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DESIGN, MODELLING AND FATIGUE LIFE ANALYSIS OF A 450 KG LPG CYLINDER

SUBMITTED BY

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(079MSMDE021)

A THESIS

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The undersigned certify that they have read, and recommended to the Institute of Engineering for acceptance, a thesis entitled **" Design, Modelling and Fatigue Life Analysis of a 450 kg LPG cylinder "** submitted by Swikrit Dahal in partial fulfillment of the requirements for the degree of Master of Science in Mechanical System Design and Engineering.

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ABSTRACT

With the increasing use of large-capacity LPG cylinders in Nepal's industrial sector, evaluating their structural safety and fatigue performance has become necessary. Unlike small domestic cylinders, 450 kg LPG cylinders operate under higher internal pressure and are exposed to repeated loading during filling, transportation, and discharge. Because of these repeated pressure cycles, fatigue becomes an important factor in their design and long-term reliability.

The present study focuses on the design, modelling, and fatigue life analysis of a 450 kg LPG cylinder based on the requirements of EN 13445 and ASME Section VIII Division 2. A detailed three-dimensional CAD model of the cylinder was developed and analysed in ANSYS Workbench using finite element analysis to study the stress distribution under a design pressure of 1.76 MPa. The analysis also considered the effect of self-weight, nozzle loading, and fixed tripod supports.

From the analytical calculations, the hoop stress and longitudinal stress in the cylindrical shell were found to remain within the allowable design limits. However, the finite element analysis showed that higher stresses developed near geometric discontinuities. The maximum equivalent stress was observed around the bracket welds and nozzle regions, where the peak stress reached 369.04 MPa. While the main shell experienced relatively uniform stress, these localized regions were identified as the most critical areas of the structure.

The fatigue evaluation was performed using both the stress-life (S-N) method and the ASME fatigue assessment procedure. The results showed that the support bracket areas were the most fatigue-sensitive regions of the cylinder. The minimum predicted fatigue

life was approximately 70,266 cycles. In addition, the cumulative fatigue damage factor remained well below unity, indicating that the cylinder can safely withstand the expected service loading conditions throughout its design life.

Overall, the study shows that the pressure-retaining shell of the cylinder performs safely under both static and cyclic loading conditions. At the same time, the fatigue performance of the vessel is strongly influenced by the geometry of the support structure and the quality of welds at attachment locations. The findings of this research can be useful for improving the design, fabrication, and inspection practices of large-capacity LPG cylinders in Nepal.

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LIST OF SYMBOLS

P	Internal pressure (MPa)
D	Internal diameter (mm)
T	Thickness of shell (mm)
D_i	Internal diameter of cylinder (mm)
R	Internal radius (mm)
L	Length of cylindrical shell (mm)
V	Volume of cylinder (m^3)
h_i	Height of ellipsoidal head (mm)
R	Crown radius (mm)
r_k	Knuckle radius (mm)
σ_{eq}	Equivalent (Von Mises) stress (MPa)
σ_a	Alternating stress amplitude (MPa)
σ_y	Yield strength (MPa)
σ_u	Ultimate tensile strength (MPa)
E	Modulus of elasticity (MPa)
ν	Poisson's ratio
f	Allowable stress (MPa)
η	Joint efficiency
Z	Joint factor
C	Corrosion allowance (mm)
C_i	Internal corrosion allowance (mm)
C_o	External corrosion allowance (mm)
AR	Aspect ratio of ellipsoidal head

K	Geometric factor
β	Shape factor
N	Number of cycles
D_f	Fatigue damage factor
S	Stress amplitude (MPa)
X, Y	Fatigue parameters (ASME formulation)
δ_r	Radial deformation (mm)

LIST OF ABBREVIATIONS

LPG	Liquefied Petroleum Gas
FEA	Finite Element Analysis
FEM	Finite Element Method
CAD	Computer-Aided Design
NDT	Non-Destructive Testing
ASME	American Society of Mechanical Engineers
BPVC	Boiler and Pressure Vessel Code
EN	European Standard
DBF	Design by Formula
DBA	Design by Analysis
UTS	Ultimate Tensile Strength
YS	Yield Strength
S-N Curve	Stress Number of Cycles Curve
MPa	Mega Pascal
mm	Millimetre
kg	Kilogram
L	Liter

CHAPTER 1 INTRODUCTION

1.1 Background

Liquefied Petroleum Gas (LPG) is one of the major energy sources used in Nepal for domestic, commercial, and industrial purposes. The commonly used 14.2 kg household cylinder has already been studied extensively and is covered by established safety regulations and standards. However, the same level of technical study is not available for 450 kg industrial LPG cylinders. These large-capacity cylinders are gradually being introduced in Nepal for industrial and commercial applications, especially in hotels, restaurants, factories, and heating systems. Their increasing use, together with limited technical research and regulatory experience, makes it necessary to study their structural safety and long-term performance.

Nepal depends heavily on LPG imported from India to satisfy its growing energy demand. According to the Department of Customs, Nepal (2081/82), the country imported around 554.6 million kilograms of LPG during the fiscal year 2081/82. The total import value was approximately NPR 625.9 million, while the import revenue was about NPR 11.7 million (Department of Customs, Nepal, 2081/82). These figures show the increasing use of LPG across different sectors in Nepal. Since the country relies mainly on imported LPG, maintaining the safety and reliability of storage cylinders has become an important issue. With the gradual rise in the use of large industrial cylinders such as 450 kg cylinders, proper design standards and safety evaluation procedures are becoming even more necessary in Nepal.

LPG cylinders basically function as pressure vessels and are subjected to repeated pressure loading during filling, transportation, and discharge operations. These repeated loading and unloading cycles generate fluctuating stresses in the shell, head junctions, nozzles, and support regions of the cylinder. Over a long service period, repeated stress variation can initiate fatigue cracks, even when the stresses are below the yield strength of the material. International standards such as EN 13445-3 provide guidelines for evaluating fatigue behaviour in pressure vessels, but only limited studies are available specifically for large-capacity LPG cylinders such as the 450 kg type (CEN, 2021).

In recent years, finite element analysis has become an effective tool for predicting stress distribution and fatigue behaviour in complex pressure vessel structures. Compared to conventional analytical methods, numerical simulation can identify localized stress concentration regions with better accuracy. Non-destructive testing methods further help in assessing the condition of the vessel without causing damage to the structure. However, most published research related to LPG cylinders mainly focuses on smaller domestic cylinders, while studies related to industrial-scale cylinders are still limited (Zienkiewicz et al., 2013).

Therefore, the present research focuses on the design, finite element modelling, and fatigue life assessment of a 450 kg LPG cylinder using both analytical calculations and numerical simulation techniques. The analysis is carried out according to the requirements of EN 13445 and ASME Section VIII Division 2 (CEN, 2021). The study mainly aims to evaluate the stress distribution and fatigue performance of the cylinder and provide useful technical information for improving the safety and reliability of large-capacity LPG cylinders in Nepal.

1.2 Statement of problem

The 450 kg LPG cylinder is mainly intended for industrial and bulk storage applications, and its use in Nepal has only recently started receiving attention under national standards. While 14.2 kg cylinders are generally sufficient for household use, industries, restaurants, hotels, and commercial facilities require a much larger and continuous LPG supply. Because of this, users often need to replace smaller cylinders repeatedly or depend on large LPG bank systems. Frequent cylinder replacement is not only inconvenient but also inefficient for industrial operations. At the same time, poorly managed LPG bank systems may create additional safety risks if proper guidelines and standards are not followed.

Although the use of 450 kg LPG cylinders is increasing, there are still very limited technical studies related to their structural strength, fatigue behaviour, and long-term performance. Unlike smaller domestic cylinders, these large-capacity cylinders operate under higher internal pressure and experience repeated loading during filling, transportation, and emptying. Such repeated pressure cycles can gradually produce fatigue damage in critical regions such as welded joints, nozzles, and support attachments (Stephens et al., 2001).

At present, only limited research is available regarding the structural and fatigue behaviour of 450 kg LPG cylinders under cyclic loading conditions. Without detailed analysis and proper validation, there may be uncertainties regarding their safety and service reliability during long-term operation. Therefore, a detailed study on the design, stress distribution, and fatigue life assessment of 450 kg LPG cylinders is necessary to support their safe and reliable application in Nepalese industrial sectors.

1.3 Objectives

1.3.1 Main objective

To design, model and perform fatigue life analysis of 450 kg LPG cylinder and to validate analytical solutions with finite element simulation

1.3.2 Specific objectives

The specific objectives of this research are:

- To design and model a 450 kg LPG cylinder
- To analyse stress distribution of 450 kg LPG cylinder
- To analyse fatigue life under cyclic loading conditions of 450 kg LPG cylinder
- To compare the analytical solutions with finite element simulations for stress and fatigue life analysis

1.4 Limitations of the study

Although a comprehensive study was undertaken to determine the study's outputs, there were certain limitations that arose during the research. These limitations include:

- The research focuses only on the 450 kg LPG Cylinders which is yet to be used in Nepali commercial & industrial Sectors
- Due to resource and equipment constraints for large-scale vessels, the study relies on numerical simulations and analytical modelling rather than experimental destructive testing
- The finite element analysis is subject to potential truncation errors in the ANSYS software

CHAPTER 2 LITERATURE REVIEW

2.1 Overview of LPG Cylinders

Liquefied Petroleum Gas (LPG) is one of the most commonly used alternative fuels for domestic, commercial, and industrial applications due to its high calorific value, clean-burning nature, and ease of storage in pressurized containers. LPG mainly consists of hydrocarbon gases produced during petroleum refining and natural gas processing. Its primary components are propane (C_3H_8) and butane (C_4H_{10}), although small quantities of propylene, butylene, and other hydrocarbons may also be present (World LPG Association, 2022).

The composition of LPG generally varies depending on climatic conditions and regional fuel standards. In colder regions, LPG usually contains a higher proportion of propane because propane can vaporize more easily at low temperatures. In warmer regions, a greater percentage of butane is preferred because of its higher energy content and improved storage efficiency. In Nepal, LPG used for household and industrial purposes typically contains a propane-butane mixture ranging from about 40–60% propane and 60-40% butane, depending on seasonal demand and refinery supply conditions (Nepal Oil Corporation, 2021).

Propane mainly helps maintain proper vaporization under low-temperature conditions, whereas butane contributes to higher calorific value and better storage efficiency. The combined properties of these gases make LPG a suitable and efficient fuel for a wide range of applications.

The physical properties of propane and butane that influence LPG storage and cylinder design are shown below.

Table 1: Physical Properties of Propane and Butane

Property	Propane	Butane
Chemical Formula	C_3H_8	C_4H_{10}
Boiling Point	-42 °C	−0.5 °C
Density (Liquid)	493 Kg/ m³	584 Kg/ m³
Heating Value	46.4 MJ/Kg	45.7 MJ/kg

These properties determine the vapor pressure inside the cylinder and influence the structural design of LPG storage vessels.

2.2 Types of LPG Cylinders Used in Industry

LPG cylinders are available in several configurations depending on storage capacity and industrial requirements. Small cylinders used in households generally have capacities between 5 kg and 14.2 kg, whereas industrial cylinders may range from 50 kg to 450 kg or larger storage vessels (Kumar, Singh, & Chaudhary, 2016).

Large industrial LPG cylinders such as 450 kg cylinders are commonly used in restaurants, hotels, factories, and commercial heating systems. These cylinders are designed as pressure vessels capable of safely storing LPG at pressures that may exceed 10-15 bar depending on temperature conditions.

Two major configurations of large LPG cylinders are commonly used.

1. Vertical cylinders

- Used when floor space is limited.

2. Horizontal cylinders

- Used for bulk storage and easier transport.

Figure 1: Industrial 450 KG LPG Cylinder
[Image Source: supergas.com]

Major Components of LPG Cylinder

The major structural parts of an LPG cylinder include:

Table 2: Major Structural Part and Function

Components	Function
Cylinder Shell	Main pressure containing body
Dome/Head	Distributes internal pressure uniformly
Valve	Controls LPG flow in and out of cylinder
Stand/Base Ring	Supports cylinder during storage
Protective Cap	Protects valve from mechanical damage
Weld Joint	Connects shell and head sections
Lifting Lugs	Used for safe handling and transportation

The cylinder shell is the primary structural component that withstands internal pressure. The dome or head helps distribute stresses uniformly and prevents localized stress concentration. The base ring or stand provides stability when the cylinder is placed vertically.

2.3 Pressure Vessels Used for LPG Storage

Liquefied Petroleum Gas (LPG) is extensively used in domestic, commercial, and industrial sectors because of its high energy content and convenient transportation and storage characteristics. In most applications, LPG is stored in pressurized vessels where it remains in liquid form under moderate pressure conditions. Since LPG is highly flammable, the storage vessels used for handling and transportation must be designed and manufactured according to strict engineering standards to ensure safe operation.

A pressure vessel is generally defined as a closed container designed to store gases or liquids at pressures significantly different from atmospheric pressure. During service, these vessels are subjected to internal pressure, temperature changes, and various mechanical loads. Because of this, the design of pressure vessels must carefully consider factors such as structural strength, material properties, fatigue resistance, and fabrication techniques to maintain safety and reliability during operation (Megyesy, 2008).

The shape and geometry of a pressure vessel play an important role in determining how stresses are distributed throughout the structure. Cylindrical shells with hemispherical or ellipsoidal end caps are commonly preferred in pressure vessel design because these geometries provide a more uniform stress distribution compared to flat surfaces. This helps reduce stress concentration and improves the overall structural performance of the vessel.

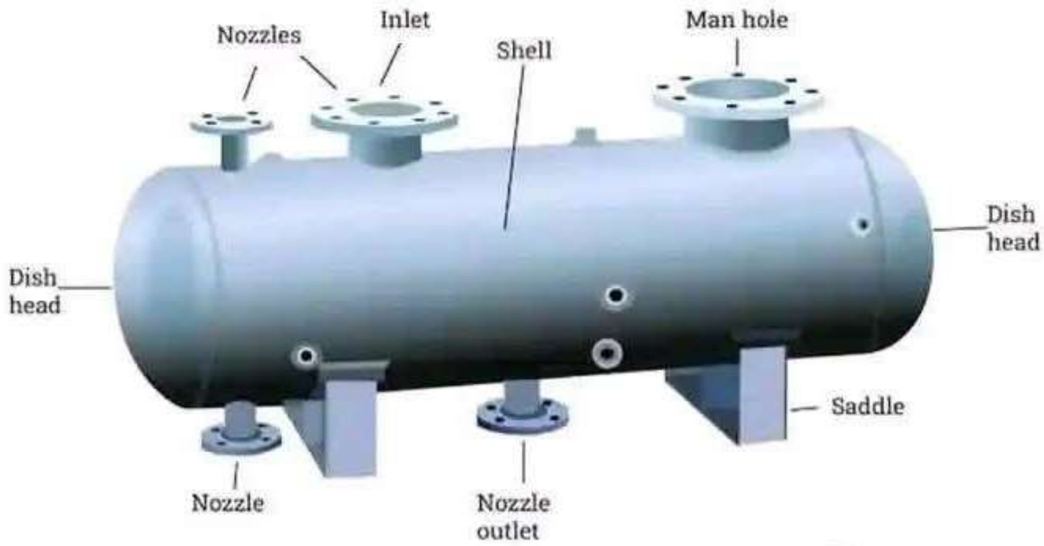


Figure 2: Typical pressure vessel showing cylindrical shell, heads, nozzles, and support structures
[Image Source: iqsdirectory.com]

2.4 Design Standards for Pressure Vessels

The safe design of pressure vessels requires compliance with internationally recognized codes and standards. Among the most widely used standards are the ASME Boiler and Pressure Vessel Code (BPVC) and the European Standard EN 13445.

These standards define requirements for material selection, allowable stress limits, thickness calculations, fatigue assessment, inspection procedures, and testing methods.

2.4.1 EN 13445 Standard

EN 13445 is a European standard developed for unfired pressure vessels. The standard provides design rules that ensure safe operation under various loading conditions including internal pressure, thermal stresses, and cyclic loading. EN 13445 consists of several parts covering materials, design, fabrication, inspection, and testing (CEN, 2021).

Part 3 of the standard focuses specifically on design calculations, including wall thickness determination and fatigue analysis. The standard provides two general design approaches:

- Design by Formula (DBF)
- Design by Analysis (DBA)

Design by Formula involves simplified analytical equations derived from classical pressure vessel theory. Design by Analysis, on the other hand, allows engineers to use numerical techniques such as finite element analysis to determine stress distributions more accurately.

The standard also specifies allowable stresses based on material properties. These allowable stresses are derived from yield strength or tensile strength with safety factors applied.

The allowable stress can be expressed as:

$$\sigma_{allow} = \min\left(\frac{R_e}{1.5}, \frac{R_m}{2.4}\right) \tag{2.1}$$

Where,

R_e = Yield strength

R_m = Ultimate tensile strength

All design calculations are performed considering the corroded condition of the vessel, ensuring that long-term material degradation does not compromise structural integrity (CEN, 2021).

2.5 Stress Analysis in Pressure Vessels

Stress analysis is an essential step in pressure vessel design. When internal pressure acts on the vessel wall, it generates two principal stresses:

- Hoop stress (circumferential stress)

For thin-walled cylindrical pressure vessels, these stresses can be calculated using classical equations.

Hoop Stress

$$\sigma_h = \frac{PD}{2t} \tag{2.2}$$

Longitudinal stress

$$\sigma_l = \frac{PD}{4t} \tag{2.3}$$

P = internal Pressure

t = wall thickness

The hoop stress is typically twice the longitudinal stress and therefore governs the design thickness of cylindrical pressure vessels.

However, real pressure vessels also experience additional stresses at discontinuities such as nozzles, weld joints, and supports. These stress concentrations must be analyzed carefully because they can lead to fatigue crack initiation.

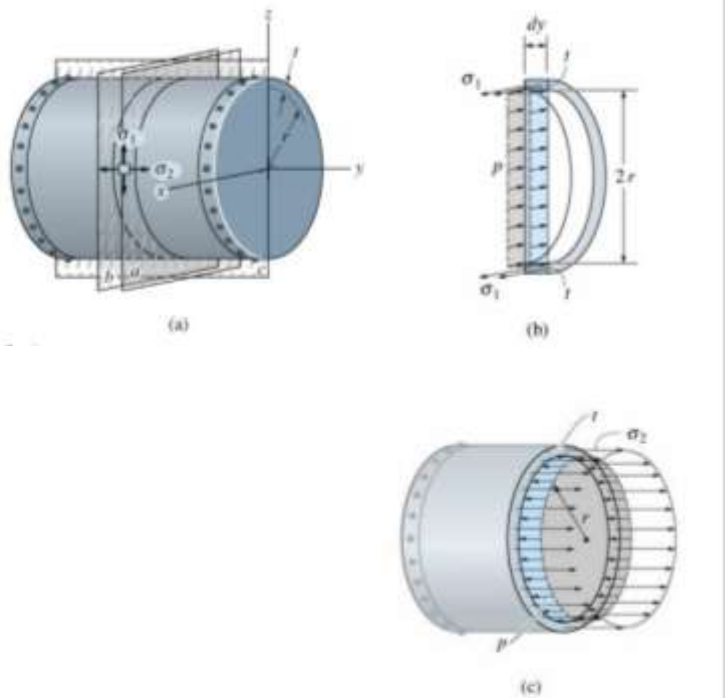


Figure 3: Hoop and longitudinal stresses generated in a cylindrical pressure vessel due to internal pressure
[Image Source: (Hibbler,2017)]

2.6 Materials and Structural Design of LPG Cylinders

Material selection is one of the most important factors in pressure vessel design because it directly affects the structural strength, fatigue performance, corrosion resistance, and weldability of the vessel. For safe operation under internal pressure, the selected material must satisfy the mechanical and metallurgical requirements specified by international design standards. According to EN 13445 for unfired pressure vessels, pressure vessel quality steels such as P355GH are commonly recommended for LPG storage vessels and cylindrical pressure containers because of their good toughness, weldability, and suitable mechanical strength (CEN, 2021).

P355GH is a normalized carbon steel that is specifically developed for pressure vessel applications operating at elevated temperatures. The material generally possesses a minimum yield strength of about 355 MPa along with good ductility and toughness characteristics. These properties make it suitable for moderate-pressure storage applications and help reduce the possibility of brittle failure under repeated loading conditions (CEN, 2021).

In some industrial applications, alternative high-strength steels are also used to improve structural performance and reduce material thickness. One such material is E450BR, which has been adopted in certain LPG cylinder designs because of its higher strength and better load-carrying capability. Compared to conventional pressure vessel steels, E450BR provides a yield strength of approximately 450 MPa, which allows the cylinder to sustain higher stress levels while still maintaining the required safety margin. Its higher strength also contributes to improved structural efficiency in large-capacity LPG cylinder applications.

The allowable design stress for pressure vessel materials is determined using the criteria provided in pressure vessel standards. According to EN 13445 design rules, the allowable stress f is calculated as the minimum of the following limits:

$$\mathrm{f} = \min\left(\frac{\mathrm{R_e}}{1.5}, \frac{\mathrm{R_m}}{2.4}\right) \tag{2.4}$$

Where,

$\mathrm{R_e} = \text{yield strength}$

$\mathrm{R_m} = \text{ultimate tensile strength}$

This approach ensures that the vessel operates well below the plastic deformation limit and prevents structural failure during pressure loading (CEN, 2021).

In addition to shell materials, pressure vessel components such as couplings and nozzles are often manufactured using SA-350 Grade LF2 Class 1 steel, which provides excellent toughness at low temperatures and good weldability. This material is commonly used in pressure vessel components exposed to low-temperature LPG environments because it maintains adequate mechanical properties under cryogenic or sub-zero conditions (ASME, 2021).

Large LPG cylinders are generally constructed as cylindrical shells with elliptical heads. Elliptical heads are widely used because they provide improved stress distribution compared with flat heads and require less material thickness than hemispherical heads. The geometry reduces stress concentration near the shell-to-head junction, which is typically a critical region in pressure vessel structures (Megyesy, 2008).

LPG cylinders are typically manufactured from high-strength carbon steel plates that provide sufficient ductility, toughness, and resistance to cyclic loading. In Nepal, cylinder materials must comply with national standards such as NS 367 and NS 368, which specify chemical composition and mechanical properties suitable for pressure vessel applications.

Manufacturing involves several processes including:

- Plate rolling
- Deep drawing
- Circumferential welding
- Heat treatment
- Hydrostatic pressure testing

Welding is a particularly critical step in cylinder fabrication because weld joints are common locations for stress concentration and fatigue crack initiation. Proper welding procedures and inspection methods are therefore essential to ensure the long-term reliability of LPG cylinders (Patil & Yadav, 2019).

Table 3: Chemical Composition of E450BR Grade(wt%)(IS 2062)

Grade	**%C**	**%Mn**	**%S**	**%P**	**%Si**	**CE, max**	**Mode of Deoxidation**
E 450 BR	0.22 max	1.65 max	0.045 max	0.045 max	0.45 max	0.52	Semi-killed/Killed

Table 4: Mechanical Properties of E450BR Grades

Grade	UTS MPa min	YS, MPa min	%EL min
E450BR	588	519	38

2.7 Analysis of Failure Modes and Geometric Imperfections

The EN 13445-3 standard identifies several important failure modes that need to be considered during the Design by Analysis (DBA) procedure. Although internal pressure is the main source of primary membrane stress in a pressure vessel, localized geometric discontinuities and imperfections can create regions of high stress concentration that may affect the structural safety of the vessel (CEN, 2021).

2.7.1 Peeking and Angular Distortion

Peaking refers to a localized deviation from the ideal circular geometry of a pressure vessel, which usually occurs near longitudinal or circumferential welded joints. According to EN 13445-3, such angular distortions produce additional bending stresses that cannot be fully evaluated using simple membrane stress equations (CEN, 2021). In large-capacity vessels such as 450 kg LPG cylinders, these localized distortions can increase stress concentration near weld toes and may accelerate fatigue crack initiation under repeated loading conditions. Because of this, fatigue strength reduction factors are generally considered during numerical fatigue assessment to account for the effect of weld geometry and local stress intensification (CEN, 2021).

2.7.2 Structural Discontinuities

The standard distinguishes between pressure-bearing and non-pressure-bearing attachments. EN 13445-3, Annex G highlights that stress concentrations at support brackets such as the tripod mounting system used in this study are primary drivers of secondary stress. These discontinuities create stress risers where the localized equivalent stress can exceed the yield strength of the material, leading to cyclic plastic deformation or ratcheting (CEN, 2021).

2.8 Fatigue Failure Criteria

Unlike static failure, fatigue failure in pressure vessels occurs due to the progressive growth of cracks under cyclic pressure loads. The EN 13445-3 standard provides specific S-N curves for different weld classes. For a 450 kg cylinder, the interface between the shell and the base support is categorized as a high-stress region where the total number of allowable cycles is limited by the localized peak stress rather than the nominal hoop stress (CEN, 2021, Stephens et al., 2000).

2.9 Fatigue Behaviour of Pressure Vessels

Fatigue failure occurs when a material experience repeated cyclic loading over a period of time. Even when the applied stresses are below the material yield strength, fatigue cracks can develop due to cyclic stress variations (Stephens et al., 2001).

Pressure vessels used for LPG storage experience cyclic loading due to repeated filling and discharge operations. Each filling cycle produces changes in internal pressure, which results in stress fluctuations in the vessel walls.

Fatigue life is generally evaluated using S-N curves, which describe the relationship between stress amplitude and number of cycles to failure.

The fatigue relationship is commonly expressed as:

$$N = C(\Delta\sigma)^{-m} \tag{2.5}$$

Where,

N = number of cycle to failure

$\Delta\sigma$ = stress range

C and m = material constant

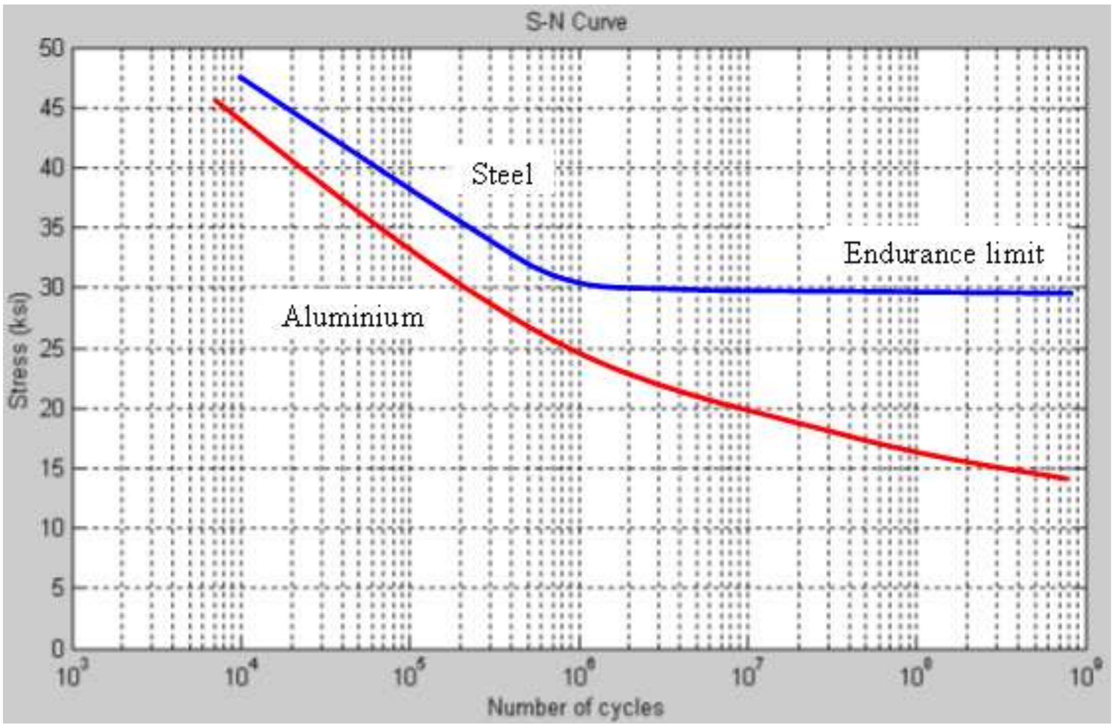


Figure 4: Typical stress-life (S-N) curve showing the relationship between stress amplitude and fatigue life
[Image Source: Dressel, A. (2008)]

2.9.1 Fatigue Behaviour of LPG Cylinders under Cyclic Loading

EN 13445 provides specific guidelines for evaluating the fatigue behaviour of pressure vessels that are exposed to repeated or cyclic loading conditions. According to the standard, fatigue assessment becomes necessary when a vessel experiences more than approximately 500 equivalent full pressure cycles during its service life (CEN, 2021).

The standard includes fatigue design curves developed from experimental fatigue data obtained through laboratory testing of structural materials. These curves are intentionally conservative so that the possibility of fatigue failure during operation remains very low.

In fatigue analysis, the expected fatigue life of the vessel is estimated by comparing the calculated equivalent stress range with the fatigue design curves specified in the standard. This approach helps determine whether the pressure vessel can safely withstand the required number of loading cycles throughout its intended service period (CEN, 2021).

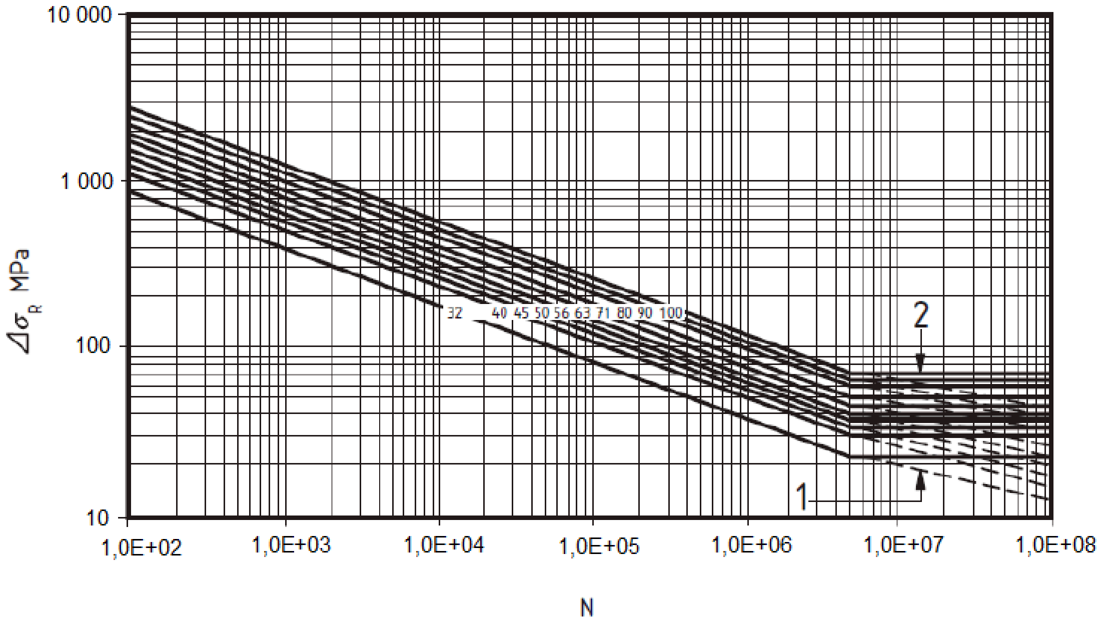


Figure 5: EN 13445 Fatigue Design Curve
[Image Source: [CEN,2021]

2.9.2 Fatigue Evaluation According to ASME Section VIII Division 2

ASME Section VIII Division 2 provides detailed procedures for fatigue evaluation using stress results obtained from analytical or numerical methods. The fatigue evaluation process typically includes the following steps:

1. Determination of cyclic stresses
2. Calculation of alternating stress amplitude
3. Comparison with fatigue design curves
4. Calculation of fatigue damage factor

The alternating stress amplitude is expressed as

$$S_a = \frac{\Delta S}{2} \tag{2.6}$$

The cumulative fatigue damage is evaluated using Miner's rule:

$$D = \sum_{0}^{i} \frac{n_i}{N_I} \tag{2.7}$$

If the damage factor is less than unity, the vessel is considered safe for the specified loading conditions.

2.10 Finite Element Analysis in Pressure Vessel Design

Finite Element Analysis (FEA) is now widely used to study pressure vessels that have complicated shapes and loading conditions. Unlike traditional analytical methods, numerical simulation helps engineers understand how stresses are distributed throughout the vessel with greater accuracy and detail (Reddy, 2019).

In FEA, the entire geometry of a pressure vessel is divided into many small interconnected elements. These elements work together to represent the behaviour of the complete structure. By applying equilibrium equations to each element, engineers can calculate stresses, strains, and deformations caused by different loading conditions (Cook et al., 2002).

Modern engineering software such as ANSYS and Abaqus is commonly used to create three-dimensional finite element models of pressure vessels. During the simulation process, operating conditions such as internal pressure, self-weight, and boundary constraints are applied so that the model closely represents real service conditions (Moaveni, 2015).

For LPG storage vessels, FEA is especially useful for locating areas where stresses become concentrated. Critical regions are often found around nozzles, support structures, and welded joints, as these areas are more vulnerable to fatigue failure and structural damage over time (Singiresu, 2018).

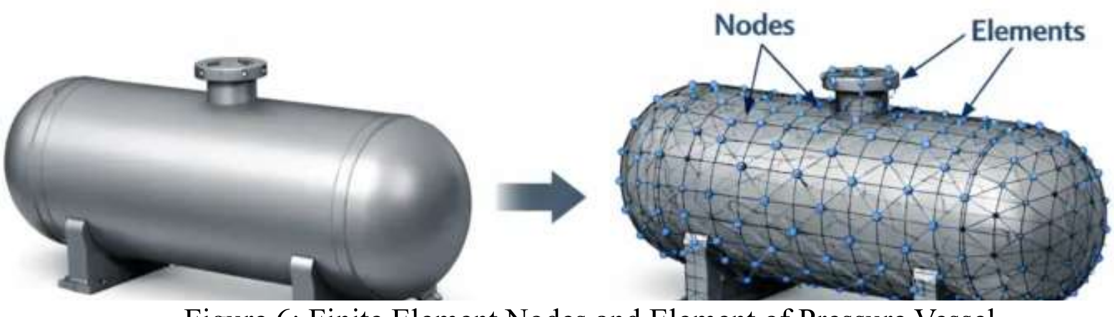


Figure 6: Finite Element Nodes and Element of Pressure Vessel

[Image Source: (Zienkiewicz et al., 2013)]

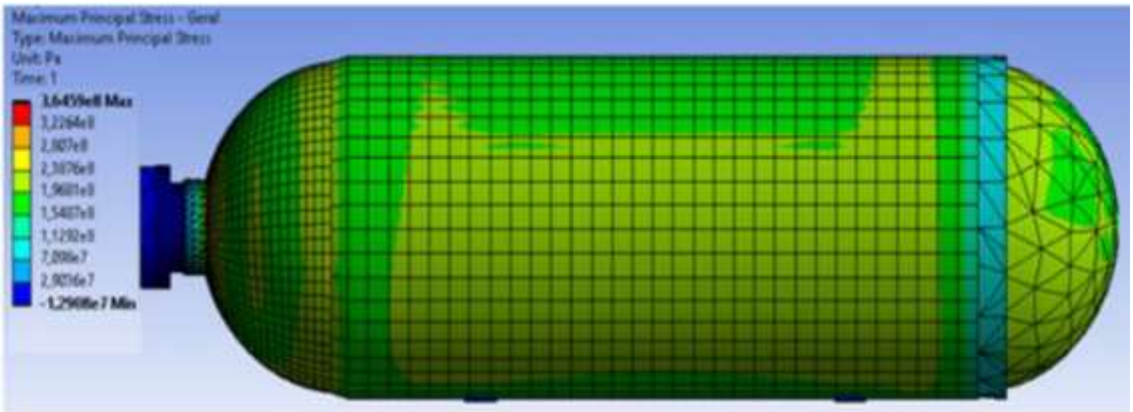


Figure 7 Finite Element Mesh of a Pressure Vessel
[Image Source: (Wang & Zhao, 2018)]

2.11 Validation Techniques

Validation is a key step to ensure the accuracy and reliability of design and analysis results. In this study, analytical solutions based on classical pressure vessel theory will be used alongside finite element analysis (FEA) to validate stress distribution and fatigue life predictions for the 450 kg LPG cylinder. Analytical calculations provide a baseline for comparison, helping to verify the assumptions and setup of the FEA model. Using both analytical calculations and numerical simulations helps make the results more reliable and makes it easier to spot any differences that need to be checked further.

2.12 Research gaps

While there are several studies that use both testing and simulation to analyze LPG cylinders, most of them focus on smaller or medium-sized cylinders. There is not enough research specifically on the large 450 kg LPG cylinders. More work is needed to fully understand the stress and fatigue behavior of these large cylinders, particularly using a combination of analytical methods, simulations, and validation techniques. This is important as Nepal begins to adopt EN 13445 standards and increases the use of these industrial cylinders.

2.13 Summary of Literature and Justification for Present Study

The literature reviewed in this chapter establishes both the theoretical background and the practical basis for the present study, while also highlighting several gaps in existing research. A number of important themes emerge from the review and help justify the methodology adopted in this work.

In terms of design methodology, previous studies show that the EN 13445 Design by Formula approach provides dependable estimates for shell thickness in standard cylindrical vessels. However, it has limitations when it comes to identifying local stress concentrations that occur around discontinuities and structural attachments (CEN, 2021). Because of this limitation, finite element analysis (FEA) has become a standard complementary tool in pressure vessel design and analysis, particularly for evaluating stress concentrations in critical regions (Zienkiewicz et al., 2013; Wang & Zhao, 2018). In this study, both approaches are combined: EN 13445 is used for the preliminary analytical design, while detailed stress behaviour is investigated using ANSYS Workbench finite element simulations.

The reviewed literature on fatigue assessment also demonstrates that fatigue failure in pressurised cylindrical vessels is usually controlled not by the general membrane stress in the shell, but by stress concentrations near weld toes, supports, and attachments (Dong, 2001; Stephens et al., 2000). Research conducted by Shariati and Janghorban (2017) found that support regions in similar vessels experience the lowest fatigue life, further supporting the need for detailed fatigue evaluation in these locations. Based on these findings, the present study adopts the ASME Section VIII Division 2 fatigue assessment procedure, which evaluates fatigue damage using peak alternating stress amplitudes together with Miner's cumulative damage rule (ASME, 2021).

Another important aspect discussed in the literature is mesh convergence in finite element modelling. The Grid Convergence Index (GCI) method proposed by Celik and Karatekin (1997), and later formalised by Roache (1994), provides a systematic and quantitative way to verify whether a mesh is sufficiently refined. Following this methodology, the present study confirms that a medium mesh containing approximately 420,000 elements produces stable and mesh-independent results.

With regard to material selection, the literature identifies both P355GH and E450BR steels as suitable materials for LPG cylinder applications (CEN, 2021). Although

E450BR offers higher yield strength and an improved strength-to-weight ratio, comparatively fewer fatigue studies have been published for this material. Therefore, this research uses E450BR as the primary construction material and additionally develops an S-N curve for the material based on theoretical fatigue relationships. This contributes further to the existing knowledge on the fatigue behaviour of E450BR steel.

Overall, the reviewed literature confirms that the methodology selected for this study is appropriate and technically justified. At the same time, it highlights how the present work extends previous research by integrating analytical design methods, finite element stress analysis, and ASME fatigue assessment procedures for a 450 kg class LPG cylinder representative of Nepal's growing industrial sector. The study also contributes through the systematic validation of finite element results against established closed-form analytical solutions.

CHAPTER 3 RESEARCH METHODOLOGY

3.1 Introduction

Designing a pressure vessel for LPG service is not simply an exercise in satisfying a static pressure criterion. The vessel must maintain its structural integrity across a service life that involves thousands of pressure cycles as it is filled, transported, and discharged. Each of those cycles imposes a fluctuating stress state on the shell wall, head-to-shell junctions, and support attachments. If those stress variations are not accounted for explicitly in the design process, fatigue damage can accumulate silently until a crack propagates to a size that compromises the vessel's safe operation.

This study addresses those concerns directly for a 990-litre (450 kg) LPG storage cylinder. The methodology brings together three complementary tools: analytical pressure vessel design based on established code formulas, three-dimensional CAD modelling, and finite element analysis to capture the spatial stress distribution that analytical methods alone cannot resolve.

Pressure vessel design is governed by internationally recognized codes that ensure safety, reliability, and standardization in engineering practice. In this study, the design procedures are primarily guided by:

- EN 13445-3:2021 - Unfired Pressure Vessels
- ASME Boiler and Pressure Vessel Code Section VIII Division 2

EN 13445 provides the primary structural design framework, setting out allowable stress limits, thickness requirements, and fatigue assessment procedures for unfired pressure vessels (CEN, 2021). ASME Section VIII Division 2 is used alongside it for the fatigue evaluation, bringing in the ASME fatigue curves and damage accumulation procedure that are particularly well suited to vessels subjected to defined pressure cycles (ASME, 2021).

Finite element analysis is essential here because the stress concentrations that govern fatigue life those at nozzle connections, head-to-shell transitions, and support bracket welds fall outside what membrane theory can predict with adequate accuracy. FEA provides the spatial detail needed to locate and quantify these peaks, which then feed directly into the fatigue damage calculation.

The overall research workflow adopted in this study is illustrated in Figure 8

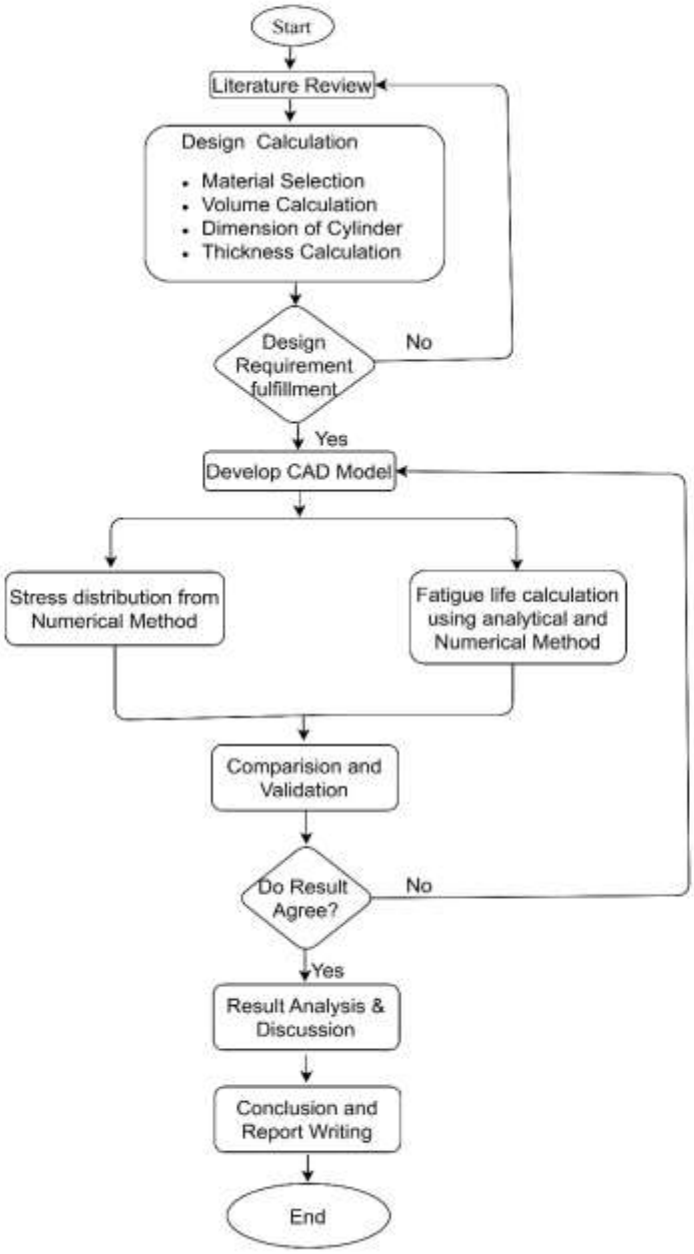


Figure 8 Methodology of Research

The research process consists of the following major stages:

1. Identification of design parameters and operating conditions
2. Analytical pressure vessel design calculations

3. Determination of shell and head thickness
4. Nozzle reinforcement calculations
5. Development of a three-dimensional CAD model
6. Finite element stress analysis
7. Fatigue life estimation

This combined analytical–numerical methodology allows both code-based verification and detailed stress evaluation of the LPG cylinder.

3.2 Design Parameters

The LPG cylinder considered in this study is a vertical cylindrical pressure vessel with ellipsoidal heads. Cylindrical pressure vessels are commonly used for LPG storage because they provide an efficient geometry for resisting internal pressure while minimizing material usage (Moss & Basic, 2013).

3.2.1 Determination of Cylinder Capacity and Dimensions for a 450 kg LPG Cylinder

The design of the LPG cylinder begins with the required storage capacity. In this study, the cylinder is required to store 450 kg of liquefied petroleum gas (LPG). Since LPG is stored in liquid form under pressure, the internal volume of the cylinder must first be determined from the mass of LPG.

3.2.1.1 Determination of Water Capacity

For LPG cylinders, the storage limit is commonly expressed using the filling ratio, which relates the mass of LPG to the water capacity of the cylinder. The filling ratio is defined as

$$\mathrm{F} = \frac{\text{Mass of LPG (kg)}}{\text{Water capacity (L)}} \tag{3.1}$$

For LPG cylinders, the recommended value of the filling ratio is approximately

$$\mathrm{F} = 0.425\ \mathrm{kg/L}$$

According to Table 2 of IS 3710:1978, the filling ratio for liquefied hydrocarbon gases depends on the density of the LPG mixture. For typical LPG densities ranging from

0.52-0.54 g/ml at 15°C, the corresponding filling ratios vary from 0.415 to 0.442. Therefore, an approximate design filling ratio of about 0.42–0.43 kg/L is commonly adopted in cylinder sizing calculations.

This value ensures that the cylinder is not completely filled with liquid LPG and sufficient vapour space remains for thermal expansion of the liquid.

Therefore, the required water capacity of the cylinder is calculated as

$$\text{Water capacity} = \frac{\text{Mass of LPG}}{\text{F}} \tag{3.2}$$

$$\text{Water capacity} = \frac{450}{0.425}$$

$$\text{Water capacity} = 1058.8 \text{ L}$$

$$\text{Water capacity} \approx 1.059 \text{ m}^3$$

Thus, a cylinder designed to store 450 kg of LPG requires an internal volume of approximately 1.06 m^3.

3.2.1.2 Selection of Cylinder Geometry

In practice, large LPG cylinders are commonly manufactured with an outer diameter of about 1000 mm and 2:1 ellipsoidal heads, as this geometry provides good structural strength and is convenient for fabrication.

Assuming

Outer diameter

$$\text{OD} = 1000 \text{ mm}$$

Shell thickness

$$\text{t} = 5.1 \text{ mm(Calculated)}$$

$$\text{D}_\text{i} = \text{OD} - 2\text{t}$$
$$\text{D}_\text{i} = 1000 - 2(5.1)$$
$$\text{D}_\text{i} = 989.8 \text{ mm}$$

3.2.1.3 Calculation of Vessel Volume

The total internal volume of the vessel consists of the cylindrical shell and two ellipsoidal heads.

$$V = V_{shell} + 2V_{head} \tag{3.3}$$

The volume of the cylindrical shell is

$$V_{shell} = \frac{\pi D_i^2}{4} L \tag{3.4}$$

For a 2:1 ellipsoidal head, the internal volume is given by

$$V_{head} = \frac{\pi}{24} D_i^3 \tag{3.5}$$

$$V_{head} = \frac{\pi}{24} (989.8)^3$$

$$V_{head} = 126\,985\,972\ mm^3$$

$$V_{head} = 0.127\ m^3$$

For two heads

$$2V_{head} = 0.254\ m^3$$

Required Shell Volume

Total required volume

$$V = 1.059\ m^3$$

Therefore

$$V_{shell} = 1.059 - 0.254$$

$$V_{shell} = 0.805\ m^3$$

3.2.1.4 Determination of Shell Length

Using

$$V_{shell} = \frac{\pi D_i^2}{4} L \tag{3.6}$$

$$0.805 = \frac{\pi (0.9898)^2}{4} L$$

$$L = 1.047\ m$$

$$L \approx 1047\ mm$$

Final Cylinder Dimensions

The calculated dimensions required to store 450 kg of LPG are summarized below.

Table 5: Design Dimensions

Parameter	Value
LPG capacity	450 kg
Water capacity	1059 L
Outer diameter	1000 mm
Internal diameter	989.8 mm
Shell length	1047 mm
Head type	2:1 ellipsoidal

These dimensions provide an internal volume of approximately 1.059 m^3, which corresponds to the required storage capacity for 450 kg of LPG based on the recommended filling ratio.

The primary design parameters used in the analytical calculations are summarized in Table 6.

Table 6: Design Parameters

Parameter	**Value**
Water Capacity	1059 L
Design Pressure	1.76 MPa
Hydrostatic Test Pressure	2.52 MPa
Design Temperature	65°C
Internal Diameter	989.8 mm
Joint Efficiency	0.85
Tensile Strength	588 MPa
Yield Strength	519 MPa
Allowable Stress	245 MPa
Corrosion Allowance	0.5 mm

These parameters form the basis for the structural design calculations performed according to pressure vessel design standards.

3.3 Material Selection

Material selection is a critical aspect of pressure vessel design because the mechanical properties of the material determine the allowable stresses, fatigue resistance, and long-term reliability of the vessel.

The EN 13445 standard commonly recommends pressure vessel steels such as P355GH, which is widely used for high-temperature and pressure applications due to its good weldability and mechanical strength (CEN, 2021).

However, in this research the material E450BR (IS 2062) steel was selected for the LPG cylinder design. E450BR is a high-strength structural steel with improved yield strength compared with conventional pressure vessel steels. The higher strength allows thinner wall sections while maintaining adequate safety margins.

The mechanical properties of E450BR relevant to pressure vessel design include:

- High yield strength
- Good toughness
- Adequate weldability
- Improved fatigue resistance

The allowable stress used in the design calculations was 245 MPa, which satisfies the allowable stress limits recommended in pressure vessel design codes.

The selection of high-strength steel for LPG cylinders has been discussed in several studies, Zhang et al. (2019) reported that high-strength steels can significantly reduce vessel weight while maintaining structural integrity when proper fatigue analysis is performed.

3.4 FEM Setup

3.4.1 Mesh Generation

To ensure numerical stability, convergence and accuracy of the results, accurate mesh generation is the crucial steps in FEM analysis. In the present study, a three-dimensional mesh was generated for the static structural analysis of 450 kg LPG cylinder using the meshing module of ANSYS Workbench.

Figure 9 3D Model of the 450 kg LPG Cylinder

Considering the complex geometry of the LPG cylinder, particularly the ellipsoidal ends and the handle region, a hybrid meshing strategy was adopted. The mesh was primarily generated using adaptive meshing technique with maximum element size of 20 mm all across the domain. To capture the geometric curvature and stress concentration effects in critical regions, local mesh controls were implemented.

- Body sizing for Ellipsoidal Ends

The top and bottom ellipsoidal bodies, stand, and nozzles of the LPG cylinder exhibit high curvature and are expected to experience significant stress gradients during loading. To accurately capture these effects, body sizing was applied to these regions with a refined element size of 10 mm.

- Curvature-Based Refinement in ellipsoidal faces

A curvature-based refinement was applied using the patch independent meshing method. This approach is particularly suitable for complex geometries where mesh conformity to curved surfaces is essential as experienced in ellipsoidal faces.

Element type: Tetrahedral elements Curvature normal angle: 40° Maximum element size: 20 mm Minimum element size: 10 mm

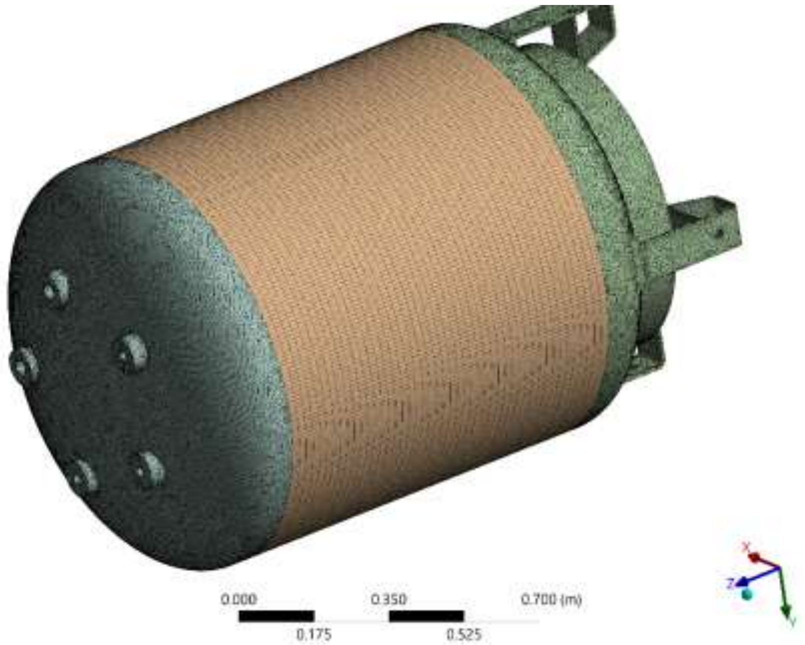


Figure 10 Generated Mesh

3.4.2 Mesh Characteristics

The final unstructured mesh consists of predominantly tetrahedral and hexahedral elements, which are well suited for complex three-dimensional elements. The mesh quality parameters as per industrial practice with their acceptable limits and achieved values in FEM model is shown in Table 7.

Table 7: Mesh Quality Parameters

Quality Check	**Acceptable Value**	**Average Achieved Value with Standard Deviation**
Aspect Ratio	<5	4.369
Skewness	<0.7	0.695
Element Quality	<0.1	0.143
Number of Nodes	-	874,707
Number of Elements	-	418726

3.4.3 Boundary Conditions and Loading Conditions

The simulation incorporates four primary types of boundary conditions: Acceleration due to gravity, Fixed Supports, Concentrated Forces in Nozzle, and Internal Pressure as shown in Table 8.

Table 8: Boundary Conditions

Boundary Condition	**Type**	**Magnitude**	**Direction / Locatio**
Fixed Support	Constraint	0 mm Displacement	Base Support Brackets (3 Locations)
Force 1	Load	1892.4 N	Z-axis (Positive) / Top Nozzle A
Force 2	Load	747.86 N	Z-axis (Positive) / Top Nozzle B
Pressure	Load	1.76 MPa	Normal to Internal Cylindrical Face
Acceleration due to gravity	Load	9.81	Z-axis(negative)

3.4.3.1 Fixed Support

To simulate the cylinder bolted or resting on a solid foundation, Fixed Supports (Labels A, B, and C) are applied to the bottom faces of the tripod mounting brackets as shown in Figure 11.

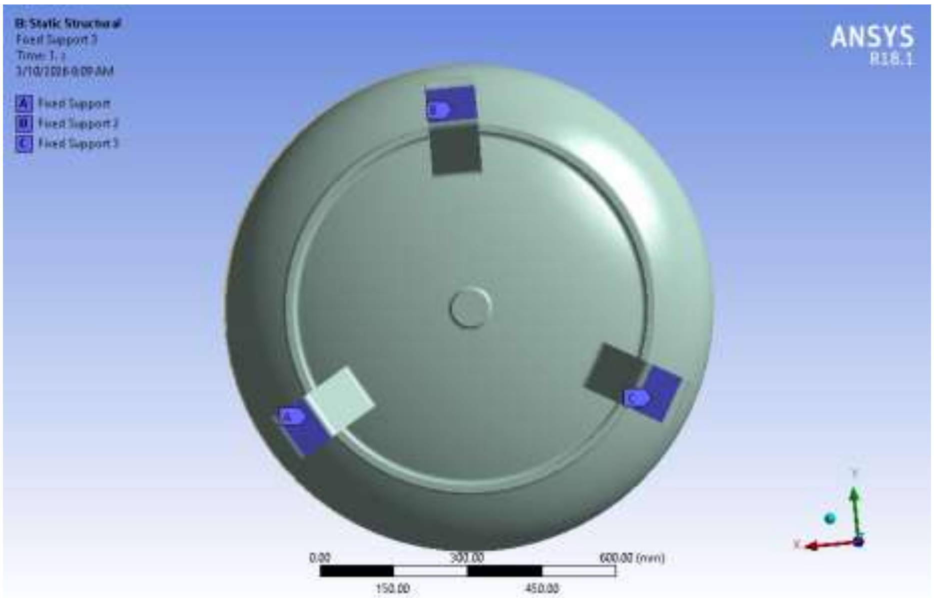


Figure 11 Fixed Support in Tripods

- Location: The three rectangular pads at the base of the cylinder.
- Effect: Prevents all translation (X, Y, Z) and rotation, ensuring the model is statically determinate and grounded.

1. Standard Earth Gravity (Self-Weight)

The addition of gravity ensures that the mass of the steel shell (and potentially the 450 kg of LPG, depending on your material density settings) is factored into the stress profile.

- Magnitude: 9.8066 m/s^2.
- Direction: Applied in the Negative Z-direction.
- Impact: This creates a vertical compressive force on the bracket supports

2. External Forces (Nozzle Loading)

Two distinct forces are applied to the top boss/nozzle attachments. These simulate the weight of the valve assembly, piping connections, or external mechanical tension.

Force 1: A vertical load of 1892.4 N is applied to the first nozzle. The component breakdown indicates the force is directed primarily along the positive Z-axis 584 N, with negligible components in X and Y as shown in Figure 4(a).

- Force 2: A vertical load of 747.86 N is applied to the second nozzle. Similar to Force 1, this acts in the positive Z-axis 747.86 N, representing a significantly higher load, possibly due to heavier connected auxiliary equipment as shown in Figure.

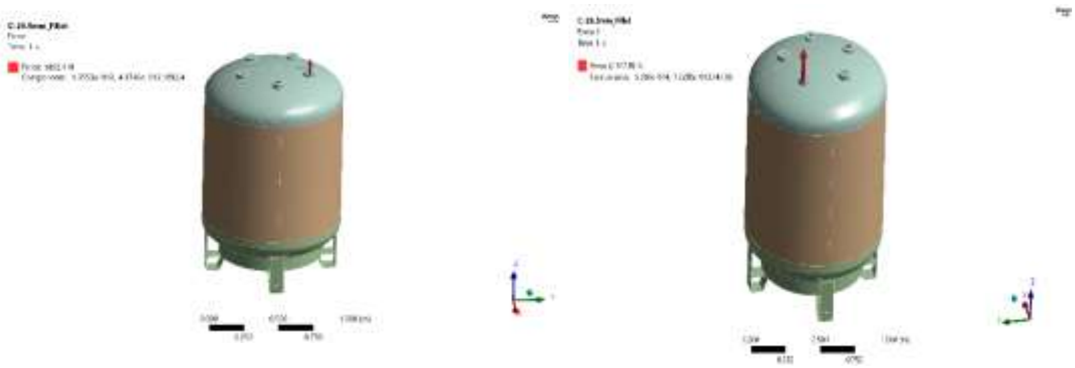


Figure 12 Nozzle Loading

(a)Force 1 (b) Force 2

a. Internal Pressure

The primary stress driver for a pressure vessel is the internal fluid pressure.

- Value: 1.76 MPa (approx. 17 bar).
- Location: Applied to the internal "wetted" surface of the cylindrical shell as shown in Figure 13.
- Direction: The pressure acts Normal to the interior faces, pushing outward. This induces hoop (circumferential) and longitudinal stresses within the cylinder walls.

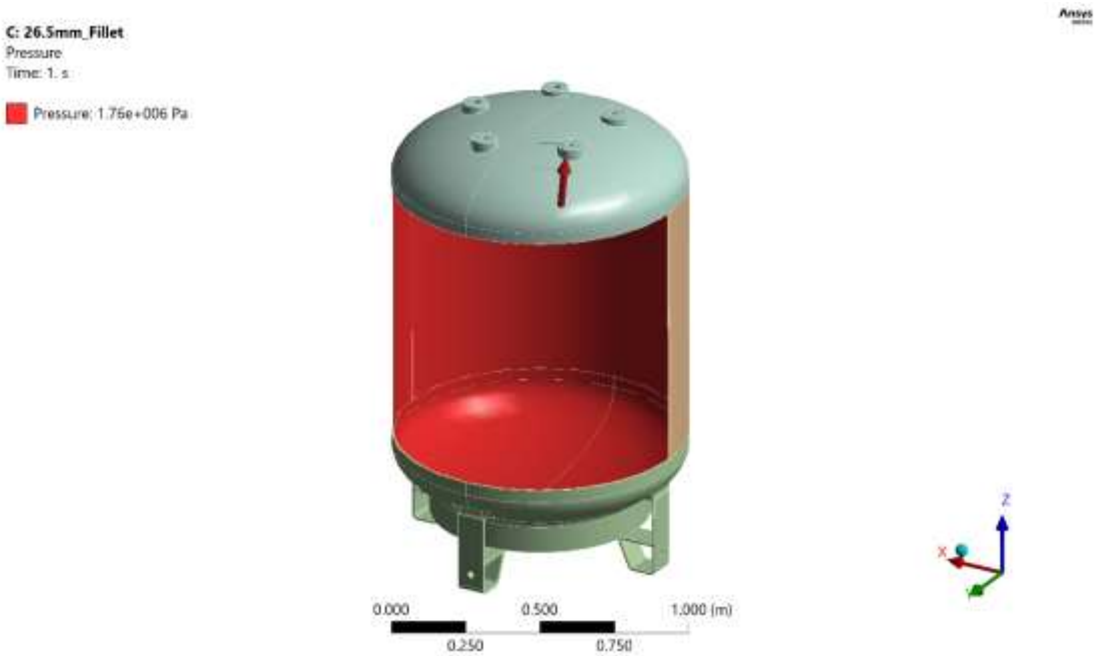


Figure 13 Pressure Loading

After creating the finite element mesh, appropriate boundary conditions and loading conditions were applied to simulate the operating environment of the LPG cylinder.

The primary loading condition considered in the analysis is internal pressure, which represents the pressure exerted by liquefied petroleum gas stored inside the cylinder. The design pressure used in the analysis corresponds to the maximum allowable operating pressure specified in the design documentation.

The internal pressure load acts uniformly on the inner surface of the cylindrical shell and heads. The applied pressure generates two principal stresses in the vessel wall:

Hoop stress

$$\Sigma_h = \frac{PD}{2t} \tag{3.7}$$

Longitudinal stress

$$\sigma_l = \frac{PD}{4t} \tag{3.8}$$

Where,

P= internal pressure

D= internal diameter

t= wall thickness

These stresses are responsible for the structural loading experienced by the vessel during operation.

In addition to internal pressure, the analysis also considered the self-weight of the vessel and the weight of the LPG contents, which contribute to the overall loading condition. Boundary constraints were applied at the support locations to represent the physical support conditions of the cylinder during operation.

Such realistic loading conditions are necessary to obtain accurate stress predictions from finite element simulations (Makwana & Patel, 2017).

CHAPTER 4 DESIGN CALCULATIONS AND ANALYSIS

4.1 Pressure Vessel Design Calculations

Pressure vessels subjected to internal pressure develop stresses in the vessel wall that must remain below the allowable limits of the material to prevent yielding or failure. Analytical design calculations are therefore used to determine the required thickness of the vessel components.

These calculations are based on thin-walled pressure vessel theory, which assumes that the wall thickness is small relative to the vessel diameter. Under this assumption, the stress distribution across the thickness is considered uniform (Moss & Basic, 2013).

The analytical design calculations performed for the LPG cylinder include:

- Cylindrical shell thickness determination
- Ellipsoidal head thickness calculation
- Stress verification
- Nozzle reinforcement design

4.1.1 Cylindrical Shell Thickness Calculation

1. Longitudinal stress

$$\sigma_h = \frac{PD}{2t} \tag{4.1}$$

Where,

The formula for the required thickness t to resist internal pressure is given by BS EN 13445-3:2021, Clause 7.4.2:

$$t = \frac{PD}{2f\eta - P} \tag{4.2}$$

Where,

f = allowable stress
η = joint efficiency

Substituting the design parameters:

$$t = \frac{1.76 \times 989.8}{2 \times 216.6 \times 0.85 - 1.76}$$
$$t = \frac{1742.05}{366.46}$$
$$t = 4.75 \text{ mm}$$

Thus, the minimum required shell thickness is approximately 4.75 mm.

Figure 14: Cylindrical Shell 3D Mode

4.1.2 Ellipsoidal Head Thickness Calculation

Table 9: Design Value

Parameter	Symbol	Value
Internal diameter	Di	989.8 mm
Head aspect ratio	AR	2.0
Corrosion allowance (internal& external)	c = ci + co	0.5 mm
Joint efficiency	Z	1.0

1. Head height (hi)

$$h_i = 0.5 \frac{D_i}{AR} + c \tag{4.3}$$

Where c = ci + co (corrosion allowances).

$$h_i = 0.5 \cdot \frac{989.8}{2.0} + 0.5 = 248 \text{ mm}$$

2. K factor (ratio for ellipsoidal head)

$$K = \frac{D_i}{2h_i} \tag{4.4}$$

$$K = \frac{989.8}{2 \cdot 248} = 2.0$$

3. Knuckle radius ®

$$r = D_i \left(\frac{0.5}{K} - 0.08\right) \tag{4.5}$$

$$r = 989.8 \left(\frac{0.5}{2} - 0.08\right) = 168.27 \text{ mm}$$

4. Crown radius (R)

$$R = D_i(0.44 \cdot K + 0.02) \tag{4.6}$$

$$R = 989.8(0.44 \cdot 2 + 0.02) = 890.82 \text{ mm}$$

5. Thickness Factors
    a. Y factor (relative crown thickness)

$$Y = \min\left(\frac{e}{R}, 0.04\right) \tag{4.7}$$

$$Y = \min\left(\frac{4.57588}{890.82}, 0.04\right) = 0.00514$$

    b. Z factor

$$Z = \log_{10} \frac{1}{Y} \tag{4.8}$$

$$Z = \log_{10} \frac{1}{0.0051367} = 2.289$$

X factor (relative knuckle radius)

$$X = \frac{r}{D_i} = \frac{168.27}{989.8} = 0.17 \tag{4.9}$$

N factor (empirical correction)

$$N = 1.006 - \frac{1}{6.2 + (90 \cdot Y)^4} \tag{4.10}$$

$$N = 1.006 - \frac{1}{6.2 + (90 \cdot 0.0051367)^4} = 0.8459$$

Beta coefficients (β_{01} and β_{02})

$$\beta_{01} = N(-0.1833Z^3 + 1.0383Z^2 - 1.2943Z + 0.837) \tag{4.11}$$

$$\beta_{01} = 0.8459(-0.1833 \cdot 2.289^3 + 1.0383 \cdot 2.289^2 - 1.2943 \cdot 2.289 + 0.837)$$
$$= 0.9443$$

$$\beta_{02} = \max\left(0.95(0.56 - 1.94Y - 82.5Y^2), 0.5\right) \tag{4.12}$$

$$\beta_{02} = \max\left(0.95(0.56 - 1.94 \cdot 0.00514 - 82.5 \cdot 0.00514^2), 0.5\right) = 0.5205$$

$$\beta = 10((0.2 - X)\beta_{01} + (X - 0.1)\beta_{02}) \tag{4.13}$$

$$\beta = 10((0.2 - 0.17) \cdot 0.9443 + (0.17 - 0.1) \cdot 0.5205) = 0.6476$$

Required Thicknesses

1. e_s**,** Thickness due to membrane stress (crown thickness)

$$e_s = \frac{P \cdot R}{2fZ - 0.5P} \tag{4.14}$$

$$e_s = \frac{1.767 \cdot 890.82}{2 \cdot 216.6 \cdot 1.0 - 0.5 \cdot 1.767} = 3.641 \text{ mm}$$

2. e_y**,** Thickness using Beta factor for knuckle yielding (empirical)

$$e_y = \beta \frac{P(0.75R + 0.2D_i)}{f} \tag{4.15}$$

$$e_y = 0.6476 \frac{1.767(0.75 \cdot 890.82 + 0.2 \cdot 989.8)}{216.6} = 4.576 \text{ mm}$$

3. e_b**,** Thickness due to Knuckle buckling

$$e_b = (0.75R + 0.2D_i)\left[\left(\frac{P}{111 f_b}\right)\left(\frac{D_i}{r}\right)^{0.825}\right]^{1/1.5} \tag{4.16}$$

$$e_b = (0.75 \cdot 890.82 + 0.2 \cdot 989.8)\left[\left(\frac{1.767}{111 \cdot 196.74}\right)\left(\frac{989.8}{168.27}\right)^{0.825}\right]^{1/1.5}$$
$$= 4.294 \text{ mm}$$

Required Thickness

$$e_{\text{required}} = \max(e_s, e_y, e_b) + c \tag{4.17}$$

$$e_{\text{required}} = \max(3.641, 4.576, 4.294) + 0.5 = 5.076\text{mm}$$

In practice, pressure vessel plates are selected from standard thickness values to ensure manufacturability and provide additional safety margin. Therefore, the selected shell thickness is 5.1 mm.

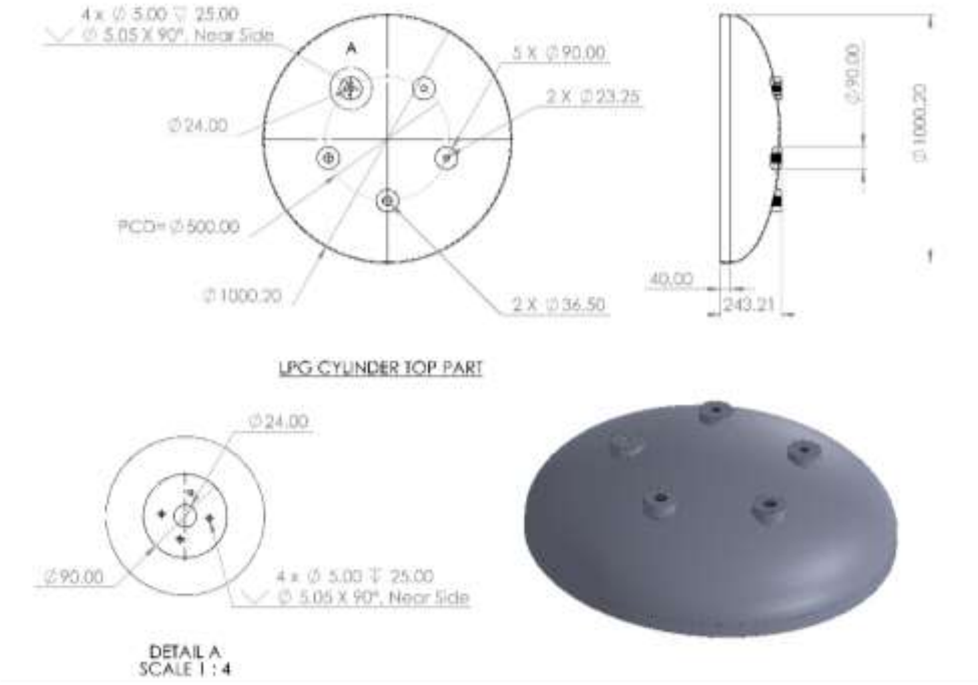


Figure 15: Ellipsoidal Head Geometry

4.2 Stress Verification

The maximum stress in the cylindrical shell at design pressure is evaluated to ensure it remains below the allowable stress.

The stress equation is

$$\sigma = \frac{P(D+t)}{2t\eta} \tag{4.18}$$

Substituting values

$$\sigma = \frac{1.76(989.8 + 5.1)}{2 \times 5.1 \times 0.85}$$

$$\sigma = 202.6 \text{ MPa}$$

Allowable stress:

$$\sigma_{allow} = 245 \text{ MPa}$$

Since

$$202.6 < 245$$

the vessel design satisfies the allowable stress requirement.

4.3 Finite Element Analysis and Fatigue Assessment

Although analytical pressure vessel equations provide reliable estimates of required thickness, they cannot capture localized stress concentrations that occur near structural discontinuities such as nozzle openings or head-to-shell junctions.

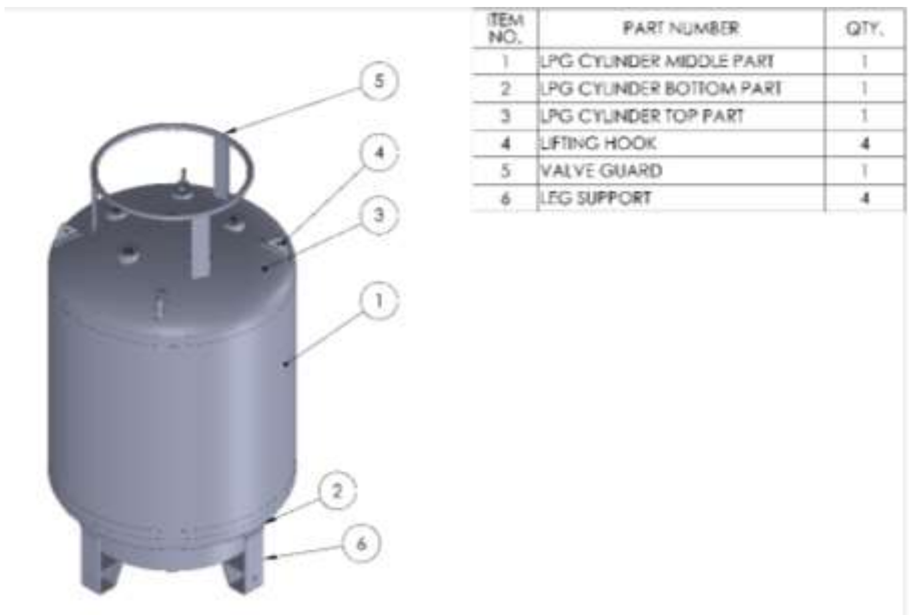
Finite Element Analysis (FEA) provides a numerical technique for evaluating the stress distribution throughout complex geometries. The method divides the structure into smaller elements, allowing the governing equations of elasticity to be solved numerically (Cook et al., 2002).

4.3.1 Finite Element Model

A three-dimensional CAD model of the LPG cylinder was created using the geometry obtained from the design calculations.

The model includes:

- cylindrical shell
- ellipsoidal heads
- nozzle openings
- support legs



ITEM NO.	PART NUMBER	QTY.
1	LPG CYLINDER MIDDLE PART	1
2	LPG CYLINDER BOTTOM PART	1
3	LPG CYLINDER TOP PART	1
4	LIFTING HOOK	4
5	VALVE GUARD	1
6	LEG SUPPORT	4

Figure 16: CAD Model

4.4 Development of S-N Curve for E450 BR Steel

4.4.1 Stress-Life Fatigue Method

Fatigue failure in metallic materials subjected to cyclic loading is typically analysed using the stress-life (S-N) approach. This method relates the stress amplitude to the number of cycles to failure using the Basquin equation:

$$\sigma_a = AN^b \tag{4.19}$$

Where,

- σ_a= stress amplitude (MPa)
- N= number of cycles to failure
- Aand b= material constants derived from experimental data or empirical approximations

The stress-life approach is particularly suitable for high-cycle fatigue (10^3-10^6 cycles), where applied stresses remain below the material's yield strength (Dowling, 2013). In cases where experimental S-N data are unavailable, approximate S-N curves can be constructed using empirical anchor points based on material properties such as ultimate tensile strength (Budynas & Nisbett, 2020).

4.4.2 Material Properties

The material considered in this study is E450BR structural steel, supplied by Tata Steel. Mechanical properties were obtained from the manufacturer's test certificate (Tata Steel Limited, 2023).

Table 10: Material Property

Property	Value
Yield strength S_y	519 Mpa
Ultimate tensile strength S_u	588 Mpa
Elongation	38%

These properties form the basis for estimating fatigue strength and constructing the S-N curve.

4.4.3 Estimation of Endurance Limit

For steels with ultimate tensile strength below approximately 1400 Mpa, the rotating-beam endurance limit can be approximated as:

$$S_e' = 0.5S_u \tag{4.20}$$

Substituting the ultimate tensile strength:

$$S_e' = 0.5 \times 588 = 294 \text{ MPa}$$

Thus, the theoretical endurance limit is:

$$S_e = 294 \text{ MPa}$$

This represents the stress level below which the material can theoretically withstand an infinite number of cycles without fatigue failure (Budynas & Nisbett, 2020).

4.4.3.1 Determination of Anchor Points

In the absence of experimental S-N data, two empirical anchor points are used to construct the fatigue curve (Dowling, 2013):

1. High-stress region at 10^3cycles
2. Endurance limit at 10^6cycles

First Anchor Point (N_1, σ_1)

At 10^3cycles, the fatigue strength of steel is approximately 90% of the ultimate tensile strength:

$$\sigma_1 = 0.9S_u \tag{4.21}$$

$$= 529 \text{ MPa}$$

$$(N_1, \sigma_1) = (10^3, 529 \text{ Mpa})$$

Second Anchor Point (N_2, σ_2)

At 10^6cycles, the fatigue strength is equal to the endurance limit:

$$\sigma_2 = S_e = 294 \text{ Mpa}$$

$$(N_2, \sigma_2) = (10^6, 294 \text{ Mpa})$$

4.4.3.2 Determination of Basquin Constants

The Basquin equation:

$$\Sigma_a = AN^b \tag{4.22}$$

can be linearized by taking logarithms:

$$\log\ \sigma_a = \log\ A + b\log\ N \tag{4.23}$$

The slope b is calculated using the two anchor points:

$$B = \frac{\log\ \sigma_2 - \log\ \sigma_1}{\log\ N_2 - \log\ N_1} \tag{4.24}$$

Slope Calculation

$$\log_{10}\sigma_2 = \log_{10} 294 \approx 2.468$$
$$\log_{10}\sigma_1 = \log_{10} 529 \approx 2.723$$
$$\log_{10} N_2 = 6, \log_{10} N_1 = 3$$
$$b = \frac{2.468 - 2.723}{6 - 3} = -0.085$$

Constant Calculation

$$A = \frac{\sigma_1}{N_1^b} \tag{4.25}$$

$$N_1^b = 10^{3\cdot -0.085} = 10^{-0.255} \approx 0.5556$$
$$A = \frac{529}{0.5556} \approx 952$$

Final S-N Equation

The final stress-life relation for E450 BR steel:

$$\sigma_a = 952N^{-0.085}, 10^3 \leq N \leq 10^6$$

For cycles beyond 10^6:

$$\sigma_a = 294\ \text{MPa}$$

The stress amplitude for selected cycles is:

Table 11: SN Values

Cycles (N)	Stress amplitude σ_a(Mpa)
10^3	529
2×10^3	499
5×10^3	462
10^4	434
2×10^4	411
5×10^4	380
10^5	358
2×10^5	338
5×10^5	312
10^6	294

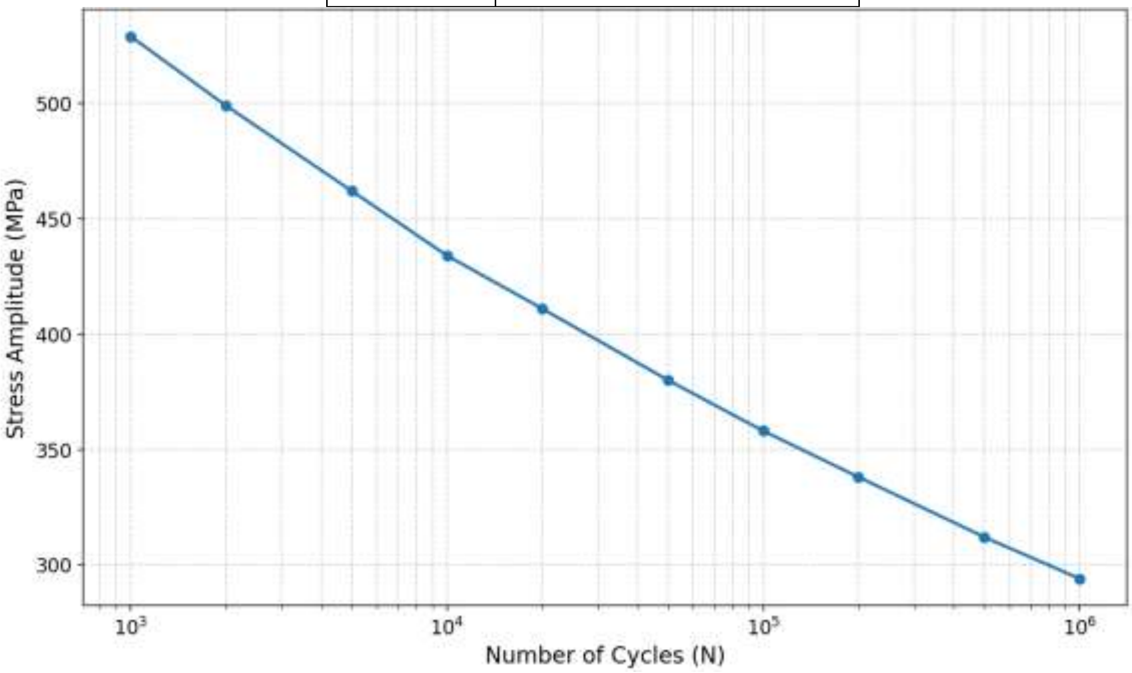


Figure 17: SN Curve of E450BR

4.5 Fatigue Life Assessment of the 990-Litre LPG Cylinder

Pressure vessels used for the storage and transportation of liquefied petroleum gas (LPG) are subjected to repeated pressurization and depressurization cycles during their operational life. These cyclic loads can lead to fatigue damage, even if the stresses remain below the yield strength of the material. Therefore, assessing fatigue life is a critical step in ensuring the structural integrity and safety of the vessel throughout its service period.

The fatigue life of the LPG cylinder in this study is evaluated according to the methodology described in ASME Boiler and Pressure Vessel Code (BPVC) Section VIII Division 2, Part 5, Annex 3-F (American Society of Mechanical Engineers, 2021). This methodology allows for the estimation of fatigue life based on the elastic alternating stress obtained from stress analysis without requiring a material-specific S-N curve. The approach ensures compliance with pressure vessel design codes while providing a conservative assessment of fatigue performance.

The analytical procedure followed in this study includes:

1. Identification of material mechanical properties
2. Determination of the modulus of elasticity at design temperature
3. Calculation of alternating stress amplitude
4. Calculation of fatigue parameters Yand Xusing ASME Annex 3-F equations
5. Determination of the allowable number of fatigue cycles
6. Calculation of the fatigue damage factor using Miner's rule
7. Evaluation of fatigue safety based on the allowable cycles

4.5.1 Material Properties

The LPG cylinder is fabricated from high-strength structural steel. The relevant mechanical properties are summarized in given table

Table 12: Mechanical Properties of E450BR

Property	Symbol	Value
Yield strength	S_y	519 Mpa
Ultimate tensile strength	S_u	588 Mpa

4.5.2 Modulus of Elasticity at Design Temperature

$$T = 65^{\circ}C$$

The modulus of elasticity at this temperature, E_T, is obtained from ASME BPVC Section II Part D- Material Properties, which provides temperature-dependent elastic properties for engineering steels (ASME, 2021). For structural steel at 65 °C, the modulus is:

4.5.3 Alternating Stress Amplitude

The maximum stress range resulting from cyclic loading was obtained from finite element analysis (FEA), considering the following loads:

- Internal pressure of 1.76 Mpa
- Self-weight of the vessel
- Weight of the contained LPG (450 kg)
- Nozzle thrust loads due to internal pressure

The maximum equivalent stress range obtained from FEA is:

$$\Delta S_p = 369.04 \text{ Mpa}$$

According to ASME fatigue methodology, the effective alternating stress amplitude, S_a, is calculated as half of the stress range:

$$S_a = \frac{\Delta S_p}{2} \tag{4.26}$$

$$S_a = \frac{369.04}{2} = 184.52 \text{ MPa}$$

4.5.4 Determination of Fatigue Parameter Y

The fatigue parameter Yis a logarithmic scaling parameter used in ASME Annex 3-F to relate the alternating stress to the elastic modulus. It is calculated using Equation 3-F.1:

$$Y = \log\left[28.3 \times 10^3 \left(\frac{S_a}{E_T}\right)\right] \tag{4.27}$$

Substituting the values:

$$S_a = 184.52 \text{ MPa}$$

$$E_T = 199860 \text{ MPa}$$

$$\frac{S_a}{E_T} = \frac{184.52}{199860}$$

$$= 0.0009232$$

Now,

$$28300 \times 0.0009232$$
$$= 26.12$$

Therefore,

$$Y = \log(26.12) \tag{4.28}$$
$$Y = 1.425$$

Thus,

$$Y = 1.416$$

The value of 10^{Y} is 26.06

4.5.6 Selection of Fatigue Equation

- If $10^{Y} \geq 20$, Equation 3-F.2 applies
- If $10^{Y} < 20$, Equation 3-F.3 applies

Additionally, ASME provides separate equations for different ultimate tensile strength ranges. For materials with $S_u \leq 552\text{MPa}$, Equations 3-F.2/3-F.3 are used, while higher-strength steels use Equations 3-F.4/3-F.5. Since the ultimate tensile strength of the present material is 588 MPa (intermediate between 552 and 793 MPa), ASME recommends interpolation. To maintain a conservative estimate, the equation for $S_u \leq 552\text{MPa}$ (Equation 3-F.2) was adopted.

Equation 3-F.2 is:

$$X = -4706.5245 + 1813.6228Y + \frac{6785.5644}{Y} - 368.12404Y^2 - \frac{5133.7345}{Y^2} + 30.708204Y^3 + \frac{1596.1916}{Y^3} \tag{4.29}$$

Substituting $Y = 1.416$:

$$Y^2 = 2.0078, Y^3 = 2.839$$
$$\frac{1}{Y} = 0.706, \frac{1}{Y^2} = 0.498, \frac{1}{Y^3} = 0.352$$

Now

$$X = -4706.5245 + 1813.6228(1.416) + 6785.5644(0.706) - 368.12404(2.845) - 5133.7345(0.497) + 30.708204(2.839) + 1596.1916(0.352)$$

$$X \approx 4.563$$

4.5.7 Allowable Fatigue Life

The allowable number of fatigue cycles is calculated as:

$$N = 10^{X} \tag{4.30}$$

$$N = 10^{4.563} \approx 36{,}559 \text{ cycles}$$

4.5.8 Fatigue Damage Factor

The fatigue damage factor, D_f, is calculated using Miner's cumulative damage rule:

$$D_f = \frac{n}{N} \tag{4.31}$$

Where n is the number of expected service cycles. Assuming the cylinder undergoes 365×15×2=10,950 pressure cycles during its operational life:

$$D_f = \frac{10950}{30{,}854} \approx 0.3548$$

4.5.9 Fatigue Design Evaluation

According to ASME Section VIII Division 2, the fatigue design is considered acceptable if $D_f < 1$. Since $D_f = 0.3548 < 1$, the LPG cylinder satisfies the fatigue design requirements. This indicates that the vessel is expected to safely withstand the specified cyclic loading conditions throughout its service life.

4.6 Analytical Stress and Deformation Analysis of the LPG Cylinder

4.6.1 Thin Cylinder Assumption

This section presents the governing equations and final results for the thin-walled stress and deformation analysis. Full step-by-step numerical working is provided in Annex C.

The thin cylinder assumption is valid when

$$\frac{t}{D} < 0.1 \tag{4.32}$$

where tis the wall thickness and Dis the internal diameter of the cylinder.

For the LPG cylinder used in this study,

$$t = 5.1 \text{ mm}$$

$$D = 990 \text{ mm}$$

Therefore,

$$\frac{t}{D} = \frac{5.1}{990} = 0.005151$$

Since this ratio is significantly smaller than 0.1, the cylinder satisfies the thin-wall condition and the thin pressure vessel theory can be applied. Under this assumption the stresses across the wall thickness are considered uniform (Budynas & Nisbett, 2020).

4.6.2 Design Parameters

The design parameters used for the analytical calculations are summarized below.

Table 13: Design Parameters

Parameter	Value
Internal pressure P	1.76 MPa
Cylinder wall thickness t	5.1 mm
Inner diameter D	990 mm
Elastic modulus E	199860 MPa
Poisson's ratio ν	0.30

The temperature-dependent modulus of elasticity was obtained from ASME Boiler and Pressure Vessel Code Section II Part D.

4.6.3 Circumferential (Hoop) Stress

When internal pressure acts inside a cylindrical vessel, the wall experiences a circumferential stress acting tangentially around the circumference of the cylinder. This stress tends to split the cylinder along its longitudinal axis and is therefore the most critical stress component in cylindrical pressure vessels.

For a thin-walled cylinder, the circumferential stress is expressed as

$$\sigma_h = \frac{PD}{2t} \tag{4.33}$$

Substituting the design parameters into the equation,

$$\sigma_h = \frac{1.76 \times 990}{2 \times 5.5}$$

$$\sigma_h = 158.40 \text{ MPa}$$

Thus, the circumferential stress developed in the cylinder wall is approximately 158.40 MPa.

4.6.4 Longitudinal Stress

In addition to hoop stress, the cylinder wall also experiences a longitudinal stress due to the pressure acting on the end caps of the cylinder. This stress acts along the axis of the cylinder.

For thin cylindrical vessels, the longitudinal stress is given by

$$\sigma_l = \frac{PD}{4t} \tag{4.34}$$

Substituting the values,

$$\sigma_l = \frac{1.76 \times 990}{4 \times 5.5}$$

$$\sigma_l = 79.2 \text{ MPa}$$

Therefore, the longitudinal stress acting in the cylinder wall is approximately 79.2 MPa.

As predicted by thin cylinder theory, the circumferential stress is approximately twice the longitudinal stress.

4.6.5 Equivalent (Von Mises) Stress

Since the cylinder wall experiences stresses in more than one direction, it is useful to evaluate the equivalent stress using the distortion energy theory. The Von Mises criterion is widely used for ductile materials such as steel and is also the stress criterion used in finite element software such as ANSYS Workbench.

For a biaxial stress state where radial stress is negligible, the Von Mises equivalent stress is expressed as

$$\sigma_v = \sqrt{\sigma_h^2 + \sigma_l^2 - \sigma_h\sigma_l} \tag{4.35}$$

Substituting the calculated stresses,

$$\sigma_v = \sqrt{158.4^2 + 79.2^2 - (158.4 \times 79.2)}$$

Now,

$$\sigma_v = \sqrt{25090.56 + 6272.64 - 12545.28}$$

$$\sigma_v = \sqrt{21931}$$

$$\sigma_v \approx 137 \text{ MPa}$$

Thus, the equivalent Von Mises stress in the cylinder wall is approximately 137 MPa.

4.6.6 Radial Deformation of the Cylinder

The internal pressure causes the cylinder wall to expand slightly in the radial direction. The radial deformation of a thin cylindrical shell can be estimated using elasticity theory.

The radial deformation is expressed as

$$\delta_r = \frac{Pr^2}{Et}(1 - \nu) \tag{4.36}$$

Where,

r= internal radius of the cylinder
E= modulus of elasticity
ν= Poisson's ratio.

The internal radius is obtained from the diameter as

$$r = \frac{D}{2} \tag{4.37}$$

$$r = \frac{990}{2}$$

$$r = 495 \text{ mm}$$

Substituting the values into the deformation equation,

$$\delta_r = \frac{1.76 \times 495^2}{199860 \times 5.1}(1 - 0.3)$$

The radial deformation is obtained as approximately

$$\delta_r = 0.30 \text{ mm}$$

Thus, the cylinder expands radially by about 0.30 mm under the operating pressure.

4.6.7 Finite Element Modelling of the LPG Cylinder

Finite Element Analysis (FEA) is widely used in pressure vessel design to evaluate stress distribution and deformation under complex loading conditions. Analytical formulas provide an estimation of membrane stresses in cylindrical shells; however, they cannot fully capture localized stress concentrations near discontinuities such as nozzles, supports, and weld joints. For this reason, numerical methods such as FEA are commonly used to complement analytical calculations (Dowling, 2013).

In the present work, a finite element model of the LPG cylinder was developed using ANSYS Workbench. The geometric model was created based on the dimensional specifications obtained from the engineering drawing used for the vessel design. The model consists of the cylindrical shell, ellipsoidal heads, and nozzle connections.

CHAPTER 5 RESULTS AND DISCUSSION

5.1 Mesh Independence

Before any stress or fatigue results could be trusted, it was necessary to confirm that the finite element solution had converged that is, that further refining the mesh would not materially change the answer. Three refinement levels were tested: a coarse mesh of approximately 130,000 elements, a medium mesh of 450,000 elements, and a fine mesh of nearly 1,500,000 elements. Maximum total deformation was chosen as the convergence metric because it reflects the integrated stiffness response of the whole structure and is sensitive to mesh quality everywhere, including at the nozzle junctions and tripod support attachments where stress concentrations are sharpest. This approach follows the grid convergence index (GCI) methodology widely used in computational structural mechanics to objectively quantify discretisation error (Celik & Karatekin, 1997).

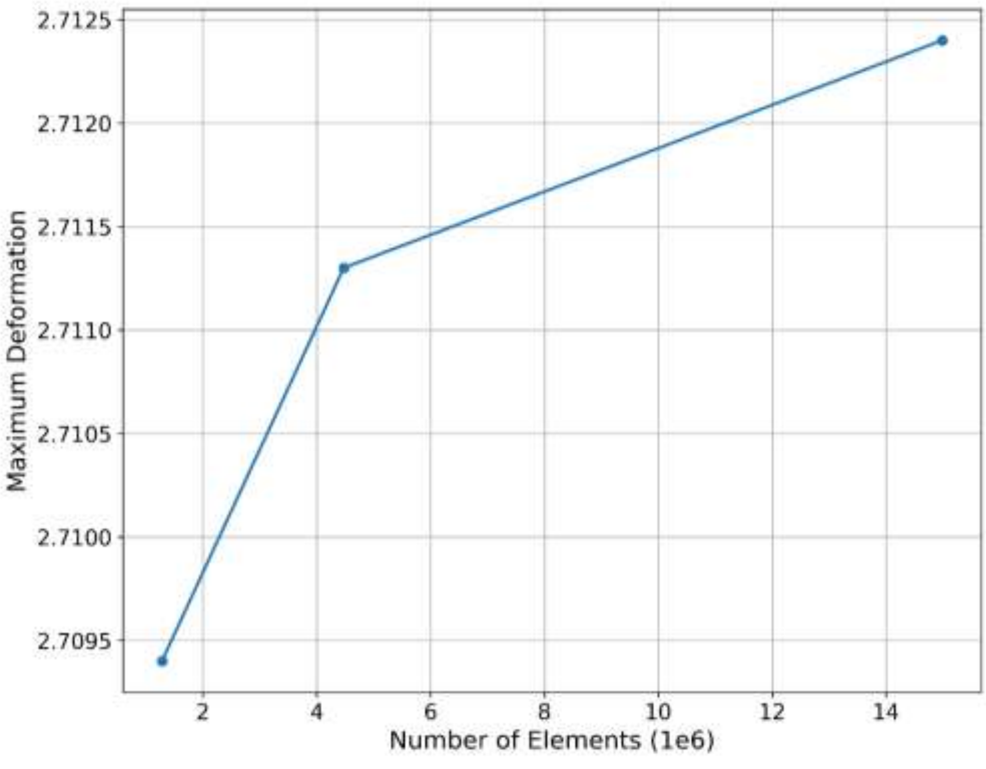


Figure 18 Mesh Independence Study

The results of the FSI mesh independence study are presented in Figure 18. The maximum deformation exhibited a monotonically increasing trend with mesh refinement, rising from approximately 2.7094 mm at the coarse level (130,000 elements) to approximately 2.7113 mm at the medium level (420,000 elements), a change of 1.9e-3 mm (0.07%), and further to approximately 2.7124 mm at the fine level (1,500,000 elements), a change of only 1.1e-3 mm (0.04%) relative to the medium mesh. The GCI computed between the medium and fine mesh levels for the structural domain was GCI = 0.08, confirming that the medium mesh of approximately 420,000 elements is adequate for the structural analyses performed in this study and results are independent of the mesh.

5.2 Stress Distribution and Structural Response

One of the key advantages of finite element analysis over closed-form solutions is precisely the ability to capture what thin-walled theory smooths over: the sharp local stress variations that arise wherever the geometry changes. While analytical formulas yield a single membrane stress for the cylindrical shell body, the FEA model resolves the full stress field, including the elevated values at nozzle penetrations, head-to-shell junctions, and support bracket attachment points.

Results are evaluated using the von Mises equivalent stress, which consolidates the three principal stress components into a single scalar measure that can be directly compared with material yield and allowable stress values. For ductile steels, the von Mises (distortion energy) criterion is the standard choice and is the basis for structural assessments in both ASME and EN 13445 (Hibbeler, 2017). The von Mises stress is defined as

$$\sigma_{vm} = \sqrt{\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]} \tag{5.1}$$

Where,

$\sigma_1, \sigma_2, \sigma_3$ are the principal stresses.

The von Mises stress provides an equivalent stress value that can be directly compared with the material yield strength to evaluate structural safety.

The stress results obtained from the finite element analysis indicated that the maximum equivalent stress occurred near the nozzle connection region, which is expected because geometric discontinuities tend to produce localized stress concentrations.

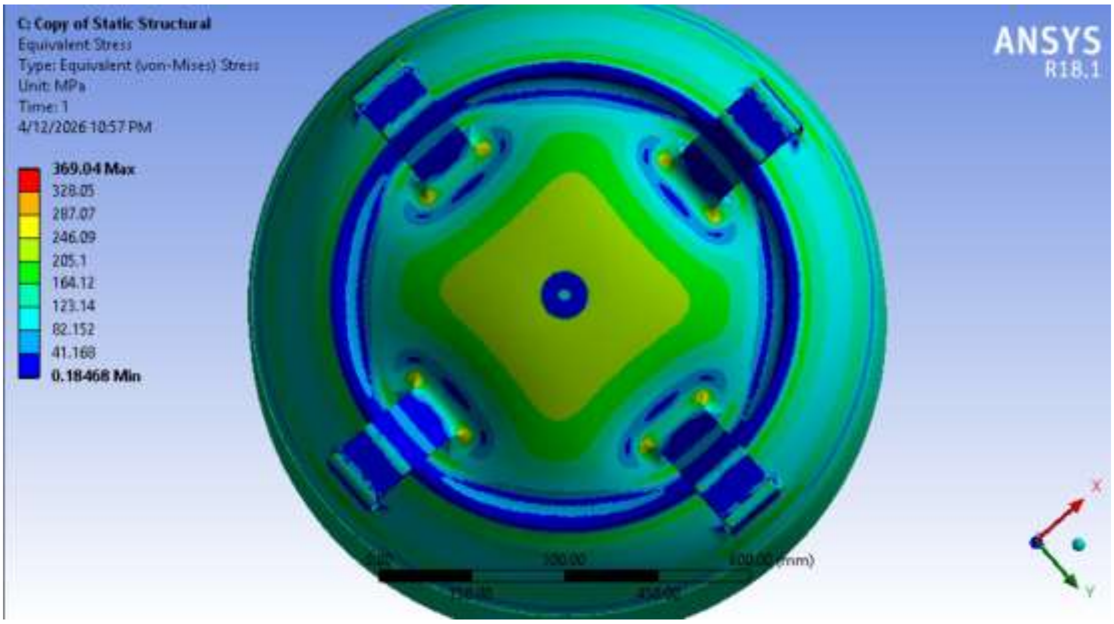


Figure 19 Equivalent stress distribution in LPG cylinder under internal pressure

The stress analysis identifies the critical areas where material failure is most likely to initiate. The observed maximum stress is 369.04 MPa. The peak stresses are concentrated at the welding seams and the attachment points of the base supports as shown in Figure 18. The bulk of the cylindrical wall maintains a stress level between 82 MPa and 164 MPa.

Which corresponds to the stress range used later in the fatigue analysis calculations. These results indicate that although the majority of the vessel experiences relatively uniform membrane stress, localized regions may experience higher stresses due to geometric discontinuities. Such stress concentrations must be carefully evaluated during fatigue assessment because they may become potential sites for crack initiation (Zhang et al., 2019).

5.3 Result of Total Deformation Analysis

The total deformation of the 450 kg LPG cylinder was obtained from static structural analysis in ANSYS. The maximum deformation is found to be 2.7113 mm. From the contour plot, the deformation is highest at the top dome region near the nozzle openings. The cylindrical part shows smaller deformation, and the bottom region has very little deformation because it is fixed by the supports.

The main reason for higher deformation at the top is the shape of the dome, which is more flexible than the cylindrical body. Also, the nozzle openings reduce stiffness in that area, making it deform more. The forces applied at the nozzles further increase the deformation locally. It can also be seen that deformation increases gradually from the bottom to the top. This is because the base is fixed and cannot move, while the upper part is free to deform under internal pressure. Overall, the deformation pattern is smooth, and the results show that the top dome region is the most critical area in terms of deformation.

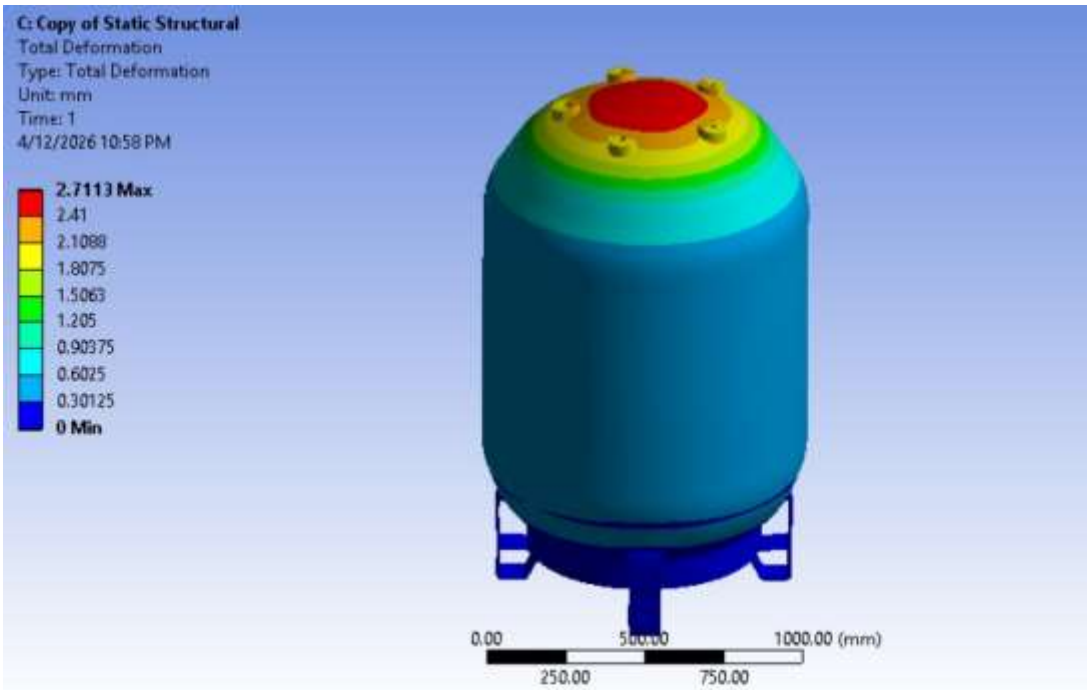


Figure 20 Total deformation contour of 450 kg LPG cylinder

5.4 Effective Alternating Stress Amplitude

With the peak stress range established from the FEA results, the next step is to convert it into the alternating stress amplitude that the ASME fatigue curves require. ASME Section VIII Division 2 (Part 5, Clause 5.5.3.2) defines this as

$$S_{alt,k} = \frac{K_f \times K_e \times \Delta S_{P,k}}{2} \tag{5.2}$$

Where,

K_f= fatigue strength reduction factor

K_e= fatigue penalty factor

$\Delta S_{P,k}$= equivalent stress range

For the LPG cylinder analysis:

$$\Delta S_{P,k} = 369.04 \text{ MPa}$$

The fatigue strength reduction factor and penalty factor were taken as

$$K_f = 1$$

$$K_e = 1$$

Substituting the values into the equation:

$$S_{alt,k} = \frac{1 \times 1 \times 369.04}{2}$$

$$S_{alt,k} = 184.52 \text{ MPa}$$

Thus, the effective alternating stress amplitude for fatigue evaluation is approximately

$$S_{alt} = 184.52\text{MPa}$$

This value represents the cyclic stress amplitude experienced by the vessel during pressure loading cycles.

5.5 Fatigue Life Estimation

Using the alternating stress amplitude of 184.52 MPa, the allowable fatigue life is read from the ASME design curves (Annex 3-F). These curves relate alternating stress to cycles-to-failure for carbon and low-alloy steels and incorporate a factor of safety on both stress and cycles, making them inherently conservative (ASME, 2021).

The modulus of elasticity for the selected material HSFQ-450 steel at the design temperature of 65 °C is

$$E_T = 199.86 \times 10^3 \text{ MPa}$$

Using the fatigue evaluation procedure described in the ASME standard, the following parameters were obtained:

$$Y = 1.416$$

$$10^Y = 26.06$$

Substituting the value of Y into the fatigue curve equation gives

$$X = 4.563$$

The number of cycles to failure is therefore

$$N = 10^X \tag{5.3}$$

$$N = 10^{4.563}$$

$$N = 36559 \text{ cycles}$$

5.5.1 Fatigue Damage Assessment

To determine whether the vessel satisfies fatigue life requirements, the fatigue damage factor is calculated using Miner's cumulative damage rule.

$$D_f = \frac{n}{N} \tag{5.4}$$

Where,

n= number of expected operating cycles
N= allowable fatigue cycles

For the LPG cylinder:

$$n = 1$$
$$N = 36559$$

Therefore

$$D_f = \frac{12000}{36559}$$
$$D_f = 0.3282$$

Since

$$D_f < 1$$

The fatigue damage accumulated during the design life is within acceptable limits.

Therefore, the LPG cylinder design satisfies the fatigue strength requirements for cyclic pressure service.

5.6 Fatigue Life Estimation using numerical method

The fatigue analysis predicts the number of cycles the vessel can withstand before crack initiation occurs due to repeated pressurization. The observed minimum life is 70,266 cycles and maximum life is 1e6 cycles. The fatigue critical regions are located at the inner radius of the base support legs as shown in Figure 18. This indicates that the interface between the vessel body and its supports is the primary life-limiting factor, rather than the pressure vessel wall itself. The design is robust for the main body, but the support bracket geometry creates stress risers that significantly reduce the total cycle count.

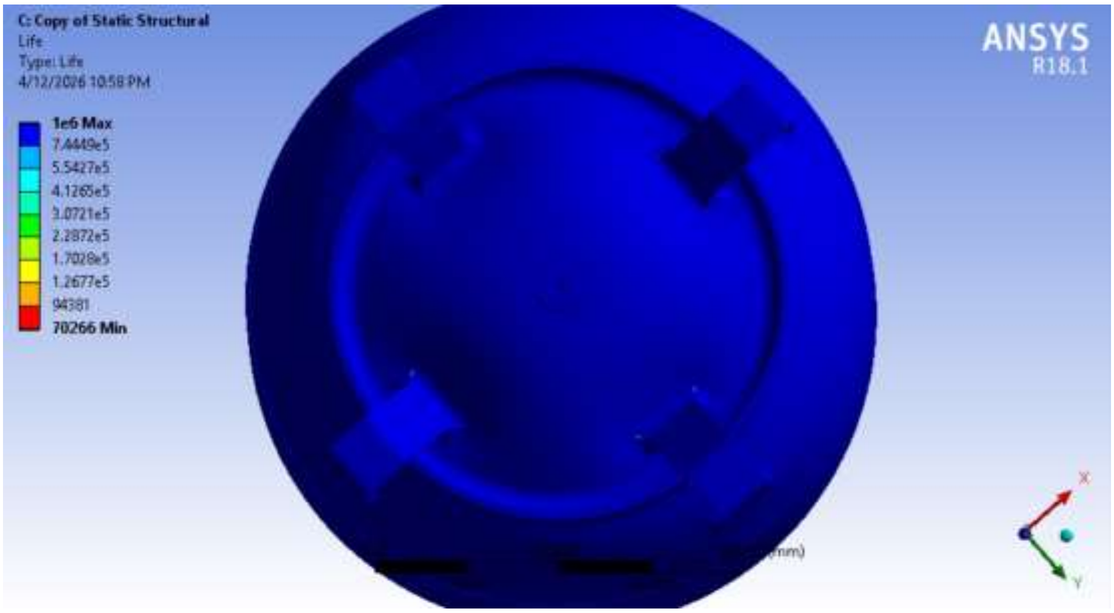


Figure 21 Fatigue life estimation

5.7 Comparison of Analytical and FEA Results

To evaluate the reliability of the analytical predictions, the key results obtained from thin-walled pressure vessel theory were compared against the finite element analysis (FEA) outputs from ANSYS. This comparison helps clarify where the two approaches agree, where they diverge, and why such differences are physically expected. A summary of the comparison is presented in Table 14.

Table 14: Comparison of Analytical and FEA Results

Parameter	Analytical	FEA	Remarks/ Limitation
Von Mises Stress (σv)	148 MPa	123-205 MPa (shell body), 369 MPa (bracket weld)	The analytical stress value of 148 MPa closely matches the stress range obtained from the FEA results for the vessel shell, indicating good agreement between the two approaches. However, the much higher stress concentration of 369 MPa observed at the bracket weld is not predicted by the analytical method. This is because thin-walled pressure vessel theory

			assumes a smooth and uniform geometry and therefore cannot capture the localised stress concentrations that develop around welded joints and support brackets.
Radial Deformation (δr)	0.30 mm (cylindrical shell, pressure only)	0.30-0.60 mm (shell), 2.71 mm max at top dome	The analytical deformation value for the shell (0.30 mm) is very close to the minimum deformation predicted by the FEA model (0.301 mm), showing strong agreement between the two methods in the cylindrical shell region. However, the maximum deformation obtained from the FEA analysis, 2.71 mm at the top dome, is significantly higher because the numerical model considers the combined influence of internal pressure, axial stresses, and gravitational loading. In contrast, the analytical equation only accounts for radial expansion caused by internal pressure. For this reason, directly comparing the analytical result with the dome deformation predicted by FEA is not entirely appropriate.

Min. Fatigue Life	36,559 cycles (ASME; Df = 0.35)	70,266 cycles (min. at support region)	The two approaches cannot be compared directly because they are based on different fatigue assessment methods. The ASME procedure evaluates fatigue using a stress-amplitude approach derived from the maximum stress range obtained from the FEA results. In contrast, the ANSYS fatigue analysis uses an S-N curve together with the full spatial distribution of cyclic stresses throughout the structure. As a result, the FEA-based fatigue prediction gives a higher life estimate because it captures the actual stress distribution in greater detail, particularly in regions away from the support structure where stresses are lower. Despite these methodological differences, both assessment approaches indicate that the vessel design is safe and capable of withstanding the expected service life of 10,950 operating cycles.

The comparison between the analytical calculations and the numerical FEA results shows strong overall agreement, particularly in the cylindrical shell region of the vessel. The analytical Von Mises stress value of 148 MPa falls within the FEA stress range of 123-205 MPa for the shell, suggesting that thin-walled pressure vessel theory provides reliable membrane stress predictions for the main cylindrical body. Similarly, the difference in radial deformation between the analytical and FEA results is less than 0.4%, which further supports the accuracy and validity of the finite element model.

The largest difference between the two methods occurs near the bracket weld regions, where the FEA predicts a peak stress of 369 MPa, significantly higher than the analytical shell stress. This outcome is expected because analytical equations based on thin-walled assumptions consider the vessel geometry to be smooth and uniform. As a result, they cannot account for local stress concentrations caused by welded supports, geometric discontinuities, or abrupt structural transitions. In contrast, finite element analysis is capable of capturing these localised effects in detail. For this reason, the present study uses the FEA peak stress rather than the analytical stress as the basis for the ASME fatigue assessment, making the evaluation both more conservative and more consistent with design code requirements.

A similar trend can be observed in the fatigue life assessment. The ASME analytical fatigue procedure predicts an allowable life of 36,559 cycles with a damage factor of 0.35, whereas the ANSYS fatigue simulation estimates a minimum fatigue life of 70,266 cycles at the support region. Although the values differ, they are not contradictory because the two methods evaluate fatigue differently. The ASME approach is based on the maximum stress amplitude in the structure and therefore provides a conservative lower-bound estimate. On the other hand, the ANSYS fatigue model considers the full spatial distribution of cyclic stresses, which generally results in longer predicted fatigue lives in areas experiencing lower stress levels. Importantly, both methods indicate fatigue lives that are far greater than the expected operational requirement of 10,950 service cycles over a 15-year design life, confirming the structural safety and reliability of the vessel design (Shariati & Janghorban, 2017).

CHAPTER 6 CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

This study aimed to thoroughly evaluate the structural integrity and fatigue performance of a 450 kg LPG cylinder under realistic operating conditions by combining analytical calculations with finite element analysis (FEA) in accordance with EN 13445 and ASME Section VIII Division II standards. The findings provide important insights into both the overall strength of the vessel and the critical regions that influence its long-term durability.

The study began with analytical design calculations based on thin-walled pressure vessel theory. Using this approach, the required thicknesses for the cylindrical shell and end heads were determined, and the resulting stresses were checked against the allowable limits for E450BR steel. The calculated hoop stress and longitudinal stress were approximately 158.4 MPa and 79.2 MPa, respectively, showing that the vessel can safely withstand the design internal pressure of 1.76 MPa. These results confirm that the basic design satisfies the required strength criteria.

To gain a more detailed understanding of the vessel behaviour, finite element analysis was then carried out using ANSYS. The simulation results showed that stresses across most of the cylindrical shell remain relatively uniform and within safe limits. However, higher localised stresses were identified near structural discontinuities such as nozzle connections, welded joints, and support brackets. The maximum equivalent (von Mises) stress predicted by the simulation was 369.04 MPa, occurring mainly around the support bracket interfaces and welded regions.

Fatigue performance was evaluated using both the S-N curve method and the ASME fatigue assessment procedure. The alternating stress amplitude was obtained from the stress range calculated through FEA, and this was used to estimate the fatigue life of the cylinder. The analysis predicted a minimum fatigue life of approximately 70,266 cycles in the most critical regions, while less critical areas of the vessel showed fatigue lives extending up to 10^6 cycles. In addition, the calculated fatigue damage factor remained below unity, indicating that the cylinder is expected to perform safely under the anticipated cyclic loading conditions throughout its service life.

The results further reveal that, although the main pressure-retaining shell performs safely under both static and cyclic loading, the overall fatigue behaviour of the cylinder is largely controlled by the support structures and welded attachments. The most critical locations identified in the analysis were the inner radii of the support brackets and the welded attachment regions, where stress concentrations were significantly higher than in the rest of the vessel.

In summary, the study demonstrates that the proposed 450 kg LPG cylinder design is structurally sound and satisfies both strength and fatigue requirements. At the same time, the findings emphasise the importance of geometric discontinuities in influencing the long-term reliability and durability of pressure vessels. Careful attention to the design of supports, welds, and attachment regions is therefore essential for improving fatigue performance and extending service life.

6.2 Recommendations

1. The present study identifies support brackets as the most critical regions governing fatigue life. Therefore, further research can be conducted to optimize the geometry of the support structures. Modifications such as increasing fillet radius, improving weld profiles, or redesigning the support configuration may significantly reduce stress concentration and enhance fatigue life.

2. The current analysis does not include detailed modelling of weld geometry and residual stresses. Since welded joints are potential sites for crack initiation, future studies can incorporate weld bead geometry and residual stress effects to obtain a more realistic fatigue life prediction.

3. The nozzle regions were subjected to loading in the present study; however, a more detailed investigation considering nozzle-shell interaction, reinforcement pads, and stress intensification factors can be carried out to better understand their influence on structural integrity.

4. The study is limited to numerical and analytical approaches due to practical constraints. Experimental validation such as hydrostatic pressure testing and fatigue testing is recommended for future research to verify the simulation results and improve reliability.

5. Since LPG cylinders operate under varying temperature conditions, further studies can be conducted to evaluate the effect of temperature on internal pressure, material properties, and induced stresses.

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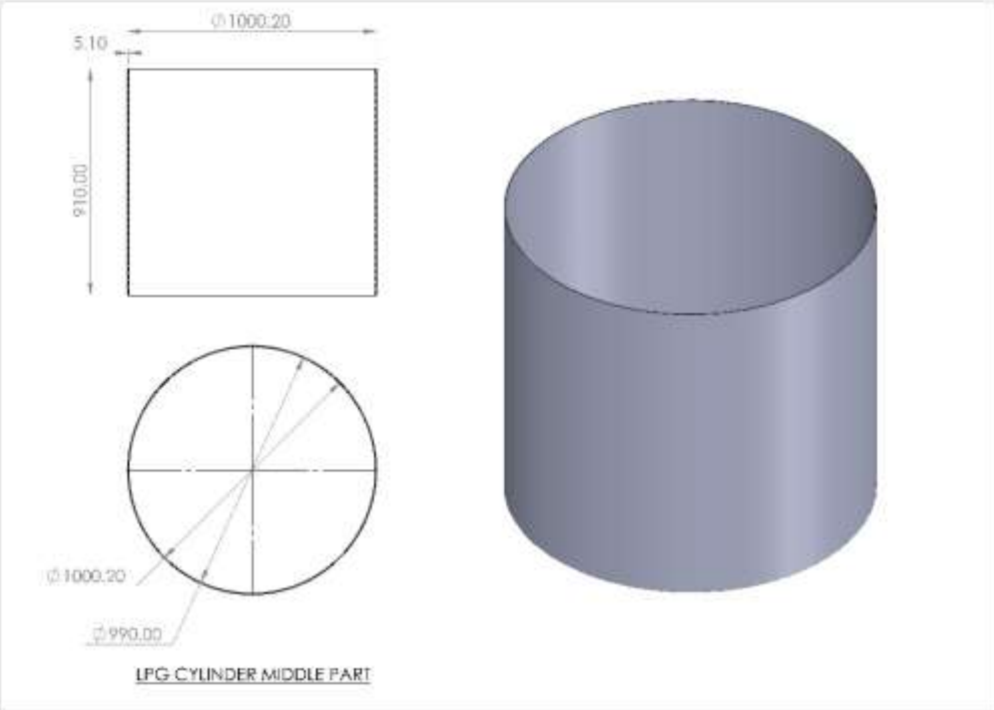
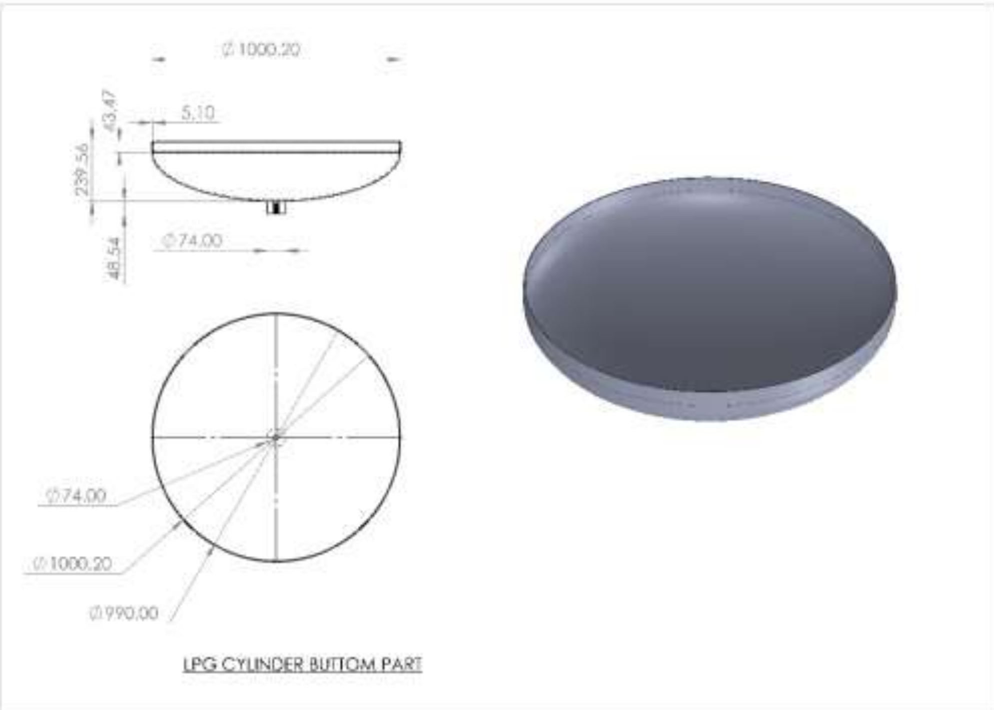
ANNEX

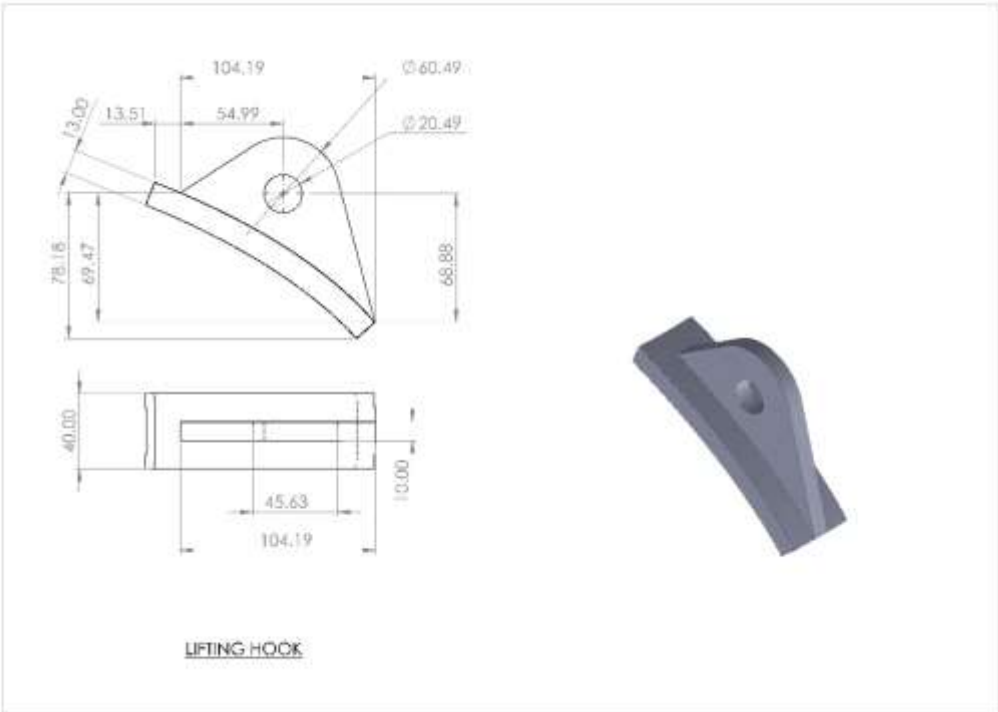


Ø1000.20
5.10
910.00
Ø1000.20
Ø990.00
LPG CYLINDER MIDDLE PART



Ø1000.20
43.47
5.10
239.56
Ø74.00
48.54
Ø74.00
Ø1000.20
Ø990.00
LPG CYLINDER BUTTOM PART


4 x ∅ 5.00 ▽ 25.00
∅ 5.05 X 90°, Near Side
A
5 X ∅90.00
2 X ∅23.25
∅24.00
PCD= ∅500.00
∅1000.20
2 X ∅36.50
∅90.00
∅1000.20
40.00
243.21
LPG CYLINDER TOP PART
∅24.00
∅90.00
4 x ∅ 5.00 ▽ 25.00
∅ 5.05 X 90°, Near Side
DETAIL A
SCALE 1 : 4


104.19
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13.00
13.51
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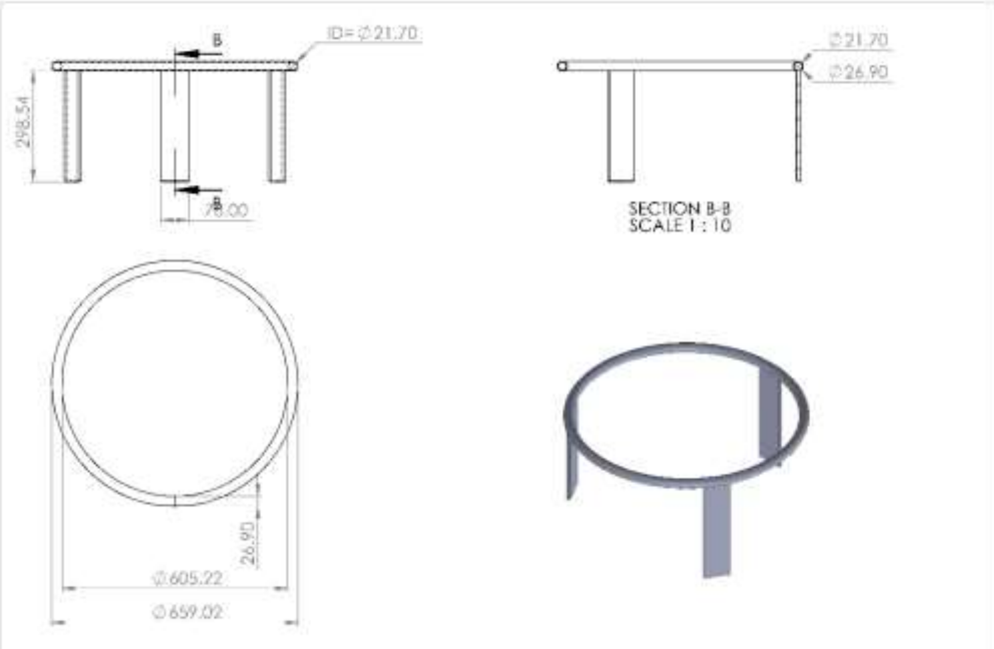
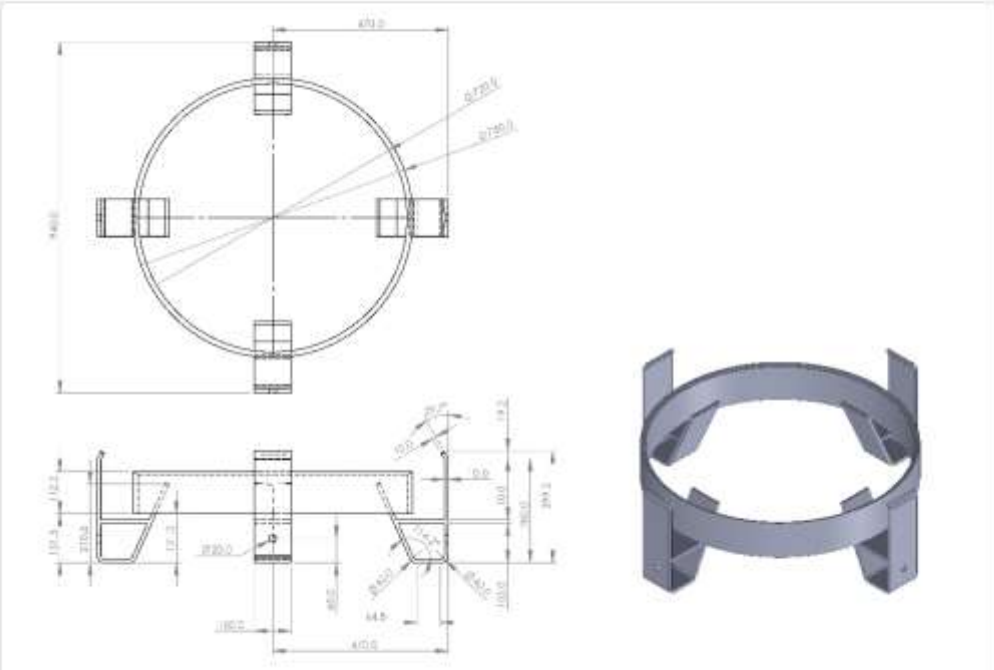
78.18
69.47
68.88
40.00
10.00
45.63
104.19
LIFTING HOOK



VALVE GUARD



LEG SUPPORT