KAUNAS UNIVERSITY OF TECHNOLOGY

MECHANICAL ENGINEERING AND DESIGN FACULTY

Edgaras Gubskis

DESIGN AND MODELLING OF VESSEL WITH HALF-PIPE COIL

Master's Degree Final Project

Supervisor
Assoc. prof. dr. Povilas Krasauskas

KAUNAS, 2016

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Industrial Engineering and Management (code M5106M21)

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FACULTY OF MECHANICAL ENGINEERING AND DESIGN

INDUSTRIAL ENGINEERING AND MANAGEMENT (M5106M21)

DESIGN AND MODELLING OF VESSEL WITH HALF-PIPE COIL

DECLERATION OF ACADEMIC INTEGRITY

20 MAY 2016

I confirm that the final project of mine, **Edgaras Gubskis**, on the subject “Design and modelling of vessel with half-pipe coil” is written completely by myself; all the provided data and research results are correct and have been obtained honestly. None of the parts of this thesis have been plagiarized from any printed, internet-based or otherwise recorded source. All direct and indirect quotations from external resources are indicated in the list of references. No monetary funds (unless required by low) have been paid to anyone for any contribution to this thesis.

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____
(Signature, date)

Kazimieras Juzėnas
(Name, Surname)

MASTER STUDIES FINAL PROJECT TASK ASSIGNMENT
Study program INDUSTRIAL ENGINEERING AND MANAGEMENT

The final project of Master studies to gain the master qualification degree, is research or applied type project, for completion and defence of which 30 credits are assigned. The final project of the student must demonstrate the deepened and enlarged knowledge acquired in the main studies, also gained skills to formulate and solve an actual problem having limited and (or) contradictory information, independently conduct scientific or applied analysis and properly interpret data. By completing and defending the final project Master studies student must demonstrate the creativity, ability to apply fundamental knowledge, understanding of social and commercial environment, Legal Acts and financial possibilities, show the information search skills, ability to carry out the qualified analysis, use numerical methods, applied software, common information technologies and correct language, ability to formulate proper conclusions.

1. Title of the Project

Design and modelling of vessel with half-pipe coil

2. Aim of the project

Design pressure vessel with half-pipe coil according ASME Boiler & Pressure Vessel Code, Section VIII, Division 1

3. Structure of the project

Introduction: general overview, aim of the work, objectives.
Literature analysis: related to analysis of construction and design of “jacketed” pressure vessel.
Construction and design: construction investigation for “jacketed” pressure vessel, material selection, design data and calculations.
Experimental part: standard tensile test is used to determine materials mechanical properties.
Modelling: finite element model of pressure vessel with half-pipe coil.
Testing: pressure vessel hydraulically tested to ensure construction safety.
Conclusions: overview and results.

4. Requirements and conditions

5. This task assignment is an integral part of the final project

6. Project submission deadline: 2016 May 20.

Given to the students: Edgaras Gubskis

Task Assignment received: Edgaras Gubskis

Supervisor: Assoc. prof. Dr. Povilas Krasauskas

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Edgaras Gubskis. *Design And Modelling Of Vessel With Half-pipe Coil: Master's* thesis in industrial engineering and management / supervisor assoc. prof. Povilas Krasauskas. The Faculty of Mechanical Engineering and Design, Kaunas University of Technology.

Study area and field: Production and Manufacturing Engineering, Technological Sciences

Key words: *Design and Modelling of Vessel, „Jacketed" pressure vessel, Standard tensile test, Hydraulic test.*

Kaunas, 2016. 40 p.

SUMMARY

This project is aimed to engineers who model and design vessels with applied internal or external pressure by ASME boiler and Pressure Vessel Code, section VIII, Division 1. The aim of the project is to design and verify safety of pressure vessel with half-pipe coil according mentioned standard.

Every standard or code provides its own safety requirements. The main safety requirement for pressurised vessels is based on determination of vessels components minimal thicknesses. After thicknesses are set and vessel fabricated pressure test must be applied to pressurised components to ensure safety of pressure vessel.

The main problem of research is that ASME code provides only recommendations what minimal thickness for vessel's conical bottom should be used in case if pressurised half-pipe coil is welded on it. Every engineer can follow this rule and use recommended minimal thickness but in our case we bypass this rule and reduce minimal required thickness by determining actual materials yield strength, making Finite Element model and applying pressure test. These changes of vessel design are important because it reduces materials costs. Also, production time is reduced because thinner steel sheet is better wieldable and formable.

Therefore the objectives of Master project were defined as follow:

- "Jacketed" pressure vessel analysis;
- Construction and design of vessel;
- Design calculations;
- Standard tensile test;
- Pressure test of vessel.

Edgaras Gubskis. Talpyklos Su Gyvatuku Projektavimas ir Modeliavimas *Magistro* baigiamasis projektas / vadovas doc. dr. Povilas Krasauskas; Kauno technologijos universitetas, Mechanikos inžinerijos ir dizaino fakultetas.

Studijų kryptis ir sritis: Gamybos inžinerija, Technologijos mokslai.

Reikšminiai žodžiai: *slėginio indo projektavimas ir modeliavimas, Slėginis indas su „marškiniais", Standartinis tempimo bandymas, Hidraulinis bandymas.*

Kaunas, 2016. 40 p.

SANTRAUKA

Šis tiriamasis darbas skirtas inžinieriams, kurie susiduria su slėginių indų projektavimu pagal ASME boilerių ir slėginių indų kodo taisykles. Šio projekto tikslas – pagal paminėtą kodą suprojektuoti ir patikrinti pagamintos talpyklos su privirintu gyvatuku saugumą taikant hidraulinį bandymą.

Kiekvienas standartas ar kodas turi savo nustatytus saugumo reikalavimus. Slėginių indų ar jo komponentų, minimalių sienelių storių nustatymas yra vienas iš pagrindinių slėginių indų saugumo reikalavimų.

Pagrindinė šio darbo problema yra tai, jog ASME taisyklės pateikia tik rekomendacijas, koks minimalus indo kūginio dugno sienelės storis turi būti naudojamas, jeigu jis yra paveiktas išorino slėgio. Mūsų atveju, minėtas dugnas yra veikiamas slėgio, kurį sukelia privirintas gyvatukas. Rekomendaciniai storiai yra pateikiami su nemaža atsarga, norint užtikrinti 100% talpyklos eksploatacijos saugumą. Kiekvienas inžinierius gali sekti ASME kodo pateiktomis taisyklėmis ir projektavimo metu taikyti rekomendacinius storius, bet mūsų tiriamajame darbe surasime būdą, kaip išvengti ASME kodo rekomendacijų ir sumažinti talpyklos dugno sienelės storį. Šie pakeitimai indo gamybai sumažina naudojamą medžiagų kiekį ir padidiną gamybos našumą, kadangi plonesnis metalas yra lengviau formuojamas ir suvirinamas.

Norint išspręsti paminėta problemą, buvo suformuotos sekančios užduotys:

- Slėginių indų su „marškiniais" analizė;
- Slėginių indų konstrukcija ir projektavimas;
- Projektiniai skaičiavimai;
- Standartinis tempimo bandymas;
- Indo hidraulinis bandymas;
- Ekonominiai skaičiavimai;
- Išvados ir rezultatai.

Introduction

This project is aimed to engineers who modelling and design vessels with applied internal or external pressure. First of all, what is pressure vessel should be understood. Vessels, containers, or pipelines that maintain, store, transfer or receive fluids under internal or external pressure over 0,05 barg are called pressure vessels. Pressure vessels are used in a number of industries for example, the dairy industry for cream production, the brewery industry for beer maturation.

Jacketed pressure vessels are pressure vessels with applied second pressure reservoir which is called “Jacket”. “Jackets” can be welded on entire vessel’s body in order to provide heating or cooling to the vessel contents. For example, the dairy industry using “jacketed” vessels to keep milk products at constant temperature during all year seasons.

However, the components under pressure are dangerous and during the year various fatal accidents have been recorded. Nowadays, pressure vessel manufacturing, exploitation and design are controlled by engineering authorities backed by legislation. This is why every country has their own design codes/standards of pressure vessel. Each design code/standard has their own requirements; some of them are the same and some of them different.

Following pressure vessel design codes/standards each engineer who model or design pressure vessel must evaluate parameters such as maximum temperature and safe operating pressure, safety factor, corrosion allowance and involve non-destructive testing, hydraulic testing. When the design parameters have been established, suitable materials and design code/standard are selected.

The design calculation in the various codes/standards of construction always implies factors of safety. The factors of safety are generally applied to the materials of pressure vessel, so that pressure vessel can safely operate. If the pressure vessel operating in a simple environment a safety factor is based entirely on yield strength, therefore European and American pressure vessel codes/standards sets a factor safety of 1.5 for the yield strength.

The aim of this research

- Design pressure vessel with half-pipe coil welded on conical bottom and cylinder acc. ASME Boiler & Pressure Vessel Code, Section VIII, Division 1.

Problem of research

- ASME Boiler & Pressure Vessel Code, Section VIII, Division 1 provides only recommendations what minimal thickness for vessel’s conical bottom should be used in case if half-pipe coil is welded on it.

Therefore the objectives of Master project were defined as follow:

- Analysis of "jacketed" pressure vessels;
- Pressure vessel design philosophy;
- Design calculations of pressure vessel;
- Standard tensile test;
- Modelling of pressure vessel;
- Hydraulic test of pressure vessel;
- Economic calculations;
- Results and conclusions.

1. Literature Analysis

1.1. Pressure vessel analysis

Vessels, containers, or pipelines that maintain, store, transfer or receive fluids under internal or external pressure over 0,05 barg are called pressure vessels. Pressure vessels are used in a number of industries for example, the dairy industry for cream production, the brewery industry for beer maturation. The example of pressure vessel is shown in Fig. 1.1 [1].

Fig. 1.1 Pressure vessel for brewery industry

When discussing about pressure vessels, we must also consider tanks. Pressure vessels and tanks are similar in both design and construction the only main differences between mentioned positions is that tanks are limited to atmospheric pressure [1]

The inside pressure of pressure vessel is usually higher than the outside, except cases then external pressurised "jacket" is applied. The contents inside the vessel may suffer a change in state as in the case of steam boilers. Most of pressure vessels together with high pressures has high temperatures and in rare cases even radioactive materials. Because of mentioned hazards it is necessary that the design of vessel be such that no leakage can occur [2].

The basic construction of any pressure vessel or atmospheric tank consists of one cylinder and two heads. The fundamental differences of vessel construction are the type of bottom is used. There are many bottom types which can be attached to a vessel, column or drum such as hemispherical, semi-hemispherical, elliptical, conical, etc. Examples of pressure vessel construction are shown in Fig. 1.2.

a **b**

Fig. 1.2 Examples of pressure vessel construction

a) Vertical conical vessel [3]; b) Horizontal spherical vessel [4].

Vessels used to store liquids at atmospheric or low pressure are usually constructed with flat end sections. This is the cheapest option of enclosing the ends of a vessel. For fluids storage fluids at higher pressures, the ends are usually domed to reduce mechanical stresses [5]. There are five main types of vessel heads that can be manufactured and purchased, each suited for its own level of pressure. These five main types of vessel heads are described below:

- Flat head shown in Fig. 1.3 is only used for vessels that are containing liquids that are kept at atmospheric or low pressure. If they are used to store liquids under high pressure they won't to handle the overpressure and may slot or otherwise be damaged under the strain. These heads are also extremely cheap to produce and should be used whenever low pressure liquids are being stored in order to increase profit margins [6].

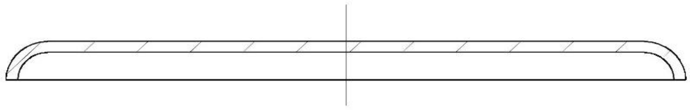

Fig. 1.3 Flat vessel head

- A torispherical head shown in Fig. 1.4 is a head that is shaped like a dome, with a flat edge all around the area where the dish ends. This flat area, called the "knuckle," is used because if a dome shape were directly welded to a pressure vessel, the dome would wrap and put undue stress on certain areas of the head. The weakest part of this type of bottom is the knuckle so many manufacturers increase the thickness of the whole bottom. This is an expensive process because it requires more raw material, meaning that torispherical heads are best for relatively medium pressure liquids [5].

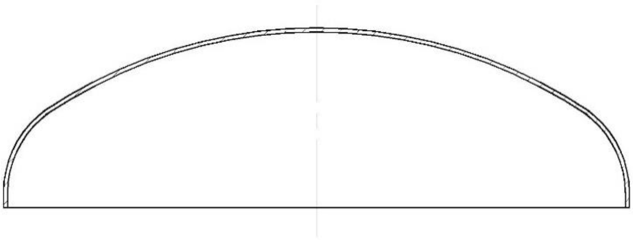

Fig. 1.4 Torispherical vessel head

- Ellipsoidal head showed in Fig. 1.5 are usually used if the pressure of a liquid being stored goes over 10 bar. This type of head looks like an ellipse making it deeper than a torispherical bottom, with its radius varying continuously. Ellipsoidal head can be directly welded to the pressure vessel without any important change in the strength of the bottom. These tend to be more expensive than the two heads previously mentioned [5].

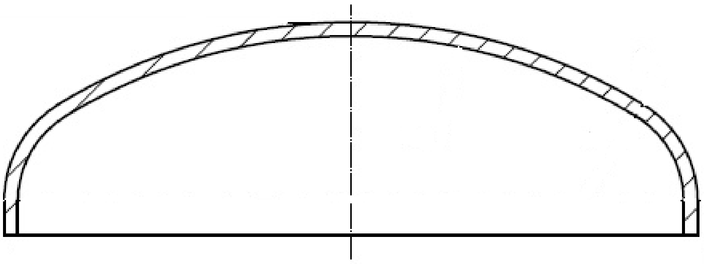

Fig. 1.5 Ellipsoidal vessel head

- Hemispherical heads shown in Fig. 1.6 are shaped like hemispheres, are easily the most expensive type of head to make due to the amount of material used, as well as the level of craftsmanship [7]. This type of bottom is expensive to fabricate and their ability to handle high pressure contents they are usually used to store liquefied natural gas. Otherwise the expense would not be worth it.

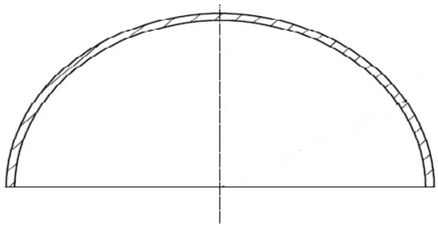

Fig. 1.6 Hemispherical head

- Conical heads with knuckle shown in Fig. 1.7 are shaped like cone and its edges ends by “knuckle” which size is at least 6% of the inside diameter. These type of heads can hold quite high pressure, doesn’t require any skilled craftsmanship or lots or raw material, manufacturing doesn’t take long and they provide fast liquid flow-out from the vessel [7].

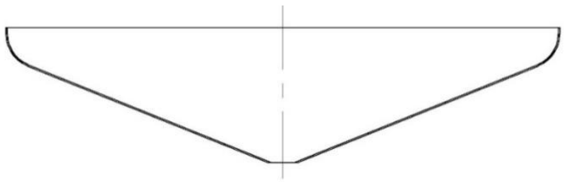

Fig. 1.7 Conical head with knuckle

1.2. “Jacketed” pressure vessel analysis

To cool or heat contents inside vessel external “jackets” or internal coils are installed. “jackets” are most common in the following industries: beverage, food, alcohol, chemistry industries. There are three main types of “jackets” – Spiral baffle “jackets”, half-pipe coil, dimpled “jackets”. These heating or cooling “jackets” are most popular to use because of the following reasons [8]:

- It is easy to clean and maintain by applying “clean” media;
- All types of liquids can be used to heat or cool contents inside vessel. To heat up contents steam is perfect to apply and to cool glycol or cold water are ideal.
- It provides easiest way to control velocity, circulation and temperature of cooling or heating content.
- These types of “jackets” can be produced from weaker mechanical properties materials than vessel itself.

Spiral baffle “jackets”

The biggest advantage of this type of “jackets” is that the heating/cooling liquid inside “jacket” covers the full base surface and the construction is most simple of other “jackets” and they can handle high speed velocities but to be able to fabricate this type of “jackets” gap between the baffies and the “jacket” must be applied. However mentioned gap is very small so main flow area per baffle is significant compared with cross sectional flow [3].

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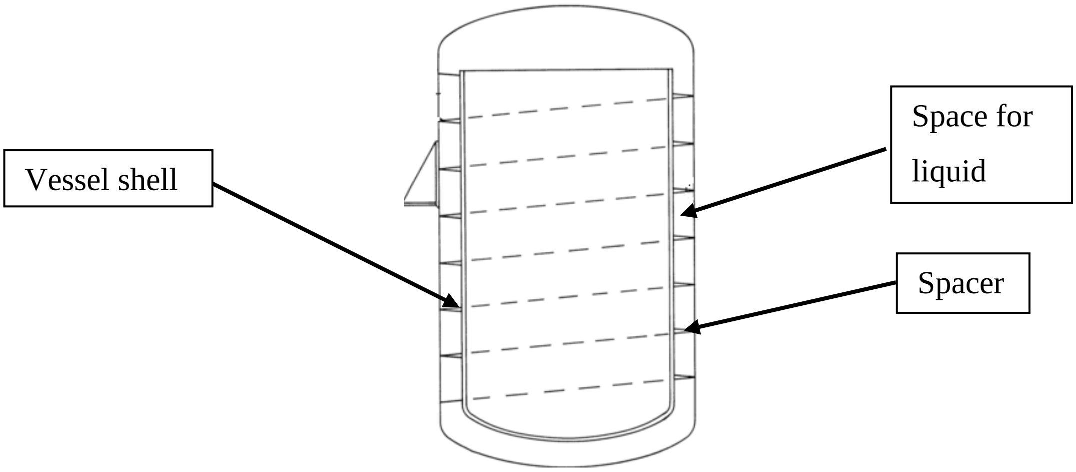


Fig. 1.8 Jacket with a spiral baffle [3]

Spiral half-pipe coil “jackets”

This type of design creates high turbulence and velocity in the halfpipe coil. The half-pipe coil is usually used to keep inside contents ate higher than 100°C temperature. These type of heating-cooling coil is more superior to conventional “jackets” because drop appeared by pressure can be easily monitored. There are no restrictions to the number of outlet and inlet nozzles so it can be divided into multiple zones for maximum efficiency and flexibility. The biggest disadvantage half-pipe coil that it is not practical and very expensive to apply on smaller diameter vessels. Example of Half Pipe Coil welded to the shell shown in Fig. 1.7 [3].

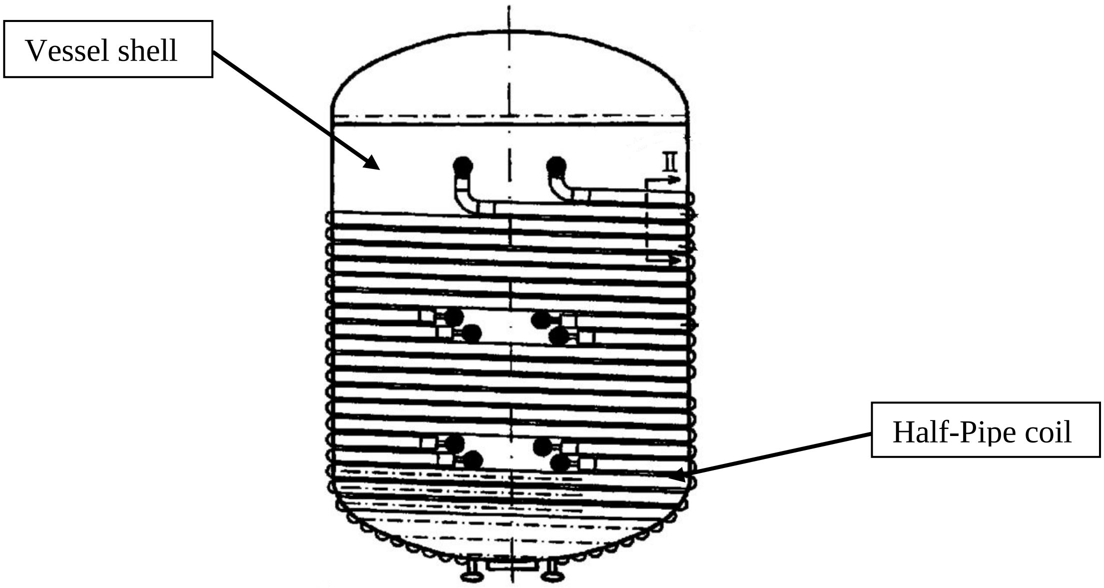


Fig. 1.9 Spiral half-pipe coil welded to the shell [4]

Dimpled "jackets"

A thin additional cylinder is applied to the vessel cylinder with dot welds located in a regular grid often about 60 mm on centre both horizontally and vertically. The dimple jacket interior sees various kinds of heating and cooling media, including water, steam, and water-glycol. Some media may include corrosive components, for example chlorinated city water and some are not. The dimple jackets is usually insulated with mineral wool or polyurethane foam and then cladded with a stainless steel or galvanized carbon steel sheets. The dimple jackets are limited to many heating/cooling cycles as the vessels are operates through the numerous production processes. Example of dimpled jacket welded to the vessel shell shown in Fig. 1.8 [3]

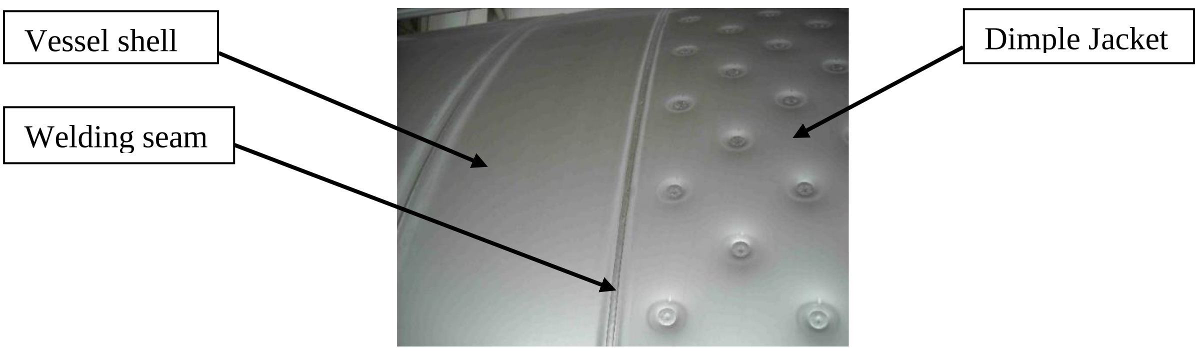


Fig. 1.10 Dimple jacket welded and pickled on vessel shell

1.3. Pressure vessel design philosophy

For genuine pressure vessels and similar equipment, preliminary design may still be influenced by heat transfer and fluid flow requirements. Although the aspect of thermal hydraulic design is intricately related to the structural design, especially for thermal transient loadings [8]. In this research work we will not be discussing them in any detail. We assume that the temperature distribution associated with a particular thermal transient has already been evaluated in a typical design application [9].

The basic design of a pressure vessel part would cause looking at the geometry and manufacturing construction details and also on the loads appeared by the part. The loads caused by the pressure vessel is connected to factors such as design pressure, design temperature, and mechanical loads (due to dead weight and thermal expansion) along with the postulated transients (typically those due to temperature and pressure) that are anticipated during the life of the plant. These transients generally reflect the fluid temperature and pressure excursions of the mode of operation of the equipment. The type of fluid that will be contained in the pressure vessel of course is an important design parameter, especially if it is radioactive or toxic. Also site location information must be involved to provide loads due to earthquake (seismic), and other accident loads [10].

ASME Boiler & Pressure Vessel Code

In general, pressure vessels designed in accordance with the ASME Code, Section VIII, Division 1, are designed by rules and do not require a detailed evaluation of all stresses. It is specified that secondary and high localized bending stresses may exist but are allowed for by use of a higher design equity and design rules for details. It is also required that all loadings (the forces applied to a vessel or its structural elements) must be considered [6].

ASME code Section VIII, Division 1 presents formulas and calculation instructions for thicknesses and stresses of vessel components, it is up to the designer to select acceptable analytical procedures for analysing other vessel’s components and to combine the calculated loadings and stresses in a method appropriate with the intended process of the equipment for an most economical and safe design. For the supporting structures, the designer must also evaluate the load combinations specified by the applicable building code [6].

ASME code Section VIII, Division 1 sets allowable stresses that the maximum general primary membrane stress must be less than allowable stresses outlined in material sections. Further, it states that the maximum primary membrane stress plus primary bending stress may not exceed 1.5 times the allowable stress of the material sections [6].

In overall practice when preparing a more detailed stress analysis to apply higher allowable stresses. Actually, analysis of stresses allows improve knowledge of localized stresses and the use of higher allowable stresses in place of the larger design margin used by the Code. This larger design equity really reflected lack of knowledge about actual stresses [8].

Structural and material

Materials with high mechanical properties created by alloying elements, production processes, or heat treatments, are developed to satisfy economic or engineering requirements for example material with high mechanical properties can reduce minimal demanded thicknesses of pressure vessel. They are constantly tested to demand design limits consistent with their higher strength and adjusted to vessel design as experimental and fabrication knowledge justifies their use.

There is no perfect material for pressure vessels suitable for all environments, but material selection must match application and environment. This is very important in pressured chemical reactors because of the embrittlement effects of gaseous absorption, and in nuclear reactors because of the irradiation damage from neutron bombardment [11].

The most common materials that are used in construction of pressure vessel are [11]:

- Nonferrous materials such as aluminium and copper;
- Specialty metals such as titanium and zirconium;
- Steels;
- Non-metallic materials, such as, plastic, composites and concrete;
- Metallic and non-metallic protective coatings.

The mechanical properties that generally are of interest are:

- Yield strength;
- Ultimate strength;
- Reduction of area (a measure of ductility);
- Fracture toughness;
- Resistance to corrosion.

Stress analysis of pressure vessel

The most pressure vessel fatal accident occurs because of stresses created by pressure so the design of pressure vessel must confirm pressure vessel safety. Before every design start it is necessary to set maximum and operating allowable pressures inside pressure vessel and its components. To evaluate stresses created by pressure analytical or experimental tests must be applied. For example the finite element model helps for designers to analyse stress distribution of vessel and its components. By stress analysis we can determine relationship between applied loads to the vessel and related responses like deformations in dimension strains or stresses [12].

We can obtain stress-strain curve of material by applying standard tensile test. An example of standard tensile test specimen is shown in Fig. 1.11. In order to determine the yield and ultimate strength of a specimen stress-strain curve is used Fig. 1.12. Yield stress $R_{p0.2}$ is defined at the stress level at which the specimen achieves a specified deviation from a linear stress-strain relationship. An offset of 0.2% deformation is used for determine yield stress, as shown in the figure.

Ultimate tensile strength R_m is determined as the maximum tensile stress of the material which can sustain it without fracture. It is calculated by dividing the maximal load applied during the tensile test by the original cross sectional area of the specimen [13].

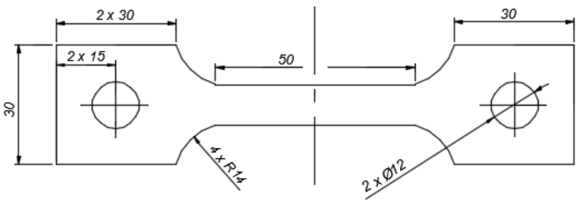


Fig. 1.11 Tensile test specimen

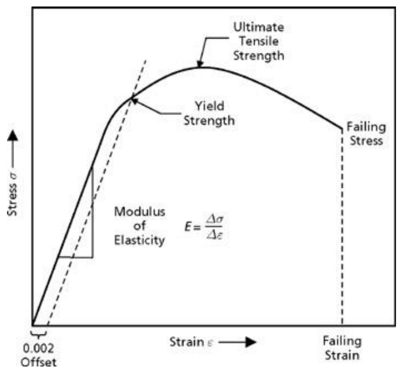


Fig. 1.12 Typical stress-strain curve [12]

1.4. Pressure test

Pressure test is based on ASME Code Section VII Div. and can be applicable to all equipment and their parts made of steels in accordance with mentioned requirements. The vessel should be subjected to a proof test to demonstrate the integrity of the finished product.

Scope

In this research work we will present basic procedure of hydraulic test which enables to confirm the mechanical resistances and verify safety of the welded or joined components. In our case pressure is designed according ASME code section VIII Div. 1 so the pressure test must comply code requirements. During hydraulic test entire vessel and its components should be unpainted and uncovered [14].

To ensure safety pressure test is common applied using cold water because testing by gas is potentially more dangerous.

Combined hydrostatic / pneumatic test – in some cases it may be desirable to test a vessel when it is partly filled with liquid. This is as dangerous as pneumatic test. The pressure test shall be done using adequate tools and equipment and ensuring that the required safety rules and regulations are fulfilled [14].

After all fabrication has been completed, inspections shall be made prior to pressure test to ensure that in all respects the design, materials, manufacturing and testing comply with the requirements [6].

Safety requirements

Prior the test starts on site a toolbox safety meeting shall be conducted to ensure all parties involved know exactly what they have to do. All pressure test personnel shall attend, with details of the work, the hazards and the precautions being discussed. All personnel involved in the operations shall attend to the toolbox safety talks sign report [6].

Pressure tests to be performed in designated test area. The pressure test area shall be proper condoned. Safety signs shall be installed. The minimum distance between barrier and vessel to be pressured shall be minimum 5m. During pressure test no work shall be performed on or near the vessel under any circumstances while the test is in progress [6].

Pressure gauge cannot be dismantled while the system is under pressure. Prior to start any pressurization, all hoses, valves, flanges, fittings, studs, nuts, etc. shall be visually inspected by operator. If any defects found, then shall be removed. Tightening or loosening of plugs, bolts, hoses, or test fixtures while equipment is pressurized is strictly prohibited [6].

No vessel shall be subject to any form of shock loading such as hammer testing when undergoing proof testing. After pressure testing pressure will be reduced in a controlled manner though a drain hose. Attention must be paid to the direction where the pressure will be released [7].

Pressure test equipment

The equipment shall be verified or calibrated with valid certificate. Pressure gauges, pressure recorder, temperature recorder, pressure sensor, thermometer shall be verified and calibrated by external accredited 3rd party after 1 year. Testing components and equipment shall have working pressure rating greater to the test pressure. All elements that cannot withstand the pressure shall be removed [14].

All equipment shall be loaded on site in a manner that leaves suitable and safe escape routes and shall comply with pressure test. All equipment shall be visually checked prior use and shall be in good conditions and shall be re-calibrated at any time that there is reason to believe they are in error. If any defects are found then the dedicated equipment shall be removed from service, places in a quarantined area [6].

The pressure gauge shall be chosen in order that the test pressure shall be within 1/3 and 2/3 of the pressure gauge range. The scale should indicate a range of approx. the double value of the intended test pressure. In no case the scaling should be lower than 1,5 times the test pressure or higher than 4 times the test pressure gauge range [14].

2. Construction and design

Design and modelling of “jacketed” pressure vessel purpose is intended to use in energy industry. Position of the vessel – vertical. The vessel has cylindrical form and is welded from the stainless steel SA-240 type 304L sheets with the conical bottom and flat top. To heat-up and keep constant temperature of the product inside vessel it has welded Half-Pipe jacket on the cylinder and conical bottom. General sketch of vessel is shown in Fig. 2.1. More detailed drawing of vessel is presented in appendix no 1.

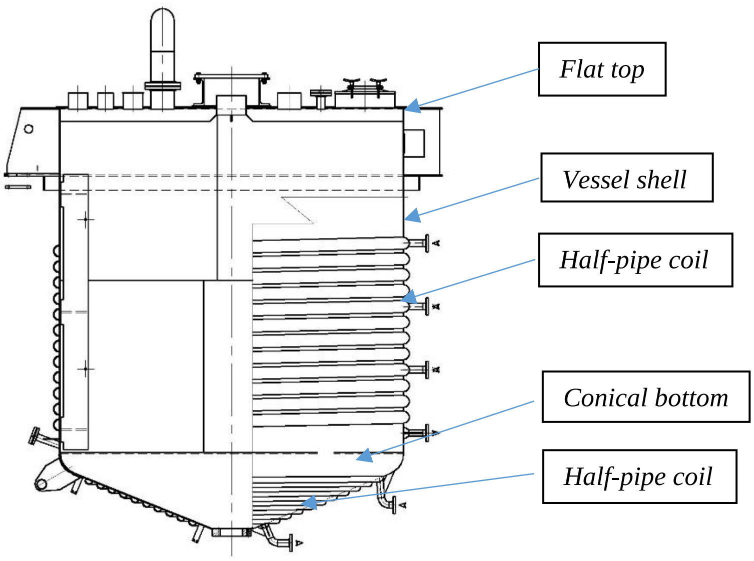


Fig. 2.1 Design of „jacketed“ pressure vessel

2.1. Material

For this project we use stainless steel grade AISI 304l. This grade of steel also known as '18-8' stainless because of it has 18 percent chromium content and 8 percent nickel content. It is the most universal and most widely used stainless steel, available in a wide range of products, forms and superior than any other. This type of stainless steel also has excellent forming and welding characteristics. The balanced austenitic structure of Grade 304 is readily alloyed or roll formed into a variety of components for applications in the industrial, architectural, and transportation fields. It also has outstanding welding characteristics and post-weld annealing is not required when welding thin sections [15].

Chemical composition of stainless steel grade 304l according manufacturer is presented in table no 2.1. appendix no. 2

Table 2.1 Chemical composition of stainless steel grade 304l

Elements, %	C	Si	Mn	P	Cr	Ni	Nb	Cu	Co	N
Heat	0.020	0.32	1.57	0.38	18.30	8.14	0.008	0.38	0.20	0.061

Mechanical properties of stainless steel grade 304l according manufacturer are presented in table no 2.2. appendix no. 2

Table 2.2 Mechanical properties of stainless steel grade 304l

Material properties	Yield stress $R_{p0.2}$, MPa	Ultimate tensile strength R_m, MPa	Elongation after fracture, *A*, %		Hardness HRB
			A_5, %	50mm, %	
min.	230	450	45	40	Not specified
max.	330	580	Not specified		201

2.2. Construction of vessel

Construction of vessel is based on bottom selection of vessel. When choosing the bottom for a pressure vessel, it is important to take the pressure of the liquid that will be stored into consideration. If the incorrect head is applied, it might be impossible to maintain the pressure of the vessel and a dangerous situation might develop.

My current target is to design a vessel with applied jacket on bottom to heat the vessels contents at 110°. To reach this temperature vessel should be heated by steam which creates 6 bar pressure. For my project I have chosen to use conical bottom because it can be made of thinner steel sheet than hemispherical or elliptical heads, takes less time to manufacture and most important, because of its shape half-pipe coil can be welded faster.

2.3. Construction of "jacket"

Literature overview of vessel "jackets" showed that less complex and most efficient "jacket" types are dimpled and half-pipe coil "jackets" welded on outside of vessel shell. Half-pipe coil and dimple jacket options offer optimal economic advantages over the conventional jacket for steam medium handling.

For our pressure vessel half-pipe jacket is applied. Dimple jackets are more economic and faster to manufacture than half-pipe coil but half-pipe coil is superior to the dimple jacket because of its increased strength, provides high velocity and turbulence within the jacket, the drop can be carefully controlled and more than one service can be supplied to different sections of the wall.

2.4. Design data and calculations

Design calculations of our pressure vessel are made according ASME Boiler and Pressure Vessel Code Section VIII Division 1.Mentioned code presents calculation methodology and recommendations to determine required minimal vessel body thicknesses to ensure safety.

According design task initial design data are presented in table 2.3.

Table 2.3 Design data

Description	Vessel	Half-pipe coil
Design code	ASME VIII Div.1	
Fluid, m3	Oils	Steam / Condensate
Volume gross, m3	13.5	0.085x3 / 0.049
Volume net, m3	10	0.085x3 / 0.049
Operating pressure, barg	0.2	5
Max. allowable working pressure, barg	0.5	6
Design pressure, barg	Hydrostatic	6
Operating temperature, °C	0 ÷ 110	0 ÷ 110
Hydraulic test	Water, 10...20 °C, 30 min	
Corrosion and erosion allowance, mm	0	0
Vessel inside diameter, mm	2488	-
Height of the cylinder part, mm	2516	-
Main material	S.s.type 304L	S.s.type 304L

Design scheme of vessel component under pressure are shown in Fig. 2.2.

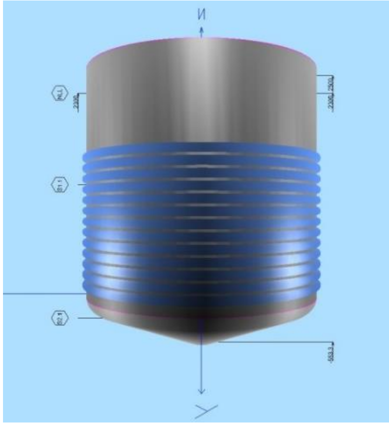


Fig. 2.2 Vessel design scheme

Design calculations are applied on components in direct contact with pressure see Table 2.4.

Table 2.4 Vessel components in contact with pressure

ID	Component Type	Component Description
HP.1	Half Pipe Jacket	Jacket on cylindrical shell
S1.1	Cylindrical Shell	Main Shell
S2.1	Conical Shell	Conical bottom

For design calculations Visual Vessel Design by OhmTech Ver:15.01“ is used. Calculations report located in appendix no 3. The most important output data are shown in Table 2.5.

Table 2.5 Results of design calculations

Description	Output data
Top head of vessel	Minimal required thickness, 4 mm
Vessel cylinder	Minimal required thickness, 6mm
Vessel bottom	Recommended minimal thickness, 6mm
Half-pipe coil (cylinder)	Minimal test pressure 10.2 barg
Half-pipe coil (bottom)	Minimal test pressure 10.2 barg

2.5. Results

ASME Pressure Vessel Code doesn‘t provide any method how minimal conical bottom wall thickness should be calculated if it is under external pressure. Code only gives recommendation to use same conical bottom wall thickness as vessel cylinder. In our case min. required cylinder thickness is 6mm. Long practice says that if conical bottom is made from 6mm thickness sheet it should operate in extreme conditions under high pressures. In our case vessel will be installed on site there are simple environment and design pressure inside heating coil is quite low – 6bar. Long practice says that for such design most common bottom wall thickness is 4mm and to use this thickness we must prove that it is safe to use. According ASME code Section VIII, Division 1 all pressure vessels and its pressurized parts design thicknesses must be chosen by not exceeding 2/3 of material yield strength so in the next paragraphs we determine actual yield strenght of mateial and make FE model of bottom half-pipe coil.

3. Experimental part

3.1. Standard tensile test

Three specimens were produced in order to examine the mechanical property of the material - yield stress $R_{p0.2}$.

For this reason, three standard virgin plate specimens were cut from the stainless steel grade 304l sheet with thickness of 4 mm (Fig 3.1). Three specimens were used to get more accurate results.

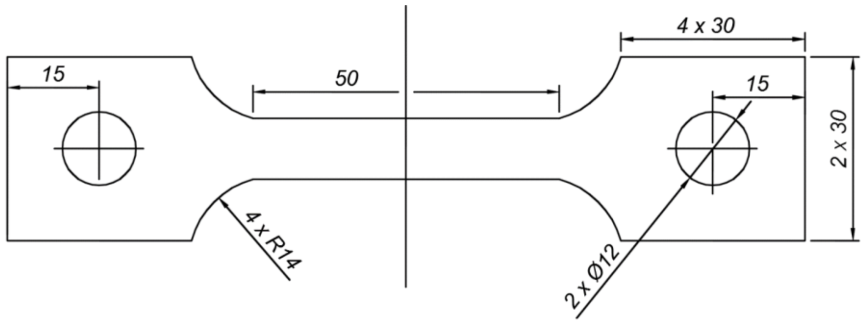


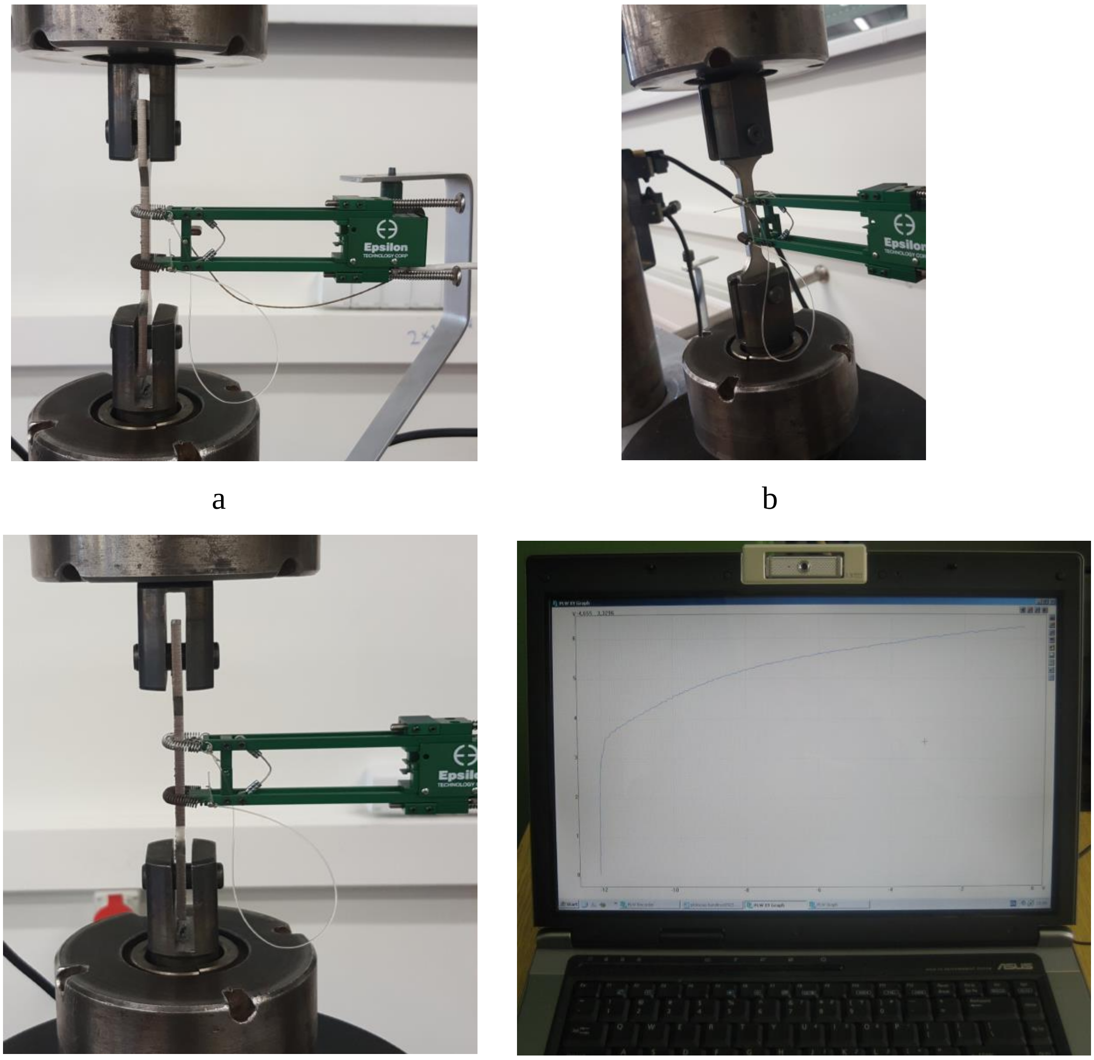
Fig. 3.1 Shape and dimensions of the specimens

The tensile test has been carried out on 10 kN capacity standard universal low-cycle tension-compression testing machine UME-10TM with the stress rate 20 MPa/s, which is in accordance to the requirements [ISO 6892-1] to keep stress loading rate in the limits 2-20 MPa/s. Following to these requirements specimen loading rate was set 1 mm/min.

The grips with cylindrical hardened loading pins with the diameter of 12 mm were used to maintain normal tensile stress perpendicular to the specimen symmetrical axis during the tension.

Before the testing, machine elastic load measuring element, connected to the upper specimen clamping grip by rigid joint was calibrated using two standard force reference dynamometers: DOP 3-20, which is used for testing machine calibration in the range of 20 – 200 kN. [Appendix 4].

Axial displacement of the specimen was measured using extensometer “*Epsilon*” with gauge length of 25mm, Strain extensometer “*Epsilon*” was calibrated using micrometer-caliper MI- 25 with resolution accuracy of 0.001 mm [Appendix 3]. The extensometer was fixed on the centre of specimen working zone as showed in Fig. 3.2.



a b

c d

Fig. 3.2 Photos of specimen with grips and strain gauge:

a - virgin plate specimen no. 1; b – - virgin plate specimen no. 2; c - virgin plate specimen no. 3; d - force-displacement record

The measurements via oscilloscope "*Picoscope 3204 Osciloscope*" were recorded by the computer and force scale m_F = 47732,7 N/mV and displacement scale m_ε = 1.2563 mm/mV was determined.

Experimental investigation of the mechanical properties

The test was performed according to the standards [ISO 6892-1: 2009, LST EN 10002-1:2003]. According to the testing program, by tress standard virgin plate specimens (marked as 1, 2 and 3) were tested. Thus in total, during the tensile test 3 specimens were tested in order to examine main mechanical property of the steel - Yield stress $R_{p0.2}$

The photos of virgin plates after fracture are showed in Fig. 3.3

Fig. 3.3 Virgin plates after fracture

Engineering stress-deformation curves of each specimen are presented in Figs. 3.4 – 3.6.

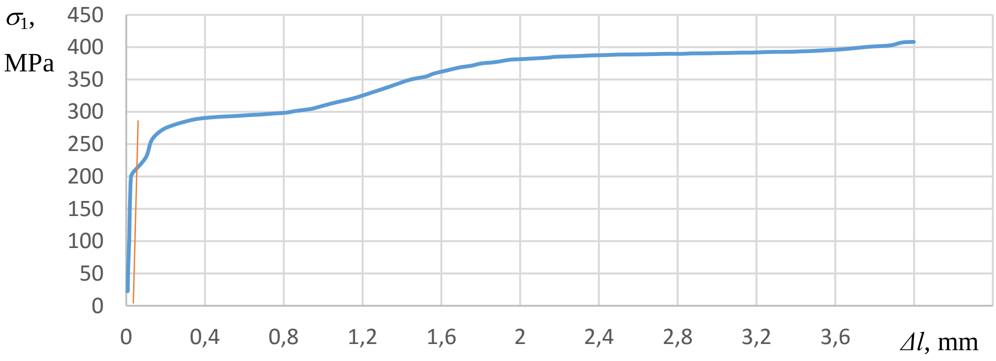


Fig. 3.4 Stress-deformation curve of specimen no. 1

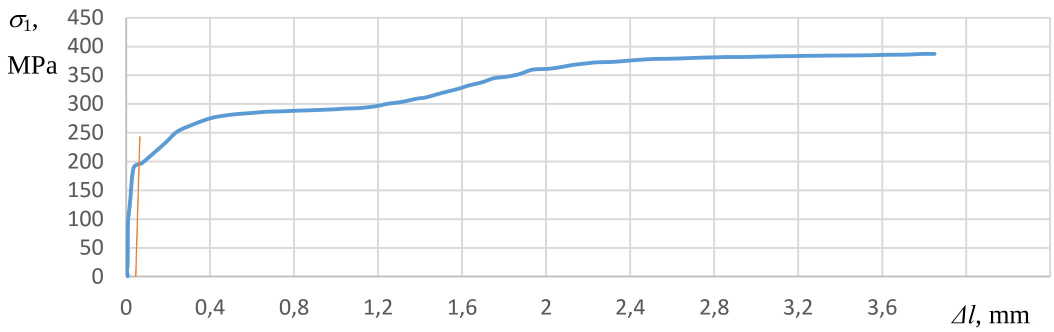


Fig. 3.5 Stress-deformation curve of specimen no. 2

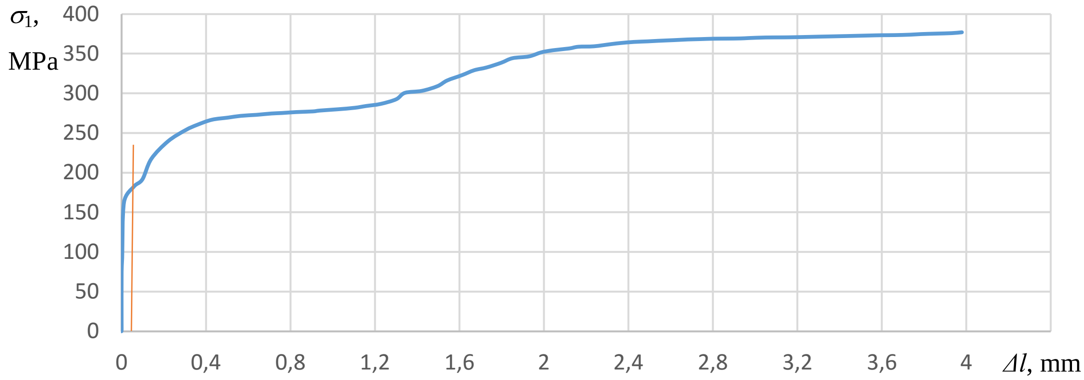


Fig. 3.6 Stress-deformation curve of specimen no. 3

An offset of 0.2% deformation is used for determine yield stress, as shown in the Figs. 3.4-3.6. Yield stress for virgin plate is presented in table no. 3.1.

3.2. Results

The obtained mechanical properties obtained by standard tensile test are shown in Table 3.1.

Table 3.1 Mechanical properties of each specimen

Specimen mark	Dimensions of specimen ($a \times b \times l$)	Length of working part L_i, mm	Yield stress $R_{p0.2}$, MPa
1	4 × 30 × 120	50	210
2	4 × 30 × 120	50	195
3	4 × 30 × 120	50	185
Average			196.66

4. Modelling

The objective of this part is to verify the strength of the heating coil welded on conical head with respect to overpressure. A 2D axisymmetric linear elastic structural analysis is performed to find stresses. The analysis executed using the FE package included to Solid works 3D CAD design software.

The stresses acting in heating coil where checked with respect to stress categorization approach (ASME Section VIII, Division 1) using a linear elastic analysis. The analysis is carried out in the FE analysis program SolidWorks Simulation 2013.

Structural model

The section of heating coil is modelled according to appendix no. 1 drawings, however only details of importance with respect to stiffness are included. The principal scheme of Half-Pipe coil is shown in Fig. 3.1. It is simplified by omitting small parts which have no significant influence on stiffness of the tank. Planar 2D mesh has been used for numerical analysis of the tanks. The geometric model and FE model used in the analysis is shown in Fig. 4.1, 4.2, 4.3.

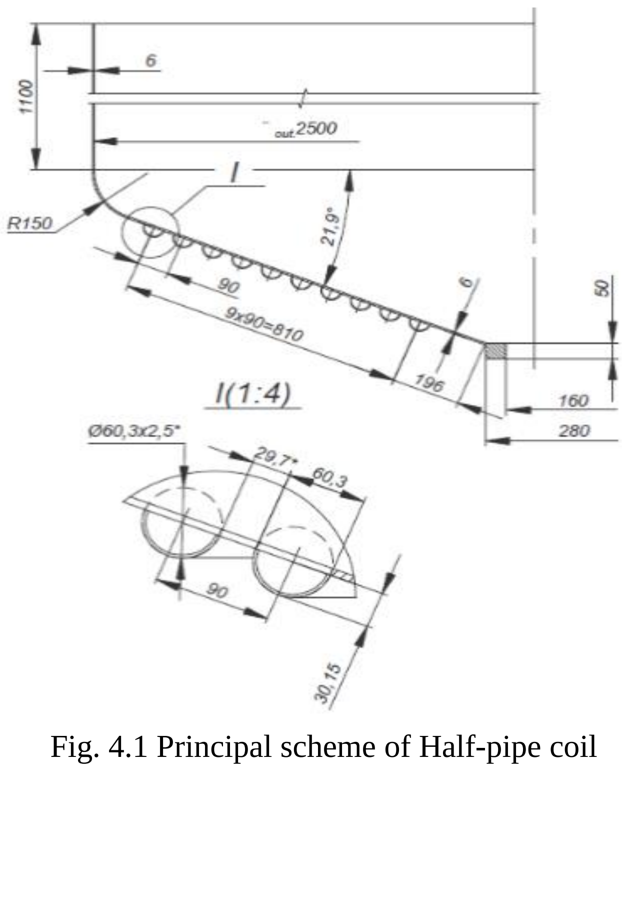


Fig. 4.1 Principal scheme of Half-pipe coil

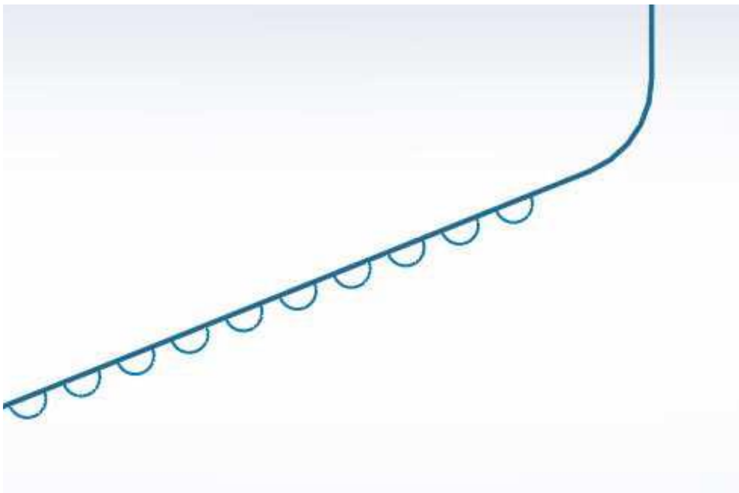


Fig. 4.2 Geometric model of hating coil

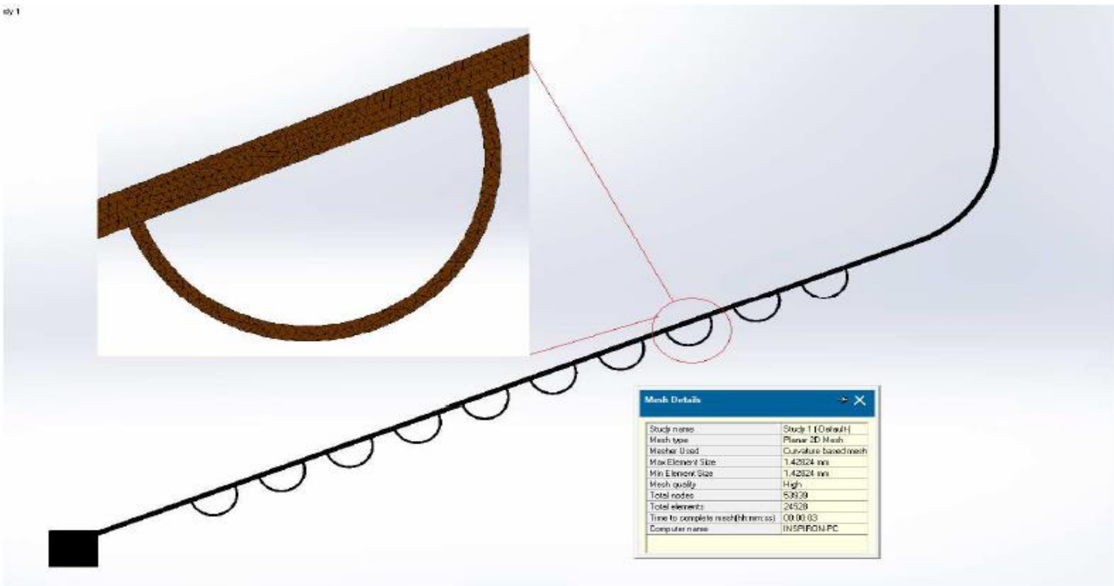


Fig. 4.3 FE model of heating coil

Boundary conditions and applied actions

The simulation has been carried out by using linear elastic structural analysis. Pressure of 0,6 MPa is applied to heating coil Fig. 4.4. Edge of cylindrical shell free end is constrained in all directions (all degree of freedom is constrained) Fig. 4.5.

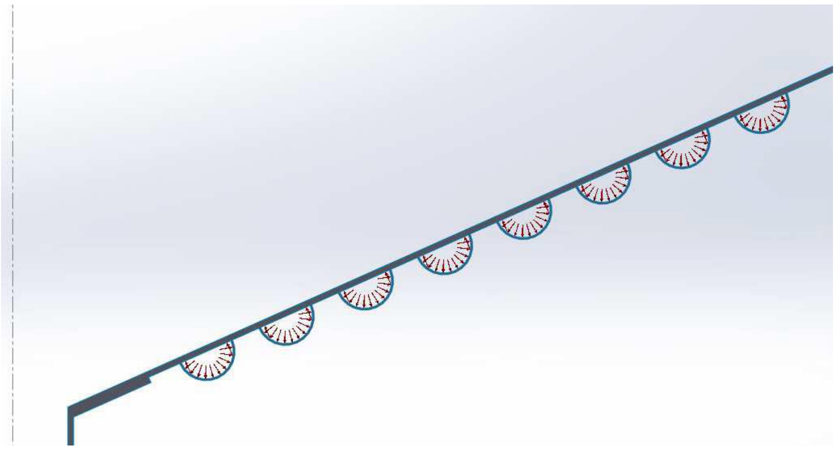


Fig. 4.4 Internal pressure load

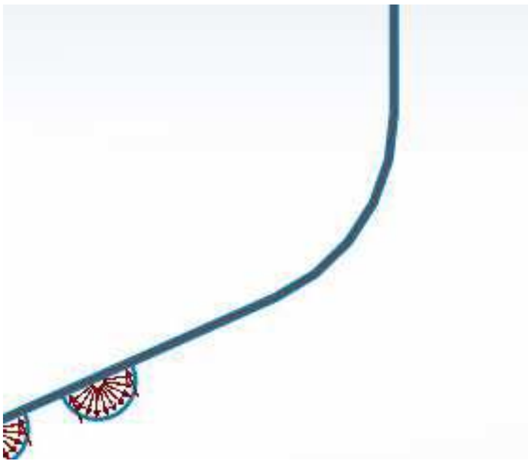


Fig. 4.5 Constrains

Assessment criteria

To verify strength of the half-pipe coil welded on conical head with respect to overpressure we need to know allowable (yield) stress of material. The yield strength of vessel bottom material is 196.66MPA.

According ASME code Section VIII, Division 1 all pressure vessels and its pressurized parts design thicknesses must be chosen by not exceeding 2/3 yield strength, so our allowable stresses is 134.11 MPA.

4.1. Results

The maximum calculated equivalent stress of the heating coil is 26.4 MPa Fig. 5.1. Maximum calculated displacement of the tank is 0.037 mm Fig. 4.6.

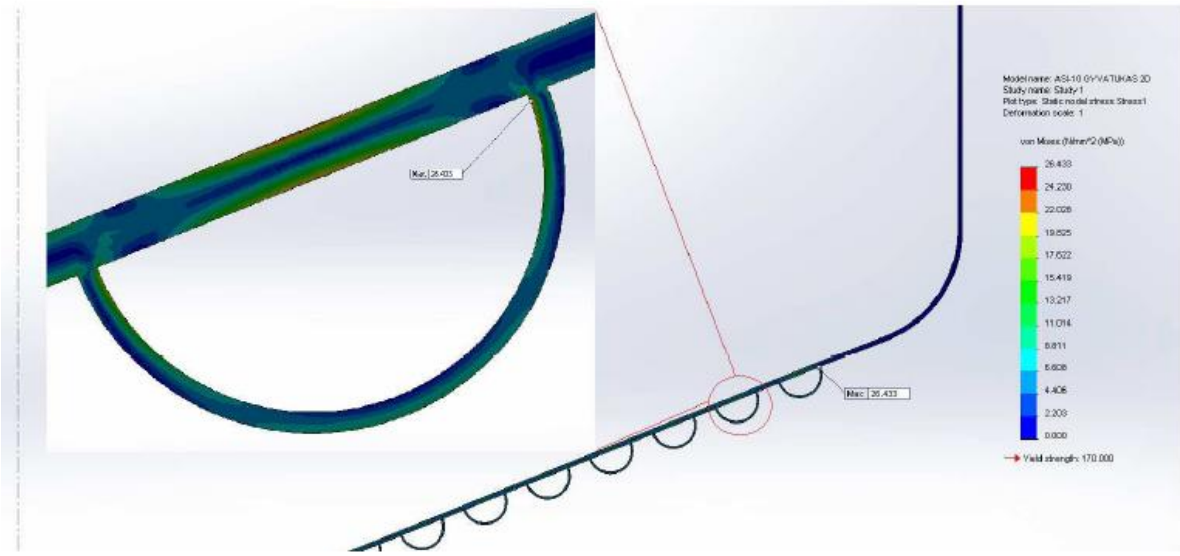


Fig. 4.6 Equivalent stress distribution

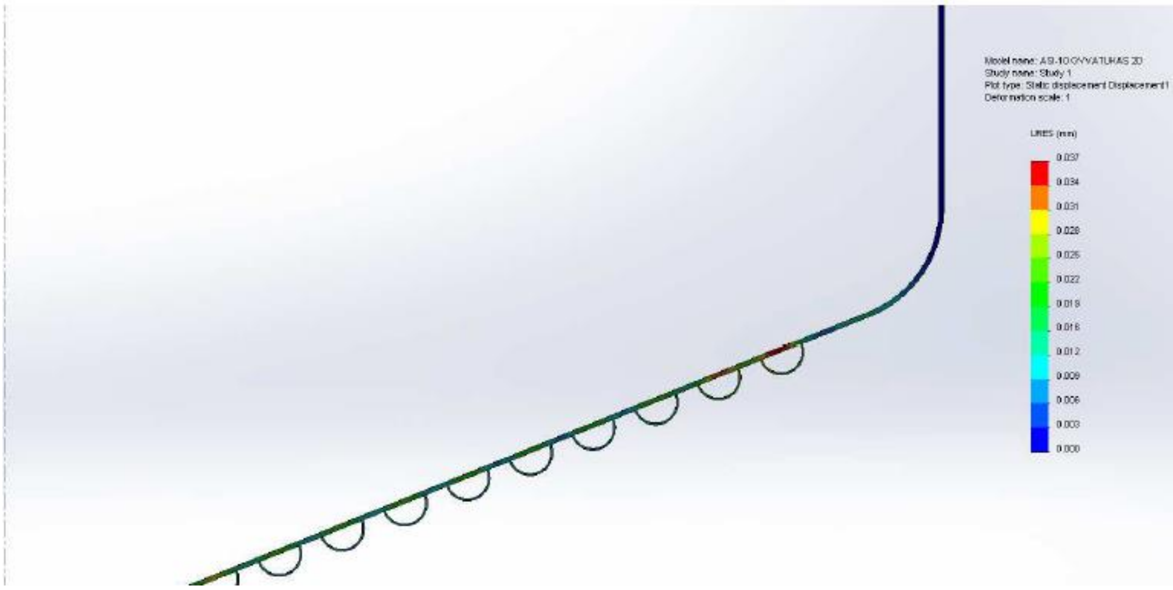


Fig. 4.7 Displacements

The FE model of heating half-pipe coil was developed. The FE pressure vessel analysis has been executed by taking into consideration the design pressure.

The maximum stresses in all cases are developed in vicinity of a sharp edge of heating coil and can be considered as occurred at areas of stress concentration. The general primary membrane stress (26.4 MPa) developed due to internal pressure do not reach the assessment criteria (131.11 MPa). Maximum calculated displacement of structure 0.037 mm.

5. Testing

Hydraulic test must be applied. To verify and confirm the mechanical resistance of the vessel and its half-pipe coil

In this research work procedure of hydraulic test is based on ASME code Section VIII Div. 1 and all equipment and their parts made of steel in accordance with mentioned code.

Hydraulic test applied after all fabrication has been completed, inspections made prior pressure test to ensure that in all aspects the design, materials, manufacturing and testing comply with requirements. The test performed and assessed by authorized personal having habit of such operation. For safety reasons the pressure test conducted under cold water conditions. The manufactured pressure vessel is shown in figure 5.1.

Fig. 5.1 Manufactured vertical conical pressure vessel with half-pipe coil

The pressure test was applied by using “Rothenberger RP Pro II” pump (Fig. 5.2) constructed for hydraulic tests. To measure pressure P certified electronic monometer was used (Fig. 5.3). To record water temperature during hydraulic test E&H electronic temperature transmitter is used. Recorder temperature 13°C (Fig. 5.4)

Fig. 5.2 “Rothenberger RP Pro II” pump used for pressure test

Fig. 5.3 Electronic monometer used for pressure test

Fig. 5.4 Water temperature under pressure test

To verify safety of vessel and its half-pipe coil design calculations by ASME code showed us that half-pipe coil must be over pressured by water under 10.2 bar and kept at that pressure for 5 hours (300 min) Fig. 5.5. After 5 hours operator of pressure test reduced pressure to 6 bar to inspect if no leakage occurred. Further operator fully reduced pressure and made dimension control which is presented in appendix no. 5.

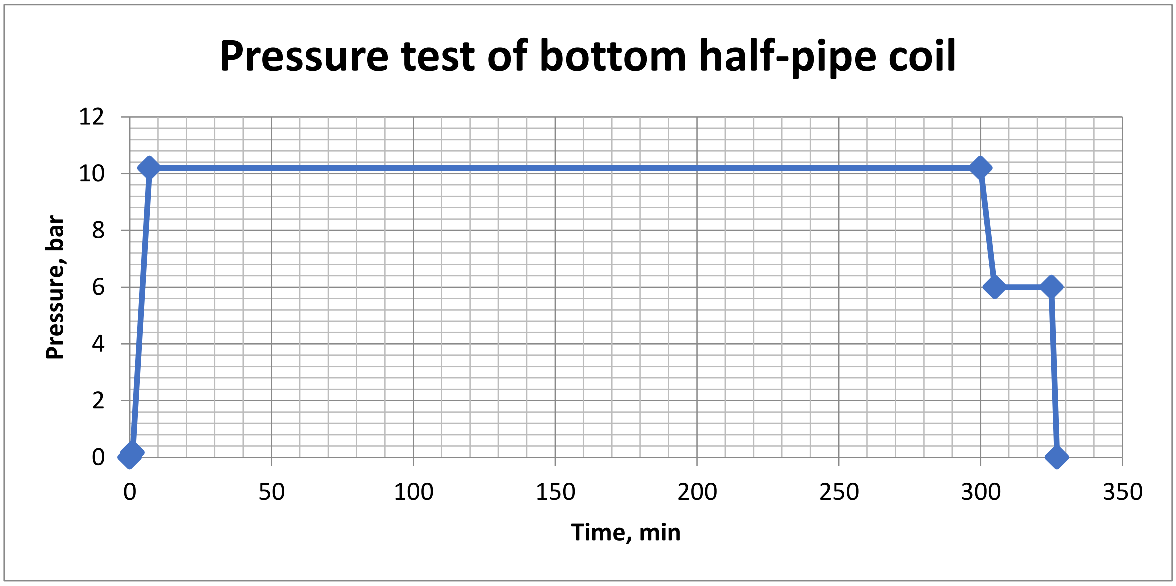


Fig. 5.5 Graph of half-pipe coil pressure test

5.2. Results

After hydraulic test no deformation in dimensions and leakage appeared. From the obtained results we can assume that all aspects the design, materials, manufacturing comply with requirements. Summary of results are shown in table 5.1.

Table 5.1 Results of pressure test

Description	Working pressure, bar	Test pressure, bar	Liquid	Liquid temp., °C	Leakage detected	Deformation detected
Bottom coil	6	10.2	water	13	no	no
Cylinder coil	6	10.2	water	13	no	no

6. Economic calculations

ASME code provides methodology how minimal vessel shell wall thickness must be calculated in case if it is under external pressure but for conical bottom with applied external pressure it gives only recommendation what minimal wall thickness should be used. In this case recommended minimal thickness is 6mm. After FE analysis and pressure testing we proved that minimal required thickness can be reduced to 4 mm. These changes in vessel's bottom thickness not even reduced material costs but also reduced production time because thinner material is more wieldable and formable. The approximate savings of material and working hours are shown in Table 6.1 and 6.2.

Table 6.1 Comparison of material consumption and costs

	S. steel sheet dimensions, mm	Quantity, pcs.	Weight, kg	Price of 1 kg, EUR	Total price, EUR
Vessel's bottom made of 6mm	6 x 2000 x 4000	2pcs.	768	2.10	1075.20
Vessel's bottom made of 4mm	4 x 2000 x 4000	2pcs.	1024	2.10	1612.80
Difference			256		537.60

Table 6.2 Comparison of production costs

	Required labour hours for production, Hours	Price of 1 labour hour, EUR	Costs to fabricate 1pc. Bottom, EUR
Vessel's bottom made of 6mm	6	16	96
Vessel's bottom made of 4mm	5	16	80
Difference			16

Tables 6.1 and 6.2 shows that our improvement in design reduced amount of materials and production time. Total saving of one vessel's bottom production equals to 537.60 EUR + 16 EUR = 553.60 EUR.

Conclusions

1. Literature analysis showed that vessel construction and design should be chosen after evaluation of vessel purpose, environment in which vessel will operate, safety requirments.
2. ASME Pressure Vessel code provides methodology how minimal vessel shell wall thickness must be calculated in case with applied external pressure but for conical bottom with applied external pressure it gives only recommendation what minimal wall thickness should be used. In my case recommended bottom thickness was 6 mm and after Finite Element analysis it reduced to 4 mm. Reduced thickness means lower material costs and faster fabrication.
3. The Yield strength of materiali s presented in material certificate provided by manufacturer. The problem is that Yield strength is presented by value intervals, in our case from 230 MPa to 330 MPa. In our case, to make precise FE model true material Yield strenght must be known. This machanical property was obtained by applying standard tensile test and results showed that in fact materials yield strenght is 196.66 MPa, not 230MPa as specified in manufacturers certificate.
4. FE model including ASME code safety factor showed that conical bottom minimal wall thickness can be reduced from recommended 6 mm to allowable 4 mm. This improvement reduced materials costs, thinner material means that it can be welded and formed easier and faster.
5. Pressure test confirmed that vessel has enough mechanical resistance to operate with applied overpressure inside half-pipe coil welded on conical bottom.
6. Improvement in design reduced amount of materials and production time. Total savings of one vessel’s bottom production equals to 537.60 EUR + 16 EUR = 553.60 EUR.

Literature

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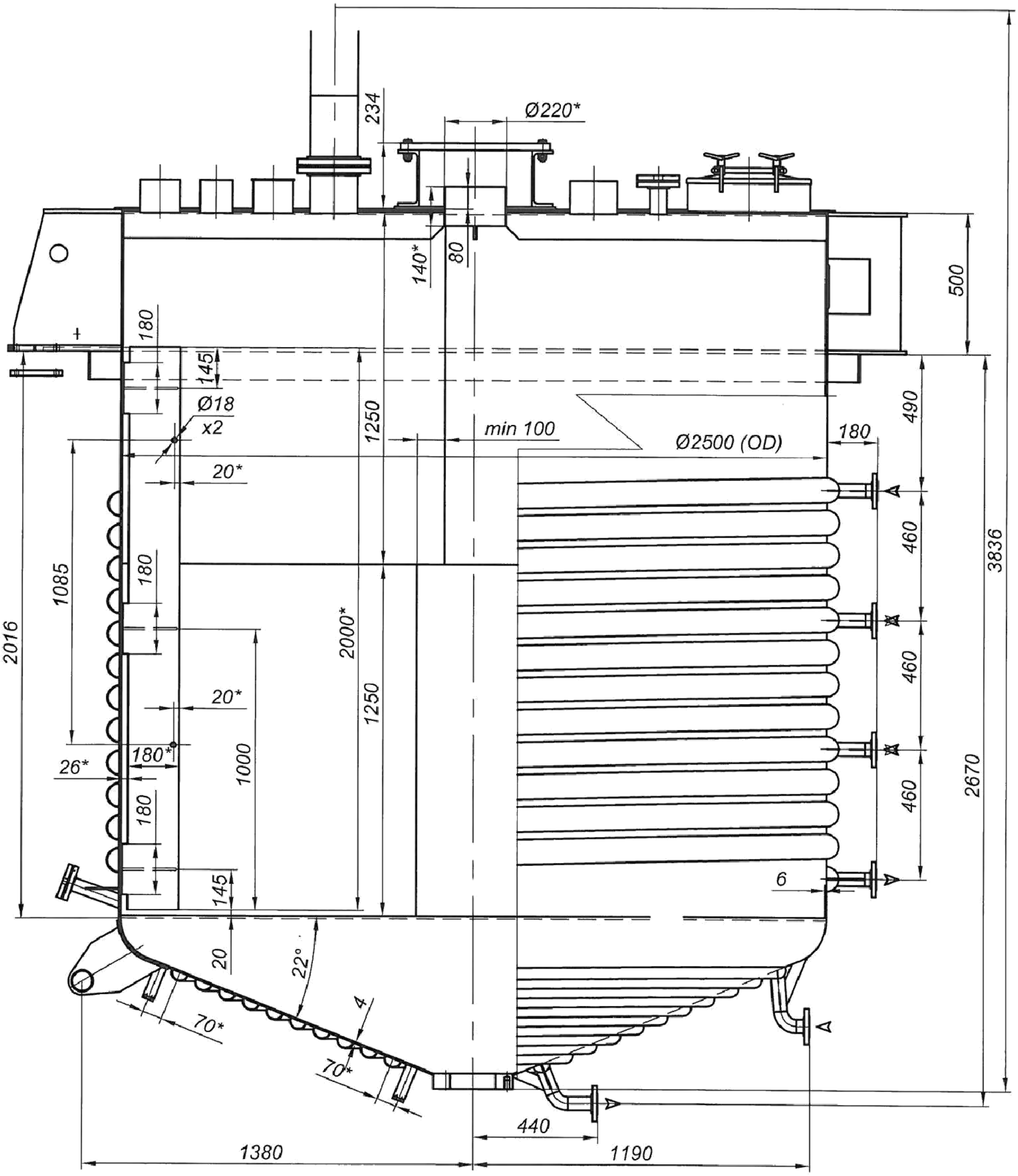

Pak.	Lapas	Dokum. Nr.	Parašas	Data		Raidė	Masė	Mastelis
Projektavo		Gubskis						1:20
T.kontrolė						Lapas		Lapų 1
N.kontrolė						Astra		

Drawing

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3D View of Vessel (alter by using the Save User Specified View command)
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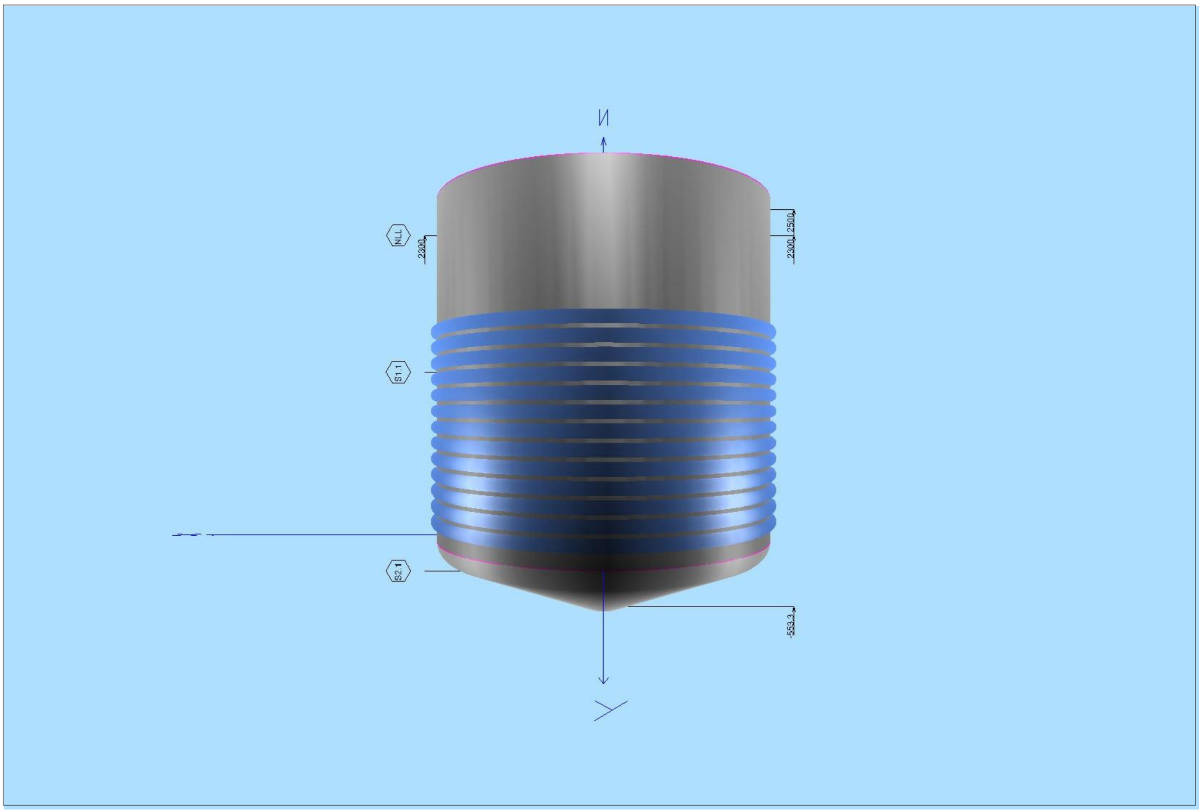

History of Revisions

Rev	ID	Component Type	Comp. Description	DATE & TIME
A	HP.1	Half Pipe Jacket	Jacket on cylindrical shell	15 Apr. 2015 14:57
A	S1.1	Cylindrical Shell	Main Shell	15 Apr. 2015 14:55
A	S2.1	Conical Shell	Conical bottom	15 Apr. 2015 14:55

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A  First Issue
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Design Data & Process Information

Description	Units	Design Data
Process Card		General Design Data
Design Code & Specifications		ASME VIII Div.1
Internal Design Pressure (MPa)	MPa	0.05
External Design Pressure (MPa)	MPa	0
Hydrotest Pressure (MPa)	MPa	
Maximum Design Temperature ('C)	'C	164
Minimum Design Temperature ('C)	'C	0
Operating Temperature ('C)	'C	
Corrosion Allowance (mm)	mm	0
Content of Vessel		
Specific Density of Oper.Liq		1
Normal Liquid Level NLL (mm)	mm	2300

Weight & Volume of Vessel

ID	No.	Wt-UnFinish.	Wt-Finished	Tot.Volume	Test.Liq.Wt	Oper.Liq.Wt
HP.1	1	305.0 kg	305.0 kg	0.254 m3	254.0 kg	253.7 kg
S1.1	1	923.0 kg	923.0 kg	12.160 m3	12160.0 kg	11182.0 kg
S2.1	1	251.0 kg	251.0 kg	0.996 m3	996.0 kg	995.8 kg
Total	**3**	**1479.0 kg**	**1479.0 kg**	**13.410 m3**	**13410.0 kg**	**12431.5 kg**

Weight Summary/Condition	Weights
Empty Weight of Vessel incl. 5% Contingency	1553 kg / 1.6 Tons
Total Test Weight of Vessel (Testing with Water)	14963 kg / 15.0 Tons
Total Operating Weight of Vessel	13984 kg / 14.0 Tons

Center of Gravity

ID	X-Empty	Y-Empty	Z-Empty	X-Test	Y-Test	Z-Test	X-Oper	Y-Oper	Z-Oper
HP.1	0	0	836	0	0	836	0	0	836
S1.1	0	0	1250	0	0	1250	0	0	1150
S2.1	0	0	-201	0	0	-201	0	0	-201

CENTER OF GRAVITY AT CONDITIONS BELOW	X	Y	Z
Empty Vessel	0	0	918
Test Condition of Vessel (Testing with Water)	0	0	1113
Operating Condition of Vessel	0	0	1023

Max. Allowable Pressure MAWP

ID	Comp. Type	Description	Liq.Head	MAWP New & Cold	MAWP Hot & Corr.
S1.1	Cylindrical Shell	Main Shell	0.023 MPa	0.345 MPa	0.341 MPa
S2.1	Conical Shell	Conical bottom	0.023 MPa	0.127 MPa	0.125 MPa
	MAWP			**0.127 MPa**	**0.125 MPa**

```
Note : Other components may limit the MAWP than the ones checked above.
Note : The value for MAWP is at top of vessel, with static liquid head subtracted.
```

Test Pressure

UG-99(b) REQUIRED MINIMUM TEST PRESSURE.
TEST PRESSURE OF VESSEL - NEW & COLD -

```
Design Pressure........................:     0.050 MPa
Design Temperature.....................:   164.0 C
```

ID	Description	Samb	Sdes	Samb/Sdes	Pd	Pt	PtMax
S1.1	Cylindrical Shell-Main Shell	115.0	113.6	1.01	0.050	0.066	0.699
S2.1	Conical Shell-Conical bottom	115.0	113.6	1.01	0.050	0.066	0.284
S2.1	Conical Shell-Conical bottom	115.0	113.6	1.01	0.050	0.066	0.284

HYDRO-TEST

REQUIRED TEST PRESSURE AT TOP OF VESSEL PtReq(Hydro Test) ......: 0.0658 MPa
MAXIMUM TEST PRESSURE AT TOP OF VESSEL PtLim(Hydro Test) ....: 0.2837 MPa

Note : Other components may limit Ptlim than the ones checked above.

NOMENCLATURE:
Samb - is the allowable stress at room temperature.
Sdes - is the allowable stress at design temperature.
SR - is the Stress Ratio SR = Samb / Sdes.
LSR - is the Lowest Stress Ratio for the materials of which the vessel is constructed.
PtMax - is the maximum allowed test pressure determined for the part under consideration.
Pt - is the required test pressure determined for the part under consideration Pt = 1.3 * Pd * SR.
PtReq - is the required minimum test pressure PtReq = 1.3 * Pd * LSR.
PtLim - is the maximum allowed test pressure (minimum value for PtMax) for the listed components.

Bill of Materials

ID	No	Description	Component Dimensions	Material Standard
HP.1	1	Half Pipe Jacket-Jacket on cylindrical shell	3" dc= 88.9, ecb= 3.05, L= 94247.8, Pitch= 115	ID 2, SA-312(M) Gr.TP304L, S30403 Smls. & wld. pipe, PNo=8
S1.1	1	Cylindrical Shell-Main Shell	Do= 2500, t= 6, L= 2500	ID 1, SA-240(M) Gr.304L, S30403 Plate, PNo=8
S2.1	1	Conical Shell-Conical bottom	DiL= 2488, DiS= 250, Lc= 553.28, t= 6, rL= 150	ID 1, SA-240(M) Gr.304L, S30403 Plate, PNo=8

Notes, Warning & Error Messages

ID & Comp. Description	Notes/Warnings/Error Messages
HP.1 Half Pipe Jacket Jacket on cylindrical shell	
-	NOTE: For cyclic service a full penetration groove weld is recommended.

TOTAL No. OF ERRORS/WARNINGS : 0

Maximum Component Utilization - Umax

ID	Comp.Type	Umax(%)	Limited by
HP.1	Half Pipe Jacket	41.2%	MAWPHC (shell & jacket)
S1.1	Cylindrical Shell	43.8%	Shell - Min.thickness to UG-16
S2.1	Conical Shell	54.9%	Internal Pressure Large End

Component with highest utilization Umax = 54.9% S2.1 Conical bottom

Average utilization of all components Umean= 46.6%

ID	Material Name	Temp	ST	SY	SYd	S_d	Sr	ftest	E-mod	Note
1	SA-240(M) Gr.304L, S30403 Plate, PNo=8 , SG=7.85	164	485	170	128.6	113.6	115	153	182478	G5,G21,T4
2	SA-312(M) Gr.TP304L, S30403 Smls. & wld. pipe, PNo=8 , SG=7.85	164	485	170	128.6	113.6	115	153	182478	G5,G21,T4,W12,W14

Notation:
Thickness in mm, stress in N/mm2, temperature in deg.C

TG   : Test Group 1 to 4
Max.T: Maximum thickness for this stress set, 0 or 999 = No limit specified
S/C : CS = Carbon Steel, SS = Stainless Steel
SG   : SG = Specific Gravity (Water = 1.0)
ST   : MIN.TENSILE STRENGTH at room temp.
SY   : MIN. YIELD STRENGTH at room temp.
SYd : MIN. YIELD STRENGTH at calc.temp.
S_d : DESIGN STRESS at calc.temp.
Sr   : DESIGN STRESS at room temp.

Note : G5 = Due to the relatively low yield strength of these materials, these higher stress values were established at temperatures where the shorttime tensile properties govern to permit the use of these alloys where slightly greater deformation is acceptable. The stress values in this range exce
ed 662/3% but do not exceed 90% of the yield strength at temperature. Use of these stresses may result in dimensional changes due to permanent strain. These stress values are not recommended for the flanges of gasketed joints or other applications where slight amounts of distortion can cause leakage
or malfunction. For Section III applications, Table Y-2 lists multiplying factors that, when applied to the yield strength values shown in Table Y-1, will give allowable stress values that will result in lower levels of permanent strain.

Note : G21 = For Section I, use is limited to PEB-5.3. See PG-5.5 for cautionary note.

Note : T4 = Allowable stresses for temperatures of 480°C and above are values obtained from time-dependent properties.

Note : W12 = These S values do not include a longitudinal weld efficiency factor. For Section III applications, for materials welded without filler metal, ultrasonic examination, radiographic examination, or eddy current examination, in accordance with NC-2550, shall provide a longitudinal weld eff
iciency factor of 1.00. Materials welded with filler metal meeting the requirements of NC-2560 shall receive a longitudinal weld efficiency factor of 1.00. Other longitudinal weld efficiency factors shall be in accordance with the following: (a) for single butt weld, with filler metal, 0.80; (b) for
single or double butt weld, without filler metal, 0.85; (c) for double butt weld, with filler metal, 0.90; (d) for single or double butt weld, with radiography, 1.00.

Note : W14 = These S values do not include a weld factor. For Section VIII, Division 1, and Section XII applications using welds made without filler metal, the tabulated tensile stress values shall be multiplied by 0.85. For welds made with filler metal, consult UW-12 for Section VIII, Division 1,
or TW-130.4 for Section XII, as applicable.

Comp.Location in Global Coord.System

ID	Comp. Type	X	Y	Z	Teta	Phi	ConnID
HP.1	Half Pipe Jacket	0	0	0	0.0	0.0	S1.1
S1.1	Cylindrical Shell	0	0	0	0.0	0.0	
S2.1	Conical Shell	0	0	0	0.0	0.0	S1.1

The report above shows the location of the connecting point (x, y and z) for each component referenced to the coordinate system of the connecting component (ConnID). The connecting point (x, y and z) is always on the center axis of rotational symmetry for the component under consideration, i.e. the connecting point for a nozzle connected to a cylindrical shell will be at the intersection of the nozzle center axis and the mid thickness of the shell referenced to the shell s coordinate system. In addition the orientation of the the center axis of the component is given by the two angles Teta and Phi, where Teta is the angle between the center axis of the two components and Phi is the orientation in the x-y plane
The basis for the coordinate system used by the software is a right handed coordinate system with the z-axis as the center axis of rotational geometry for the components, and Teta as the Polar Angle and Phi as the Azimuthal Angle

MDMT Minimum Design Metal Temperature

Table :

ID-Description	Material Name	tn(mm)	tg(mm)	Ratio	E(*)	Curve
HP.1 Jacket on cylindrical shell - Shell	SA-240(M) Gr.304L, S30403 Plate, PNo=8	6.0	6.0	0.41	1.00	
S1.1 Main Shell - Shell	SA-240(M) Gr.304L, S30403 Plate, PNo=8	6.0	6.0	0.20	0.70	
S2.1 Conical bottom - LargeShell	SA-240(M) Gr.304L, S30403 Plate, PNo=8	6.0	6.0	0.52	0.70	

Table Continued

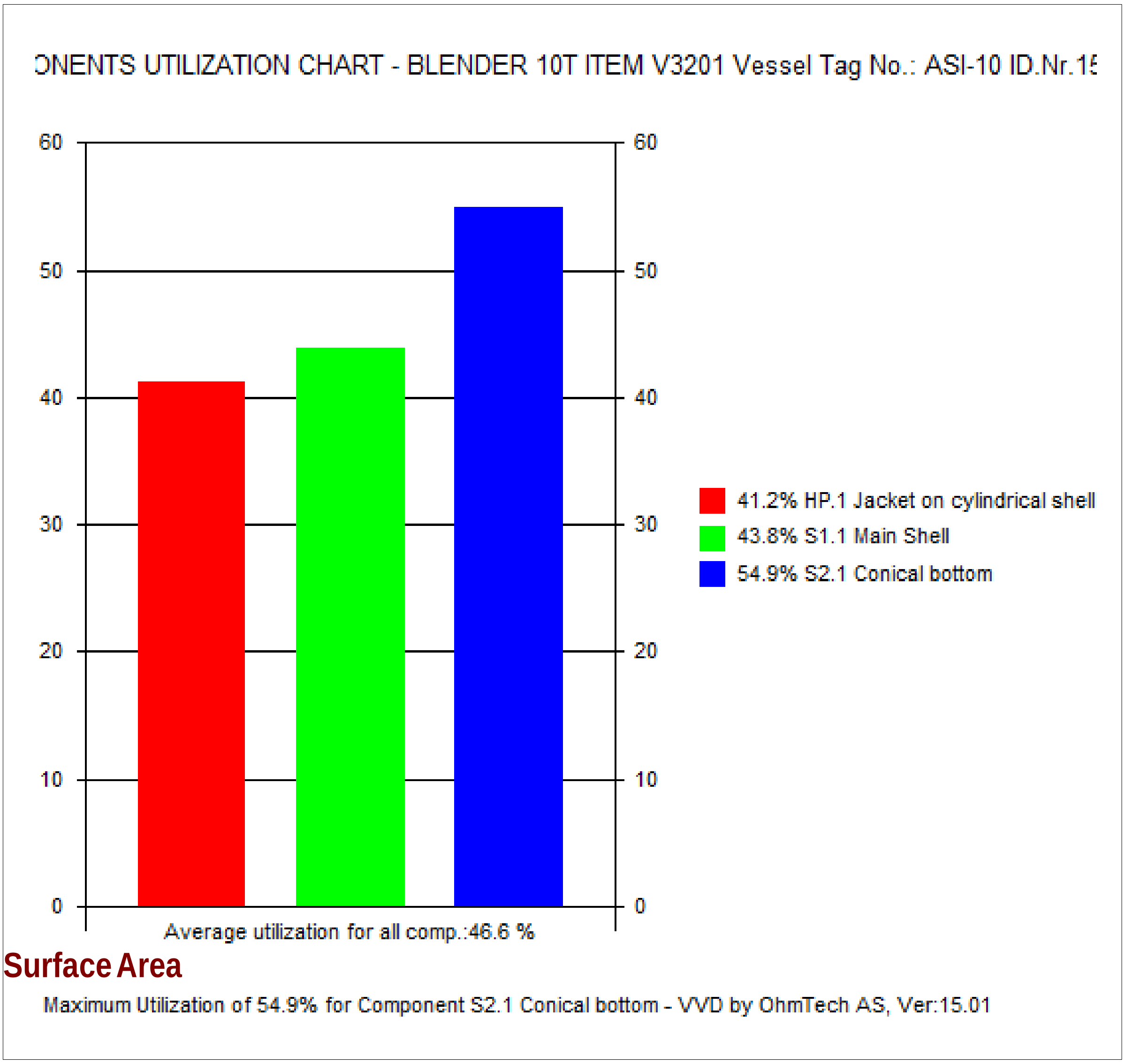
ID-Description	T1(C)	T2(C)	MDMT(C)	Comments
HP.1 Jacket on cylindrical shell - Shell			-196	For thermally treated materials, ref. is made to UHA-51(c)
S1.1 Main Shell - Shell			-	NOTE: UHA-51(g) Material is exempted from impact testing due to low stress. Ratio of design stress to allowable tensile stress is less than 0.35.
S2.1 Conical bottom - LargeShell			-196	For thermally treated materials, ref. is made to UHA-51(c)

```
MDMT CALCULATIONS PER UCS-66, UG-20(f), UHA-51 and Appendix JJ
MDMT Required        :   0.0 C
MDMT Lowest Allowable: -196 C

NOMENCLATURE:
  tn   - Nominal thickness of component under consideration(including corr.
allow.).
  tg   - Governing thickness of component under consideration.
  Ratio- tr*E(*)/(tn-c), utiliztion of component for given process conditions.
  tr   - Required minimum thickness of component at calculation temperature of
MDMT.
  E(*) - Joint efficiency factor, not lower than 0.8.
  Curve- Applicable curve A, B, C or D in Figure UCS-66.
  T1   - Unadjusted MDMT/Lowest allowable temperature for given part, value taken
from Figure UCS-66 based on curve A, B, C or D.
  T2   - Reduction in MDMT without impact testing per Figure UCS-66.1.

NOTES:
  UCS-68(c) If postweld heat treatment is performed when it is not otherwise a
requirement, a 17C reduction in impact test exemption temp. may be given to the
min. permissible temp. for P.no.1 materials.
  The maximum general primary stress in the pads are conservatively assumed to be
the same as that in the corroded shell.
  Note: For austenitic stainless steels, reference is made to UHA-51(c) for
thermally treated materials.
```

NOTE: LOWEST MDMT = -196 C (Warmest Value)



Surface Area

ID	No.	Description	Area Outside(m2)	Area Inside(m2)
HP.1	1	Half Pipe Jacket, Jacket on cylindrical shell	0.000	0.000
S1.1	1	Cylindrical Shell, Main Shell	19.635	19.545
S2.1	1	Conical Shell, Conical bottom	5.416	5.369
Total	**3**		**25.051**	**24.914**

INPUT DATA

COMPONENT ATTACHMENT/LOCATION

GENERAL DESIGN DATA

```
PRESSURE LOADING: Design Component for Internal Pressure Only
PROCESS CARD:
General Design Data : Temp= 164°C, P=0.0500 MPa, c=0.0 mm, Pext=0.0000 MPa
SPECIFIC DENSITY OF OPERATING LIQUID................:SG        1.0000
LIQUID HEAD.........................................:LH       2300.00 mm
```

SHELL DATA

```
CYLINDER FABRICATION: Plate Material
DIAMETER INPUT: Base Design on Shell Outside Diameter
SA-240(M) Gr.304L, S30403 Plate, PNo=8 164'C
ST=485 SY=170 SYd=128.64 S=113.6 Sr=115 Stest=153  (N/mm2)
WELD JOINT EFFICIENCY FACTOR: None RT UW-11(c) Type 1 (E=0.7)
OUTSIDE DIAMETER OF SHELL...........................:Do       2500.00 mm
LENGTH OF CYLINDRICAL PART OF SHELL.................:Lcyl     2500.00 mm
NOMINAL WALL THICKNESS (uncorroded).................:tn        6.0000 mm
NEGATIVE TOLERANCE/THINNING ALLOWANCE...............:th        0.3000 mm
Calculate minimum shell thickness due to internal pressure at different elevations
with steps of 1000 mm.: NO
Split shell into several shell courses and include welding information: NO
```

CALCULATION DATA

UG-27-CYLINDRICAL SHELLS UNDER INTERNAL PRESSURE

```
Outside Radius of Shell
Ro = Do / 2 =2500/2=                                        1250.00 mm
```

*»Thin Cylinder Check P=0.0726 <= 0.385 * S * E=30.62[MPa] «» OK«*

```
Required Minimum Shell Thickness Excl.Allow. tmin :
tmin = P * Ro / (S * E + 0.4 * P)                              (APP.1-1 (1))
=0.0726*1250/(113.6*0.7+0.4*0.0726)=                        1.1408 mm
```

*»Thin Cylinder Check tmin=1.14 < 0.5 * R=625[mm] « » OK«*

```
Required Minimum Shell Thickness Incl.Allow. :
tmina = tmin + c + NegDev =1.14+0+0.3=                       1.4408 mm

Analysis Thickness
ta = tn - c - NegDev =6-0-0.3=                               5.7000 mm
```

Internal Pressure tmina=1.44 <= tn=6[mm]	24.0%	OK
Shell - Min.thickness to UG-16 Thk=5.7 >= UG-16(b)(4)(2.5mm)=2.5[mm]	43.8%	OK

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP:

```
Inside Diameter of Shell
Di = Do - 2 * ta =2500-2*5.7=                               2488.60 mm
Inside Radius of Shell
R = Di / 2 =2488.6/2=                                       1244.30 mm
MAWP HOT & CORR. (Corroded condition at design temp.)
MAWPHC = S * E * ta / (R + 0.6 * ta)
=113.6*0.7*5.7/(1244.3+0.6*5.7)=                            0.3633 MPa

MAWP NEW & COLD (Uncorroded condition at ambient temp.)
MAWPNC = Sr * E * (ta + c) / (R - c + 0.6 * (ta + c))
=115*0.7*(5.7+0)/(1244.3-0+0.6*(5.7+0))=                    0.3678 MPa
```

ASME VIII Div.1:2013 - UG-27 CYLINDRICAL SHELL
S1.1 Main Shell

MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)

```
MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)
Ptmax = SYtest * Etest * (ta + c) / (R + 0.6 * (ta + c))
=153*1*(5.7+0)/(1244.3+0.6*(5.7+0))=                    0.6990 MPa
```

UG-99(b) REQUIRED MINIMUM TEST PRESSURE: NEW AT AMBIENT TEMP. Ptmin

```
Ptmin = 1.3 * Pd * Sr / S =1.3*0.05*115/113.6=           0.0658 MPa
```

Test Pressure Ptmin=0.0658 <= Ptmax=0.699[MPa]	9.4%	OK

CALCULATION SUMMARY

UG-27-CYLINDRICAL SHELLS UNDER INTERNAL PRESSURE

```
Required Minimum Shell Thickness Excl.Allow. tmin :
tmin = P * Ro / (S * E + 0.4 * P)                                (APP.1-1 (1))
=0.0726*1250/(113.6*0.7+0.4*0.0726)=                    1.1408 mm
Required Minimum Shell Thickness Incl.Allow. :
tmina = tmin + c + NegDev =1.14+0+0.3=                   1.4408 mm
```

Internal Pressure tmina=1.44 <= tn=6[mm]	24.0%	OK
Shell - Min.thickness to UG-16 Thk=5.7 >= UG-16(b)(4)(2.5mm)=2.5[mm]	43.8%	OK

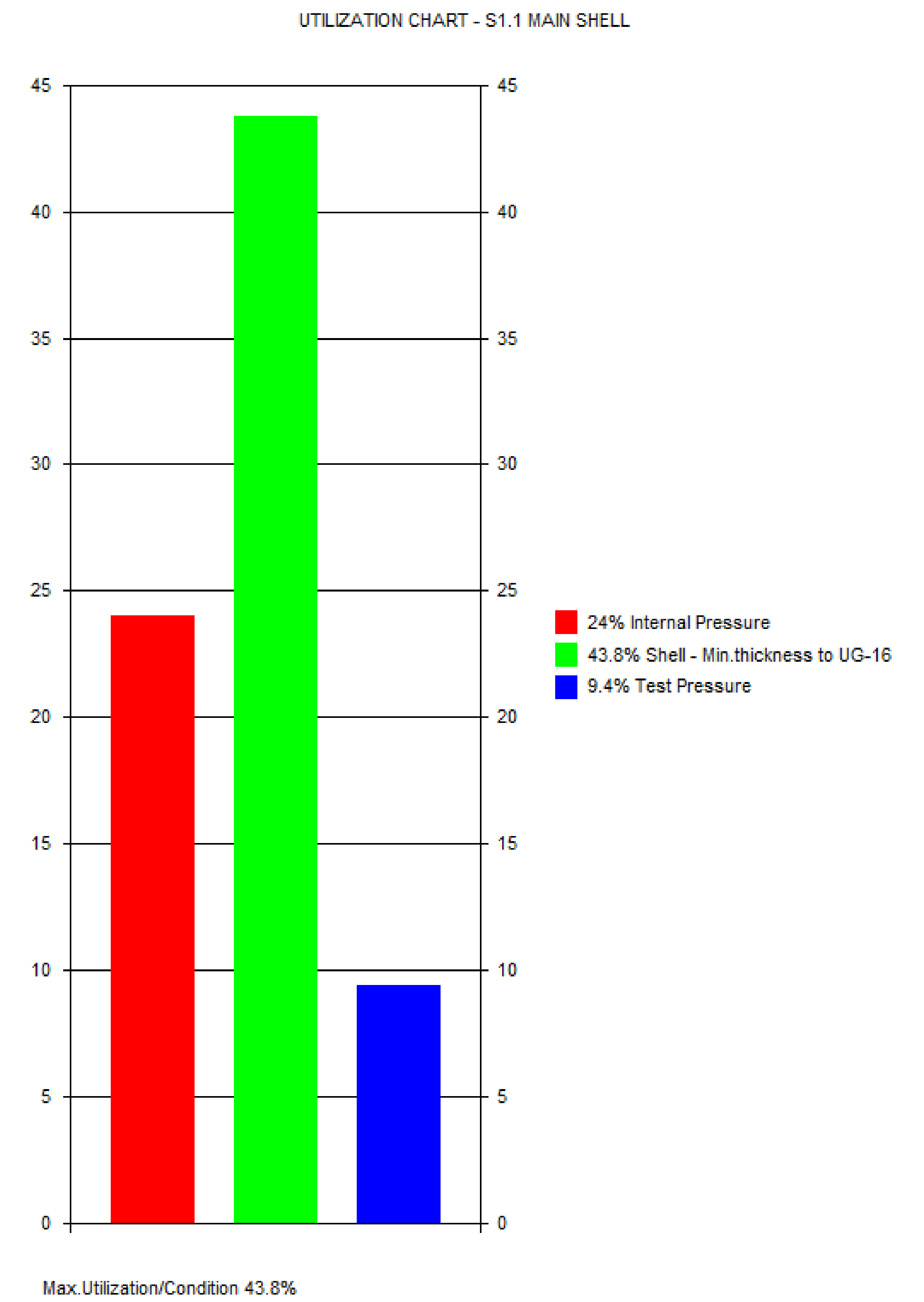
```
MAWP HOT & CORR. (Corroded condition at design temp.)
MAWPHC = S * E * ta / (R + 0.6 * ta)
=113.6*0.7*5.7/(1244.3+0.6*5.7)=                         0.3633 MPa
MAWP NEW & COLD (Uncorroded condition at ambient temp.)
MAWPNC = Sr * E * (ta + c) / (R - c + 0.6 * (ta + c))
=115*0.7*(5.7+0)/(1244.3-0+0.6*(5.7+0))=                 0.3678 MPa
```

MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)

```
MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)
Ptmax = SYtest * Etest * (ta + c) / (R + 0.6 * (ta + c))
=153*1*(5.7+0)/(1244.3+0.6*(5.7+0))=                    0.6990 MPa
```

Test Pressure Ptmin=0.0658 <= Ptmax=0.699[MPa]	9.4%	OK

Volume:12.16 m3 Weight:922.6 kg (SG=7.85)


UTILIZATION CHART - S1.1 MAIN SHELL
45
40
35
30
25
20
15
10
5
0
24% Internal Pressure
43.8% Shell - Min.thickness to UG-16
9.4% Test Pressure
Max.Utilization/Condition 43.8%

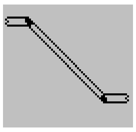
INPUT DATA

COMPONENT ATTACHMENT/LOCATION

```
Attachment: S1.1    Cylindrical Shell    Main Shell
Location: Along z-axis zo=  0
```

GENERAL DESIGN DATA

```
PRESSURE LOADING: Design Component for Internal Pressure Only
PROCESS CARD:
General Design Data : Temp= 164°C, P=0.0500 MPa, c=0.0 mm, Pext=0.0000 MPa
```



```
Type of Transitions: Cylindrical Shells on Both Large & Small Base
SPECIFIC DENSITY OF OPERATING LIQUID................:SG        1.0000
LIQUID HEAD.........................................:LH       2817.20 mm
```

DATA FOR CONE

```
Base of Cone Connecting Cylinder: Connected Cyl.S1.1 is at LARGE base of cone
Specify the Semi-Angle at Apex of the Conical Section
WELD JOINT EFFICIENCY FACTOR: None RT UW-11(c) Type 1 (E=0.7)
SA-240(M) Gr.304L, S30403 Plate, PNo=8 164'C
ST=485 SY=170 SYd=128.64 S=113.6 Sr=115 Stest=153  (N/mm2)
SEMI-ANGLE AT THE APEX OF THE CONICAL SECTION.......:alfa       68.00 degr.
MODULUS OF ELASTICITY at design temp................:E      1,8516E05 N/mm2
NOMINAL THICKNESS OF THE CONE.......................:tn        6.0000 mm
NEGATIVE TOLERANCE/THINNING ALLOWANCE...............:th        0.3000 mm
```

DATA FOR TRANSITION AT LARGE BASE OF CONE

```
This Cone to Cylinder Junction is a Line-of-Support: NO
NOT connected to a cylindrical shell or length of cylinder is less than
2*SQR(RL*ts): NO
Include Knuckle: YES
SA-240(M) Gr.304L, S30403 Plate, PNo=8 164'C
ST=485 SY=170 SYd=128.64 Ss=113.6 fl20=115 fltest=153  (N/mm2)
INSIDE  DIAMETER  AT  LARGE  BASE  OF  CONE(corroded).....:DiL   2488.00  mm
NOMINAL  THK.OF  CYLINDER  AT  LARGE  JUNCTION(uncorr.)..:tnL     6.0000  mm
WELD JOINT EFFICIENCY FACTOR........................:El        0.7000
INSIDE RADIUS OF CURVATURE(large base of cone)......:rL        150.00 mm
NOMINAL THICKNESS OF KNUCKLE(Large End).............:tnLk      6.0000 mm
```

DATA FOR TRANSITION AT SMALL BASE OF CONE

```
This Cone to Cylinder Junction is a Line-of-Support: NO
NOT connected to a cylindrical shell or length of cylinder is less than
2*SQR(RL*ts): NO
Include Knuckle: NO
Include Stiffener Ring at Transition: NO
SA-240(M) Gr.304L, S30403 Plate, PNo=8 164'C
ST=485 SY=170 SYd=128.64 Ss=113.6 fs20=115 fstest=153  (N/mm2)
INSIDE DIAMETER AT SMALL BASE OF CONE(corroded).....:DiS       250.00 mm
NOMINAL THK.OF CYLINDER AT SMALL JUNCTION(uncorr.)..:tnS        15.00 mm
WELD JOINT EFFICIENCY FACTOR........................:Es        0.7000
AXIAL GLOBAL FORCE AT SMALL END OF CONE.............:Ws         0.00 kN
GLOBAL MOMENT AT SMALL END OF CONE..................:Ms         0.00 kNm
```

CALCULATION DATA

UG-32 CONICAL SHELLS - INTERNAL PRESSURE

UG-32(h) - TORICONICAL HEADS AND SECTIONS

```
Inside Diameter of Conical Shell at the Knuckle to Cone Junction Di
Di = DiL - 2 * rL * (1 - Cos( alfa))
=2488-2*150*(1-Cos(68))=                              2300.38 mm
Required Minimum Cone Thickness at Large End Excl.Allow. tmin :
tminL = P * Di / (2 * Cos( alfa) * (S * E - 0.6 * P))          (2)
=0.0777*2300.38/(2*Cos(68)*(113.6*0.7-0.6*0.0777))=     3.0019 mm
Required Minimum Cone Thickness Incl.Allow. tmina :
tmina = tminL + c + th =3.+0+0.3=                       3.3000 mm
```

Internal Pressure Large End tmina=3.3 <= tn=6[mm]	54.9%	OK

```
Analysis Thickness
ta = tn - c - th =6-0-0.3=                                 5.7000 mm
```

CONE - Min.thickness to UG-16 Thk=5.7 >= UG-16(b)(4)(2.5mm)=2.5[mm]	43.8%	OK

```
Required Minimum Cone Thickness at Small End Excl.Allow. tmin :
tminS = P * DiS / (2 * Cos( alfa) * (S * E - 0.6 * P))          (2)
=0.0777*250/(2*Cos(68)*(113.6*0.7-0.6*0.0777))=         0.3260 mm
```

Appendix 1-4(d) Required Thickness of Knuckle at Large End

```
L = Di / (2 * Cos( alfa)) =2300.38/(2*Cos(68))=          3070.40 mm
M = 0.25 * (3 + Sqr( L / rL))
=0.25*(3+Sqr(3070.4/150))=                              1.8811 mm
Required Thickness of Knuckle at Large End Incl.Allow. tminfL :
tminfL = P * L * M / (2 * S * E - 0.2 * P) + c + th
=0.0777*3070.4*1.88/(2*113.6*0.7-0.2*0.0777)+0+0.3=     3.1220 mm
```

Internal Pressure Knuckle Thickness Large End tminfL=3.12 <= tnLk=6[mm]	52.0%	OK

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP :NEW & COLD

```
Large End Cone PmaxL:
PmaxL = 2*S*E*ta*Cos(alfa)/(Di+1.2*ta*Cos(alfa))              ((2))
=2*115*0.7*5.7*Cos(68)/(2300.38+1.2*5.7*Cos(68))=       0.1493 MPa
Large End Knuckle Pmaxk:
Pmaxk = 2 * S * E * ta / (M * L - ta * (M - 0.2))             ((2))
=2*115*0.7*5.7/(1.88*3070.4-5.7*(1.88-0.2))=            0.1592 mm
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.1493,0.1592)=           0.1493 MPa
```

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP :HOT & CORR

```
Large End Cone PmaxL:
PmaxL = 2*S*E*ta*Cos(alfa)/(Di+1.2*ta*Cos(alfa))              ((2))
=2*113.6*0.7*5.7*Cos(68)/(2300.38+1.2*5.7*Cos(68))=     0.1475 MPa
Large End Knuckle Pmaxk:
Pmaxk = 2 * S * E * ta / (M * L - ta * (M - 0.2))             ((2))
=2*113.6*0.7*5.7/(1.88*3070.4-5.7*(1.88-0.2))=          0.1572 mm
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.1475,0.1572)=           0.1475 MPa
```

MAX TEST PRESSURE (Uncorroded cond. at ambient temp.)

```
Large End Cone PmaxL:
PmaxL = 2*S*E*ta*Cos(alfa)/(Di+1.2*ta*Cos(alfa))              ((2))
=2*153*1*5.7*Cos(68)/(2300.38+1.2*5.7*Cos(68))=         0.2837 MPa
Large End Knuckle Pmaxk:
Pmaxk = 2 * S * E * ta / (M * L - ta * (M - 0.2))             ((2))
=2*153*1*5.7/(1.88*3070.4-5.7*(1.88-0.2))=              0.3025 mm
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.2837,0.3025)=           0.2837 MPa
```

UG-99(b) REQUIRED MINIMUM TEST PRESSURE: NEW AT AMBIENT TEMP. Ptmin

```
Ptmin = 1.3 * Pd * Sr / S =1.3*0.05*115/113.6=              0.0658 MPa
```

Test Pressure Ptmin=0.0658 <= Ptmax=0.2837[MPa]	23.1%	OK

```
ts = tnL - c =6-0=                                            6.0000 mm
ts = tnS - c =15-0=                                            15.00 mm
```

APPENDIX 1-5(e) - REINFORCEMENT AREA AT SMALL END

```
RatioS = P / (Ss * Es) =0.0777/(113.6*0.7)=                 9,7711E-04
Delta (Table 1-5.1) = Delta =4=                               4.0000 degr.
Inside Radius of Small Cylinder
Rs = DoS / 2 =280/2=                                          140.00 mm
Since Delta < Alfa Reinforcement Check IS Required.
Axial Load at Small End
QS = P * Rs / 2 + Q1 =0.0777*140/2+0=                         5.4390 N/mm
Required Area at Small End of Cone
ArS = (kS*QS*Rs/(Ss*Es))*(1-Delta/alfa)*Tan(alfa)
=(1*5.439*140/(113.6*0.7))*(1-4/68)*Tan(68)=                  22.31 mm2
Minimum length of cylindrical shell at Small End of Cone
LcylS = 1.4 * SQR( Rs * ts) =1.4*SQR(140*15)=                  64.16 mm
Available Area at Small End of Cone
AeS = 0.78*SQR(Rs*ts)*((ts-tsminS)+(ta-tminS)/Cos(alfa))+AstiffS
=0.78*SQR(140*15)*((15-0.1222)+(5.7-0.326)/Cos(68))+0=     1044.57 mm2
```

Reinf.Area Small End-Int.Pressure AeS=1044.57 >= ArS=22.31[mm2]	2.1%	OK

```
NOTE: Appendix 1-5 Calculations are not required for the large end transition asa
knuckle is present.
```

CALCULATION SUMMARY

UG-32 CONICAL SHELLS - INTERNAL PRESSURE

UG-32(h) - TORICONICAL HEADS AND SECTIONS

```
Required Minimum Cone Thickness Incl.Allow. tmina :
tmina = tminL + c + th =3.+0+0.3=                            3.3000 mm
```

Internal Pressure Large End tmina=3.3 <= tn=6[mm]	54.9%	OK
CONE - Min.thickness to UG-16 Thk=5.7 >= UG-16(b)(4)(2.5mm)=2.5[mm]	43.8%	OK

Appendix 1-4(d) Required Thickness of Knuckle at Large End

```
Required Thickness of Knuckle at Large End Incl.Allow. tminfL :
tminfL = P * L * M / (2 * S * E - 0.2 * P) + c + th
=0.0777*3070.4*1.88/(2*113.6*0.7-0.2*0.0777)+0+0.3=        3.1220 mm
```

Internal Pressure Knuckle Thickness Large End tminfL=3.12 <= tnLk=6[mm]	52.0%	OK

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP :NEW & COLD

```
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.1493,0.1592)=              0.1493 MPa
```

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP :HOT & CORR

```
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.1475,0.1572)=              0.1475 MPa
```

MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)

```
Pmax = MIN( PmaxL, Pmaxk) =MIN(0.2837,0.3025)=              0.2837 MPa
```

Test Pressure Ptmin=0.0658 <= Ptmax=0.2837[MPa]	23.1%	OK

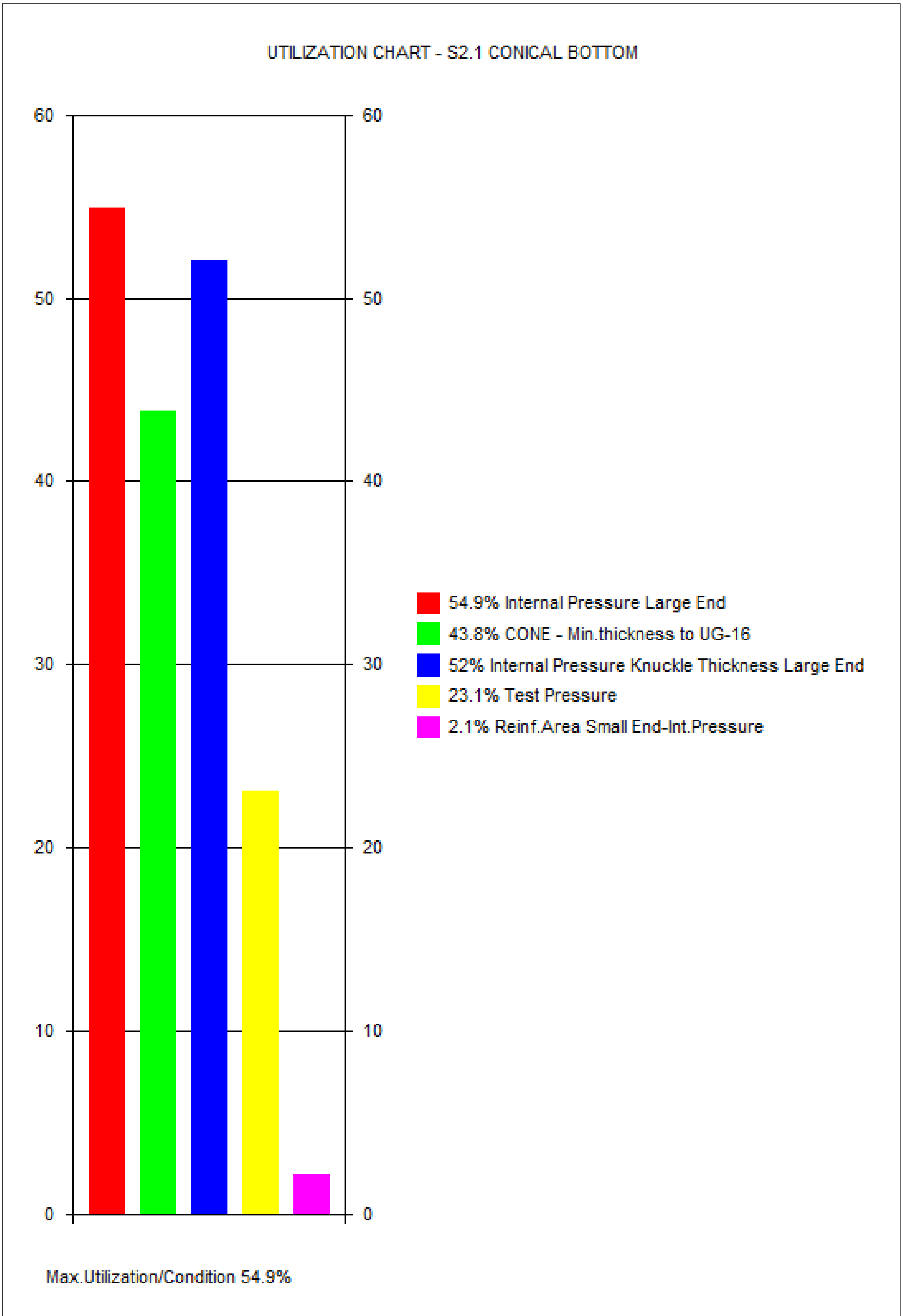
APPENDIX 1-5(e) - REINFORCEMENT AREA AT SMALL END

```
Required Area at Small End of Cone
ArS = (kS*QS*Rs/(Ss*Es))*(1-Delta/alfa)*Tan(alfa)
=(1*5.439*140/(113.6*0.7))*(1-4/68)*Tan(68)=              22.31 mm2

Available Area at Small End of Cone
AeS = 0.78*SQR(Rs*ts)*((ts-tsminS)+(ta-tminS)/Cos(alfa))+AstiffS
=0.78*SQR(140*15)*((15-0.1222)+(5.7-0.326)/Cos(68))+0=     1044.57 mm2
```

Reinf.Area Small End-Int.Pressure AeS=1044.57 >= ArS=22.31[mm2]	2.1%	OK

Volume:0.9958 m3 Weight:250.4 kg (SG=7.85)


UTILIZATION CHART - S2.1 CONICAL BOTTOM
60
50
40
30
20
10
0
54.9% Internal Pressure Large End
43.8% CONE - Min.thickness to UG-16
52% Internal Pressure Knuckle Thickness Large End
23.1% Test Pressure
2.1% Reinf.Area Small End-Int.Pressure
Max.Utilization/Condition 54.9%

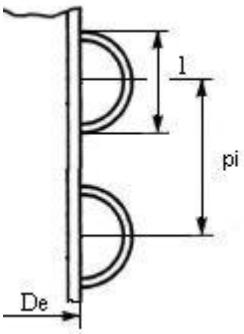
INPUT DATA

SHELL DATA (S1.1)

```
WELD JOINT EFFICIENCY FACTOR: None RT UW-11(c) Type 1 (E=0.7)
DESIGN PRESSURE....................................:Ps         0.0500 MPa
INTERNAL CORROSION ALLOWANCE........................:c           0.00 mm
OUTSIDE DIAMETER OF SHELL...........................:Do       2500.00 mm
NOMINAL  WALL  THICKNESS  (uncorroded)................:tn    6.0000  mm
NEGATIVE TOLERANCE/THINNING ALLOWANCE...............:th   0.3000  mm  SA-
240(M) Gr.304L, S30403 Plate, PNo=8 164'C
ST=485 SY=170 SYd=128.64 Ss=113.6 Ss20=115 Sstest=153  (N/mm2)
```

DATA FOR HALF PIPE

```
SA-312(M) Gr.TP304L, S30403 Smls. & wld. pipe, PNo=8 164'C
ST=485 SY=170 SYd=128.64 Sc=113.6 Sc20=115 Sctest=153  (N/mm2)
HALF PIPE/LIMPET COIL INTERNAL DESIGN PRESSURE......:P         0.6000 MPa
CORROSION ALLOWANCE FOR HALF PIPE/LIMPET COIL.......:cc         0.00 mm
OUTSIDE DIAMETER OF HALF PIPE/LIMPET COIL...........:dc         88.90 mm
Nominal Size of Pipe: 3"
Comment (Optional):
NOMINAL THICKNESS OF HALF PIPE/LIMPET COIL(uncorroded):tcb     3.0500 mm
NEGATIVE DEVIATION/TOLERANCE........................:negT       12.50 %
HALF PIPE/LIMPET COIL ATTACHMENT LENGTH.............:l          88.90 mm
TOTAL SPAN OF COILS/LENGTH ALONG SHELL..............:L        1380.00 mm
PITCH SPACING.......................................:pi        115.00 mm
```



CALCULATION DATA

APPENDIX EE, HALF-PIPE JACKETS

```
Inside Radius of Jacket
r = 0.5 * (dc - 2 * (tcb - cc))
=0.5*(88.9-2*(3.05-0))=                                   41.40 mm
Minimum thickness of half pipe/limpet coil(excl.allow.), tcmin
tcmin = P * r / (0.85 * Sc - 0.6 * P)
=0.6*41.4/(0.85*113.6-0.6*0.6)=                          0.2582 mm
Required thickness of half pipe/limpet coil(Incl.Allow.) :
tcmina = (tcmin + cc) / (1 - negT / 100)
=(0.2582+0)/(1-12.5/100)=                                 0.2951 mm
```

Half Pipe/Coil Internal Pressure tcmina=0.2951 <= tcb=3.05[mm]	9.6%	OK

MAXIMUM ALLOWABLE WORKING PRESSURE MAWP:

```
Analysis Thickness of Half Pipe/Limpet Coil, eca
tca = tcb - cc - NegDev =3.05-0-0.3813=                    2.6688 mm

Shell Analysis Thickness
ta = tn - c - th =6-0-0.3=                                 5.7000 mm
```

```
Inside Radius of Shell
R = 0.5 * (Do - 2 * (tn - c)) =0.5*(2500-2*(6-0))=          1244.00 mm
K-Factor for corroded condition
From Figure EE-2; K= 113.4
K-Factor for new/uncorroded condition
From Figure EE-2; K= 113.4
PJacketHC = 0.85 * Sc * tca / (r - 0.6 * tca)
=0.85*113.6*2.67/(41.4-0.6*2.67)=                          6.4749 MPa
PShellHC = (1.5 * Ss - Ps * R / (2 * ta)) / K
=(1.5*113.6-0.05*1244/(2*5.7))/113.4=                      1.4545 MPa
MAWP HOT & CORR. (Corroded condition at design temp.)
MAWPHC = MIN( PJacketHC, PShellHC) =MIN(6.47,1.45)=         1.4545 MPa
```

MAWPHC (shell & jacket) P=0.6 <= MAWPHC=1.45[MPa]	41.2%	OK

```
PJacketNC = 0.85*Sc20*(tca+cc)/((r-cc)-0.6*(tca+cc))
=0.85*115*(2.67+0)/((41.4-0)-0.6*(2.67+0))=                6.5547 MPa
PShellNC = (1.5 * Ss20 - Ps * (R - c) / (2 * (ta + c))) / K
=(1.5*115-0.05*(1244-0)/(2*(5.7+0)))/113.4=                1.4730 MPa
MAWP NEW & COLD (Uncorroded condition at ambient temp.)
MAWPNC = MIN( PJacketNC, PShellNC) =MIN(6.55,1.47)=         1.4730 MPa
```

MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)

```
PJacketNC = 0.85*Sctest*(tca+cc)/((r-cc)-0.6*(tca+cc))
=0.85*153*(2.67+0)/((41.4-0)-0.6*(2.67+0))=                8.7206 MPa
PtShell = 1.5 * Sstest / K =1.5*153/113.4=                   2.0238 MPa
Ptmax = MIN( PtJacket, PtShell) =MIN(8.72,2.02)=            2.0238 MPa
```

UG-99(b) REQUIRED MINIMUM TEST PRESSURE: NEW AT AMBIENT TEMP. Ptmin

```
Ptmin = 1.3 * Pd * Sr / S =1.3*0.6*115/113.6=               0.7896 MPa
```

Test Pressure Ptmin=0.7896 <= Ptmax=2.02[MPa]	39.0%	OK

CALCULATION SUMMARY

APPENDIX EE, HALF-PIPE JACKETS

```
Required thickness of half pipe/limpet coil(Incl.Allow.) :
tcmina = (tcmin + cc) / (1 - negT / 100)
=(0.2582+0)/(1-12.5/100)=                                  0.2951 mm
```

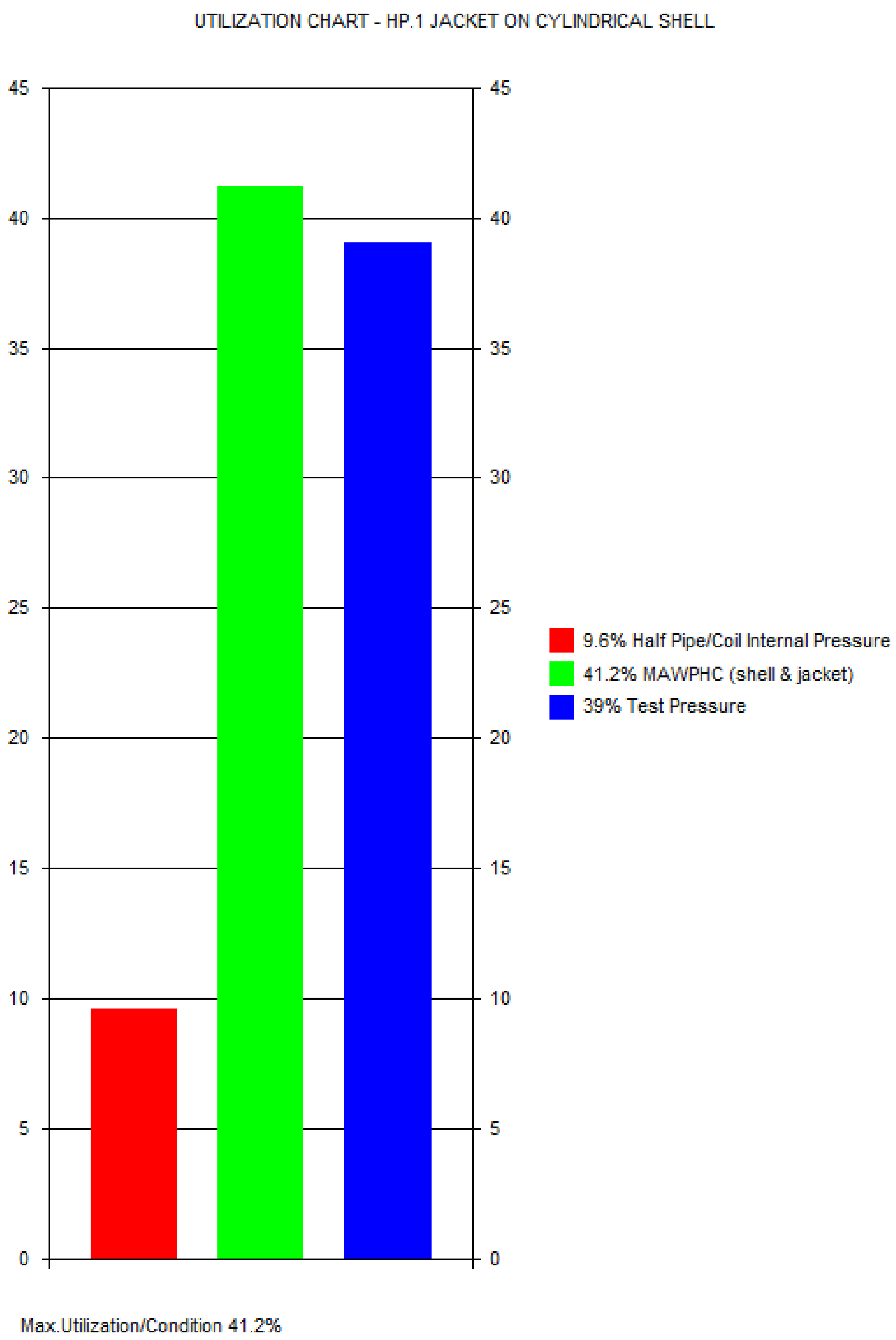
Half Pipe/Coil Internal Pressure tcmina=0.2951 <= tcb=3.05[mm]	9.6%	OK
MAWPHC (shell & jacket) P=0.6 <= MAWPHC=1.45[MPa]	41.2%	OK

MAX TEST PRESSURE (Uncorroded cond.at ambient temp.)

```
Ptmax = MIN( PtJacket, PtShell) =MIN(8.72,2.02)=            2.0238 MPa
```

Test Pressure Ptmin=0.7896 <= Ptmax=2.02[MPa]	39.0%	OK

Volume:0.2537 m3 Weight:304.3 kg (SG=7.85)


UTILIZATION CHART - HP.1 JACKET ON CYLINDRICAL SHELL
45
40
35
30
25
20
15
10
5
0
9.6% Half Pipe/Coil Internal Pressure
41.2% MAWPHC (shell & jacket)
39% Test Pressure
Max.Utilization/Condition 41.2%

outokumpu
stainless steel & high performance alloys

CERTIFICATE –ZEUGNIS - CERTIFICAT

EN 10204-3.1
2245253-EN

Invoice No. / Rechnung Nr. / N° du certificat	Page / Seite / Page
6610/1000256952	1/1

Business Unit / QCM	Date Datum Date	Load, Ladung, Charge No	Acknowledged ID, Bestätigung, Commande ID
Avesta Works / Johan Nordström	10-Apr-2014	PL/103162	6610/300289182

Your ref, Ihre Ref., Votre ref
s/100400/024/2014.

Buyer, Besteller, Acheteur
NOVA TRADING S.A.
UL. STAROTORUNSKA 5
PL 87-100, TORUN
POLAND

Consignee, Empfänger, Lieu de livraison
NOVA TRADING S.A.

Requirements, Anforderungen, Exigences
ASTM A 240M-13c
ASME SEC II PART A SA-240/SA-240M 2013
EN 10088-2:2005
AD 2000 W2, W10 & EN 10028-7
EN 10088-4:2009
EN ISO 9445-2

Mark of Manufacturer / Zeichen des Lieferwerkes / Signe de producteur	Process / Erschmelzungsart / Mode de fusion	Inspector's stamp / Zeichen des Sachverständigen / Poinçon de l'expert	Grade, Werkstoff, Nuance
outokumpu	E+AOD	AJA	Outokumpu 18-8L TYPE 304 TYPE 304L 1.4301 1.4307

Product, Erzeugnisform, Produit
Stainless Steel Cold Rolled, Coil
finish 2B, cut edge

Line / Reihe / Ligne	Item / Position / Poste	Heat-Lot No / Schmelze-Lot Nr / Coulée n° - Lot No	Size / Abmessungen / Dimensions	Pieces / Stuckzahl / Nombre	Quantity / Unit / Menge / Einheit / Quantité / Unité
1	4	441034-003	4,00 X 2000 mm	1	7330 KG

Chemical composition – Chemische Zusammensetzung – Composition chimiques

	C	Si	Mn	P	S	Cr	Ni	Nb	Cu	Co	N
Heat	.020	.32	1.57	.038	.001	18.30	8.14	.008	.38	.200	.061

Radioactive contamination check acc. IAEA recommendations: Approved

Test results – Prüfergebnisse – Results dessais (1N/mm² = 1 MPa) F = Front – Anfan – Debut B = Back – Ende – Fin T = Transverse – Quer – Travers

	Test	Ref	Temp	RP 0.2	RP 1.0	RM	A5	2"	HB
			°C	N/MM2	N/MM2	N/MM2	%	%	HB
Min			+20	230	260	540	45	40	
Max						700			201
	F	T	+20	330	368	643	55	53	178
	B	T		321	359	639	55	53	170

Corrosion acc. ASTM A 262-E, EN ISO 3651-2A: Approved
Heat treatment / Solution annealed: Material temperature 1100 °C / Quenched (forced air + water)
Steel grade verification (PMI-spectroscopic): Approved
Marking, visual insp. and gauge measurement: Approved
Approved acc. AD 2000 Merkblatt W0 by TÜV NORD Systems with renounce of countersignment
Certified acc. Pressure Equipment Directive (97/23/EC) by TÜV CERT-Certification body
for pressure equipment of the TÜV NORD Systems; notified body, reg-no. 0045.

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SWEDEN
Regoffice: Stockholm SWEDEN, Regno: 556001-8748

Telephone: + 46 (0)226 811 73
Fax: + 46 (0)226 816 46
V.A.T no: SE556001874801

This material is found to comply with order requirements

Joakim Johansson
Authorized Inspector

DNV
EN ISO 9001
EN ISO 14001

CE 0045
Certificate no.: 0045-CPD-0669

Appendix 4 List of used equipment and devices

No.	Title	Producers' serial No.	Measurement limits and accuracy	Enterprise of verification (calibration)
1.	Universal 25 t tension – compression machine	Nr.326	25kN±10 N	State Standard and Calibration Laboratory "Kaunas Metrology Center". Calibration certificate Nr. MJ-605
2.	Dynamometer - caliper DOSM 20-1 Indicator	Nr. 508 Nr. 483720	10000 ±5 N	State Standard and Calibration Laboratory "Vilnius Metrology Center". Calibration certificate Nr. 993791- J-0.1-193
3.	Micrometer, type I	Nr. 7547	25- 50 ±0.01 mm	State Standard and Calibration Laboratory "Kaunas Metrology Center". Verification certificate Nr. 1267159
4.	Picoscope 3204 PC Oscilloscope	480-6969	0-20 V Resolution 8 bits/30%	PICO Technology Limited, Great Britain
5.	Laptop „ATOMIC Action L51	10-1685-010820	B kl.	MOBILE INC, China
7.	Frequency converter SKB3400150 STD	209100256 4	0.25 -7.5 kW	EMERSON Industrial Automation, China
8.	Extensometer 3542-025M-050-HT2	E85974	25 mm +12.5/-2.5 mm	Epsilon Technology Corp., USA, Nr.1113246

Appendix 5 Dimensional check

Matmenys / Dimension, mm							
	Išmatuoti Measured	Nomin. Nominal	Nukrypim. Deflection		Išmatuoti Measured	Nomin. Nominal	Nukrypim. Deflection
$ØD_{out}$	2501	2500	+1	C	1061	1065	-4
H	498	500	-2	C1	435	440	-5
H1	582	599	-17	C2	1194	1190	+4
H2	225	234	-9	C3	1375	1380	-5
L	1996	2016	-20	X	529	530	-1
L1	2494	2516	-22	X1	450	450	0
L2	2650	2670	-20	X2	550	550	0
L3	3792	3836	-44	X3	649	650	-1
F	98:100	100	-1	X4	969	970	-1
F1	146	150	-4	Y	229	228	+1
F2	77	80	-3	Y1	499	500	-1
F3	178:182	180	-2:+2	Y2	497	500	-3
E	354	374	-20	Y3	228	228	0
E1	654	654	0	Y4	438	440	-2
E2	150	146	+4	Y5	246	248	-2
E3	460	460	0				
E4	460	460	0				
E5	462	460	+2				
E6	465	490	-35				

Išilginių sudurtinių suvirinimo siūlių išgaubtumas (įgaubtumas)/ Peaking on longitudinal butt welds of outer cylinder			
Indo išorėje / Outside cylinder			
Išgaubtumas / Outwards peaking			Įgaubtumas / Inwards peaking P
P1	P2	P=0,25(P1+P2)	
2	1	0,75	
3	2	1,25	
4	2	1,5	
3	3	1,5	
3	2	1,25	
3	3	1,5	
2	2	1,0	
2	4	1,5	
3	3	1,5	
3	4	1,75	
2	4	1,50	

Matuojama 20° šablonu intervalu 250MM /
Shall be checked by 20° gauge – intervals 250 mm

Atvamzdių išdėstymas / Nozzle positioning ASI-10.00.000SB					
Atvamzdis/ Nozzle	Brėžinys/ Drawing	Išvada/ Conclusion	Atvamzdis/ Nozzle	Brėžinys/ Drawing	Išvada/ Conclusion
A1	44,4°	OK	T1	305°	OK
A2	90°	OK	W1	130,5°	OK
A3	270°	OK	W2	119,5°	OK
A4	229,5°	OK	W3	275,5°	OK
A5	218,5°	OK	W4	264,5°	OK
A6	207,5°	OK	V1	241,5°	OK
A7	196,5°	OK	V2	202,5°	
A8	185,5°	OK	N1	center	OK
A9	174,5°	OK	NP	345°	OK
A10	163,5°	OK	LL	45°, 135°, 225°, 315°	OK
A11	152,5°	OK			
A12	141,5°	OK	B	60°,180°,300°	OK
A13	25,1°	OK	Sb	60°,180°,300°	OK
B1	center	OK	OG	0°,30°,90°,120°, 150°,210°,240°, 270°,330°	OK
C1 – C4	140°	OK			
S1 – S4	130°	OK	IG	34°, 147°, 214°, 327°	OK
M1	0°	OK			
L1	340,7°	OK			

Išvada / Conclusion ....

Tikrinama 20° šablonų. nelygumų ilgis neturi viršyti 2% šablono ilgio /Shall be checked by 20°gauge.values shall not exceed 2% of gauge length

	Išmatuotas perimetras (išorinis) / Measured perimeter (outside) C_e, mm	Paskaičiuotas išorinis skersmuo / Calculated outside diameter D^p_e, mm $D^p_e=C_e/\pi$	Nominalus išorinis skersmuo/ Nominal outside diameter D_e, mm	Nukrypimas / Deviation, %
I-I	7855	2500,32	2500	0,01
II-II	7856	2500,64	2500	0,03
III-III	7855	2500,32	2500	0,01

	Vidinis skersmuo / Inside diameter, mm	
	Min	Max
I-I	2490	2491
II-II	2488	2491
III-III	2487	2493

[

ASI-10 Korpuso geometriniai apmatavimai / Dimensional check