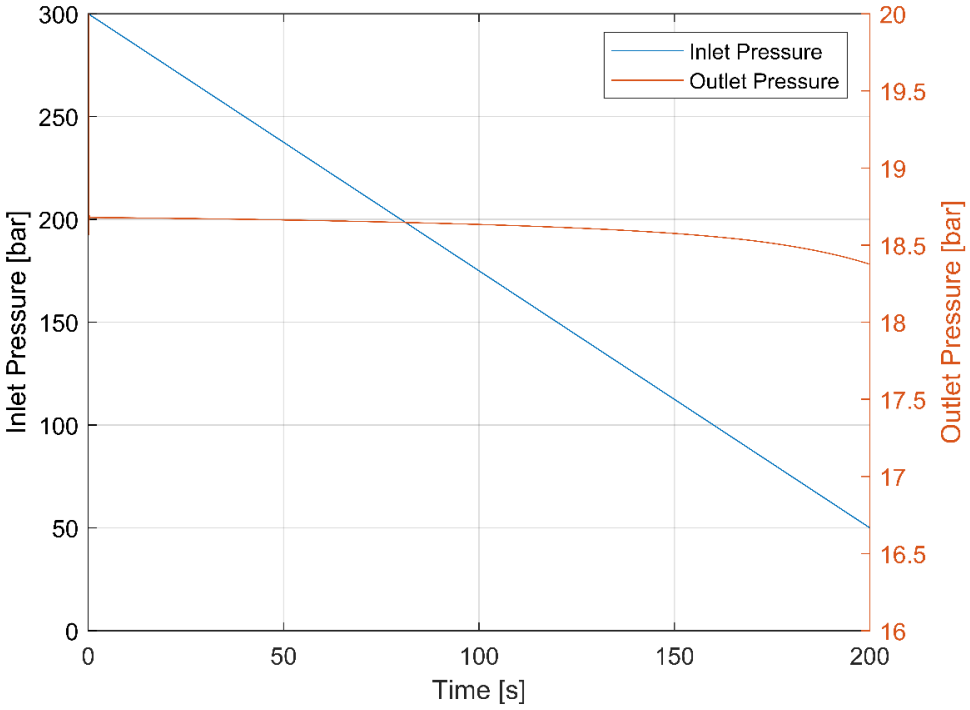

Figure 5.37: Inlet/Outlet Pressure for scenario 4 of case 2

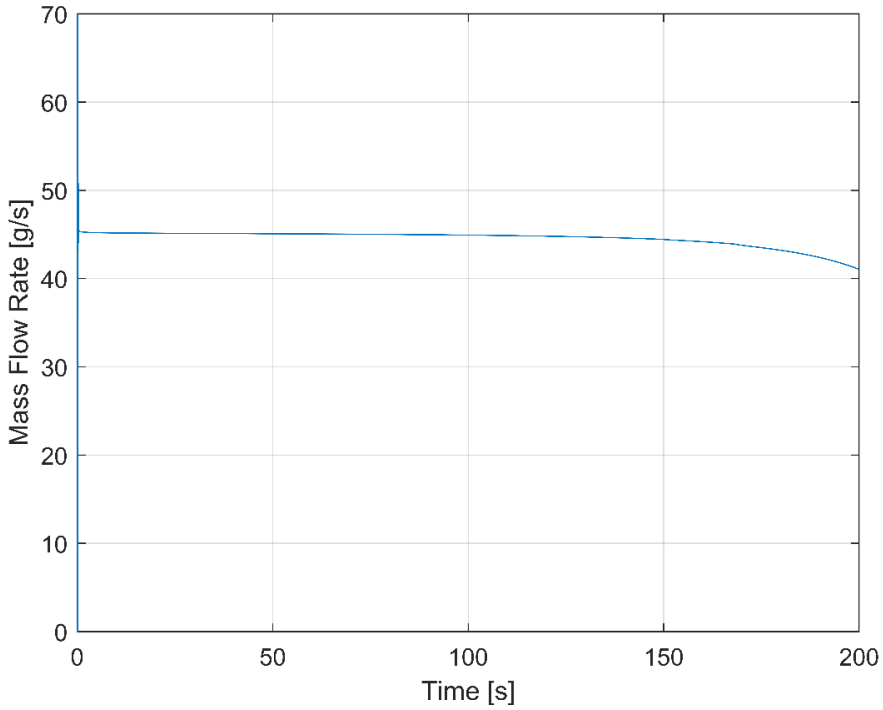

Figure 5.38: Mass flow rate for scenario 4 of case 2

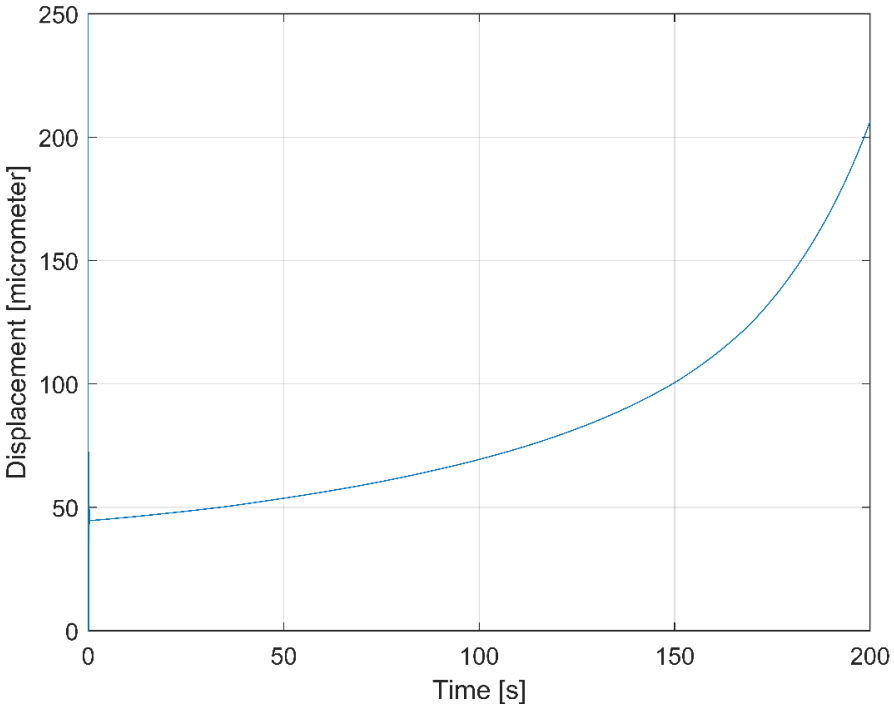

Figure 5.39: Poppet displacement for scenario 4 of case 2

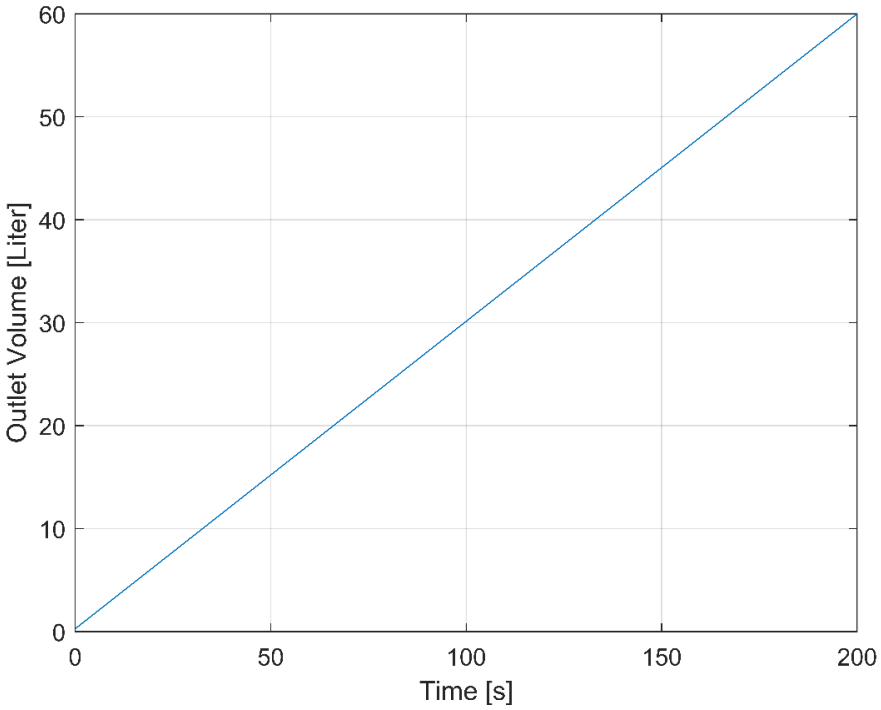

Figure 5.40: Outlet volume for scenario 4 of case 2

The results of scenario 1 and scenario 4 is very close to each other. However, in a close look the minor differences on outlet pressure can be seen.

The outlet pressure is compared for all scenarios on Figure 5.41. Scenario 1 and 4 shows the closest results to each other. Although all the scenarios start at the same point, none of them has the same value in the end.

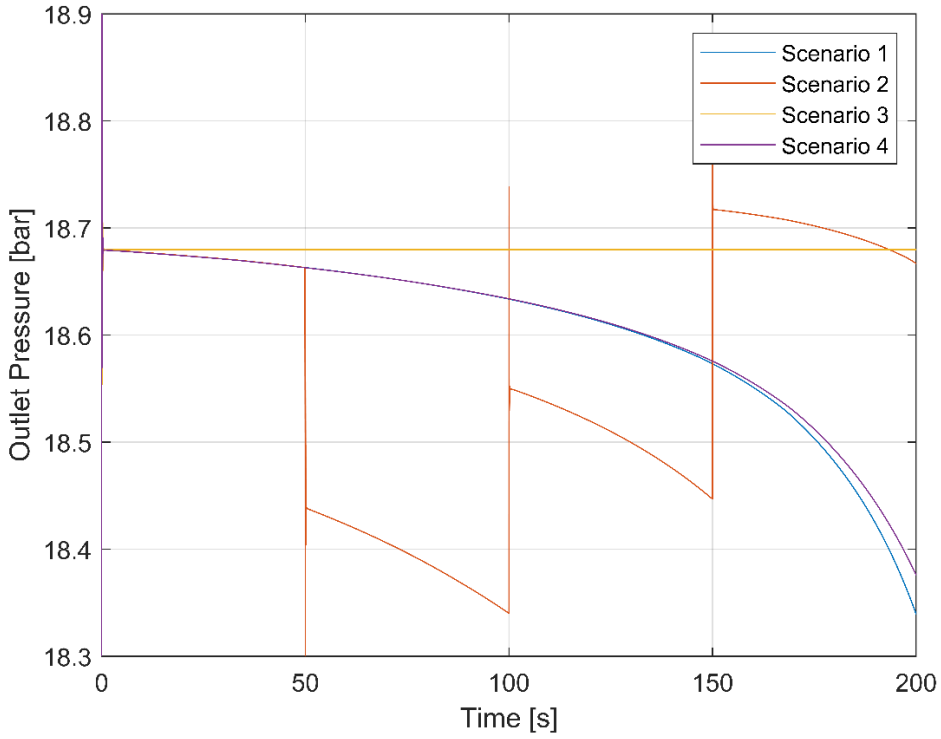

Figure 5.41: Outlet pressure comparison of all scenarios for case 2

5.3.3 Case 3

Case 3 requirements and design inputs are mentioned on Table 4.1 and Table 4.6. The main difference between the other two case and case 3 is that case 3 uses helium as fluid while the other two use nitrogen. The purpose of this case to regulate the pressure to 23.5 Bar.

Scenario 1:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. On the other hand, mass flow rate and outlet volume are taken as fixed. The values of these parameters are given in Table 5.12. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.42, Figure 5.43, Figure 5.44.

Table 5.12: Input Parameters - Case 3 Scenario 1

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	55
Outlet volume [l]	1.5

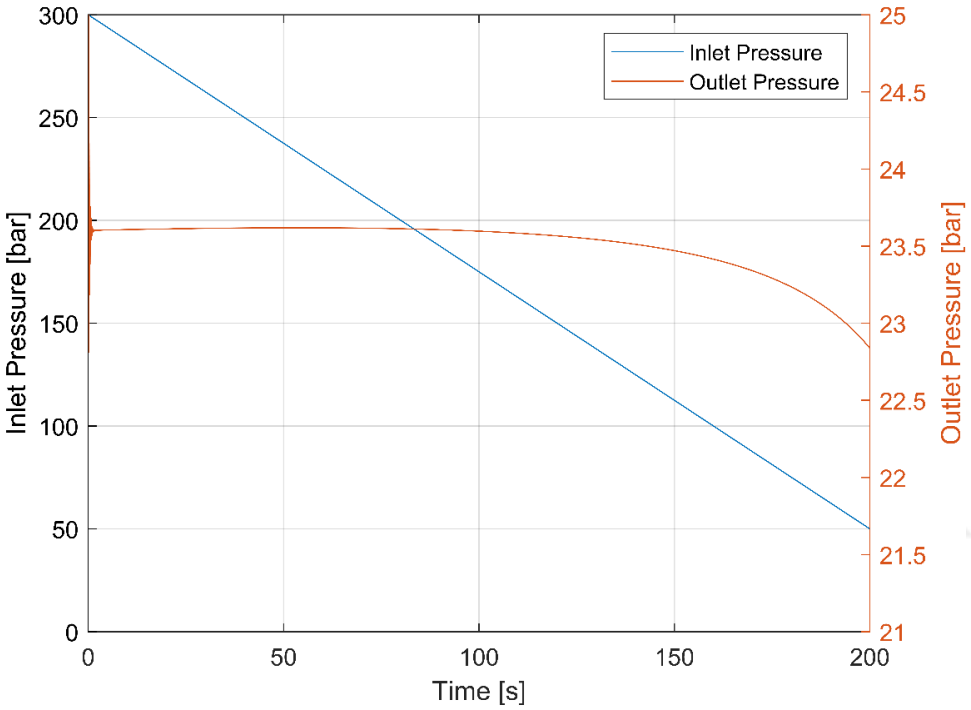

Figure 5.42: Inlet/Outlet Pressure for scenario 1 of case 3

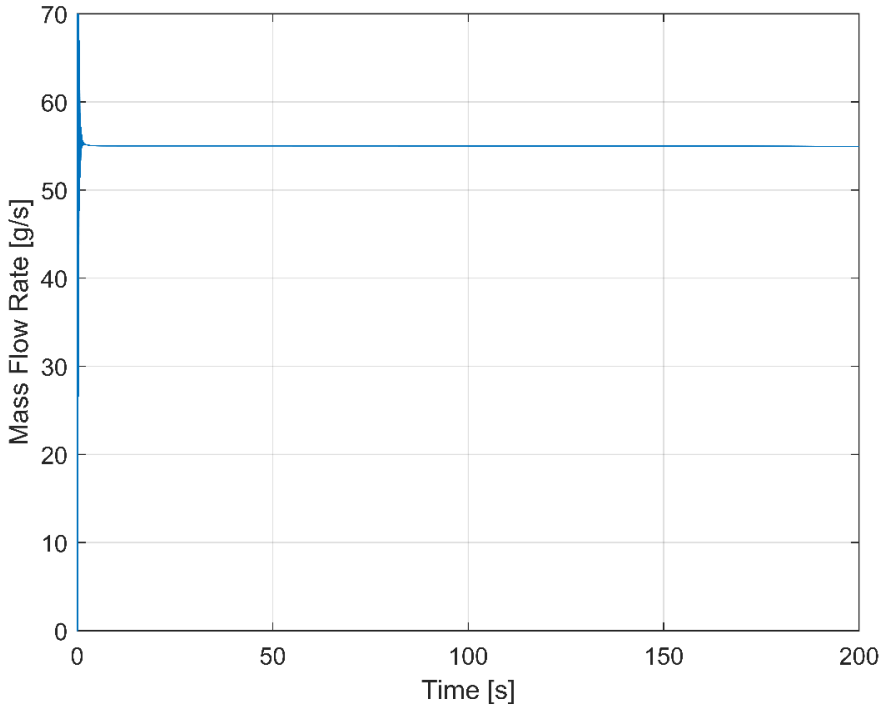

Figure 5.43: Mass flow rate for scenario 1 of case 3

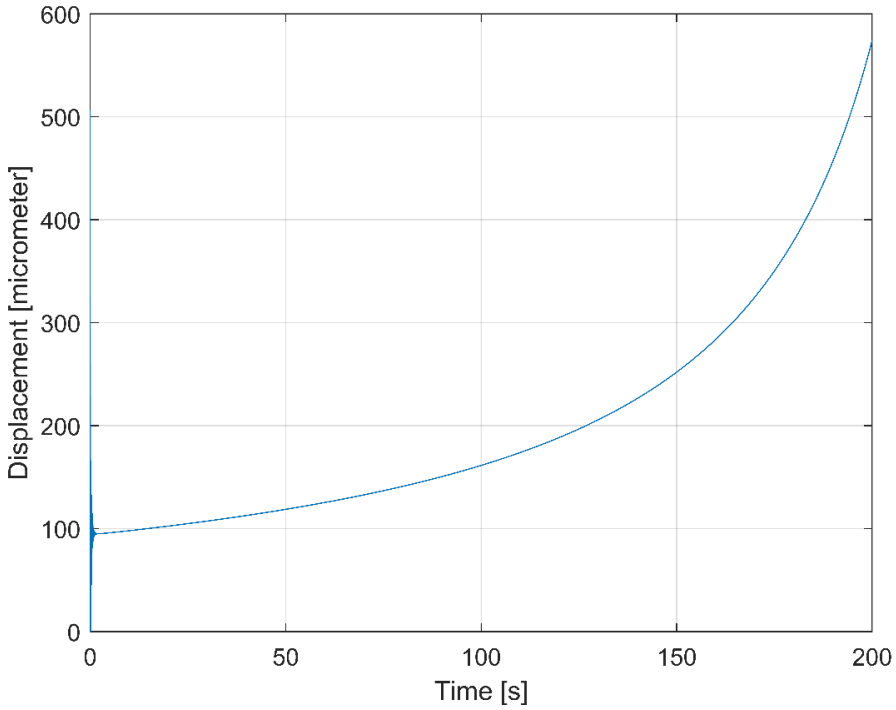

Figure 5.44: Poppet displacement for scenario 1 of case 3

Figure 5.42 shows that, the outlet pressure of the pressure regulator starts 23.55 bar and decreases as inlet pressure decreases. However, decreasing rate of the outlet pressure gradually increases. Figure 5.43 shows that, the mass flow rate stays still. On Figure 5.44, it can be seen that, the poppet displacement increases by time.

Scenario 2:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. In addition to that, mass flow rate is varying and it changes its value at every 50 s. The outlet volume is taken as fixed. The values of these parameters are given in Table 5.13. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.45, Figure 5.46, Figure 5.47.

Table 5.13: Input Parameters - Case 3 Scenario 2

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	[55 – 250 – 125 – 20 g/s] (varying)
Outlet volume [l]	1.5

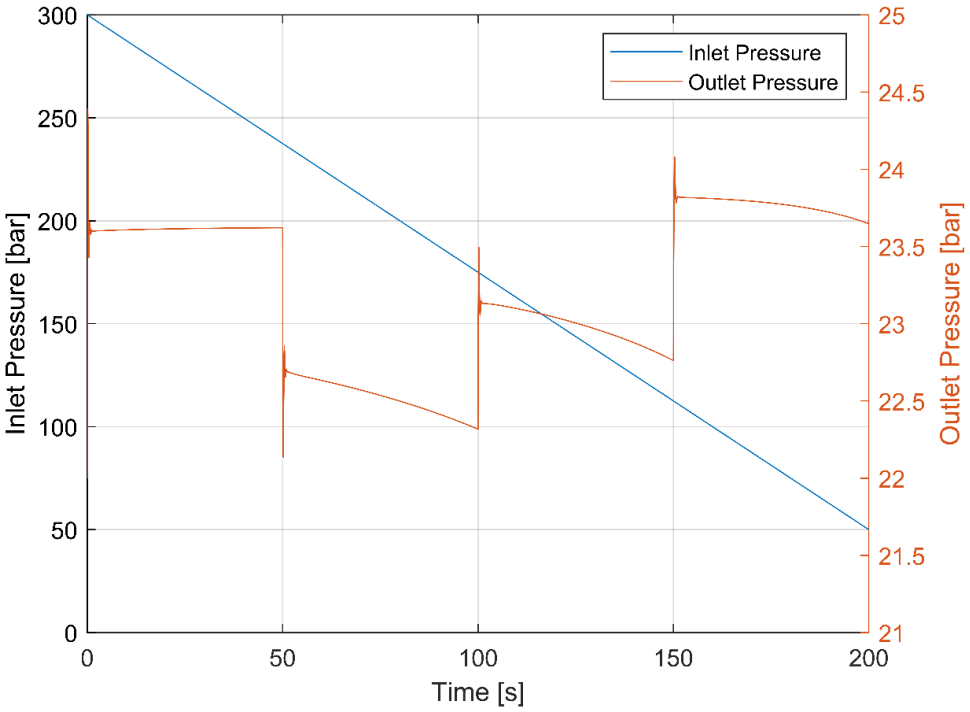

Figure 5.45: Inlet/Outlet Pressure for scenario 2 of case 3

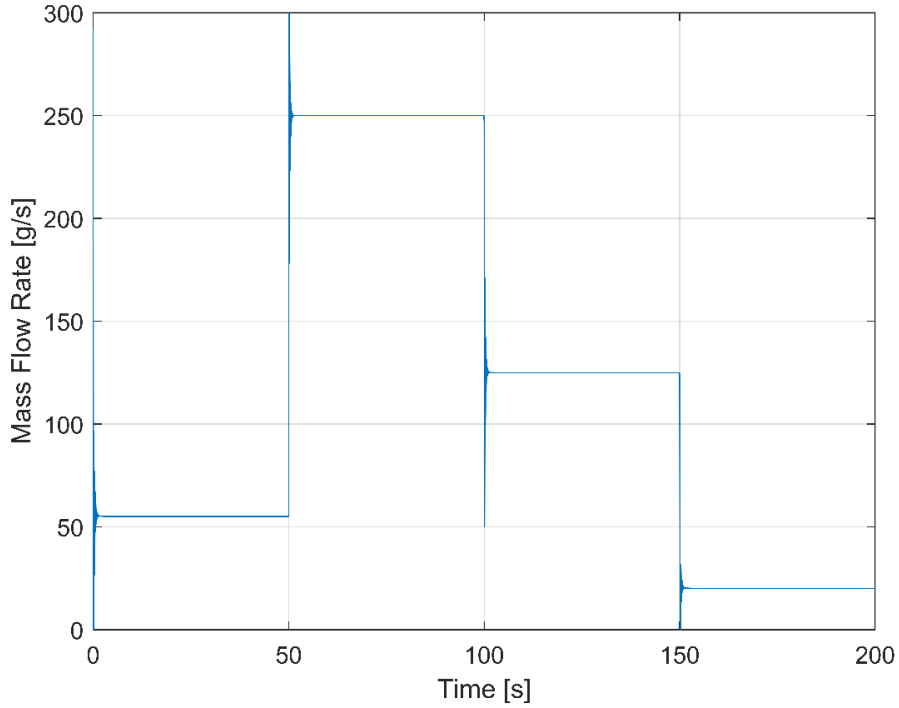

Figure 5.46: Mass flow rate for scenario 2 of case 3

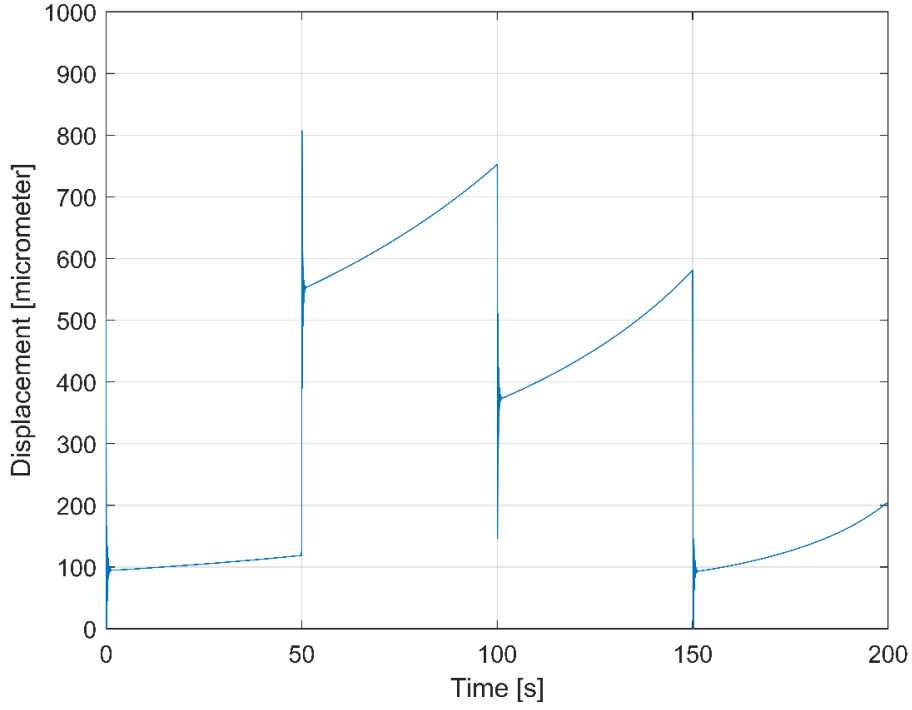

Figure 5.47: Poppet displacement for scenario 2 of case 3

Figure 5.45 shows that, the outlet pressure of the pressure regulator starts 23.55 bar and, at 50 s, it is down to 22.6 bar due to increase of mass flow rate to 250 g/s. When the mass flow increases, the outlet pressure is decreases. Figure 5.46 shows that, the mass flow rate varies. On Figure 5.47, it can be seen that, the poppet displacement increases as mass flow rate increases.

Scenario 3:

In this scenario, the inlet pressure, mass flow rate and outlet volume are fixed for all simulation time. The values of these parameters are given in Table 5.14. The simulation results of inlet/outlet pressure, mass flow rate and x displacement are shown in Figure 5.48, Figure 5.49, Figure 5.50.

Table 5.14: Input Parameters - Case 3 Scenario 3

Parameter	Value
Inlet pressure [Bar]	300
Mass flow rate [g/s]	55
Outlet volume [l]	1.5

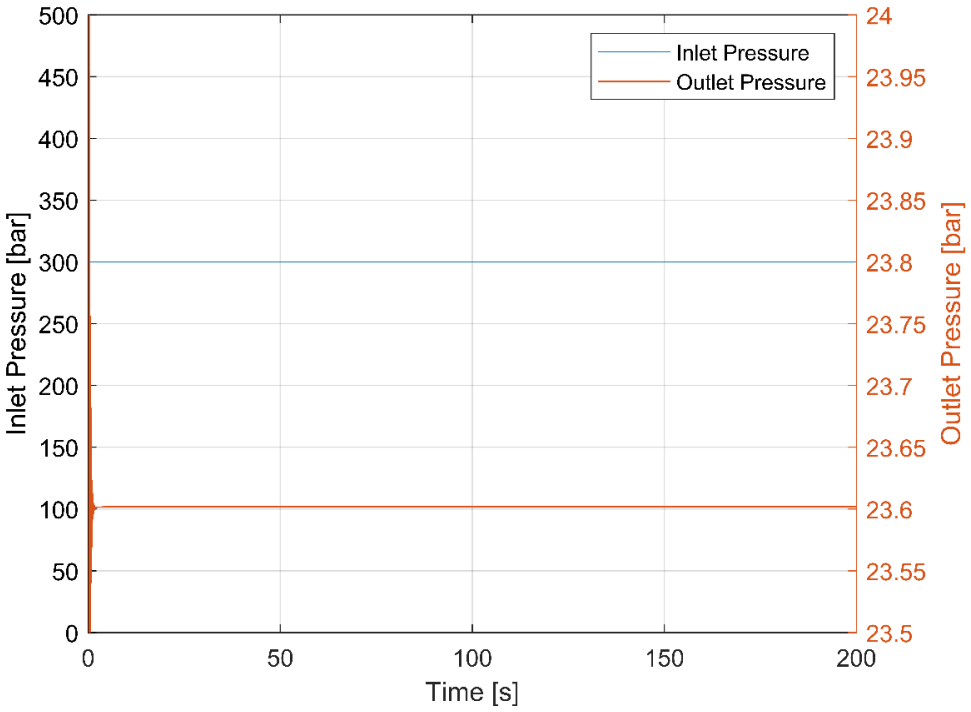

Figure 5.48: Inlet/Outlet Pressure for scenario 3 of case 3

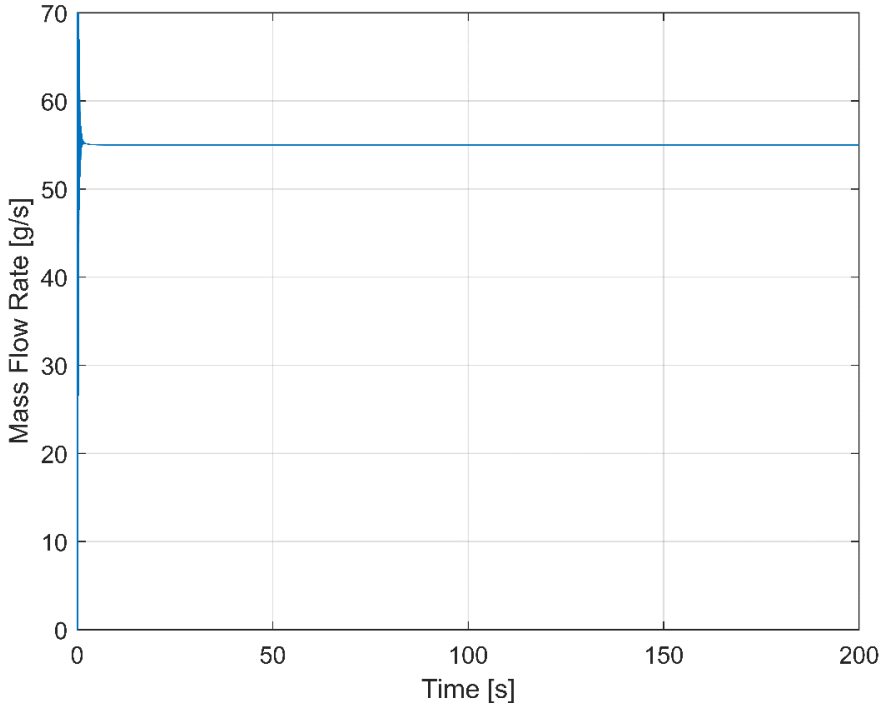

Figure 5.49: Mass flow rate for scenario 3 of case 3

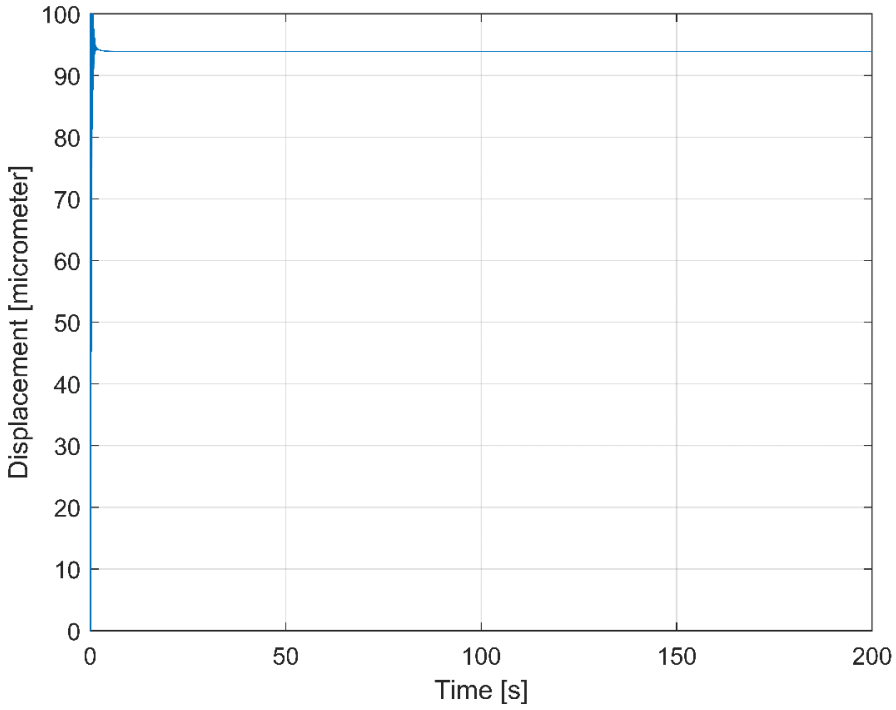

Figure 5.50: Poppet displacement for scenario 3 of case 3

Figure 5.48, Figure 5.49 and Figure 5.50 shows that, the outlet pressure and x displacement of the pressure regulator stays still when there is no change in inlet pressure and mass flow rate.

Scenario 4:

In this scenario, the inlet pressure is decreasing linearly for all simulation time. However, the outlet volume is increasing. On the other hand, mass flow rate is taken as fixed. The values of these parameters are given in Table 5.15. The simulation results of inlet/outlet pressure, mass flow rate, x displacement and outlet volume are shown in Figure 5.51, Figure 5.52, Figure 5.53, Figure 5.54.

Table 5.15: Input Parameters- Case 3 Scenario 4

Parameter	Value
Inlet pressure [Bar]	300-50 (decreasing)
Mass flow rate [g/s]	55
Outlet volume [l]	1.5-100 (increasing)

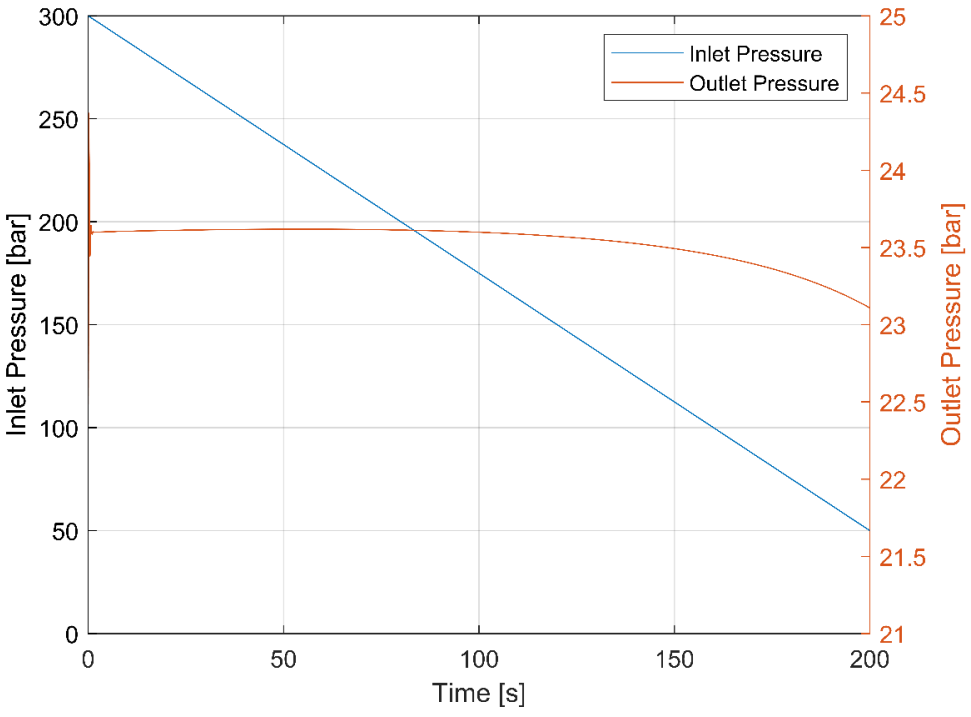

Figure 5.51: Inlet/Outlet Pressure for scenario 4 of case 3

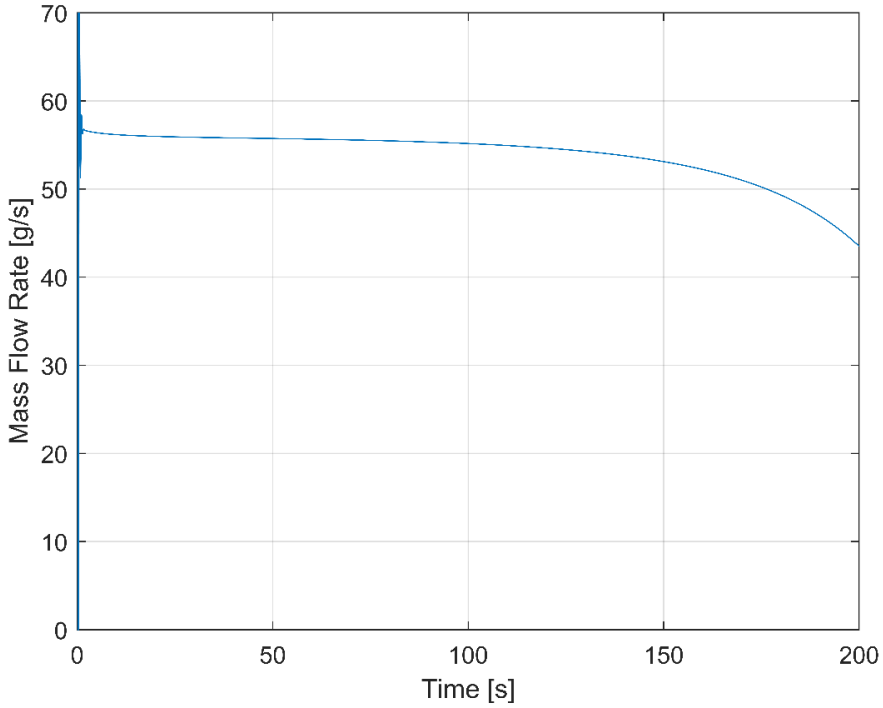

Figure 5.52: Mass flow rate for scenario 4 of case 3

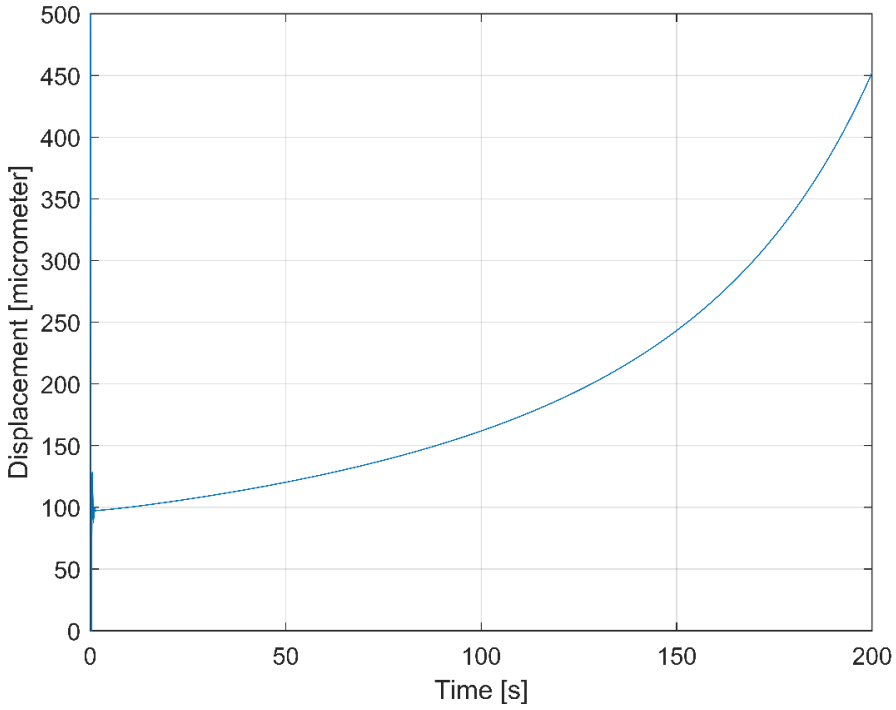

Figure 5.53: Poppet displacement for scenario 4 of case 3

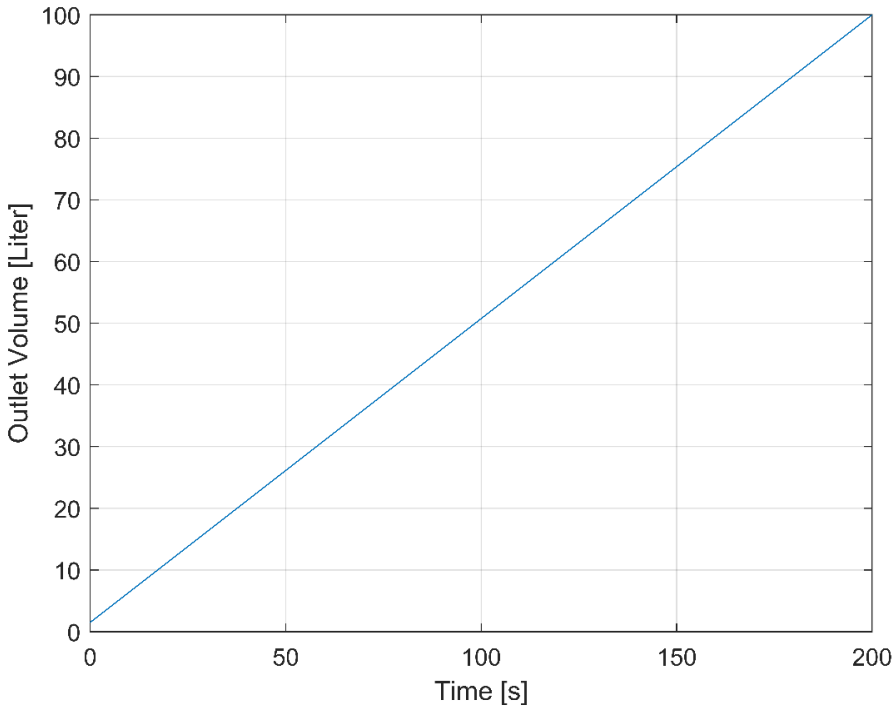

Figure 5.54: Outlet volume for scenario 4 of case 3

The results of scenario 1 and scenario 4 is very close to each other. However, in a close look the minor differences on outlet pressure can be seen.

The outlet pressure is compared for all scenarios in Figure 5.55. Scenario 1 and 4 shows the closest results to each other. Although all the scenarios start at the same point, none of them has the same value in the end.

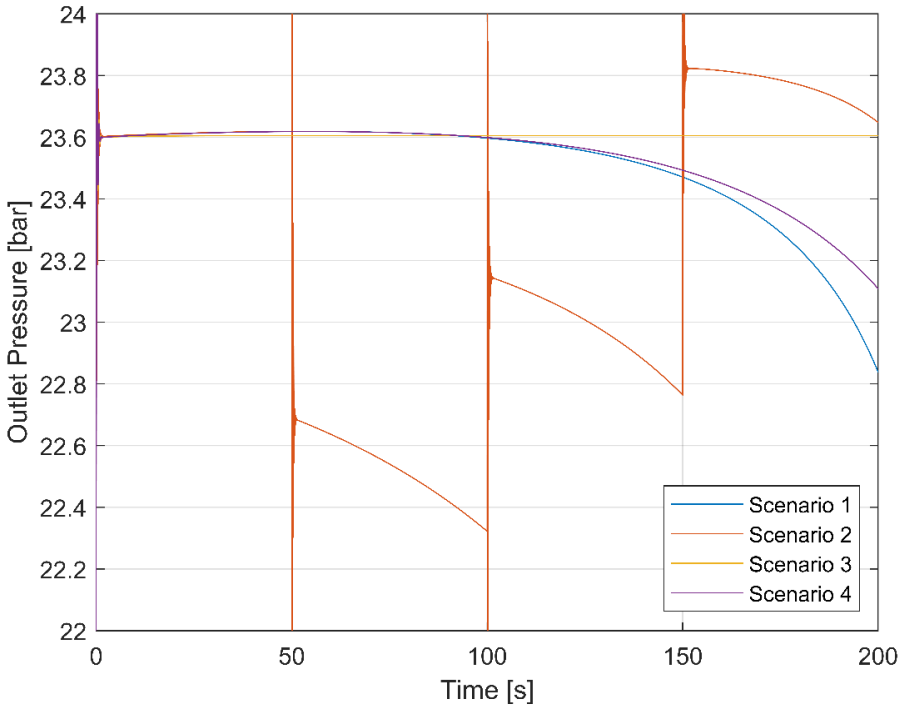

Figure 5.55: Outlet pressure comparison of all scenarios for case 3

6. CONCLUSION

In this study, the design steps of a pressure regulator for liquid propellant rocket engines, its mathematical modeling, dynamic model and the simulations performed by using these models are explained in detail. During the preparation of the mathematical model, it is emphasized that different modeling approaches can arise for different scenarios, and therefore separate models are presented.

To validate the obtained dynamic model, tests were performed with three different regulators. The test inputs for each regulator were applied to the simulation and the simulation outputs were obtained under the same conditions as the test conditions. The obtained simulation outputs and test results were compared separately for each regulator. The comparison revealed a deviation of around 2-3% in one of the three regulators, while the other two regulators exhibited an error rate of around 0.3%. These results demonstrate that the simulation outputs are satisfactory and validate the developed mathematical and dynamic models.

After confirming the validity of the designed model for the pressure regulator, simulations were performed on three different pressure regulators. The requirements and design inputs for each regulator were specified separately. These regulators differ in terms of dimensions, set pressure and mass flow rate variations, and the fluid they regulate. Simulations were performed with four different scenarios for each regulator, and the results were compared among themselves. Scenarios include variations in inlet pressure, mass flow rate, and outlet volume.

In the results obtained, it has been seen that the variable inlet pressure creates differences in the performance of the regulator. It has been observed that although the outlet pressure characteristic is preserved at different mass flow rates for the same regulator, the outlet pressure changes. It was also observed that the variable outlet volume did not affect the operating characteristics of the regulator significantly, but it caused slight deviations in the outlet pressure compared to a constant outlet volume.

In future work, an optimization study will be performed to minimize the impact of the variables such as inlet pressure, mass flow rate, and outlet volume. The optimized regulator will be produced and tested, and the validity of the optimization will be verified.

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