Journal of Applied Science and Advanced Technology
Journal Homepage : https://jurnal.umj.ac.id/index.php/JASAT

Analysis of Pressure Vessel Design on Radiator Cooling System Using Low Carbon Steel Material

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ARTICLE INFO

JASAT use only:
Received date : 21 November 2020
Revised date : 25 January 2021
Accepted date : 15 February 2021

Keywords:
Pressure vessels
Radiator cooling system
Hydrostatic test
Working pressure
Maximum allowable working pressure

ABSTRACT

This study discusses the analysis of a pressure vessel's design in the Radiator Cooling 1000 (RC-1000) system, which operates at a design temperature of 100oC. A pressure vessel is a container of gaseous, solid, or liquid material subjected to internal or external pressure and can withstand various other load variations. The pressure vessel on the RC-1000 system has a diameter of 85.4 mm or 3.36 inches and will experience an internal pressure of about 143.7 kPa or 20.84 psi, so it must be designed safely. This research method uses analytical and experimental methods. The analytical method is used to calculate the thickness of the pressure vessel material, the maximum allowable working pressure, and the hydrostatic test calculation. While the experimental method was carried out on the hydrostatic test process, the evaluation was based on the prevailing regulations in the Republic of Indonesia. Using the SPCC-SD material (JIS 3141), it was found that the minimum thickness of this pressure vessel is 1.15 mm or 0.0452 inches on the shell side and 1.10 mm or 0.0434 inches on the head/dome side. The thickness of the material used on the shell side and head/dome is 1.2 mm or 0.0472 inches in practice. This pressure vessel has passed the hydrostatic test at 1600 kPa or 232.1 psi. The test pressure is given around seven psi higher because it makes it easier to read the scale on the pressure gauge.

INTRODUCTION

This paper discusses the calculation of pressure vessel design using steel material SPCC-SD (JIS G 3141). SPCC-SD material is low carbon steel because the carbon content is below 0.3% [1][2]. Research on the analysis of the pressure vessel design on the radiator cooling system is a part of the research with the big theme "Energy and Exergy Analysis of Motorcycle Radiators at High Working Temperature Using Ethylene Glycol" which uses a pressure vessel with a total capacity of 2.3 litres. This pressure vessel input design uses a maximum working temperature of 110 ^{o}C of water. According to the water properties table, the pressure vessel's pressure is around 143.7 kPa or 20.8 psi or about 0.144 bar [3]. Regulation of the Minister of Manpower of the Republic of Indonesia Number 37 of 2016 regulates the occupational safety and health of pressure vessels and storage tanks. The criteria in Article 5 paragraph (2) state the criteria for pressure vessels in two conditions, namely a pressure vessel which has a pressure of more than 1 kg/cm^2 (or about 98.07 kPa) and a volume of more than 2.25 litres [4]. The pressure vessel is also defined as a designed and built the place to accommodate pressurized gas or liquid (fluid) [5]. Based on these criteria, the pressure vessel to be used has met the required criteria to be analyzed to meet the required safety standards.

Pressure vessels are widely used in the commercial and aerospace industries such as fuel tanks, portable oxygen storage, and compressed natural gas (CNG) pressure vessels for vehicles. This vessel usually consists of two separate parts: a cylinder section and two heads (domes or caps). The head is usually the most critical part of pressure vessel design. The desired parameters for a good head shape are to have a higher burst pressure and internal volume and a lower weight [6].

Several studies related to pressure vessels, among others, were carried out by Hardy (1990) by optimizing composite reinforcing pressure vessels.

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DOI: https://dx.doi.org/10.24853/JASAT.3.3.81-90

Optimization was carried out by modelling the finite element method using E-Glass Epoxy + Steel (EGS), Kevlar-49 Epoxy + Titanium (KET) T300-5208 Graph. Epoxy + Aluminum (GRA). The research was aimed at obtaining the optimum geometric shape and material. The results confirm that the KET material geometry has the highest pressure, followed by EG and GRA. For efficiency, the KET material also has the highest pressure vessel efficiency than the other two materials [7].

The research was also conducted by Javadi (2013) by evaluating the residual stress of welding pressure vessels with 304L stainless steel material. Residual stress was measured using the ultrasonic method. The evaluation results show that the axial residual stress distribution is more uniform than the circular stress, which can be justified by the pressure vessel surface distortion. The ring wedge position is more complicated than the axial wedge due to the welding deformation of the surface under test. Circular residual stresses are more affected by the scan path than axial stresses. It means that the difference in the results of 90 degrees and 270 degrees according to the results of hope stress is higher than axial stress results [8]. Ibrahim (2015) conducted a stress analysis on a pressure vessel with a thin wall. The soda can is analyzed as a thin wall pressure vessel. The soda can material uses aluminium material. In a thin wall pressure vessel, there are two stresses: longitudinal stress and circular stress. Longitudinal stress results from internal pressure acting on the end of the cylinder and the long stretch of the cylinder.

The circular stress results from the radical action of the internal stresses that tend to increase around the cans. The increased pressure in soda can be determined by measuring the elastic stretch of the soda's surface. The internal pressure for pressurized soda can be lowered using Hooke's fundamental law of stress and the strained relationship, which relates changes in the circle and axial strain to internal stresses. Two strain measurements (Measurements Group-CEA series gages) are attached to a soda can to measure the change in strain measured by the voltage across a calibrated Wheatstone bridge. M-bond 200 (Measurements Group, Inc) adhesive is used to glue the strain gauge to the surface of the soda can. The elastic strain of the soda's outer surface can be determined by a strain gauge mounted on the surface of the can and connected to a strain indicator. Longitudinal stresses, circular stresses, and internal stresses are determined from the general Hooke's law equations for stress and strain. Small variations were noted in the internal stresses calculated from the longitudinal strain and the circular strain [9].

Kim (2019) conducted an investigation of CNG pressure vessels with defects caused by heat treatment using numerical analysis methods. Under investigation, the pressure vessels had volumes of 122 L, 128 L 154 L, 164 L, and 188 L. The investigation results showed that the fatigue life of the containers with poor heat treatment was significantly reduced compared to regular containers; Hence, the vessel broke before reaching the specified lifetime. It has been shown that reduced fatigue life is a major cause of pressure vessel rupture accidents [10].

In contrast to previous studies, this research was conducted to produce a design that meets safety standards when used. This study's input design is a working pressure of 143.7 kPa and a volume of 2.35 L. This study uses a low carbon steel material (SPCC-SD). The research method used analytical and experimental methods. The analytical method is used to calculate the thickness of the material. At the same time, the experimental method is used to validate the pressure vessel design. Design validation is carried out by conducting a hydrostatic test concerning the applicable regulations in Indonesia.

Material Thickness and Maximum Pressure Calculation

The top and bottom dome are the cover parts on the upper and lower sides of the pressure vessel ends, respectively. The geometric shape of the top and bottom dome is ellipsoidal. The top and bottom dome also have a nipple that will be connected to the radiator. The top nozzle serves as an outlet for hot fluid to the radiator for the cooling process. The radiator's cooling process occurs by forced convection, and then it will be pumped back into the pressure vessel. In the pressure vessel, the fluid is heated to a maximum design temperature of 100^{o}C. In this condition, the top and bottom dome material are assumed to experience a maximum heat treatment of 100^{o}C. In this condition, the pressure in the pressure vessel is around 143.7 kPa or around 20.85 psi. This pressure is an operational condition (working pressure), whereas the amount of design pressure (design pressure) is to add 30 psi or 10% of the working pressure [11] [12]. In this study, a design pressure of 10% was used so that the design pressure value was 22.93 psi [11].

Using the inner diameter of Di and the type of ellipsoidal dome as a reference for calculation, the top/bottom dome material's thickness can be calculated using equation 1[11] [12].

$$t = \frac{PD}{2SE - 0.2P} \quad (1)$$

f using the outer dimensions as a reference, the top and bottom dome thickness calculations are calculated using equation 2 [11][12].

$$t = \frac{PD}{2SE + 1.8P} \quad (2)$$

While the permissible pressure on the top and bottom dome is calculated using equation 3 [11][12].

$$P = \frac{2SEt}{D + 0.2t} \quad (3)$$

If using the outer diameter as a reference for calculation, the allowable pressure on the top and bottom dome is calculated using equation 4[11][12].

$$P = \frac{2SEt}{D - 1.8t} \quad (4)$$

With the same input parameters, the material thickness calculation on the shell side is performed using equation 5 [11][12].

$$t = \frac{PR}{SE - 0.6P} \quad (5)$$

If using the outer diameter as the input parameter, the calculation of the thickness of the material on the shell side is carried out with equation 6 [11][12].

$$t = \frac{PR}{2SE - 0.4P} \quad (6)$$

While the allowable pressure on the shell is calculated using equation 7 [11][12].

$$P = \frac{SEt}{R + 0.6t} \quad (7)$$

If using the outer diameter as a reference, then the calculation of the allowable pressure on the shell is calculated using equation 8 [11][12].

$$P = \frac{SEt}{R - 0.4t} \quad (8)$$

P is design pressure or maximum allowable working pressure (MAWP) in psi, S is stress value material in psi, E is joint efficiency, R is inside or outside radius in inches, and t is the wall thickness, and CA is corrosion allowance in inches [11].

Stress Calculation on Shell

Because of the cylinder's geometry, there is uniformity in the internal or external stresses, which indicates that the pressure in the longitudinal seam is twice the circumferential seam. The amount of compressive stress on the longitudinal side of the joint due to external and internal pressure is calculated using equation 9.

$$S_l = \frac{PD}{2t} \quad (9)$$

Meanwhile, the amount of compressive stress on the circumferential joint due to external pressure and internal pressure is calculated using equation 10.

$$S_c = \frac{PD}{4t} \quad (10)$$

P is the design pressure or maximum allowable working pressure (MAWP) in psi, *S* is the material stress value in psi, *D* is the diameter in inches, and t is the wall thickness in inches [11].

Hydrostatic Test

A hydrostatic test is carried out to validate the design to meet safety regulators. Hydrostatic testing is a destructive test method. According to the ASME standard section VIII-1, UG-99, pressure vessels designed for internal pressure must be subjected to a hydrostatic test pressure which at each point in the vessel is at least equal to 1.3 times the MAWP times the ratio of the stress value S for the test temperature to the voltage value S at design temperature. The hydrostatic pressure is calculated using equation 11[11] [12].

$$P_h = 1.3.MAWP x \frac{S_t}{S_d} \quad (11)$$

MAWP is the maximum allowable working pressure in psi. S_t is the material stress value at the test temperature in psi and S_d is the material design stress value in psi. [11].

Table 1. Dimension of pressure vessel capacity 2.35 L

Position	Standard	Value	Position	Standard	Value
a	33.3 ± 0.4	33.4	e	16.2± 0.4	16.3
b	129.0 ± 0.4	129.2	f	16.2± 0.4	16.3
c	33.3± 0.4	33.4	g	Φ 85.4± 0.4	85.5
d	227.0± 0.4	127.2	h	R-8.6 ± 0.4	R-9.0

All dimension in mm.

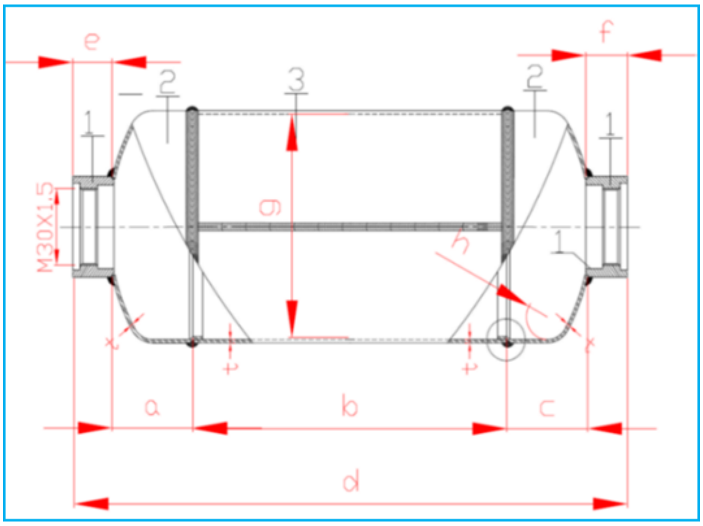


Fig. 1. Pressure vessel geometry for RC-1000 system

EXPERIMENTAL METHOD

Pressure vessel and RC-1000 System

RC-1000 is a Radiator Cooling System, a radiator cooling system that uses a heater to heat a mixture of water (water heater) and ethylene glycol. Heater installed in the heating tank using a band heater with a capacity of 1000 W in the heating tank. The heater tank has a volume capacity of 2.35 L with a maximum temperature set at 110ºC. According to the regulation of the Minister of Manpower of the Republic of Indonesia Number 37 of 2016 concerning occupational safety and health of pressure vessels and storage tanks, the heater tank is an integral part of it is the definition and pressure vessel criteria established. The pressure vessel in the RC-1000 system is shown in Figure 1.

The input design for the pressure vessel is volume and working pressure. Volume and pressure determine the optimum material thickness from the pressure vessel to meet safety standards. The pressure vessel diameter is one of the parameters that must be taken into account because it relates to the volume and thickness of the material. By using a certain thickness of the material, the diameter is calculated to meet the specified requirements. The design pressure input is calculated by understanding the physical properties of the water. With a maximum pressure of 110ºC, it is known that the pressure is about 143.7 kPa. A design drawing based on the pressure vessel input design is shown in Figure 2. While the dimensions/sizes of the pressure vessel are presented in Table 1.

The pressure vessel consists of 4 main parts: the neck-ring, top dome, shell, and bottom dome [13]. The neck ring is made of S-45 C material, while the top dome, shell and bottom dome will be designed using SPCC-SD material. Neck-ring serves to install the adapter and connect the hose to the RC-1000 system. The top and bottom dome geometric shapes are made ellipsoidal.

Pressure Vessel Material

The material used for pressure vessel designed is SPCC-SD steel which refers to the JIS 3141 standard. SPCC-SD material is equivalent to SA-1008 material.The chemical composition of SPCC-SD is presented in Table 2 [14] [15].

Table 2. Chemical composition of SPCC-SD material (%)

Element	Values (%)
C	0.15 max.
Mn	0.05 max.
P	0.04 max.
S	0.04 max.

While the mechanical properties of the SPCC-SD material are presented in Table 3[14] [15].

Table 3. Mechanical properties SPCC-SD steel sheet

Descriptions	Value
YP (MPa/psi)-max.	240/ 34809
TS (MPa/Psi)-min	270/ 39160.2
EL (%)-min	37

Some carbon steel sheet material in the ASME Section II part D standard and have tensile strength values close to SPCC-SD (JIS G3141), including SA-1008 grad CS-A, SA 1008 grade SC-B, SA-283 grade A, SA-285 grade A, SA-414 grade A and SA-414 grade B. The tensile strength and maximum allowable stress of the six meters above are shown in Table 4.

Table 4. ASME low carbon steel sheet material [16]

Type	Grade	Tensile strength (Mpa/psi)	Maximum allowable stress Mpa/psi*
SA-1008	CS-A	275 / 39885.4	78.6 / 11399.97
SA-1008	CS-B	275 / 39885.4	78.6 / 11399.97
SA-283	A	310 / 44961.7	88.9 / 12893.85
SA-285	A	310 / 44961.7	88.9 / 12893.85
SA-285	B	345 / 50038.0	88.9 / 12893.85
SA-414	A	310 / 44961.7	88.9 / 12893.85
SA-414	B	345 / 50038.0	98.6 / 14300.72

Pressure Vessel Input Design

The discussion section states that the working pressure stated in this study is 20.85 psi. The design pressure used as the basis for calculation is to add 10% of the working pressure so that the value is 22.92 psi. The pressure vessel design data to be used is listed in Table 4.

Table 5. Joint efficiency on pressure vessel welding

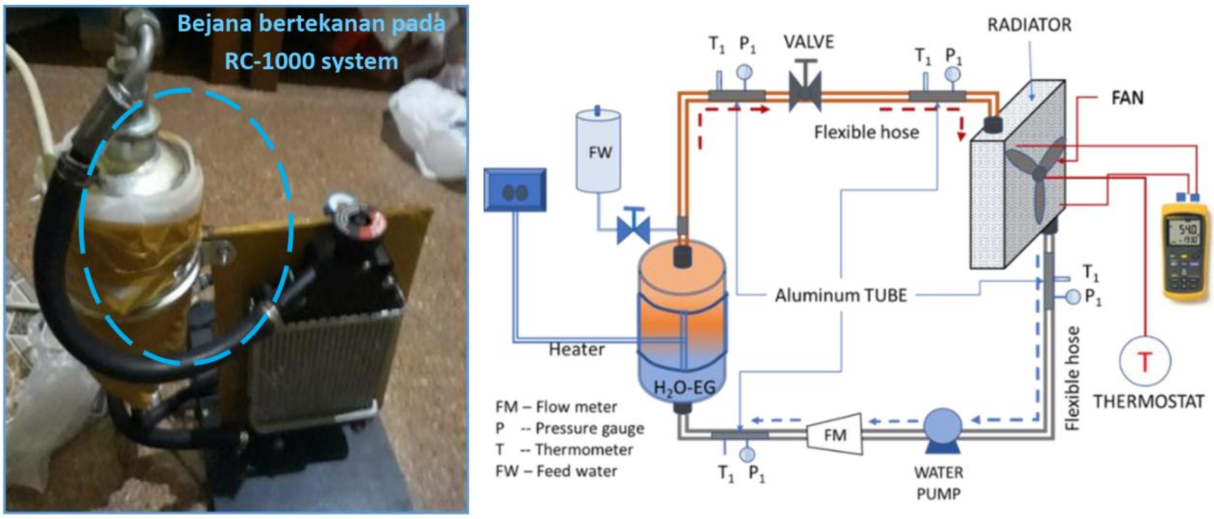
Welding Type (UW-11)	Joint efficiency, E		
	Full radiographed	Spot Examined	Non
Type 1	1.0	0.85	0.75
Type 2	0.90	0.80	0.65

Referring to the welding type code UW-11, welding on the longitudinal shell is type 1, namely butt joint, while welding between the shell and the top and bottom dome is type 2, namely, butt joint with backing plate. The joint efficiency, E in this condition is presented in Table 5 [11]. According to table 4, the joint efficiency on the shell is 0.65. The top and bottom dome are seamless, so the joint efficiency E used is 1.0. The identification of other pressure vessel design parameters listed in Table 6.

Table 6. Design input of pressure vessel

Parameter	Symbol	Value
Design pressure (MPa/psi)	P	22.92
Stress value (MPa/psi)*	S	78.6/11400
Efficiency shell/dome	E	0.65/1.0
Inside dia. (mm/ inches)	D	85.4 /3.36
Inside radius (mm/ inches)	R	42.7/1.68
Corrosion allowance [17]	$C.A$	0.04
Thickness (mm/ inches)	t	?

*) Stress value at 110ºC/230ºF.



(a) (b)

Fig. 2. (a) Pressure vessel on RC-1000 system and, (b) Schematic of RC-1000 system

RESULTS AND DISCUSSION

Analysis of the thickness material

The top and bottom dome materials thickness is calculated based on Table 4 and Table 5. The thickness analysis of the material uses SPCC-SD material, which has a minimum tensile strength specification of 10152.6 psi. Generally, the SPCC-SD material produced has a tensile strength between 270 MPa (39160.2 psi) to 345 MPa (50038.0 psi). For calculations using the inner diameter, the thickness of the shell side's material is calculated using equation one plus the corrosion allowance of 0.04 inches.

$$t = \frac{PD}{2SE - 0.2P} + CA$$

$$t = \frac{22.92\ x\ 3.36}{2x11400x0.65 - 0.2x22.92} + 0.04$$

$$= 0.0434\ inchi\ = 1.10\ mm$$

According to the availability of material, 1.2 mm or 0.0473 inches of material is used for the shell of the pressure vessel so that the maximum allowable pressure is calculated by equation 4.

$$P = \frac{2SEt}{D+0.2t} =$$

$$P = \frac{2x11400x\ 0.65\ x\ 0.0473}{D+0.2t} = 293.59\ psi$$

The data for calculating the thickness of the top and bottom dome materials with references to various materials are presented in Figure 3.

The shell thickness is calculated based on design input in Table 4 and Table 5. Material thickness analysis uses SPCC-SD material, which has a minimum tensile strength specification of 10152.6 psi. Generally, the SPCC-SD material produced has a tensile strength between 270 MPa (39160.2 psi) to 345 MPa (50038.0 psi). For calculations using the inner diameter, the shell side's material thickness is calculated using Equation 5 and added with the corrosion allowance, CA = 0.04 inch.

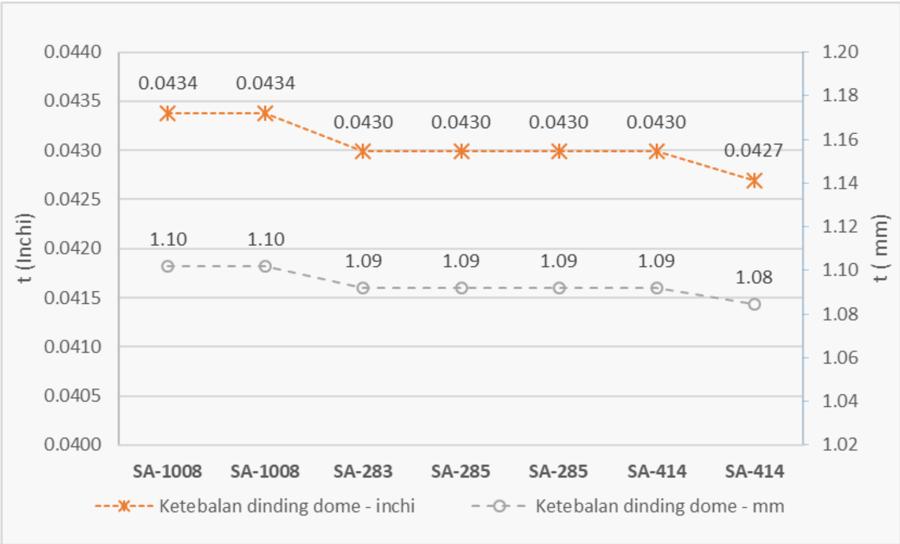


Fig. 3. Thickness material for top dan bottom dome of pressure vessel

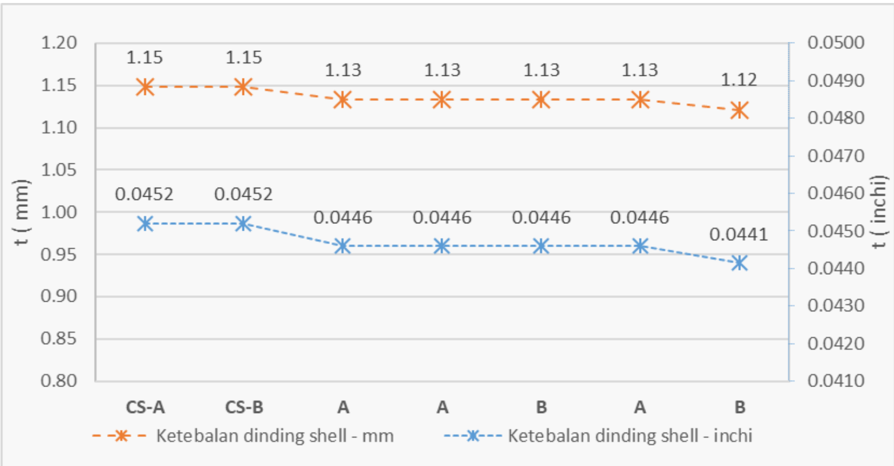


Fig. 4. Thickness material for shell side of pressure vessel

From the calculation of the maximum allowable working pressure (MAWP) on the shell and dome sides, it can be seen that the maximum allowable working pressure on the shell side is lower than on the dome side. Seeing this condition, the maximum allowable working pressure used in the calculation of the hydrostatic test is the MAWP value on the shell side, which is 196.23 psi or 1352.96 kPa.

Analysis of Stress on Shell Side

Referring to Figure 4, the SPCC-SD material used to make the dome and shell on the pressure vessel uses 1.2 mm meter with the specifications presented in Table 7.

Table 7. Mechanical properties of SPCC-SD steel sheet

Description	Standard	CSW0520B
YP (MPa/psi)-max	240/ 34809	146 / 21175.5
TS (MPa/Psi)-min	270/ 39160.2	283 / 41045.7
EL (%)-min	37	53

Because the maximum allowable stress on carbon steel with a tensile strength of 283 MPa does not exist, the SPCC-SD material is calculated by interpolating the SA-1008 grade CS-B material data SA-283 grade A. Interpolation is carried out using the following matrix:

Type	TS (psi)	MAS psi*
SA-1008	39885.4	11399.97
SPCC-SD	41045.7	x
SA-283	44961.7	12893.85

Where, X is the maximum allowable stress of the SPCC-SD material used to design the pressure vessel, whose value is calculated using the interpolation method. The value of x is calculated as follows:

$$\frac{x - 11399.97}{12893.85 - 11399.97} = \frac{41045.7 - 39885.4}{44961.7 - 39885.4}$$

$$\frac{x - 11399.97}{1493.88} = \frac{1160.3}{5076.3} = 0.23$$

$$x - 11399.97 = 0.23\,x\,341.46 = 78.0$$

$$x = 11399.97 + 78.05 = \mathbf{11478.02\ psi}$$

The value of x = 11478.02 psi and a material thickness of 1.2 mm were used to calculate the maximum compressive stress allowed to occur on the joint's longitudinal and circumferential sides. The maximum allowable compressive stress on the longitudinal side S_l , is calculated by equation 9:

$$S_l = \frac{PD}{2t} = \frac{11478.02\ x\ 3.36}{2x\ 1.2} = 16069.23\ psi$$

While the compressive stress on the circumferential joint, S_c , is calculated by equation 10:

$$S_c = \frac{PD}{4t} = \frac{11478.02\ x\ 3.36}{4x\ 1.2} = 8034.61\ psi$$

From the calculation of the compressive stress value on the longitudina, S_l and circumferential sides, S_c joint shows that the value of $S_l = 2S_c$.

Analysis Hydrostatic Test

The hydrostatic test aims to evaluate the design to meet safety requirements. The pressure used for hydrostatic is calculated using Equation 10.

$$P_h = 1.3.\,MAWPx\ \frac{S_t}{S_d}$$

From the previous data, it was found that the MAWP obtained was 196.23 psi or 1352.96 kPa. Since the maximum allowable stress at temperatures of -30 to 150 ^{o}C is the same, the ratio $\frac{S_t}{S_d} = 1$. In this case, the value of $S_t = S_d = S_c = 16069.23\ psi$, so that the amount of hydrostatic pressure to be used is:

$$P_h = 1.3\ x\ 196.23\ psi\ x\ \frac{16069.23\ psi}{16069.23\ psi}$$

$$= 225.01\ psi = 1551.4\ kPa$$

The hydrostatic test is carried out at a pressure of 1600 kPa and hold for 60 seconds. Another method used to detect leaks is multi-sensor information fusion[18]. The purpose of holding for 40 seconds is to detect a leak, indicating a pressure drop. The hydro test evaluation is carried out with the criteria; the pressure vessel must not sweat or leak; there is no permanent expansion more than 0.2% (zero points two percent) of the original volume. Furthermore, permanent expansion is calculated by the following equation [4].

$$\%\Delta v = \left(\frac{v_t - v_0}{v_0}\right) x100\%$$

Where Δv is the change in a fixed volume, v_t is the volume after testing and v_o the pre-test volume whose value is 2.3 liters. The hydrostatic test was carried out using a pressure gauge with a capacity of 6000 kPa and shown in Figure 5. Before and after hydrostatic testing, volume measurements were taken to evaluate changes in volume. The results of hydrostatic testing at a pressure of 1600 kPa obtained the value $v_t = 2{,}3\ liters$.

$$\%\Delta v = \left(\frac{v_t - v_0}{v_0}\right) x100\%$$

$$\%\Delta v = \left(\frac{2.3 - 2.3}{2.3}\right) x100\% = \left(\frac{0}{2.3}\right) x100\% = 0\%$$

Fig. 5. Pressure gauge 6000 kPa in capacity.

Regarding the Regulation of the Minister of Manpower of the Republic of Indonesia, Number 37 of 2016 regulates the occupational safety and health of pressure vessels and storage tanks. In Indonesia in 2016, the hydrostatic test is considered to have passed because the deformation remains after testing is 0%.

CONCLUSION

The design of the pressure vessel on the RC-1000 using a working pressure input design of 143.7 kPa or 20.84 psi with a working temperature of 110^o C, a shell diameter of 85.4 mm or 3.36 inches has fulfilled the operational safety requirements required in the Regulation of the Minister of Manpower of the Republic of Indonesia Number 37 of 2016 concerning safety and health at work of pressure vessels and storage tanks. Indonesia in 2016. This pressure vessel uses SPCC-SD (JIS 3141) material with a minimum tensile strength of 270 MPa and a minimum thickness of 1.15 mm. The research will continue on the performance (energy and exergy analysis) of the Radiator Cooling 1000 (RC-1000) System.

ACKNOWLEDGMENT

Thanks to “LPPM Universitas Buana Perjuangan Karawang” who has fully founded this research and publication process.

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