Design and Static Structural Analysis of a Horizontal Pressure Vessel

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الملخص.

تعتبر خزانات الضغط عنصراً مهما في الصناعة اليوم، فهي تحظى باهتمام كبير من المهندسين والبحاث. إن الهدف الرئيسي من استخدام هذه الخزانات انها تستخدم لاحتواء او حفظ بها العديد من المواد مثل: السوائل والهواء والغازات والمركبات الكيميائية والوقود. في هذه الدراسة تم تصميم خزان ضغط أفقي النوع يحتوي على 10م^3 من غاز البروبان السائل المضغوط. تمت النمذجة والمحاكاة والتحليل الانشائي على هذا الخزان بإتباع معيار الجمعية الامريكية للمهندسين الميكانيكيين. خزان الضغط المعني بالدراسة هنا هو عبارة عن خزان أسطواني أفقي برأسين بيضاويان وأربع فوهات ومسندين للتثبيت. أظهرت النتائج أن التصميم كان في مأمن من الفشل وبالتالي تم قبول النتائج، وأظهرت النتائج أيضاً أن أعلى قيمة للضغط هي الفوهة الرئيسية متبوعاً بالوعاء، بينما تعرضت الفوهات وكلا الرأسين البيضاويان والمسندان لأدنى إجهاد، وكذلك أظهرت النتائج أن التشوه الكلي يتزايد كلما ابتعد الجسم عن المسندان.

Abstract.

Pressure vessels are an important part of the industrial world and therefore hold a special interest from mechanical engineers. The main objective of using the pressure vessels are used as containers to contain many of materials such as: liquids, air, gases, chemical compounds and fuel. In this study, a horizontal pressure vessel holding 10 m^3 of pressurized Liquid Propane Gas (LPG) is designed, modeled, simulated and analyzed by following the American Society of Mechanical Engineers (ASME) standard for the design

as well as using ANSYS Static Structural for the modeling and simulation. The pressure vessel in this study is cylindrical with two elliptical heads, four nozzles, a manway and two saddles. Results have shown that the design was safe from failure and therefore, results are accepted. Results show that the highest stresses are experienced by the manway followed by the shell, while the heads, nozzles and saddles experienced the lowest stresses. Results also showed that the total deformation generally increases the further the component is from the saddle.
Keywords: Pressure vessels, ASME codes, highest stresses, ANSYS.

1.0 Introduction

The use of fluids has become an essential aspect of our everyday life, whether through direct consumption or machinery. The reason for the interest in fluids is their multiple properties such as their thermal properties, flow properties, and electrical properties of their physical properties. Sometimes there is a need to compress and pressurize the fluid, especially gases before they can be used or transported, however, some of these fluids require high pressures (or lower pressures) that require special vessels to contain, these vessels are called pressure vessels. Pressure vessels considered as important prop in petroleum and chemical industries, especially as storages for oil and chemical components. [1] Engineers design pressure vessels so, these vessels are used to contain the pressurized fluid safely and ensure that it will not fail due to the forces acting on it, this is done by adhering to strict guidelines and safe design codes and standards that are approved after rigorous procedures of experiments and studies performed by experts in the field.

1.1 Pressure Vessels

Pressure vessels are containers that are designed to contain and store fluids under certain pressure that is significantly higher or lower than the surrounding ambient pressure [2]. The higher the pressure difference is, the more dangerous the operation of the pressure vessel becomes. The design of horizontal pressure vessel resisting internal pressure with dished heads is available in codes and standards [3].

Figure1. Horizontal Pressure Vessel

Therefore, pressure vessels are designed with the safety of operation being the primary focus and therefore the design process includes parameters that ensure the best possible safety. These parameters include maximum safe operating pressure and temperature, safety factor, corrosion allowance, and minimum design temperature for brittle fracture. Pressure vessels are mostly manufactured from steel, especially carbon steel. To manufacture a cylindrical or spherical pressure vessel, rolled and possibly forged parts would have to be welded together. special precautions need to be taken since some mechanical properties of steel, achieved by rolling or forging, could be adversely affected by welding. In addition to adequate mechanical strength, current standards dictate the use of steel with a high impact resistance, especially for vessels used in low temperatures. In applications where carbon steel would suffer corrosion, special corrosion resistant material should also be used. It should be mentioned that not all pressure vessels are made from steel or metal alloys, some pressure vessels are made from filament-wound composite using carbon fiber held in place with a polymer. Due to the very high tensile strength of carbon fiber these vessels can be very light but are much more difficult to manufacture. [4].

1.2 Pressure Vessel Types

Pressure vessels can be divided into many types depending on the classification criteria and these include application, geometry and orientation [5]. Each type classification will be listed in the Table 1.

Table 1: Pressure vessels types and their classification

Pressure Vessel Type	Classification			
Types by Application	Storage Tanks	Process vessels	Heat exchangers	
Types by Geometry	Vertical pressure vessels	Horizontal pressure vessels	Conical pressure vessels	Spherical pressure vessels
Types by Orientation	Available space	Manufacturing, installation and maintenance	Seismic, wind and static head loads	Stress distribution

1.3 Pressure Vessel Testing

Once the pressure vessel is designed and built, it is tested to ensure the safety of the design, these tests are conducted using nondestructive testing methods such as ultrasonic testing, pressure tests, and radiography. Pressure tests are performed to ensure the safety, reliability, and leak tightness of pressure systems. A pressure test is required for a new pressure system before use or an existing pressure system after repair or alteration. There are mainly two pressure tests, which are hydrostatic and pneumatic tests. A hydrostatic test is performed by using water as a test medium, whereas pneumatic tests use gases such as air, nitrogen or any other nonflammable and nontoxic gas. Hydrostatic testing is preferred, because it is a safer method, as much less energy is released if a fracture occurs during the test this because water does not rapidly increase its volume when rapid depressurization occurs, unlike gases like air, which fail explosively [6]. There are other important types of tests are conducted and known as non-destructive testing (NDT) or non-destructive examination (NDE). The (NDT) tests are the most commonly conducted tests [7].

2.0 Pressure Vessel Design

The design of the pressure vessel was achieved by using the (ASME section VIII). The selected code for designing the horizontal tanks (vessels), is according to the minimum requirements of design without any flaws in the vessel parts. The specialized code for the selected type (Horizontal) is those used within ranges of (0.1MPa to 20MPa) the horizontal vessels are designed according to those ranges. The designed studied pressure vessel is composed of (Heads – Shell – Manway – Nozzles – Saddles).

2.1 Pressure Vessel Specifications

The pressure vessel chosen for this study is a pressure vessel used to contain liquid propane gas (LPG). This pressure vessel has elliptical heads and is designed to be used in fixed location on saddles. The pressure vessel will have an inner shell diameter (D_i)mm and a shell length (L_s)mm. This vessel was originally designed to hold 10m^3 of LPG at a design pressure (P_d) MPa. The pressure vessel is made of carbon steel, which is the industry standard for pressure vessel design and creation.

Table 2: pressure vessel specifications

Parameter	Value
Shell inner Diameter (D_i)	1600mm
Shell length (L_s)	4500mm
Design pressure (P_d)	1.61MPa
Vessel material	SA516 Gr 70

Table3: Mechanical properties of Gr70 carbon steel

Parameter	Value
Tensile strength (σ_t)	482.63MPa
Yield strength (σ_s)	260MPa

2.2 Shell design

The design of the pressure vessel shell includes the determination of the shell's thickness also its length. The shell’s thickness is determined by two main aspects, which are the design pressure and the corrosion allowance. The corrosion allowance dictated by the ASME standards is 3mm. Before calculating the shell thickness, it is necessary to determine the total pressure inside the vessel, which is the sum of the design pressure and static head pressure. This can be expressed into the following equations:

$$P_t = P_d + P_H \qquad (2\text{-}1)$$

$$P_H = \rho g H \qquad (2\text{-}2)$$

Where:

P_t = total pressure.
P_d = design pressure.
P_H = static head pressure.
ρ = contained fluid's density, for LPG is $1061 \frac{kg}{m^3}$
g = gravitational acceleration, which is $9.81 \frac{m}{s^2}$
H = Fluid's height.

The obtained total pressure inside the vessel by using those equations is 1.63MPa. the minimum shell thickness is determined by longitudinal section (t_a) or the shell thickness along the circumferential section (t_b), which are expressed in the following equations:

$$t_a = \frac{P_t R_i}{2SE + 0.4P_t} \qquad (2\text{-}3)$$

$$t_b = \frac{P_t R_i}{SE - 0.6P_t} \qquad (2\text{-}4)$$

Where:

R_i = shell inner radius.
E = the join efficiency which in this study is set to 1
S = Maximum allowable stress, where

$$S = \frac{\sigma_t}{s_f} \qquad (2\text{-}5)$$

σ_t = tensile strength of shell material approximately 482.63MPa.
S_f = safety factor which is set 4.0 in this study, which is ranging from 3.5 – 6.0 for this type of pressure vessels.

the maximum allowable pressure achieved by this thickness is calculated to ensure that the pressure vessel can contain the contained fluid at the design pressure. Equation (2-6) and (2-7) are used to calculate the maximum pressure at the longitudinal and circumferential sections respectively.

$$P_a = \frac{2S \times E \times t_f}{R_i - 0.4t_f} \qquad (2\text{-}6)$$

$$P_b = \frac{S \times E \times t_f}{R_i + 0.6t_f} \qquad (2\text{-}7)$$

Using both these equations results in a longitudinal maximum pressure of 425.28MPa and a circumferential maximum pressure of 208.96MPa, which are both much higher than the total pressure. This shows that the chosen shell thickness is safe and will be adopted in this study.

Table 4: Shell design parameters.

Parameter	Value
Total pressure (P_t)	**1.63MPa**
Safety factor (sf)	**4**
Joint efficiency (E)	**1**
Maximum allowable stress (s)	**120.66MPa**
Corrosion allowance (c.a.)	**3mm**
Final shell thickness (t_f)	**14mm**
Longitudinal maximum pressure (P_a)	**425.28MPa**
Circumferential maximum pressure (P_b)	**208.96MPa**

2.3 Heads design

For the design of the pressure vessel's head, the same material and allowable stress will be used. The pressure vessel used in this study has elliptical heads. The head's inner crown height (h_i) and its thickness are calculated according the following equation:

$$h_i = \frac{D_i}{4} \qquad (2\text{-}8)$$

This results in an inside crown height of 0.4m. As for the head's minimum thickness ($t_{h,min}$), equation (2-9) is used:

$$t_{h,min} = \frac{P_t \times D_i \times k}{2S \times E_L - 0.2P_t} \qquad (2\text{-}9)$$

Where E_L = the head longitude efficiency, which is ≤ 1 in this study.
K = factor which is calculated using the following equation:

$$k = 0.167\left(2 + \left(\frac{D_i}{2h_i}\right)^2\right) \qquad (2\text{-}10)$$

To calculate the maximum pressure (P_{max}) allowed by this thickness, the following equation is used:

$$P_{max} = \frac{2S \times E_l \times t_h}{k \times D_i + 0.2t_h} \quad (2\text{-}11)$$

This results in a maximum allowable pressure of 1.65MPa, which is higher than the total pressure acting on the head.

Table 5: Head design parameters

Parameter	Value
Inside crown height (h_i)	**400mm**
Head longitude efficiency (E_L)	**1**
K-factor (k)	**1.002**
Head thickness (t_h)	**14mm**

2.4 Manway design

The purpose of a manway is to provide an entry to workers for easy access to vessels or storage tanks for the cleaning and maintenance purposes. [8] The ASME standard states that all pressure vessels with diameters exceeding 450mm must have a manway but it does not provide a minimum standard size. The Canadian standard association (CSA B51) has set the minimum external diameter ($D_{o,m}$) to 16in (406.4mm) [9] , the minimum required thickness based on the pressure at the longitudinal and circumferential sections are given in the following equations:

$$t_{a,m} = \frac{P \times R_{im}}{2S \times E + 0.4 \times P_t} + c.a. \quad (2\text{-}12)$$

$$t_{b,m} = \frac{P \times R_{im}}{S \times E - 0.6 \times P_t} + c.a. \quad (2\text{-}13)$$

Where:

$t_{a,m}$ = minimum manway thickness.

$t_{b,m}$ = circumferential sections

R_{im} = internal radius of the manway, where

$$R_{im} = \frac{D_{om}}{2} - t_{nm} + c.a. \quad (2\text{-}14)$$

Using equations (2-12) and (2-14) results in minimum thicknesses of approximately 5mm and 7mm respectively, which are less than the nominal wall thickness used in this study, therefore, the current design is reliable and trusted, the table 6 is used to present all manway design parameters, the minimum area required (A_m) is calculated by using the following equation:

$$A_m = D_{im} t_{a,m} f_r + 2 t_{a,m} (t_{n,m} - c.a.)(1 - f_r) \qquad (2\text{-}15)$$

Where D_{im}= the internal diameter of the manway calculated using the following equation:

$$D_{i,m} = D_{o,m} - 2(t_{n,m} - c.a.) \qquad (2\text{-}16)$$

f_r = the ratio of allowable stress, which in this study ≤ 1.

Using these equations results in a minimum area of 8351.56mm^2 and since the manway flange covers more than this area, now the design is reliable and trusted.

Table 6: Manway design parameters

Parameter	Value
Final manway external diameter ($D_{o,m}$)	**609.6mm**
Final manway thickness ($t_{n,m}$)	**9.53mm**
Ratio of allowable stress (f_r)	**1**
No. bolt holes	**15**

2.5 Nozzles design

There are total of four nozzles used in this pressure vessel layout, each of these nozzles serves a different purpose. the design procedures and the design parameters mentioned in Table 7, also the figure 2 illustrates the nozzles positions on the vessel.

Table 7: Nozzle design parameters

	Outer diameter (D_o) (mm)	Thickness (t_f) (mm)	Minimum area (A) (mm^2)
Charge inlet	**60.33mm**	**3.91mm**	**819.1mm^2**
Outlet	**88.9mm**	**5.49mm**	**1175mm^2**
Drain	**60.33mm**	**3.91mm**	**819.1mm^2**
Pressure transmitter	**60.33mm**	**3.91mm**	**819.1mm^2**

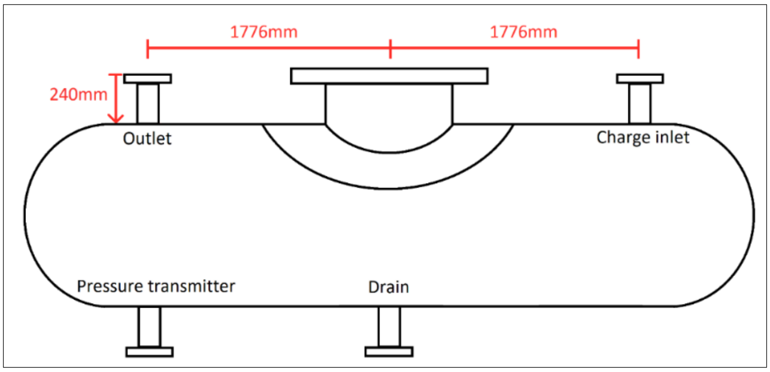

Figure 2: Nozzle positions

2.6 Saddle

The saddle is an important component of a horizontal pressure vessel since it provides a stable base to the vessel as well as spreading the weight of the vessel in a uniform manner. All the dimensions required for this study are shown in Table 8, meanwhile all criteria designing standards are shown in Figure 3.

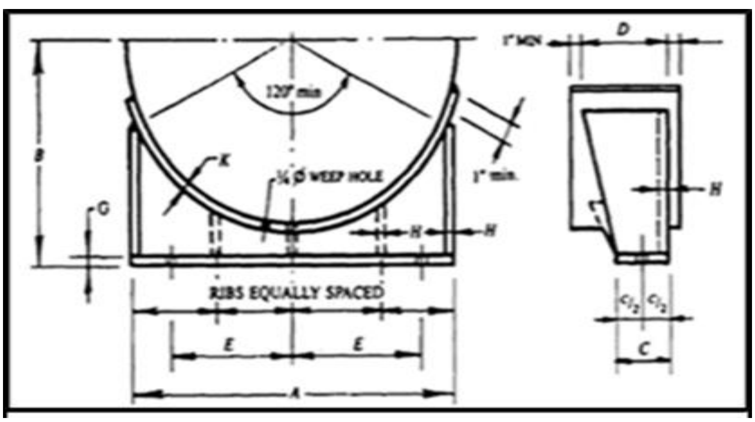

Figure 3. Main saddle dimensions and design criteria

The design of saddles is according to the ASME standards, the minimum contact angle between the shell and the saddle needs to 120° and should be supported by two saddles.

Table8: Values of saddle's dimensions

Dimension	Value
A	1460.44mm
B	1066.76mm
C	152.40mm
D	279.40mm
E	558.79mm
G	19.05mm
H	9.53mm
K	9.53mm

Equations (2-17) and (2-18) are used to calculate the weight of the vessel and contained fluid respectively:

$$m_p = V_p \rho_p \qquad (2\text{-}17)$$

$$m_f = 0.8 V_f \rho_f \qquad (2\text{-}18)$$

Where:

m_p & m_f = mass of the pressure vessel and contained fluid.
V_p & V_f = volume of the vessel and contained fluid.
ρ_p & ρ_f = density of the vessel and contained fluid.

the fluid should not exceed 80% of the maximum capacity to avoid extra stresses caused by this expansion, and because the density of LGP is $493\frac{kg}{m^3}$., Then by using the equation (2-18) with a tank capacity of $10.38m^3$, the mass of the fluid is 5115.86kg. The total volume of the vessel is $0.43m^3$ and its density is the same as steel, which is $7850\frac{kg}{m^3}$, this makes the total mass of the tank 3347.08kg, Therefore, the total mass of the vessel with the contained fluid is 8462.94kg.

it is necessary to calculate horizontal force caused by this weight, as expressed in the following equation:

$$F = K_{11} Q \qquad (2\text{-}19)$$

Where:

F = horizontal force.
K_{11} = constant based on the contact angle, which is 0.204 in this study.

Q = load on one saddle, which given by:

$$Q = gm_{saddle} \qquad (2\text{-}20)$$

Where g is the earth's gravitational acceleration is $9.81\frac{m}{s^2}$.

To calculate the stress caused by this force (σ_{saddle}), equation (2-21) is used:

$$\sigma_{saddle} = \frac{3F}{HR_i} \qquad (2\text{-}21)$$

Where H is the web plate thickness.

3.0 ANSYS Static Structural

ANSYS Static Structural is used for pressure vessels analysis, it is used to analyze the stresses and deformation caused by the inner pressure as well as the weight of the vessel and contained fluid [10]. Three-dimensional modeling is achieved by SolidWorks software. The mathematical model, which includes the boundary conditions, equations used to calculate total deformation and the equivalent stress and the numerical analysis method used as shown in the figure4.

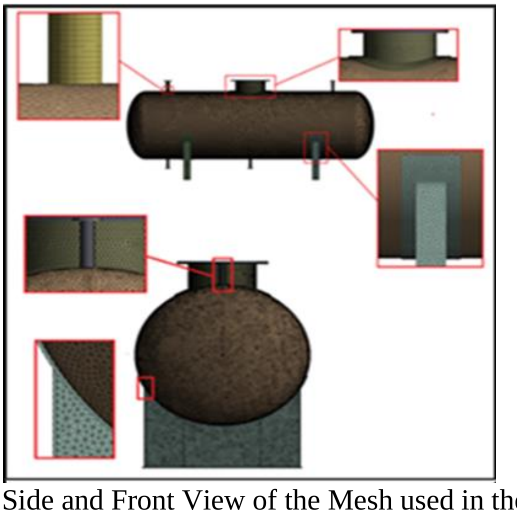

Figure 4. Side and Front View of the Mesh used in the Model

For this study, a mesh with 1176522 nodes and 596164 elements was used. This mesh was chosen based on overall mesh quality.

Results show that the cell aspect ratio was very low, with an aspect ratio not exceeding 1.16 for the vast majority of the elements i.e. 99.7% of the cells. All three mesh quality criteria were accepted, the mesh was chosen and adopted.
The boundary conditions set in this study include a constant pressure induced by the contained fluid. The pressure is set to 1.63MPa, which is the total pressure the vessel is subjected to as shown in the figure5.

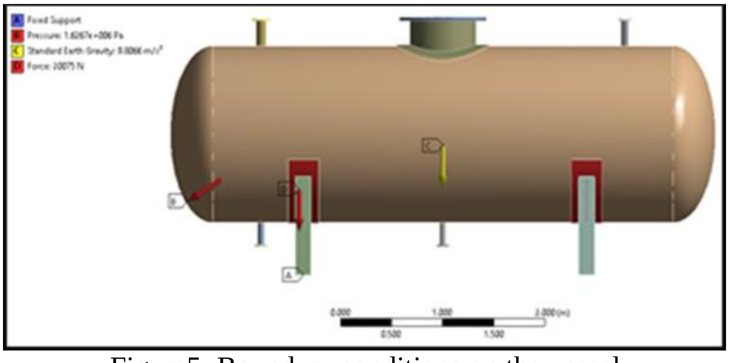

Figure5: Boundary conditions on the vessel

The highest equivalent stress experienced by the pressure vessel is 36.18% as shown in the figure 6 is lower than the maximum tensile strength of the material, which is 483MPa.

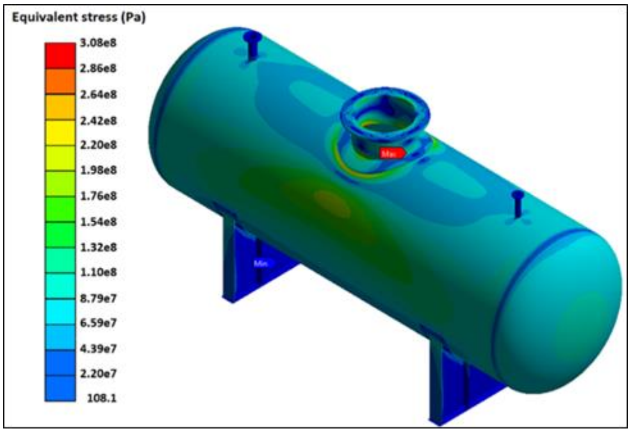

Figure 6: Equivalent stress distribution across the entire pressure vessel.

The highest stress on the saddle reached was 172MPa, which is 64.17% less than the ultimate strength of the material.

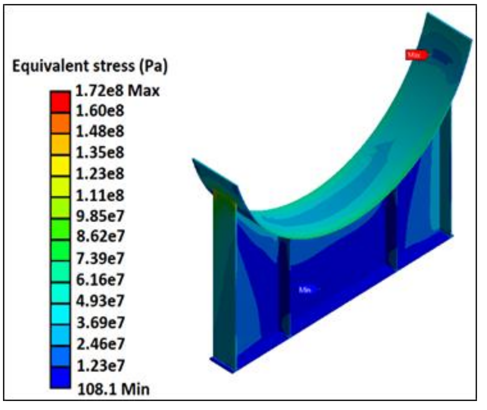

Figure 7: Stress distribution on one of the saddles.

Results on the manway show that the highest equivalent stress experienced is 244MPa, as shown in figure 8, which is 49.48% lower than the tensile strength of the material.

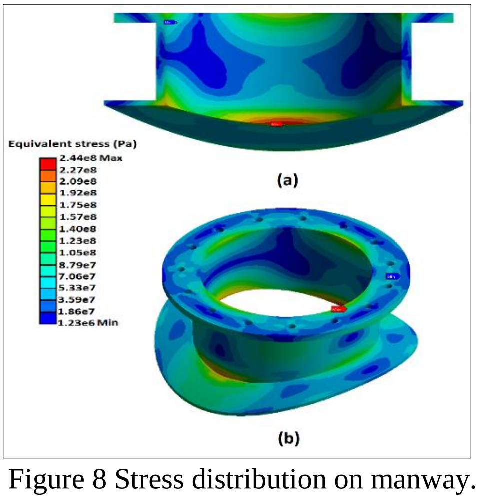

Figure 8 Stress distribution on manway.
(a) Front view of the manway and (b) the isometric view

4.0 Conclusion

During this study, a horizontal pressure vessel was designed using the ASME boiler and pressure vessel code, sec VIII standard. Once the vessel was designed based on this standard, then modeled, meshed, and simulated using ANSYS Static Structural, in-order to test the integrity of the design as well as study the stress and deformation distribution across the pressure vessel. Resulted stresses were found significantly lower than the ultimate strength of the material which was 483 MPa. Meanwhile, the highest stresses were higher than the design stress. These stresses were located at the contact points between the manway and the shell. However, the lowest stresses were located at the saddle which was 244 MPa. The highest deformation occurs at the manway and midpoints of the three sides of the shell that are not in contact with the saddle, while the saddle was experienced with no occurrence of any deformation.

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