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Design Optimization of a Low Pressure LNG Fuel Supply System

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MASTER THESIS IN MARINE MACHINERY

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FOR

STUD.TECHN. KIM NGUYEN

DESIGN OPTIMIZATION OF A LOW PRESSURE LNG FUEL SUPPLY SYSTEM

Work description

Liquefied Natural Gas (LNG) is considered to be a promising fuel for marine applications and is gradually penetrating new market segments. Used in marine gas engines, the reduction of CO_2 with 20%, NO_x with up to 85% and SO_x with almost 100% is of cause very interesting from a ship owner's point of view. However, using natural gas requires a more complicated fuel system than normally found in marine gas oil applications. The design and optimization of such systems is far more challenging. Dependent on the natural gas concept of use, the properties of the gas together with the complete fuel system can play a significant role in determining the total capabilities of the propulsion system. Different operating challenges do also play an important role in determining the design criteria for such a fuel system.

A pre-project on the topic was done last semester, fall 2014 by the student. The focus in the pre-project was the use of LNG as fuel, current status of LNG fuel system design for marine application, design criteria and operating challenges. The master thesis will be a continuation of the project thesis and will focus on design and optimization of the low pressure (LP) LNG fuel supply system with focus on the vaporizer and the heater.

It is important to investigate the process parameters and its dependency including the dependency of the parameters of the components in the system to create an optimal system. It is therefore necessary to simulate the fuel supply system to find the proper process parameter and then size the components such that the pressure, temperature and flow to the engines are in the required range.

Modeling and steady state simulations will be carried out by using the commercial simulation tool Aspen HYSYS and EDR (Exchanger Design & Rating) for different system design. The main objective on the study is to evaluate the different system designs and find a favorable design of heat exchangers based on certain constraints.

Scope of work

1. Review the relevant work presented in the pre-project and study relevant literature related to the LP LNG fuel supply system and its components with focus on the heat exchangers and the processes in the system such as regasification and heating.

2. Develop models of the LP fuel supply system using the simulation software Aspen HYSYS. The models should contain relevant mechanical components with relevant process parameters.

3. A sensitivity analysis of the different model shall be conducted by manipulating the process parameters of the heat exchanger and study the significance and dependency of the changes on the process parameters in the system.

4. A sensitivity analysis of a heat exchanger shall be conducted by manipulating some of the mechanical parameters and study the significance and dependency of the changes of the overall mechanical design including thermal and hydraulic parameters.

5. Thermal and hydraulic design calculations of the heat exchangers based on the relevant process parameters shall be conducted by using the additional design software, EDR in Aspen HYSYS.

6. Develop an overall understanding of how process designs, thermal and hydraulic calculations influence the choice of heat exchangers.

The report shall be written in English and edited as a research report including literature survey, description of mathematical models, description of control algorithms, simulations results, discussion and conclusion including a proposal for further work. Source code developed shall be provided on a CD with code listing enclosed in appendix.

The Department of Marine Technology, NTNU, can use the results freely in its research work by referring to the student's work.

The thesis should be submitted in three copies within 5 months after the registered start-up of this thesis.

Advisor: Associate Professor Eilif Pedersen

Deadline: 19.10.2015

Eilif Pedersen
Associate Professor

Preface

This master thesis was written during summer/fall of 2015 at the Department of Marine Technology at the Norwegian University of Science and Technology, NTNU. The thesis is a part of the program Marine Machinery and corresponds 30 credits.

In the process of writing this thesis I have gained an understanding of why LNG can be an interesting and preferable fuel choice, which is very relevant due to current and future emission regulations with the goal of making the world a bit greener. Based on the optimization simulations and analysis performed, I have gained a better understanding of the principles of heat exchangers and the overall system with the goal of saving energy. Working with the simulation tool Aspen HYSYS and EDR proved to be more challenging than I had expected and I often hit on problems.

I would like to thank my supervisor, Eilif Pedersen for providing me relevant papers and for introducing me to a very interesting topic. I would also like to thank PhD candidate, Erlend Liavåg Grotle for his help and guidance. Further, I would like to thank Gasnor for providing me relevant papers. A special thank to Einar for proof reading my thesis, the rest of Office A1.015 and fellow students for all the fun moments and five memorable years – It has been jolly fantastic! And last but not least, I would like to thank Ole Severin for being my knight in shining armors and for supporting me in life. Thank you.

Bergen, October 19, 2015

Kim Nguyen

Kim Nguyen

Summary

In 2014 there were 50 liquefied natural gas (LNG) fuelled ships in operation and around 70 on order worldwide. LNG proves to emit less pollution and considering the present and future emission regulations and optimistic gas fuel prices, LNG would be a preferable option as a marine fuel. The number of LNG fuelled ships is therefore likely to increase significantly the next five to ten years.

There are many ways to configure the fuel supply system. The fuel supply system consists of a tank, heat exchangers and a gas valve unit (GVU) that are connected by pipes and valves. The focus area of this study was the low-pressure (LP) LNG fuel supply system. Three different LP LNG fuel supply systems was studied and optimized in this thesis:

- CASE 1: Supply system with two shell-and tube exchangers.
- CASE 2: Supply system with one shell-and-tube exchanger.
- CASE 3: Supply system with one compact exchanger and one shell-and-tube exchanger.

The modeling, simulations and optimization for these three cases were done in the commercial simulation tool Aspen HYSYS and the additional HYSYS-software, Exchanger Design & Rating (EDR). The simulations showed how dependent the process parameters were on the other system parameters. Further, a sensitivity analysis performed gave an understanding of how the mechanical parameters of the heat exchanger affect the overall design including thermal and hydraulic parameters such as heat transfer rate and pressure drop.

For the optimization of the process parameters for the heat exchangers the inlet temperature difference of the two fluids, and type and mass flow of the heating medium was showed in this thesis to be vital. Mechanical parameters such as shell ID, tube OD and tube length were found to be vital parameters that affected not only the mechanical design but the thermal and hydraulic parameters as well.

This study only focused on the heat exchangers in the regasification system, and factors such as the pressure loss in pipes and the pressure loss through valves and fittings are not included. To get an overall understanding of the complete fuel system including tank, pressure build-up (PBU) unit, piping system and other devices concerning regasification, heating or re-liquefaction, a further study can be conducted with a system including these components.

Sammendrag

I 2014 var det 50 flytende naturgass drevne skip i drift, og rundt 70 stykker i ordrebøkene rundt om i verden. Flytende naturgass (LNG) viser seg å være mindre forurensende enn konvensjonell olje. Med tanke på de nåværende og fremtidige utslippskravene og de optimistiske drivstoffprisene for gass, vil LNG være å foretrekke. Antall LNG-drevne skip er forventet å øke betydelig de neste fem til ti årene.

Det er mange måter å konfigurere drivstoffsystemet på. Forsyningssystemet av LNG består av en trykksatt tank, varmevekslere og en gassventil-enhet (GVU) som er koblet sammen av rør og nødvendige ventiler. Fokusområdet for denne studien var et lavtrykksystem. Tre forskjellige forsyningssystemer ble studert og optimalisert i denne avhandlingen:

- CASE 1: System med to "shell-and-tube"-varmevekslere.
- CASE 2: System med en "shell-and-tube"-varmevekslere.
- CASE 3: System med en "compact"-varmeveksler og en "shell-and-tube"-varmeveksler.

Simuleringsverktøyet Aspen HYSYS og tilleggsprogrammet, Exchanger Design & Rating (EDR) ble brukt til å modellere, simulere og optimalisere systemene. Simuleringene viste hvor avhengigheten mellom prosessparameterne. I tillegg ble en sensitivitetsanalyse utført i EDR som ga en generell forståelse av hvordan de mekaniske parameterne i varmeveksleren påvirket designet inkludert de termiske- og hydrauliske parameterne som varmeoverføringsraten og trykktapet.

Innløpstemperaturforskjellen mellom de to fluidene, type varmemedium og massestrømmen av varmemediet viste seg å være avgjørende i optimaliseringen av prosessparameterne for varmeveksleren.

Mekaniske parametere som indre diameter av «shellet», ytre diameter av røret og rørlengden viste seg å være viktige parametere som påvirket ikke bare den mekaniske konstruksjon, men også de termiske- og hydrauliske parameterne.

Denne studien fokuserte kun på varmevekslerne i regassifiseringssystemet, og faktorer som trykkfallet i rørene og trykktapet gjennom ventiler og koblinger ble ikke inkludert. For å få en helhetlig forståelse av hele drivstoffsystemet inkludert tank, PBU, rørsystem og andre enheter som gjelder regassifisering, oppvarming eller re-kondensering, kan en videre studie gjennomføres ved å inkludere disse komponentene i systemet.

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Abbreviations and acronyms

CO_2	Carbon dioxide
N_2	Nitrogen
NO_x	Nitrogen oxides
PM	Particulate Matter
SO_x	Sulfur oxides

BLEVE	Boiling Liquid Expanding Vapor Explosion
BOG	Boil-Off Gas
DF	Dual Fuel
DNV	Det Norske Veritas
DNV GL	Det Norske Veritas Germanischer Lloyd
ECA	Emission Control Area
EDR	Exchanger Design & Rating
EEDI	Energy Efficient Design Index
EOS	Equation Of State
ESD	Emergency Shut Down
GHG	Greenhouse Gas
GVU	Gas Valve Unit
HFO	Heavy Fuel Oil
ID	Inner Diameter
IMO	International Maritime Organization
LMTD	Logarithmic Mean Temperature Difference
LNG	Liquefied Natural Gas
LP	Low Pressure
MDO	Marine Diesel Oil
MGH	Main Gas Heater
MGO	Marine Gas Oil
MTD	Mean Temperature Difference
NG	Natural Gas
NTU	Number of Transfer Units
OD	Outer Diameter

PBU	Pressure Build-Up
Ppmv	Parts per million by volume
TEMA	Tubular Exchanger Manufacturers Association

Symbols

General remarks: The symbols are defined both in the list of symbols and in the text. In the text, the symbols are defined in the first place they occur. A symbol can also be given several meanings.

A	Heat transfer area
A_{cross}	Cross-sectional area
B	Baffle spacing
C_p	Heat capacity
D	Pipe diameter
D_e	Equivalent diameter
D_s	Shell ID
D_t	Tube ID
F	Correction factor
f	Shell-side friction factor
f	Darcy's friction factor
G	Mass flux
g	Gravity acceleration constant
h	Heat transfer coefficient
h_i	Tube-side heat transfer coefficient
h_L	Head loss
h_o	Shell-side heat transfer coefficient
j_H	Colburn factor
K	Resistance coefficient
k	Thermal conductivity
L	Tube length
$LMTD$	Logarithmic mean temperature difference
$\dot{m}$	Mass flow
MTD	Mean temperature difference
n_b	Number of baffles
Nu	Nusselt number
n_p	Number of tube passes

p	Pressure
Pr	Prandtl number
ΔP_{sf}	Shell-side pressure drop
ΔP_{tf}	Tube-side pressure drop
$\dot{q}_x$	Heat transfer rate
$\dot{Q}$	Heat transfer rate
R_e	Reynolds number
R_N	Reynolds number
s	Fluid specific gravity
T	Temperature
U	Overall heat transfer coefficient
V	Fluid velocity
x	Wall thickness
z	Vertical elevation

ε	Effectiveness
ε	Pipe roughness
μ	Dynamic viscosity
μ_w	Viscosity at the tube wall temperature
ν	Kinematic viscosity
ρ	Fluid density
ϕ	Viscosity correction factor

1 Introduction

1.1 Motivation

In 2014, 50 liquefied natural gas (LNG) fuelled ships were in operation and around 70 on order worldwide (Wuersig et al., 2014). Today, LNG fuelled propulsion system is regarded as a proven solution for marine applications and the number of LNG fuelled ships is likely to increase significantly the next five to ten years. Future emission regulations, see Figure 1-1, and expectations of lower gas prices are the main factors for choosing LNG as fuel instead of conventional fuels like heavy fuel oil (HFO) (Wuersig, et al., 2014).

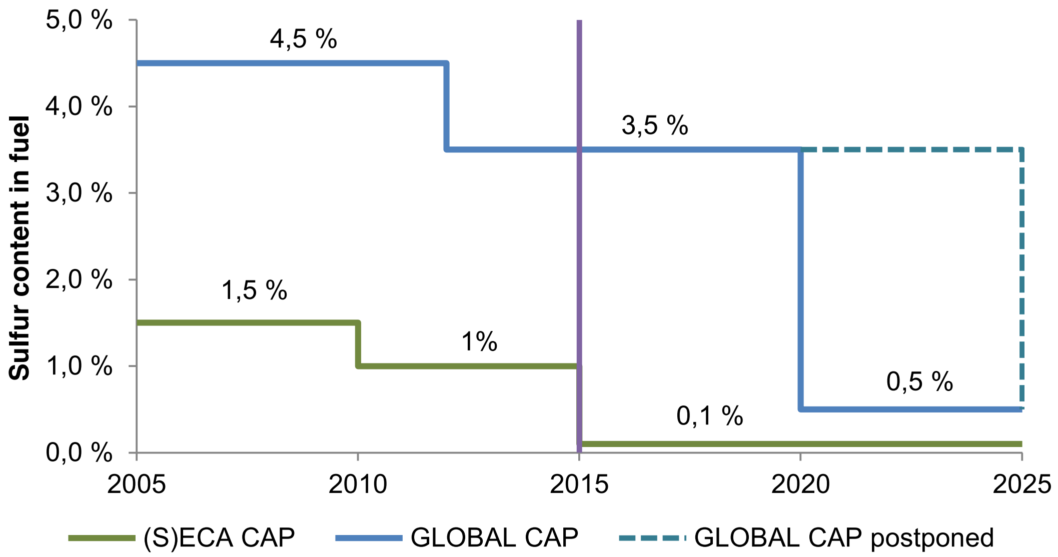

Figure 1-1: MARPOL Annex VI timeline for sulfur content limits in fuels. From 2025 ships in all global waters have sulfur content in fuel limit of 0,5%. Modified from (DNV GL, 2014a).

According to the Second IMO Green House Gas (GHG) study 2009 issued by the International Maritime Organization (IMO), emissions are projected to increase in the future due to globalization and greater energy demands (Buhaug et al., 2009). The use of LNG as fuel can limit these emissions and comply with the emission regulations. Compared to HFO, the use of LNG can nearly eliminate SOx emissions and contributes to a significant reduction of NOx and particulate matter (PM), and also a reduction of GHGs. The possible reduction of emissions by use of LNG compared to marine diesel oil (MDO) is presented in Figure 1-2.

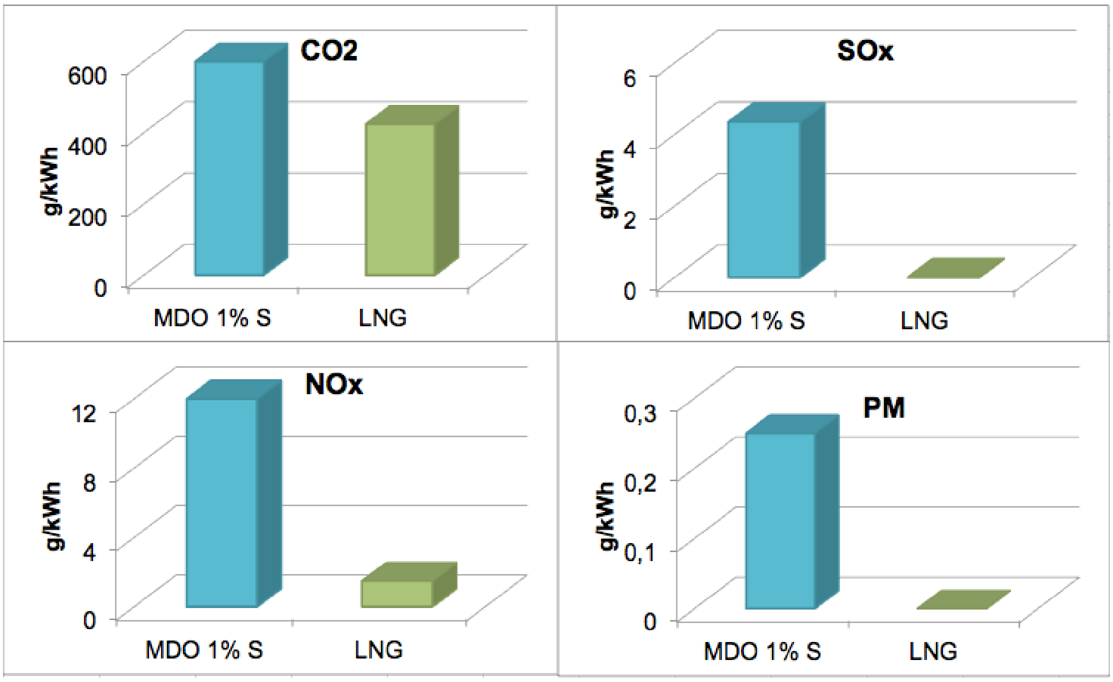

Figure 1-2: Emissions of MDO that contains 1% sulfur compared to NG from lean burn spark ignition (LSBI) engine, of the type Rolls-Royce Bergen gas engine. Modified from (Andreola et al., 2011).

1.1.1 Low-pressure (LP) LNG fuel supply system

LP LNG fuel systems mainly consist of a tank, bunkering station, pressure build-up (PBU) unit, vaporizer, main gas heater (MGH), a gas valve unit (GVU) and engines, which are all connected by valves and pipes. A possible configuration of a LP LNG fuel system is illustrated in Figure 1-3. This thesis will focus on the vaporizer and MGH components in the fuel supply system, as illustrated in the Figure 1-3.

In the fuel supply system the fuel in liquid phase from the tank is vaporized and then heated before entering the GVU that is connected to the engines. It is also possible to consume the boil-off-gas (BOG) from the top of the tank, which will be fed to the MGH and to the main circuit. The GVU is a pressure control unit, which is controlled by the engines control system, which regulates the fuel flow to the state that the engines require.

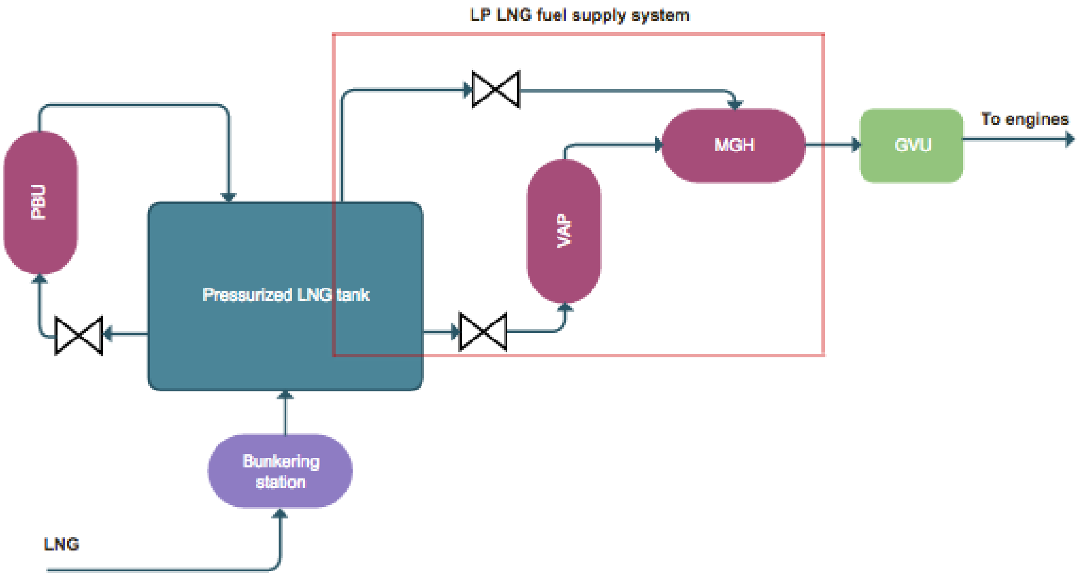

Figure 1-3: Possible configuration of a LP LNG fuel system. This thesis focus on the two heat exchangers, a vaporizer and a MGH, in the fuel supply system. Modified from (Wärtsilä, 2012).

There are many possible configurations of the LP LNG fuel supply system. For example the amount and type of heat exchangers (vaporizers and heaters) installed in the system. Figure 1-4 shows a configuration that only contains one heat exchanger. Most the vaporizers and heaters are shell-and-tube exchangers, however, there are other types of heat exchangers such as the compact heat exchanger.

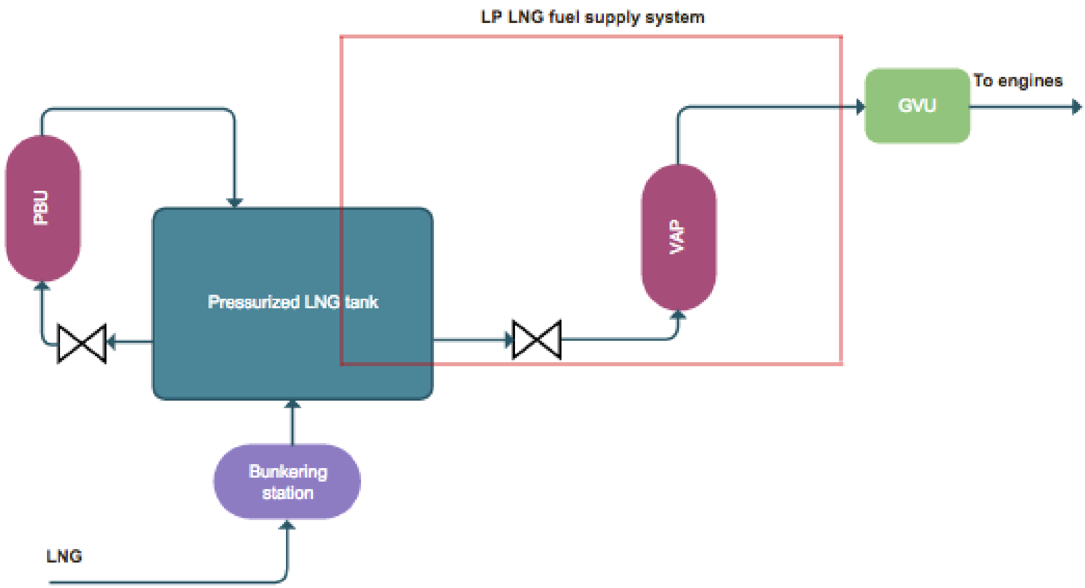

Figure 1-4: Possible configuration of the LP LNG fuel system. The fuel supply system contains only one heat exchanger, a vaporizer.

1.2 Objectives

The objective of this study is to optimize the design of the LP LNG fuel supply system with the focus on the vaporizer and the MGH. An optimized fuel supply system will reduce the energy consumption, costs and space requirements. Three different relevant systems configurations will be studied and evaluated. Further, a sensitivity analysis for a shell-and-tube exchanger will be conducted to gain an overall understanding of which parameters that can contribute to a better design and find a favorable design of heat exchangers based on certain constraints.

Simulations of the fuel supply system will be performed to find the proper process parameter. The components will then be sized, involving thermal and hydraulic calculations, such that the pressure, temperature and flow to the engines are in the required range. Modeling and steady state simulations will be carried out by using the commercial simulation tool Aspen HYSYS and EDR (Exchanger Design & Rating) for different system designs.

1.2.1 Scope of work

- Review the relevant work presented in the pre-project and study relevant literature related to the LP LNG fuel supply system and its components with focus on the heat exchangers and the processes in the system such as regasification and heating.

- Develop models of the LP fuel supply system using the simulation software Aspen HYSYS. The models should contain relevant mechanical components with relevant process parameters.

- A sensitivity analysis of the different model shall be conducted by manipulating the process parameters of the heat exchanger and study the significance and dependency of the changes on the process parameters in the system.

- A sensitivity analysis of a heat exchanger shall be conducted by manipulating some of the mechanical parameters and study the significance and dependency of the changes of the overall mechanical design including thermal- and hydraulic parameters.

- Thermal- and hydraulic design calculations of the heat exchangers, based on the relevant process parameters, shall be conducted by using the additional design software, EDR in Aspen HYSYS.

- Develop an overall understanding of how process designs, thermal- and hydraulic calculations influence the choice of heat exchangers.

1.3 Thesis structure

Chapter 1 – Introduction to the objectives of the thesis

Chapter 2 – Background information on why using LNG as fuel and issues related to it.

Chapter 3 – Overview of the LP LNG fuel supply system and its components with focus of the heat exchangers.

Chapter 4 – Theory on flow and pressure drops in pipes.

Chapter 5 – Theory on heat transfer and pressure drop in heat exchangers.

Chapter 6 – Description processes, simulation methodology in Aspen HYSYS and EDR of the three cases.

Chapter 7 – Results and discussion of the three cases, a sensitivity analysis of a shell-and-tube exchanger, evaluation of the different systems including remarks on the HYSYS-simulations.

Chapter 8 – Conclusions and proposal for further work.

2 Background

2.1 LNG as a marine fuel

Liquefied natural gas (LNG) has been used as marine fuel for more than 50 years since the first LNG carrier was launched in 1964. The boil-off gas (BOG) from the LNG cargo was used in boilers for the steam turbines (Herdzik, 2011). The liquefied fuel has proven to be a promising fuel, especially considering the current and upcoming emission regulations. Today, there are 50 LNG fuelled ships in operation that are not LNG carriers and DNV GL informs that about 69 more vessels are on order worldwide as of September 2014, which is likely to increase significantly the next five to ten years (Wuersig et al., 2014). Figure 2-1 shows the distribution of the LNG powered vessels in operation and on order on different vessel segments.

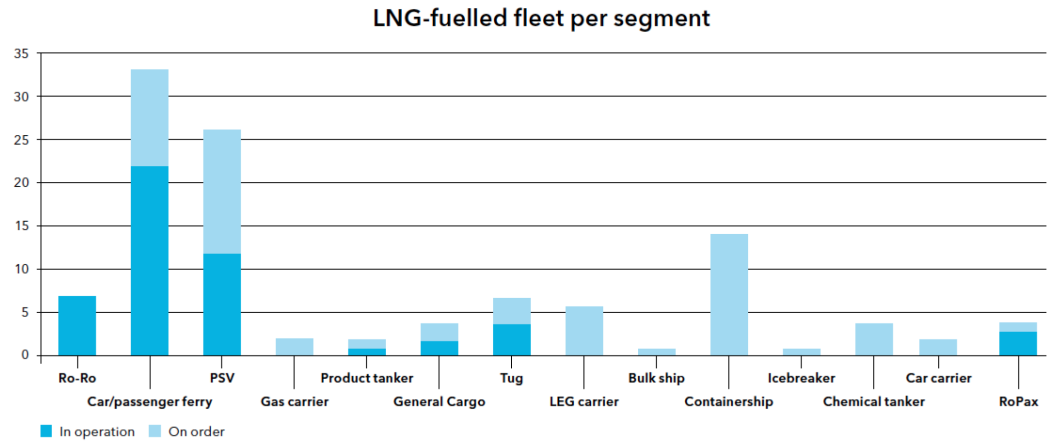

Figure 2-1: Number of LNG powered vessels in operation and on order per vessel segment as of September 2014. (Wuersig et al., 2014).

2.2 Emission regulations

According to the Second International Maritime Organization (IMO) green house gas (GHG) study 2009, emission of CO_2, NOx and SOx from international shipping was approximately 900, 22 and 12 million tons per year respectively (Buhaug et al., 2009). The study also predicts that these emissions will at least double the next 35 years without the MARPOL-regulations and DNV GL believes that the future regulations and improvements of equipment have the potential to reduce the emissions significantly. Figure 2-2, Figure 2-3 and Figure 2-4 shows the potential reduction with MARPOL annex VI in 2050 to be roughly 90%, 65% and 40% of the level of SOx, NOx and CO_2 that would have been without the regulations.

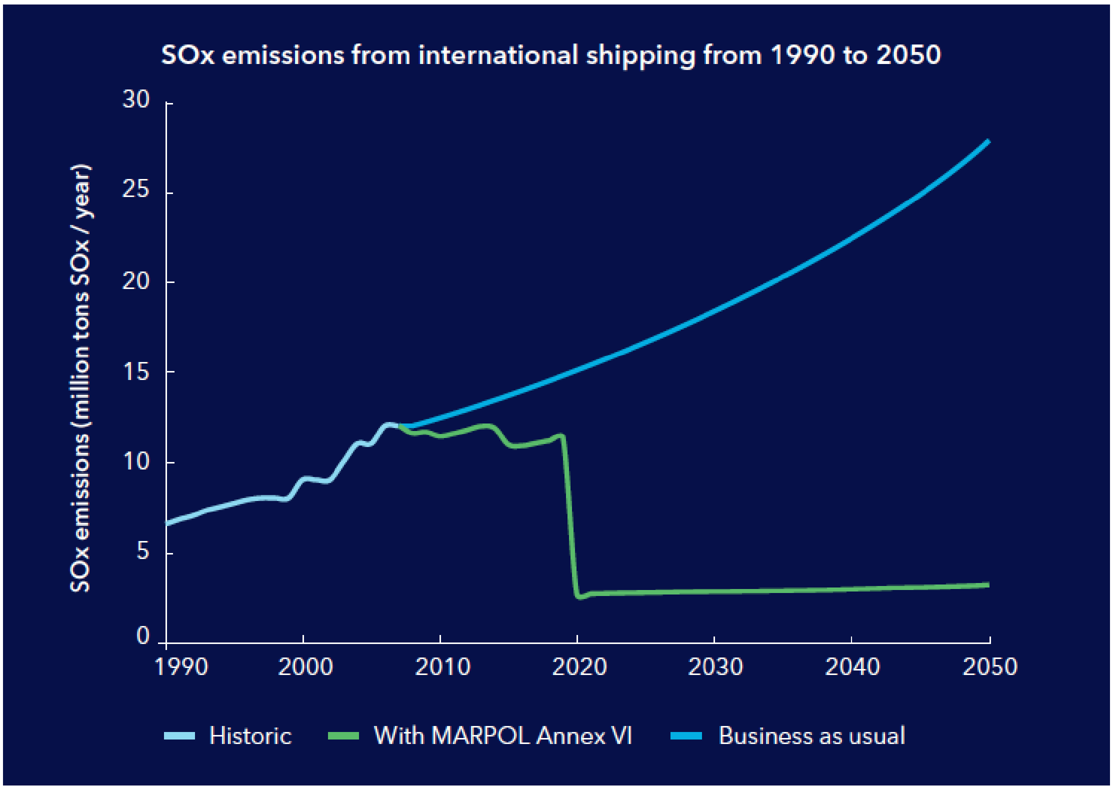

Figure 2-2: SOx emissions to air in international shipping from 1990 to 2050. (DNV GL, 2014b).

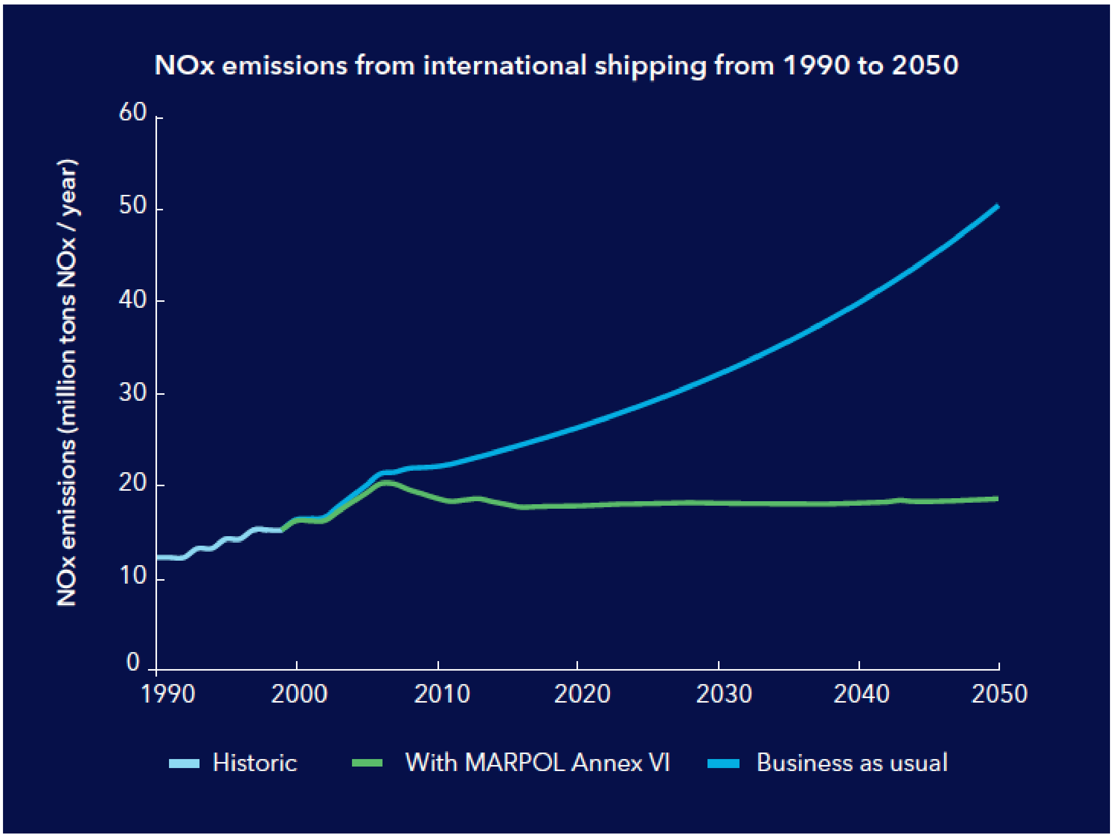

Figure 2-3: NOx emissions to air from international shipping from 1990 to 2050. (DNV GL, 2014b).

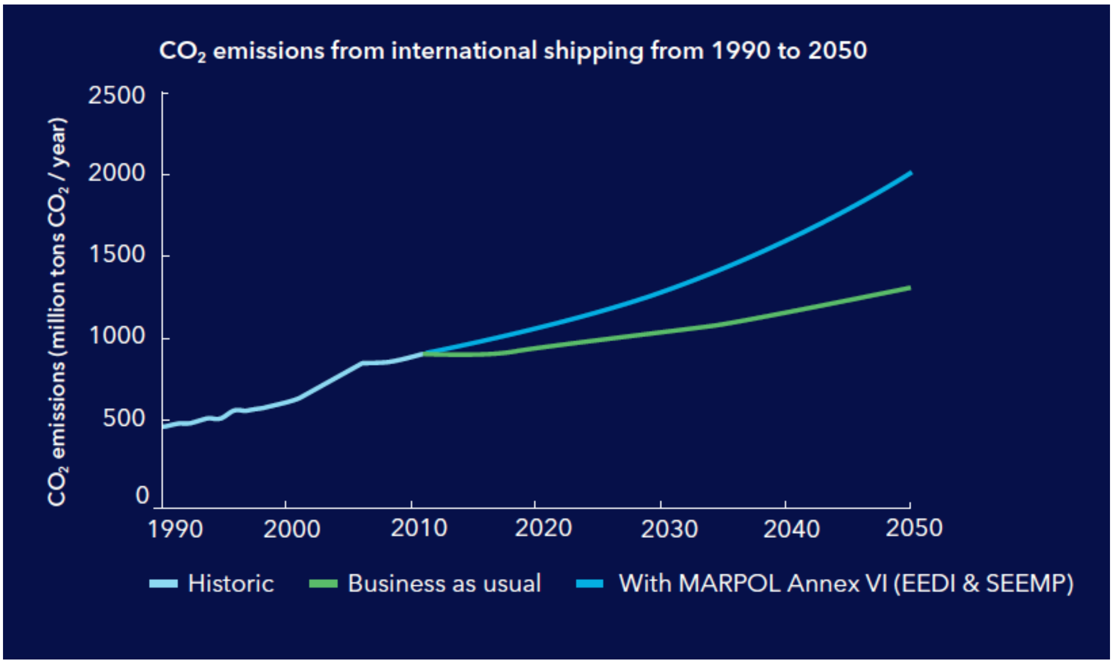

Figure 2-4: CO_2 emissions to air from international shipping from 1990 to 2050. (DNV GL, 2014b). Note the error in the color of the lines. Green line represents emission with MARPOL Annex VI and blue line represents emissions without MARPOL Annex VI.

The MARPOL convention, Annex VI by IMO concerns regulations for prevention of air pollution from ships that entered into force in 2005. A revision of the existing regulations was issued the same year (IMO, 2005). Figure 2-5 pictures the limits for sulfur content in fuel, and in what period they are valid for. January 1, 2015, the SOx-regulation entered into force and requires that the maximum sulfur content in the fuel used in Emission Control Areas (ECAs) is 0.1%. Currently, the global sulfur cap is 3.5% and will in 2020 be reduced to 0.5%, also within EU. The regulation that is set to enter into force in 2020 may be postponed to 2025 depending on an evaluation that will be made in 2018 (DNV GL, 2014a).

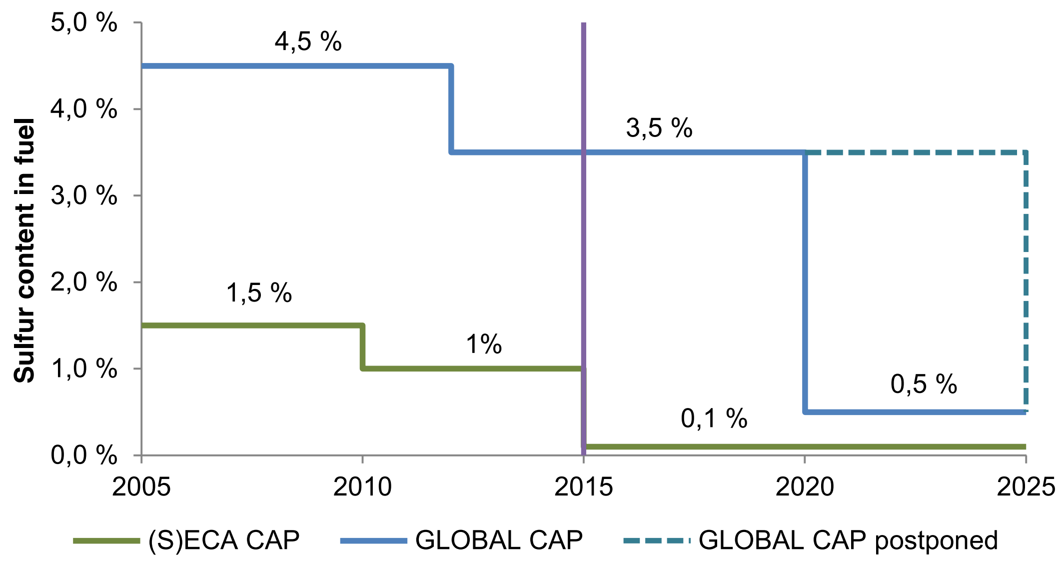

Figure 2-5: MARPOL Annex VI timeline for sulfur content limits in fuels. Modified from (DNV GL, 2014a).

The regulations for NO_x stated in annex VI, applies for new builds and vessels that sail into (N)ECAs, which is ECAs for NO_x, in North America. The current NO_x-emission requirement, Tier II, demands a reduction of 20%. Tier III will enter into force in January 1, 2016 and requires a NO_x-emission reduction of 80% (IMO, 2005). The current ECAs are presented in Figure 2-6. However, future ECAs are being discussed and proposed (Mohn, 2014).

Figure 2-6: Existing ECAs. The sulfur cap applies in all ECAs, whereas the NOx-restriction only applies in ECAs in North America. (DNV GL, 2014a).

Concerning reduction of CO_2 and other GHG emissions, IMO has adopted different phases of Energy Efficient Design Index (EEDI) that are guidelines for more energy efficient design. IMO EEDI phase 1 requirement was entered into force January 1, 2015. Phase 2 and phase 3 will take effect in 2020 and 2025 (IMO, 2005).

2.3 Advantages and disadvantages of LNG

The main emission products from a diesel engine are CO_2, NOx, SOx and particulate matter (PM). These emissions can cause increased temperature on earth, smog, formation of ground level ozone, acid rain and respiratory and other health problems for living organisms (Lloyd's Register, 2012). The use of LNG it is regarded as a clean source of energy and will contribute to a reduction of these emissions though it contains e.g. much less or no sulfur. These reductions will have significant environmental benefits such as improved local air quality, reduced acid rain and contribute to limit global warming (Lloyd's Register, 2012).

Combustion of natural gas produces emissions with lower concentrations of CO_2 and NOx, the emissions of SOx and PM are minimal and can nearly be eliminated as visualized in Figure 2-7. Compared to conventional oil-based fuels like heavy fuel oil (HFO) and marine gas oil (MGO), using LNG as fuel contributes with a CO_2 reduction up to 20% and for nitrogen, a vast reduction up to 85% (Wuersig et al., 2014). There is therefore no need for external abatement equipment to meet the emission regulations for NOx and SOx.

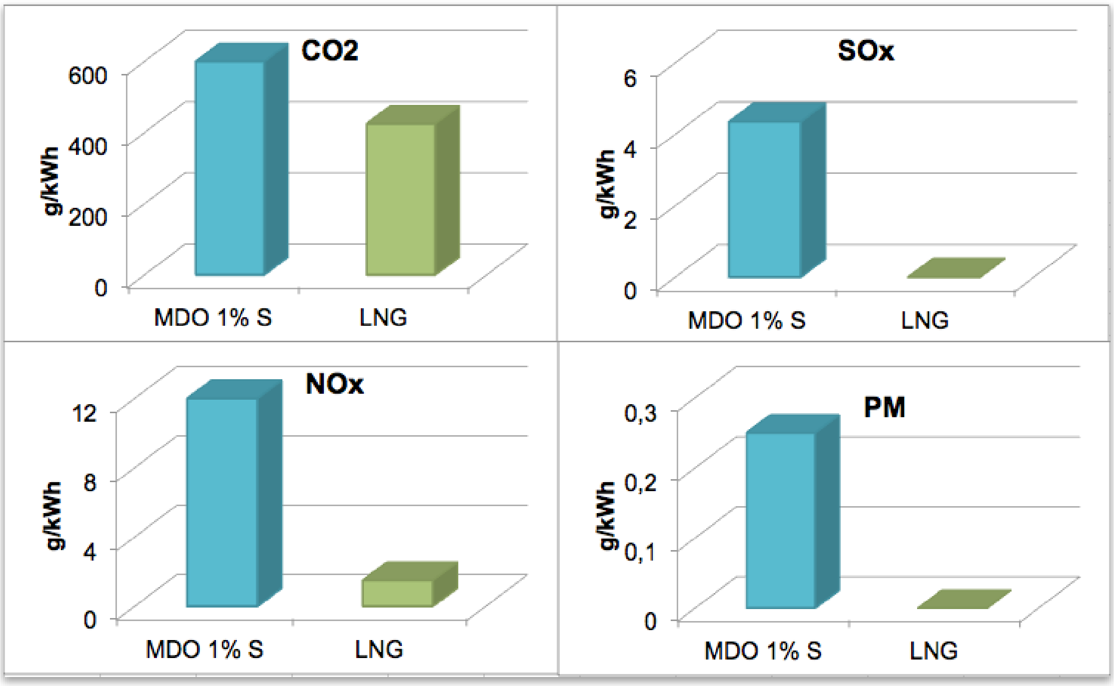

Figure 2-7: Emissions of marine diesel oil (MDO) that contains 1% sulfur compared to natural gas (NG) from lean burn spark ignition (LSBI) engine, of the type Rolls-Royce Bergen gas engine. Modified from (Andreola et al., 2011).

One of the main reasons that makes LNG the preferable fuel in some areas is the lower price compared to MGO. Optimistic price prognosis indicated in Figure 2-8 shows a reduction in LNG prices in some areas. Today the LNG production has been increased as well as the availability initiating new terminals and possible terminals in the future (Blikom, 2014). As of today, LNG facilities are available in countries like Argentina, China, The Netherlands, The USA and Norway.

Uncertain future prognosis of LNG price and availability may not persuade the ship owners to adopt the use of gaseous fuel. Installation of a more complicate system and the fact that LNG technology is a relatively new technology may cause delays of installation and construction. In case of retrofitting, the vessel may be incompatible with the LNG system. Further, reconstruction of ship to fit the system may not be possible or is too costly. In addition, the installation of a LNG system has a higher capital cost, approximately 8-15% additional capital investment than a conventional fuel system (Æsøy et al., 2011). E.g. the total building cost of the Norwegian LNG ferry, Bergensfjord was roughly 30 million EUR, which is about 11,5% higher cost than a similar diesel powered ferry (Einang, 2007).

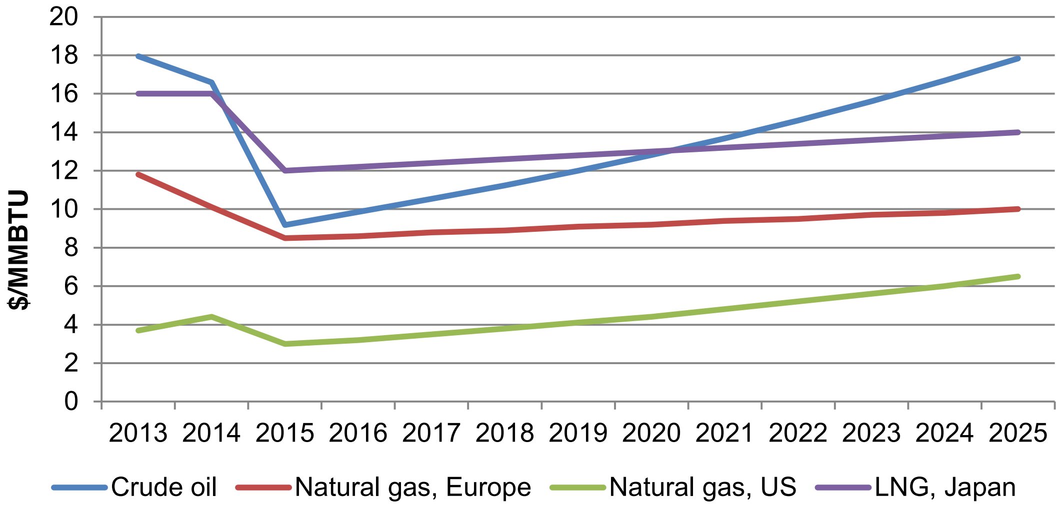

Figure 2-8: Crude oil and natural gas price forecast by The World Bank published April 22nd 2015, data retrieved from Knoema. The heat content in the fuels is measured in million British Thermal Units (MMBTU). (The World Bank, 2015). Note that the prices in Europe and the US are for natural gas, not LNG. The prices for MGO and MDO are higher and are dependent on the crude oil prices.

LNG require 1,8 more space compared to diesel oil at the same energy content but due to the bulky pressure tanks, LNG will consume three times more space than regular tanks for diesel oil (Einang, 2007). As the LNG tanks require large space and special arrangement, this may reduce the payload capacity but traditionally compartments that are utilized to store marine fuels cannot be used for payload (Æsøy et al., 2011). However, the cost of possible reduction of payload must also be taken into consideration in the evaluation process. The advantages, disadvantages and uncertainties related to using LNG as fuel for ships are shortly listed in Table 2-1.

Table 2-1: Drivers, barriers and uncertainties related for LNG powered vessels. Modified from (Balland et al., 2013).

Drivers	**Barriers**	**Uncertainties**
Reduction of SOx, NOx, CO_2 and PM	High system costs	Rate of bunker grid expansion
Comply with EEDI	Inadequate LNG bunkering grid	Future LNG fuel prices
Proven and available solution	Difficult to retrofit	
Competitive LNG prices	Size of fuel tanks	

2.4 Safety issues and design criteria

As for the physical properties written in (Wood et al., 2014). LNG is non-toxic, odor- and colorless. However leakages and spills of the liquid will vaporize and can cause suffocation due to deprivation of oxygen in non-ventilated areas. LNG is a cryogenic fluid, which means that the boiling point for LNG is very low (around -160°C). LNG can be flammable and explosive but only within the concentration range of 5 – 15 vol% if it is mixed with air. Below and above these limits, there will either not be enough methane or oxygen present for the mixture of gas or air to burn.

Rules and standards developed for LNG fueled ships are to ensure safe design and operation. The properties of LNG, especially the low temperature can cause many challenges that need cautious handling as IMO and DNV GL clarify in the report Air Pollution and Energy Efficiency from 2014 (IMO et al., 2014). The liquid gas can generate serious chill injuries and also fracture normal ship steel due to the extremely low temperature. Other safety issues concerning the large energy content in the gas tank are collision, grounding, ignition and boiling liquid expanding vapor explosion (BLEVE) (IMO et al., 2014).

The Interim guidelines for LNG fuelled ships, resolution MSC.285(86) issued by IMO was adopted on 1 June 2009. These guidelines including the listed codes and institutions serve as rules and standards for construction, safe design and –operation of equipment and installations for LNG fuelled ships.

- The Interim guidelines for LNG fuelled ships, resolution MSC.285(86)
- International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC code)
- International Code of Safety for Ships using gases or other Low-flashpoint fuels (IGF code) under the International Convention for the Safety of Life at Sea (SOLAS)
- Classification rules by e.g. DNV, Lloyd’s Register
- Local requirements

In (IMO, 2009), IMO gives guidelines on how the system is to be arranged and designed considering hazardous areas. The arrangement of equipment and surroundings must ensure an explosion proof system with sufficient ventilation and in some areas gas piping must be double walled to prevent leakages. Operational procedures must be approached in a safe way to avoid any complications and hazards. It is also important to identify the level of risk for accidents around the system installation therefore the vessel is divided in three zones, hazardous area zones, that classify the areas based on the expected presence and amount of LNG in order to make certain precautions in construction and equipment. The three hazards area zones are defined in Table 2-2.

Table 2-2: Hazardous area zones defined in (IMO, 2009).

Zone	Definition	Areas
0	LNG/NG present for longer periods or continuously	Tanks, gas pipes and all equipment containing gas
1	LNG/NG present in normal operation	Tank rooms, gas compressor rooms, enclosed and areas where gas pipes and – outlets are located
2	LNG/NG not likely to be present or for normal operation or short time presence	Areas within 1.5 m surrounding open or semi-enclosed areas of zone 1

The machinery spaces can be designed in two different ways that comply with the rules issued by IMO

- Gas safe machinery spaces
- Emergency shutdown (ESD) protected machinery spaces

Gas safe machinery spaces imply that the system is designed in a way that the machinery spaces are considered gas safe and leakages may not occur is all possible conditions. This involves gastight enclosure of the gas supply pipes in the machinery spaces as seen in Figure 2-9. The ESD protected design is considered safe during normal operation condition. An ESD is required under abnormal condition to avoid potential accidents involving gas, ignition sources and machinery. The ESD function allows exceptions for double wall piping of gas supply pipes. On the latest gas fueled vessels built, gas safe machinery spaces are applied instead of ESD protected machinery spaces (Grotle et al., 2014), though most of the vessels built between 2000 and 2010 are designed according to the ESD protected design (Æsøy et al., 2011).

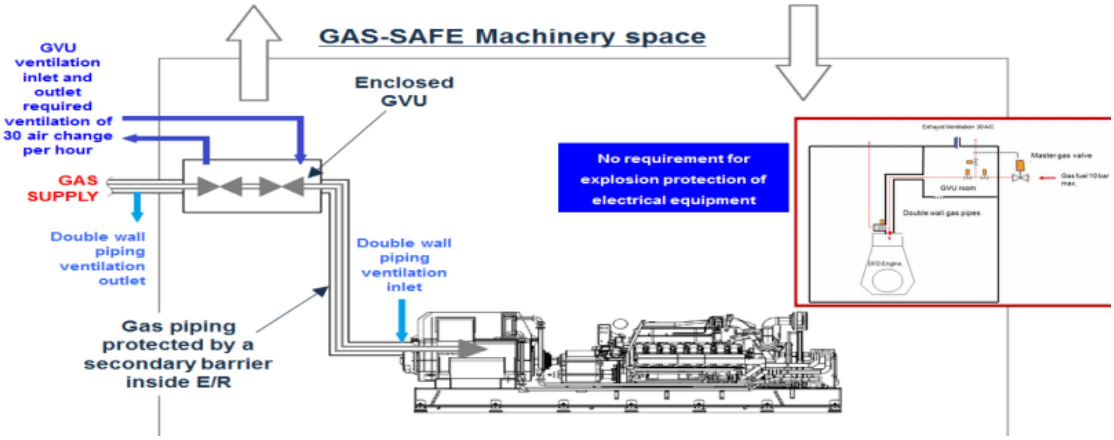

Figure 2-9: Overview of how the gas-safe machinery space is configured. The GVU is enclosed and double walled piping is required including necessary venting. However, there are no requirements for explosion protection of the electrical equipment. (Kokarakis, 2015).

3 System components in low pressure LNG fuel supply system

The low pressure (LP) LNG fuel system mainly consists of a tank, bunkering station, pressure build-up (PBU) unit, vaporizer, main gas heater (MGH), a gas valve unit (GVU) and engines connected by valves and pipes. The fuel, either in liquid or gas phase, is vaporized if needed and then heated before entering the GVU, which has certain inlet requirements. The GVU is connected to the engines and is a control unit that regulates the fuel-flow to the state that the engines require. There are also certain inlet limits for the GVU as well.

A sketch of a simplified LP LNG fuel system is given in Figure 3-1. Most LP LNG fuel systems are constructed with a PBU-unit, as illustrated in the figure. The unit is here installed in a closed loop where the liquid fuel from the bottom of the tank vaporizes through the PBU and supplied in the top of the tank. The vaporized fuel injected into the tank will keep the pressure in the tank at the required level if the pressure should fall below operation pressure. Normally there is a joint circuit with water and glycol for the vaporizer and the MGH.

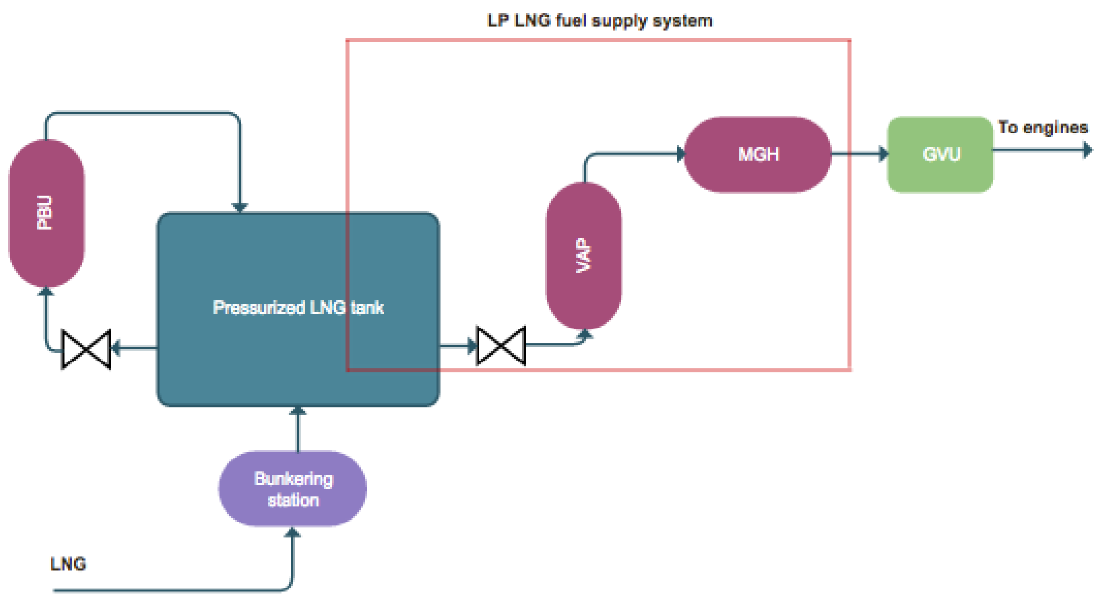

Figure 3-1: Simplified low pressure LNG fuel system with bottom filling of LNG from bunkering station. Modified from (Wärtsilä, 2012).

In this introduction-section, a brief overview of the LP fuel system has been presented. In the rest of this chapter, the relevant components for the supply system will be discussed more closely.

3.1 Pressurized tanks

There are many challenges related to the storage of LNG. One of the main challenges is the low temperature at around -160°C, which affects the choice of material. As for the shape, the tanks are designed to maintain the pressure at the required level, and also to handle challenges like boil-off gas (BOG) from the liquid fuel, which will cause the pressure within the tank to rise. There are currently three types of independent tank designed for storing LNG in accordance with IMO standards, denoted type A, B and C (IMO, 1993). The choice of tank depends on the purpose of use and the fuel system. Figure 3-2 shows a drawing of a double-shelled pressurized LNG tank type C.

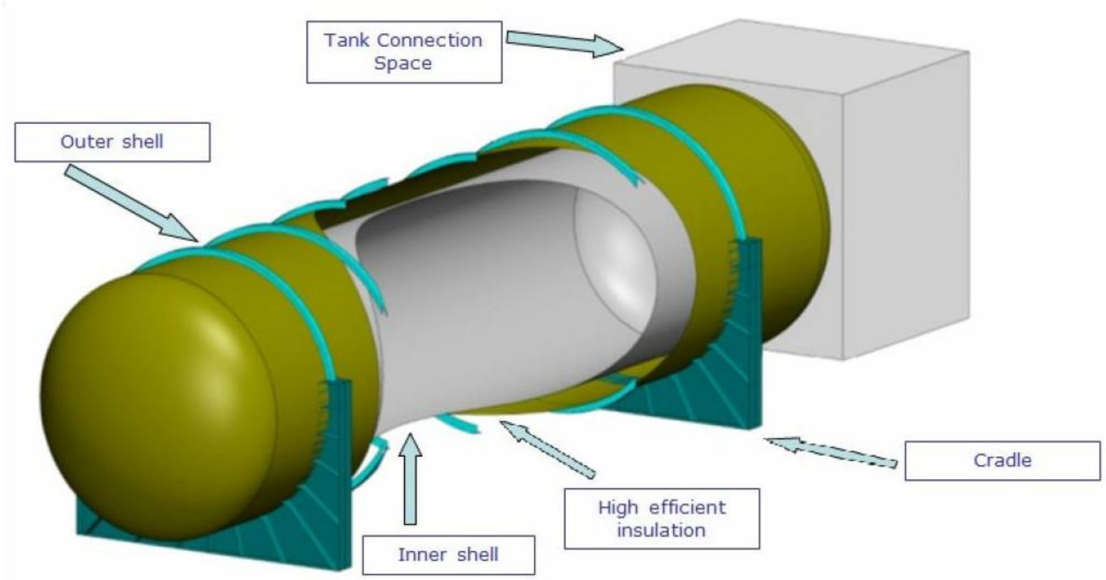

Figure 3-2: 3-dimensional drawing of a double-shelled pressurized LNG tank (type C) designed by M.E.S for Wärtsilä. (M.E.S., 2010)

For the fleet of gas fuelled vessels in operation by January 2014 only tanks of type C have been used (IMO et al., 2014). The pressurized tanks (type C) can be vacuum insulated, has a cylindrical shape with dished ends and demand larger space than the prismatic tanks as overviewed in (WPCI, 2015). In theory, the type C is leak free without secondary barrier but the leakages may occur through the valves. Normally the operation pressure for type C is in the range 4-8 bar, but they allow a pressure build-up of up to 10 bar (IMO et al., 2014). The boil-off gas (BOG) can be handled by temporarily switching the fuel outlet to consume the gas in the top of the tank (Karlsson et al., 2010). In such systems with a pressurized tank, external pumps or compressors are not necessary because the pressure will drive the fuel flow around in the system. However, a pump may be installed in the bottom of the tank to increase the pressure when it is necessary, replacing the PBU (Wärtsilä, 2012).

3.2 Pipes and valves

In this section, a brief overview of types of pipes and valves that is used for cryogenic operation such as the LNG fuel system is given. It is written based on (Loomis, 2014).

3.2.1 Pipes

Double walled piping for marine LNG fuel system is required for some critical areas where possible leakages can create a dangerous situation. Pipes with double walls can either be insulated or not insulated. In Figure 3-3, a double walled pipe with insulation is presented.

Figure 3-3: Double walled pipe with insulation. (Loomis, 2014).

There are various types of insulation options. The different insulation materials have different thermal properties. The required thickness of insulation is based on its ability to resist heat transfer. Some insulation types are listed below.

- Calcium silicate
- Mineral wool or fiberglass
- Expanded perlite
- Pyrogel XT

The inner pipe wall is often called the service pipe and is designed for the same operation-pressure and operation-temperature as the outer covering pipe. Complete venting of the annular space, the space between the service pipe and the outer pipe, is required for safety purposes. The piping system can also be designed with pressurized nitrogen in the annular space for purging.

3.2.2 Valves

In the LNG fuel system, there are valves to control and regulate the flows and for safety purposes as emergency shutdown (ESD). For cryogenic LNG systems, there are mainly three types of valves that are being used.

- Globe valves
- Ball valves
- Butterfly valves

Ball valves are often used if the diameter of the pipes in the system is small. For multiple connecting opportunities, the butterfly valves are recommended. The globe valves are the most commonly used valves in the LNG fuel system. It is operable in the temperature range of -196°C to 100°C and complies with IMO and class standards for cryogenic operation. Figure 3-4, a schematic drawing of a globe wall is presented. The globe valve can be designed for both low-pressure operation and high-pressure operation and is considered fire-safe. Compared to the other valves, the globe valve is best secured for leakages. The valve is constructed of stainless steel and its robust design allows minimal maintenance. The globe valve would be the preferable choice compared to the other types because it makes it easy to control the flow.

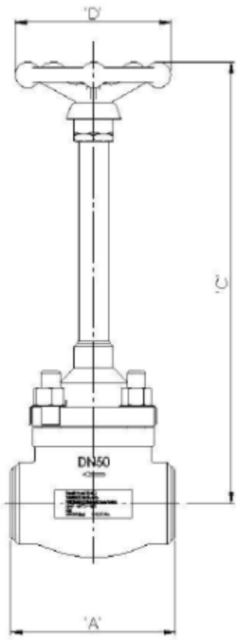

Figure 3-4: Schematic drawing of a globe valve. (Loomis, 2014)

3.3 Vaporizers and MGH

The purpose of the heat exchanger is to exchange heat between two or more fluids. The hot fluid running through the exchanger will transfer heat to the cold fluid causing a temperature rise. The fluids can maintain the phases throughout the heat exchanger and in some cases the fluids experience a phase change where the fluids vaporizes or condenses. In the fuel supply system, the LNG is vaporized to natural gas (NG) in the vaporizer and heated in the MGH, where the heating medium maintains its initial phase through both of the exchangers. The difference between these processes and the type of fluids itself must be taken into consideration when selecting a suitable type of heat exchanger.

Some various types of heat exchangers classified after construction type are listed on the next page. The heat exchangers that are relevant for the fuel supply system are the shell-and-tube exchanger and the plate-fin compact exchanger, which will further be discussed in this section. The theory on heat shell-and-tube exchangers and the compact exchanger is written based on chapter 4 and 5 in (Thulukkanam, 2013).

- Tubular heat exchanger
    - Double pipe, coiled tube, shell-and-tube
- Plate heat exchanger
    - Panel coil, spiral, welded
- Compact heat exchanger
    - Tube fin, plate-fin
- Regenerators
    - Fixed matrix, rotary matrix

3.3.1 Shell-and-tube exchanger

The shell-and-tube exchanger is the most commonly used heat exchanger due to its applicability. This type of heat exchanger mainly consists of a shell with internal tubes and baffles as illustrated is Figure 3-5. The internal baffles on the shell-side will help induce turbulent flow. Components such as front head, rear head and nozzles are also important parts of the heat exchanger. The tubes can be plain, finned or doubled shaped straight or in U-formation.

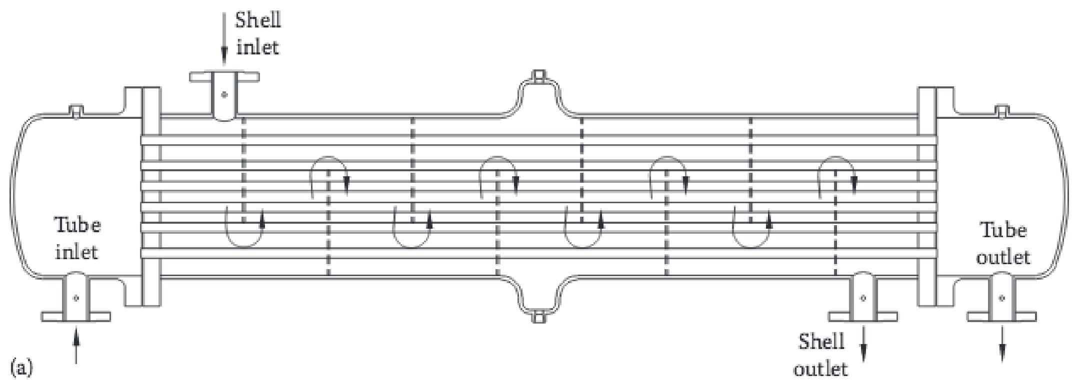

Figure 3-5: Cross-section drawing of a shell-and-tube exchanger with vertical baffles. (Thulukkanam, 2013)

The choice of the individual parts of the shell-and-tube exchanger depends on the operating parameters such as pressures, temperatures and thermal stresses. It is also important to consider factors like fouling, cleanability and corrosion characteristics and costs. The shell-and-tube exchangers are often designed after Tubular Exchanger Manufacturers Association (TEMA) standards. Different front-end type, shell-type and rear-end type after TEMA standards are presented in Figure 3-6.

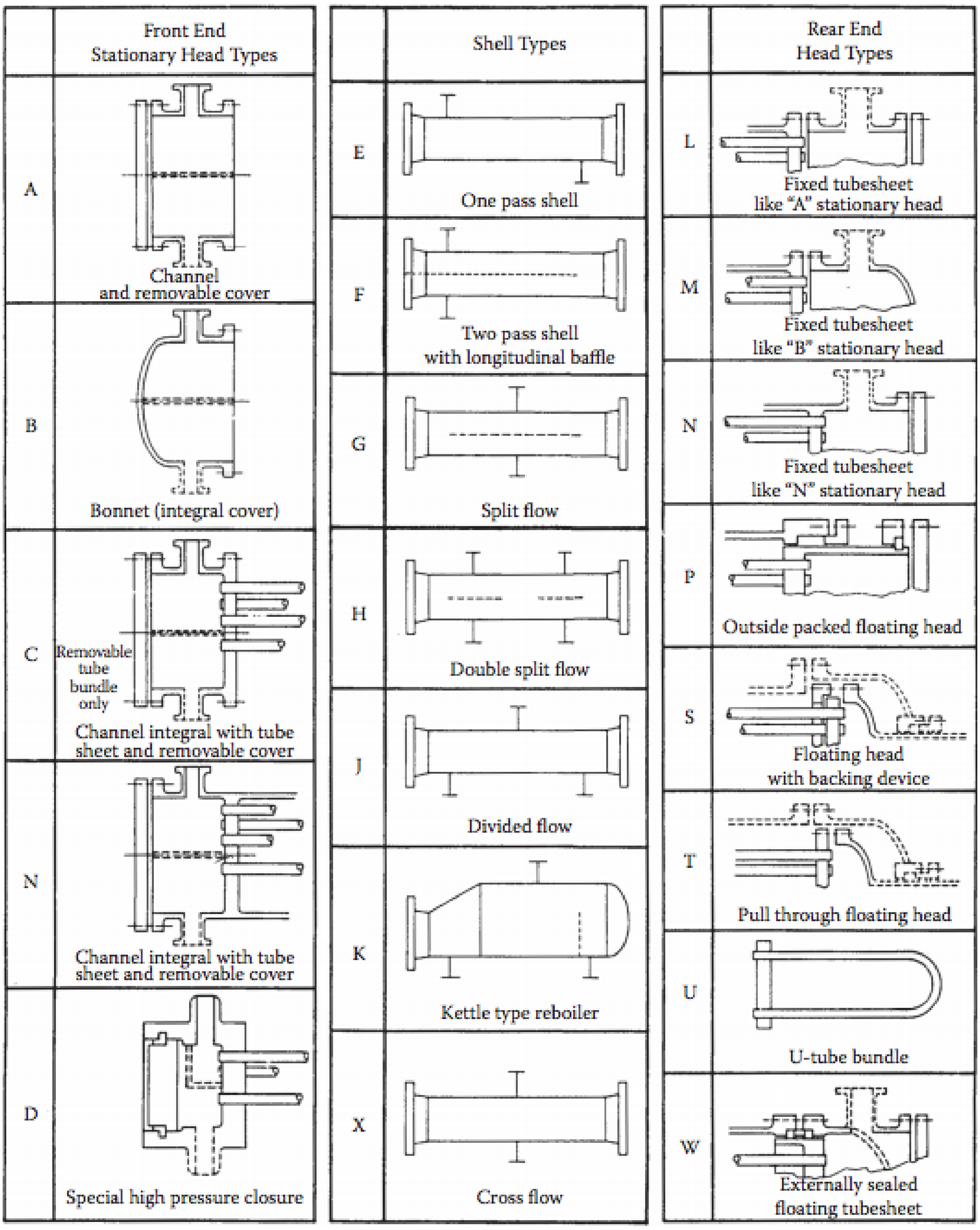

Figure 3-6: TEMA design standards for shell-and-tube exchanger. (Tubular Exchanger Manufacturers Association Inc., 1999).

The different types of the parts of the heat exchanger are suitable for different purposes and desired features. The front-end head types are channel cover and bonnet covers. These covers enclose the tubes and some of them, such as B, C and N, are integral covers. The types A, C and N allow the cover to be removed.

The shell types E, F, G, H, J and X are designed for different types of flow arrangements. In Figure 3-7 the different shell types and flow pattern on the shell-side are illustrated excluding type J.

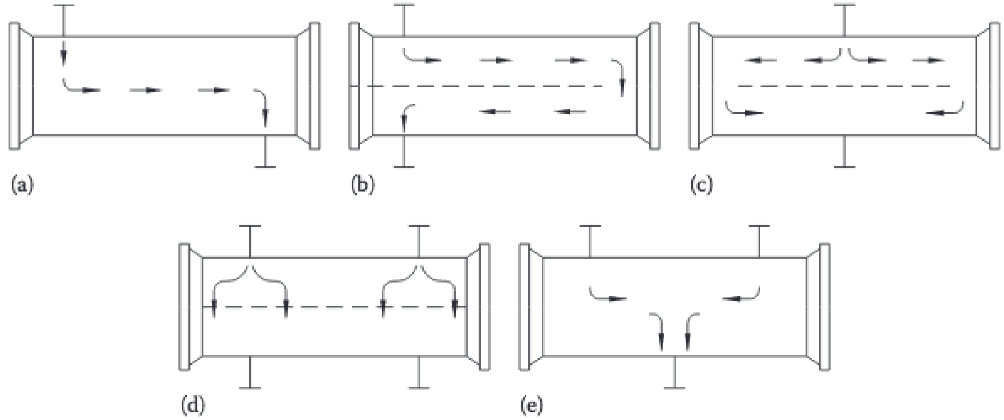

Figure 3-7: TEMA shell types and shell-side flow pattern. (a) One-pass shell, (b) two-pass shell with longitudinal baffle, (c) split flow, (d) double split flow, and (e) cross flow. Figure from p. 273 in (Thulukkanam, 2013).

The E-shell is the most common shell type with a simple design and is easy to manufacture. If the shells of this type are connected in series, it may increase the overall effectiveness of the heat exchanger unit. E-shells are also cheaper compared to the other shell types. The F-shells are rarely used due to risk of fluid leakages and other problems. F-shells also operate with a higher fluid velocity and pressure drop. G-shells and H-shells are recommended for operations with phase transition and are usually utilized as reboilers and condensers. The J-shell is suitable for low-pressure operation and is usually utilized as a condenser. Compared to the E-shell, the J-shell operates with a lower pressure drop. The E-shell and F-shell are suitable for operation with single-phase fluids.

Rear end floating heads such as P, S, T and W are suitable for operation where thermal expansion is expected due to a significant temperature difference. The tube bundle is removable for type P and W, allowing easy maintenance. Type S is recommended for high-pressure operation, while type W is recommended for low-pressure operation.

Rear-ends with fixed tubesheet, such as type L, M and N need an expansion joint to endure possible thermal expansion. Compared to the floating heads, the fixed tubesheet models are cheaper but do not allow removal of the tube bundle. They are also compatible with the stationary front head types A, B and N.

The U-type with U bundles allows the tubes to expand and contract and is suitable for high-pressure operation. However, there are many disadvantages with the U-type related to the flow, such as large pressure drops and erosion of the U-bend.

3.3.2 Compact heat exchanger

It is experienced that the compact exchanger is very energy intensive, and for cryogenic application an effectiveness of 95% or more is common. Compared to the shell-and-tube exchanger, the compact exchanger is smaller in size, more effective, but on the other hand it is more expensive, which often makes the shell-and-tube exchanger preferable. There are three types of compact heat exchangers:

- Plate-fin
- Tube-fin
- Regenerators

Figure 3-8 illustrates the basic principles of plate-fin exchangers. The exchanger is constructed with layers of plates and fins on top of each other. One of the fluids flows through the layers of fins, while the other fluid flows in the layers of plates.

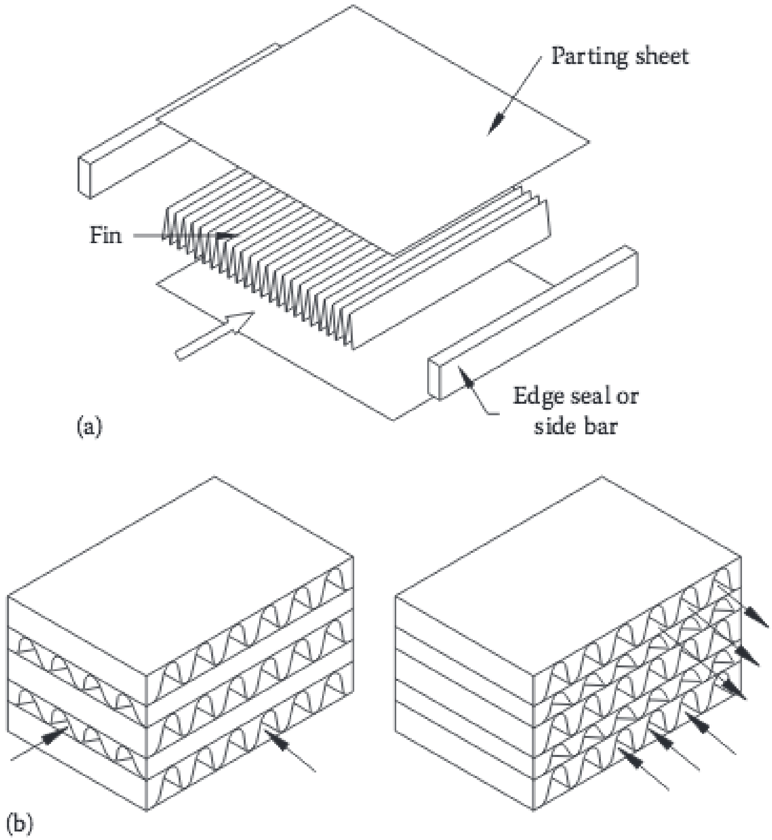

Figure 3-8: Drawings of plate-fin exchangers. Figure from p. 157 in (Thulukkanam, 2013).

Due to its compact construction, the flow distance for the fluids is shorter than for the shell-and-tube exchanger. For this type of heat exchangers it is important to design the exchanger to ensure uniform flow distribution.

4 Flow and pressure drop in pipes

In the LNG fuel supply system there are both liquid and gas flowing throughout the system. Between components in the system the fluid is transported in pipes. From the tank, the liquefied fuel flows in the pipe system to the vaporizer that transforms the liquid to a gas. The gas is further heated and flows in the pipes to the engine system.

LNG is classified as an incompressible fluid and gas as a compressible fluid. Both types of fluid flow include pressure losses. This chapter will discuss the flow and pressure loss of both incompressible and compressible fluid mainly based on (White, 2003). Flow through valves and fittings will not be discussed because it is not relevant for the thesis.

4.1 Flow regions

To date there are no general analysis of fluid motion, however there exists several methods for approximate solutions for fluid problems. The problem for general analysis appears when the properties of the flow changes dependent on its Reynolds number, Eq. 4-1.

$$R_N = \frac{\rho V D}{\mu} = \frac{VD}{\nu} \qquad \textbf{Eq. 4-1}$$

R_N: Reynolds number [-]
ρ: Fluid density [kg/m^3]
V: Fluid velocity [m/s]
D: Pipe diameter [m]
μ: Dynamic viscosity [Ns/m^2]
ν: Kinematic viscosity [m^2/s]

For a certain fluid and a fixed pipe diameter, the Reynolds number is dependent on the velocity of the flow. The flow can be divided into three different flow regions:

- Laminar flow region ($R_N < 10^3$)
- Transition flow region ($10^3 < R_N \leq 10^4$)
- Turbulent flow region ($R_N \geq 10^4$)

At low velocities the flow is smooth and steady and is termed laminar flow. At higher velocities the flow is in the transition phase between laminar- and turbulent flow. When the velocity becomes high enough, the flow will be fully turbulent, which can be characterized as a chaotic flow. In Figure 4-1, the flow lines illustrate laminar and turbulent flow.

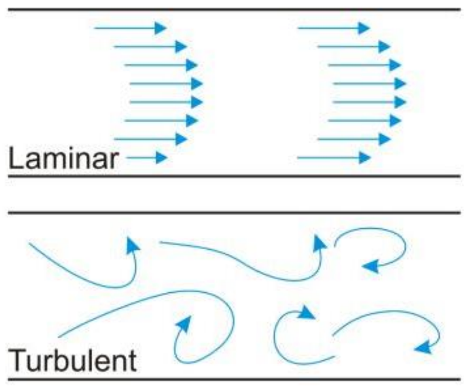

Figure 4-1: Laminar and turbulent flow. (Generalic, 2015).

It is important to note that there are other factors that affect the transition to turbulent flow such as the wall roughness inside the pipeline. The roughness will induce instability in the flow and cause fluctuations in velocity, temperature, density and pressure. For heat transfer purposes, turbulent flow is preferable due to larger surface area that makes heat transfer more effective (Serth, 2007).

4.2 LNG flow

The compressibility factor for LNG is so small that it is regarded as an incompressible fluid. Incompressible fluids have a constant density, i.e. the fluids density is neither affected by pressure- nor temperature changes. For a LNG flow in a pipeline, pressure loss will occur due to resistance, elevation and changes of kinetic energy (Rennels et al., 2012). In relevance to this thesis, this section will only discuss pressure loss in pipes due to friction.

Bernoulli’s equation, Eq. 4-2, is an energy balance equation between two points in a steady incompressible flow along a pipe with constant diameter. The equation can be used to show how pressure drop is due to friction. By assuming there are no external forces from a pump or a turbine, the velocity at point 1 equals the velocity at point 2.

Further, when there is no elevation of the piping system, the equation can be reduced to Eq. 4-3. The head loss, h_L refers to the pressure drop in the system. This pressure drop is mainly due to frictional resistance between the flow and the exposed pipe surface. The mechanical energy of the flow is converted to thermal energy and therefore the pressure is decreased (Rennels et al., 2012).

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_L \qquad \textbf{Eq. 4-2}$$

$$h_L = \frac{1}{\rho g}(p_1 - p_2) \qquad \textbf{Eq. 4-3}$$

p: Pressure [Pa]

g: Gravity acceleration constant [m/s^2]

z: Vertical elevation [m]

h_L: Head loss [m]

Darcy-Weisbach, Eq. 4-4, calculates the head loss by taking into account the frictional resistance. The friction factor, f, is also called the Darcy's friction factor. This factor depends on the flow regime. For laminar flow Eq. 4-5 yields, where the factor is fully dependent on the flow's Reynolds number. For turbulent flow, the friction factor is defined by the Colebrook-White equation, Eq. 4-6. It is also possible to use the Moody chart, Figure 4-2 to find the friction factor, which is based on the Colebrook-White equation. Darcy-Weisbach equation together with Eq. 4-3 makes it possible to calculate the pressure drop in pipes for incompressible flow.

$$h_L = f\frac{LV^2}{D2g} \qquad \textbf{Eq. 4-4}$$

$$f_{laminar} = \frac{64}{Re} \qquad \textbf{Eq. 4-5}$$

$$\frac{1}{\sqrt{f_{turbulent}}} = -2\ln\frac{\varepsilon}{3{,}7D} + \frac{2{,}51}{Re\sqrt{f_{turbulent}}} \qquad \textbf{Eq. 4-6}$$

L: Tube length [m]

f: Darcy's friction factor [-]

ε: Pipe roughness [m]

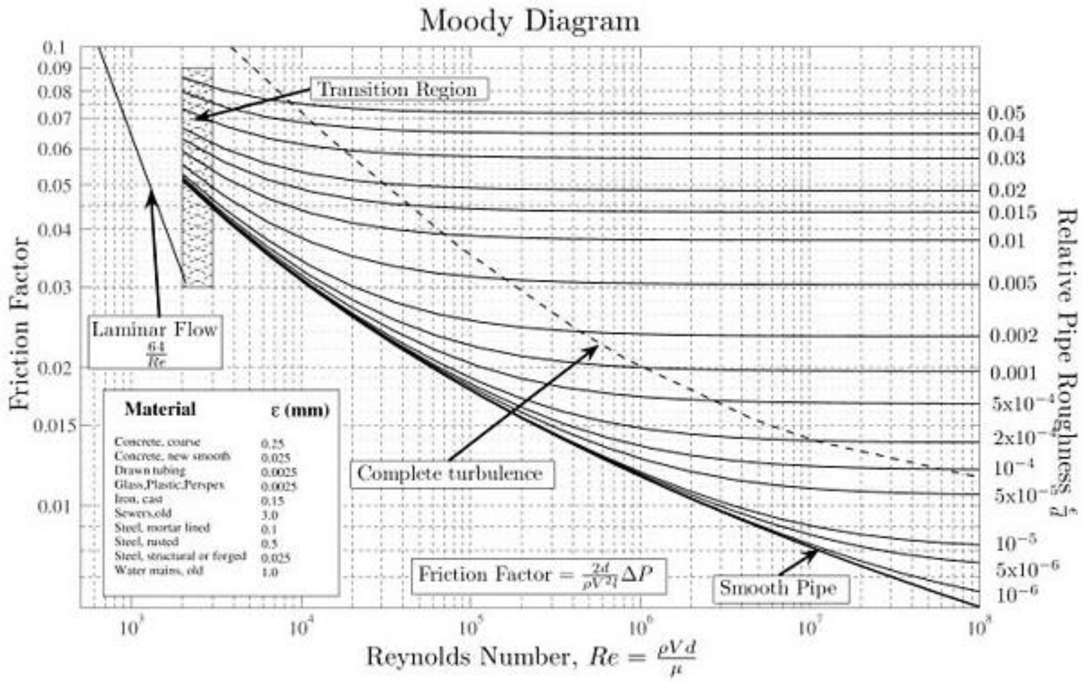

Figure 4-2: Moody chart based on the Colebrook-White equation. Utilized for finding Darcy's friction factor, *f* for a flow by its Reynolds number and relative pipe roughness. The friction factor is often used to calculate the pressure drop of the flow in the pipe. (Moody, 1944).

4.3 Natural gas flow

Unlike an incompressible fluid, the density of a compressible fluid is not constant. Natural gas is a compressible fluid, and the density is affected by both pressure- and temperature changes. For compressible fluids it is necessary to note that thermal energy is converted in mechanical energy within the gas flow (Rennels et al., 2012). This factor makes the calculation of pressure drop for compressible fluids significantly more difficult compared to incompressible fluids.

(Rennels et al., 2012) introduces six methods to calculate the pressure drop for compressible fluids. They are either by adjusting Bernoulli's equation for incompressible flow, Eq. 4-2, to approximate compressible flow or by using ideal equations for compressible flow. In these methods it is assumed that the pipe is horizontal with a constant diameter. An overview of the methods and for which type of conditions they are suitable for is shown on the next page (Rennels et al., 2012).

- Methods by adjusting Bernoulli's equation for incompressible flow to approximate compressible flow:
    - Inlet- and outlet properties
        - Suitable for pressure drops below 10% of the inlet pressure
        - Isothermal or adiabatic flow
    - Simple average properties
        - Suitable for pressure drops below 40% of the inlet pressure
        - Resistance coefficient K = 10 or greater
        - Isothermal or adiabatic flow
    - Comprehensive average properties
        - Suitable for pressure drops below 40% of the inlet pressure
        - Resistance coefficient K = 6 or greater
    - Expansion factors
        - Adiabatic flow
- Methods by using ideal equations for compressional flow for adiabatic flow with friction:
    - Mach number
        - One equation for isothermal flow
        - One equation for adiabatic flow
    - Static pressures and temperatures

The method using the **inlet- and outlet properties** of the flow is the simplest method to determine the pressure loss for a compressible flow. It is suitable for pipe systems with gas flow that has no more than 10% pressure loss along the system. Bernoulli's equation for incompressible flow can be rearranged to Eq. 4-7

$$p_1 - p_2 \approx \rho\left(h_L - \left(\frac{V_1^2 - V_2^2}{2g}\right)\right) \qquad \textbf{Eq. 4-7}$$

When the term for the head loss, Eq. 4-3, is inserted, Eq. 4-8 will represent the approximate pressure loss for compressible flow. The friction factors for the equation can be estimated by the same equations as for incompressible flow.

$$p_1 - p_2 \approx \rho\left(f\,\frac{LV^2}{D2g} - \left(\frac{V_1^2 - V_2^2}{2g}\right)\right) \qquad \textbf{Eq. 4-8}$$

5 Heat transfer and pressure drop in heat exchangers

In the low pressure (LP) fuel supply system, there are normally two heat exchangers, a vaporizer and a heater. In the vaporizer, the LNG is fully vaporized to natural gas (NG). The NG is then heated to the required inlet temperature for the gas valve-unit (GVU). The mechanisms of heat transfer occurring in these heat exchangers will be described in this chapter, including how pressure drops through a heat exchanger can be calculated and factors affecting the pressure drops.

This chapter presents the theory behind heat transfer and pressure drop in heat exchangers and is written based on chapter 1-3, 5 and 9 in (Serth, 2007).

5.1 General heat transfer

5.1.1 Conduction

Heat, or thermal energy, can be transported by conduction. Conduction is involved in most heat-transfer operations. Conduction involves heat transferring from a fluid through a solid object to another fluid, both fluids having different temperature as illustrated in Figure 5-1. In heat exchangers, if the heating medium flows in the tubes, the heat is transferred from the fluid inside the tube to the fluid outside the tube that keeps a lower temperature.

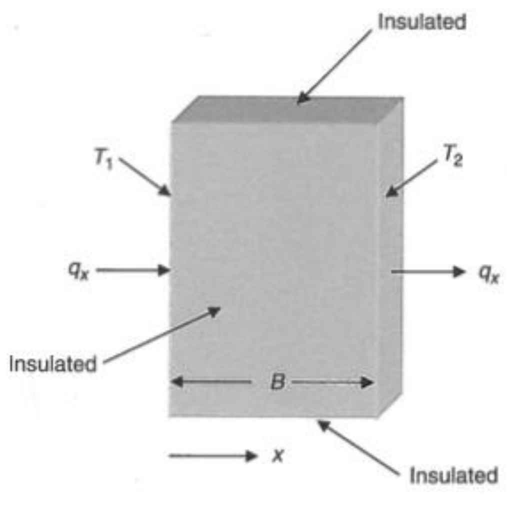

Figure 5-1: One-dimensional heat conduction in x-direction through an insulated solid object with a width, B. T_1 and T_2 is the surface temperature of the walls. Figure from p. 2 in (Serth, 2007).

In Figure 5-1 the heat transfer is one-dimensional and the heat flows in x-direction from T_1, through the wall to T_2. As indicated, the heat transfer rate is proportional to the cross-sectional area, A, whereas the temperature difference is inversely proportional to the thickness of the wall, dx. The heat transfer rate by conduction, defined by Eq. 5-1 is the rate at which heat is transferred from one fluid through a wall to another fluid.

$$\dot{q}_x = -kA_{cross}\frac{dT}{dx} \qquad \textbf{Eq. 5-1}$$

$\dot{q}_x$: Heat transfer rate [W]
k: Thermal conductivity [W/mK]
A_{cross}: Cross-sectional area [m^2]
T: Temperature [K]
x: Wall thickness [m]

Eq. 5-1 is called Fourier's law and is the general equation for one-dimensional steady state heat conduction. The thermal conductivity k is a property of the material of the solid object through which heat is transferred, and is dependent on the pressure and temperature of the material. However, for solids, liquids and gas with low-pressure, k can be assumed constant. The negative sign indicates that the transfer rate is positive when the temperature difference is negative and negative if the temperature difference is positive. Furthermore, the temperature difference in fluids often establishes convection currents, which means that the heat is transferred not only by conduction but by convection as well.

Table 5-1: Thermal conductivity at 25°C for aluminum, carbon steel and stainless steel. (The Engineering Toolbox, 2015b)

Material	Thermal conductivity at 25 °C [W/mK]
Aluminum	205
Carbon steal (1% carbon)	43
Stainless steel	16

5.1.2 Convection

Heat transfer by convection involves that thermal energy is transferred between a fluid in motion and a solid surface with a different temperature. An example of this is heat transfer from a pipe wall that is heated by electricity to the water that flows in the pipe. There are two types of convection:

- Free convection
- Forced convection

Free convection is defined such that the motion of the fluid involved in the convection process is caused by the temperature difference within the fluid. In forced convection the fluid motion is created by e.g. a pump. Figure 5-2 shows forced convection over a plate with length, L and surface temperature. T_s.

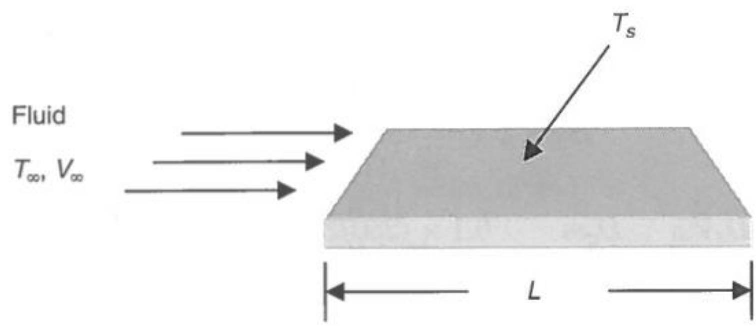

Figure 5-2: Forced convection in flow over a plate with length, L and surface temperature, T_s. Figure from p. 62 in (Serth, 2007).

The convective heat transfer rate is defined by Eq. 5-2 known as Newton's law of cooling

$$\dot{Q} = hA\Delta T \qquad \textbf{Eq. 5-2}$$

$\dot{Q}$: Heat transfer rate [W]

h: Heat transfer coefficient [W/m^2]

A: Heat transfer area [m^2]

The heat transfer coefficient, h, depends on temperature, pressure, geometry and the hydrodynamic regime. The hydrodynamic regime ranges from laminar flow to turbulent flow. If the flow is turbulent, factors such as intensity of turbulence and roughness of the surface will influence the heat transfer coefficient. In heat transfer, turbulent flow is desirable because it increases the surface area of the fluid that makes heat transfer more effective.

5.2 Heat transfer in heat exchangers

In a heat exchanger, the heat is transferred by conduction from one fluid, through a pipe wall to another fluid. However, the heat transfer between the pipe walls and the fluid can be described as convective heat transfer. The relationship between the variables in a heat exchanger is often developed by the equation of energy conservation, Eq. 5-3, and the equation of heat transfer rate, Eq. 5-4.

$$\dot{Q} = \dot{m}_1 C_{p1}(T_{1in} - T_{1out}) = \dot{m}_2 C_{p2}(T_{2in} - T_{2out}) \qquad \textbf{Eq. 5-3}$$

$\dot{m}$: Mass flow [kg/s]

C_p: Specific heat capacity [J/kg-K]

$$\dot{Q} = UA \cdot F \cdot LMTD \qquad \textbf{Eq. 5-4}$$

U: Overall heat transfer coefficient [W/m^2]

F: Correction factor [-]

$LMTD$: Logarithmic mean temperature difference [K]

$$LMTD = \frac{[(T_{1in} - T_{2out}) - (T_{1out} - T_{2in})]}{ln\left[\frac{(T_{1in} - T_{2out})}{(T_{1out} - T_{2in})}\right]} \qquad \textbf{Eq. 5-5}$$

The transfer rate and the thermal energy transported from the hot fluid to the cold fluid are constant and described by the energy Eq. 5-4. In counter flow, the driving force is the logarithmic mean temperature difference ($LMTD$), Eq. 5-5, while in parallel flow the driving force is the mean temperature difference (MTD). However, in heat exchangers, the flow pattern of the fluid varies and for design purposes the correction factor, F, should be introduced to correct for the flow pattern effects. The LMTD correction factor, F is defined by Eq. 5-6.

$$F = \frac{MTD}{LMTD} \qquad \textbf{Eq. 5-6}$$

MTD: Mean temperature difference [K]

5.2.1 Fouling factors

By the time, the heat transfer surface in the heat exchanger will collect dirt from the fluids. This dirt accumulation called fouling will settle as a film on the heat transfer area and increases the resistance. With increased resistance, the ability to transfer heat reduces and in addition the pressure loss will increase. Fouling can be caused by settlement of sludge and particles, corrosion, crystallization and biological deposits. It is necessary to include fouling factors for the fluids when designing a heat exchanger as fouling affects the thermal and hydraulic calculations (Engineering Page, 2015a) .

Table 5-2: Fouling factors for selected fluids extracted from (Engineering Page, 2015b).

Fluid	Fouling factor [m^2K/w]
Seawater	0,00009-0,00018
Engine jacket	0,00018
Treated boiler feedwater	0,00018
Gas oil	0,00009
Natural gas	0,00018
Propane	0,00018
LNG and LPG	0,00018
Glycol solutions	0,00035

5.2.2 Heat transfer coefficients

The heat transfer coefficient, h, also called the film coefficient, is the ratio of the total heat transferred and the temperature difference. It is desired with a high heat transfer coefficient in heat exchangers, which is achieved with a fully turbulent flow (Valland, 2009).

The tube- and shell-side heat transfer coefficients are useful in calculations of the clean overall coefficient. In the process of evaluating if the heat exchanger is thermally suitable, it can be necessary to compare the calculated overall heat transfer coefficient and the required overall heat transfer coefficient as well as the listed parameters:

- Heat transfer areas
- Heat transfer rates
- MTD or LMTD
- Heat transfer coefficients

For turbulent flow, the Nusselt number is given by Eq. 5-7 and yields for fully turbulent flow with Prandtl number between 0,7 and 16700 (Valland, 2009). Assuming that all tubes are exposed to the same thermal and hydraulic conditions, the tube-side heat transfer coefficient, h_i, can be calculated by Eq. 5-7 (Serth, 2007).

$$Nu = 0{,}27Re^{0{,}8}Pr^{\left(\frac{1}{3}\right)}(\mu/\mu_w)^{0{,}14} = \frac{h_i D_i}{k} \qquad \textbf{Eq. 5-7}$$

Nu: Nusselt number [-]
Pr: Prandtl number [-]
h_i: Tube-side heat transfer coefficient [W/m^{2}K]
μ_w: Viscosity at the tube wall temperature [Ns/m^2]
D_i: Tube ID [m]

Eq. 5-7 is valid for fluids with $0{,}5 \leq Pr \leq 17\,000$ and pipes $L/D > 10$. However, the right hand side of the equation is often multiplied with $\left[1 + (D/L)^{2/3}\right]$ for shorter pipes with $10 < L/D < 60$ to correct for entrance and exit effects.

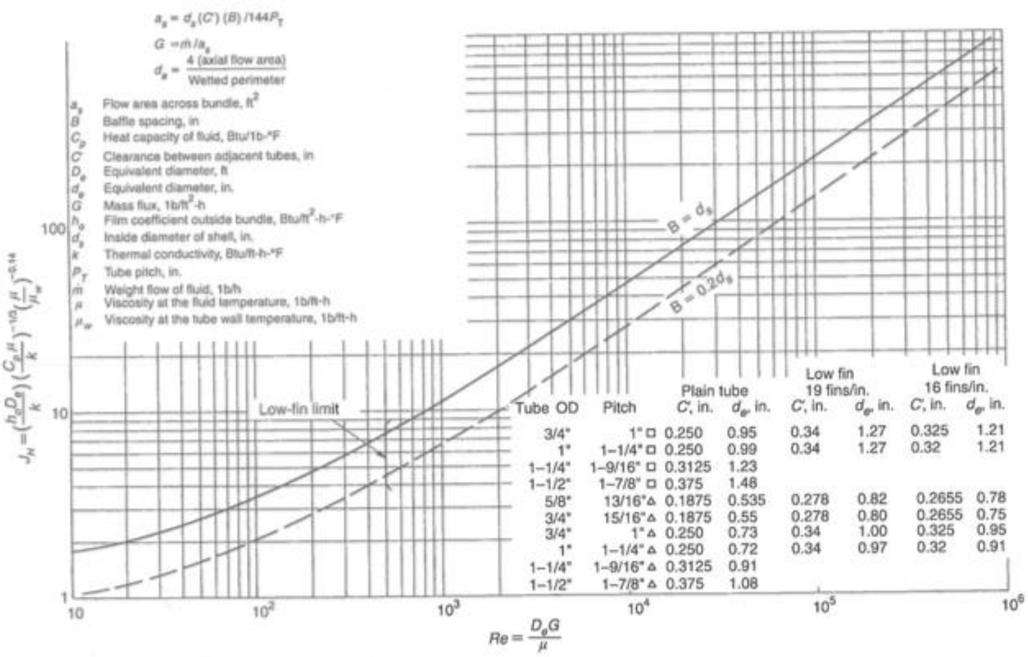

Figure 5-3: Relation between the modified Colburn factor j_H and Reynolds number, Re. Applied for finding the modified Colburn factor for further calculation of shell-side heat transfer coefficient, h_o (Henry, 1983).

The graph of modified Colburn factor, j_H, versus shell-side Reynolds number, Figure 5-3 is applied for finding j_H. It can also be calculated by Eq. 5-8. Further, the shell-side heat transfer coefficient, h_o, can be calculated by Eq. 5-9.

$$j_H = 0{,}5(1 + B/D_s)(0{,}08Re^{0{,}6821} + 0{,}7Re^{0{,}1772}) \quad \textbf{Eq. 5-8}$$

$$h_o = j_H(k/D_e) \cdot Pr^{1/3}(\mu/\mu_w)^{0{,}14} \quad \textbf{Eq. 5-9}$$

j_H: Colburn factor [-]

B: Baffle spacing [m]

D_s: Shell ID [m]

h_o: Shell-side heat transfer coefficient [W/m^{2}K]

D_e: Equivalent diameter [m]

5.2.3 Hydraulic calculations

The hydraulic calculations involve the calculations of pressure drops on both the shell-side and the tube-side. For a proper designed heat exchanger it is essential that pressure drops do not exceed the maximum allowance. In this section, the tube-side-and the shell-side pressure drop including the factors affecting the pressure drop will be discussed.

5.2.3.1 Tube-side pressure drop

The tube-side pressure drop is divided into three types of pressure drop:

- Pressure drop due to fluid friction
- Minor losses
- Nozzle pressure drop

In relevance to this thesis, only pressure drop due to fluid friction will be further discussed. The tube-side pressure drop generated by the fluid friction in the tubes can be calculated by Eq. 5-10. The pressure drop is affected by the Darcy friction factor, number of tube passes, tube length, mass flux and tube ID.

$$\Delta P_{tf} = \frac{f n_p L G^2}{2000 D_t s \phi} \qquad \text{Eq. 5-10}$$

ΔP_{tf}: Tube-side pressure drop [Pa]

n_p: Number of tube passes [-]

G: Mass flux [kg/s-m^2]

s: Fluid specific gravity [-]

ϕ: Viscosity correction factor [-]

5.2.3.2 Shell-side pressure drop

The shell-side pressure drop can be calculated by Eq. 5-11. As seen, the pressure drop is affected mainly by a friction factor, mass flux, inner shell diameter, number of baffles and the shell ID. An increase of the baffle spacing in the heat exchanger results in fewer baffles, which will decrease the pressure drop. Note that the friction factor increases with the baffle spacing and that the effect of the factors on the pressure drop depends on the relation between these two factors.

$$\Delta P_{sf} = \frac{f G^2 D_s (n_b + 1)}{2000 D_e s \phi} \qquad \textbf{Eq. 5-11}$$

ΔP_{sf}: Shell-side pressure drop [Pa]

f: Shell-side friction factor [-]

n_b: Number of baffles [-]

5.2.4 Effectiveness

Effectiveness, Eq. 5-12, of a heat exchanger is the ratio of the actual heat transfer rate and the maximum heat transfer rate possible. It expresses how effective the heat is transferred.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{(\dot{m}c_p)_{min} \Delta T_{max}} \qquad \textbf{Eq. 5-12}$$

ε: Effectiveness [-]

$\dot{m}$: Mass flow [kg/s]

This case yields for counterflow exchanger, where maximum temperature difference is the inlet temperature of the fluids. The fluid that possesses the minimum value of $\dot{m}c_p$ is the fluid that undergoes the maximum temperature change, which means that the maximum value of $\dot{m}c_p$ is the fluid that undergoes the minimum temperature change. In Table 5-3 the relation between effectiveness and the number of transfer units (NTU) for selected type of heat exchangers are presented. The value of r and NTU can be calculated by Eq. 5-13 and Eq. 5-14 respectively.

$$r = \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}} \qquad \textbf{Eq. 5-13}$$

$$NTU = \frac{UA}{(\dot{m}c_p)_{min}} \qquad \textbf{Eq. 5-14}$$

NTU: The number of transfer units [-]

Table 5-3: Effectiveness relation for selected heat exchangers.

Heat exchanger type	Effectiveness equation	
Counter flow	$\varepsilon = \dfrac{1 - exp[-NTU(1-r)]}{1 - r \cdot exp[-NTU(1-r)]}$	r < 1
	$\varepsilon = \dfrac{NTU}{1 + NTU}$	r =1
Parallel flow	$\varepsilon = \dfrac{1 - exp[-NTU(1+r)]}{1 + r}$	
Shell-and-tube	$\varepsilon = 2\left[1 + r + \beta\left(\dfrac{1 + exp[-\beta \cdot NTU]}{1 - exp[-\beta \cdot NTU]}\right)\right]^{-1}$	$\beta = \sqrt{1 + r^2}$

5.2.5 Boiling heat transfer in vaporizers

In a vaporizer, the heat transfer between the fluid in the tubes and the fluid in the shell results in boiling on the surface area that vaporizes the cold fluid. In shell-and-tube exchangers, this boiling can occur for both the fluid in the tubes and the fluid in the shell dependent on where the cold fluid flows. In Figure 5-4 a reboiler is depicted. The cold fluid is fed in the bottom of the vertical shell with horizontal tubes with hot fluid circulating. The cold fluid is heated, boiled and vaporized and the gas with lower density will leave the top of the vaporizer.

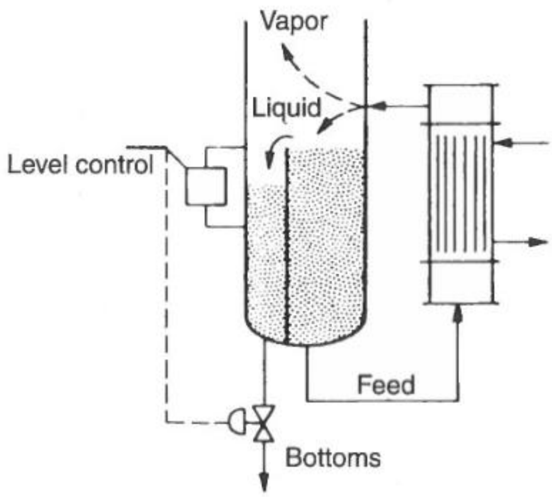

Figure 5-4: A reboiler of the type vertical thermosyphon. Liquid in the bottom of a column is fed to a vertical shell-and-tube exchanger, which vaporizes the feed to gas. The gas is then entering the column in the top. (Palen, 1988)

Boiling across a solid submerged area will occur when the surface temperature of the solid area becomes larger than the saturation temperature of the liquid flowing across. At this point, the formation of vapor bubbles occur and small bubbles will grow, detach from the surface and disperse. The growth of bubbles will induce mixing in the flow, which will increase the heat transfer rate. When most of the liquid is converted to vapor, the heat transfer rate will decrease due to the lower heat transfer rate for gases compared to liquids.

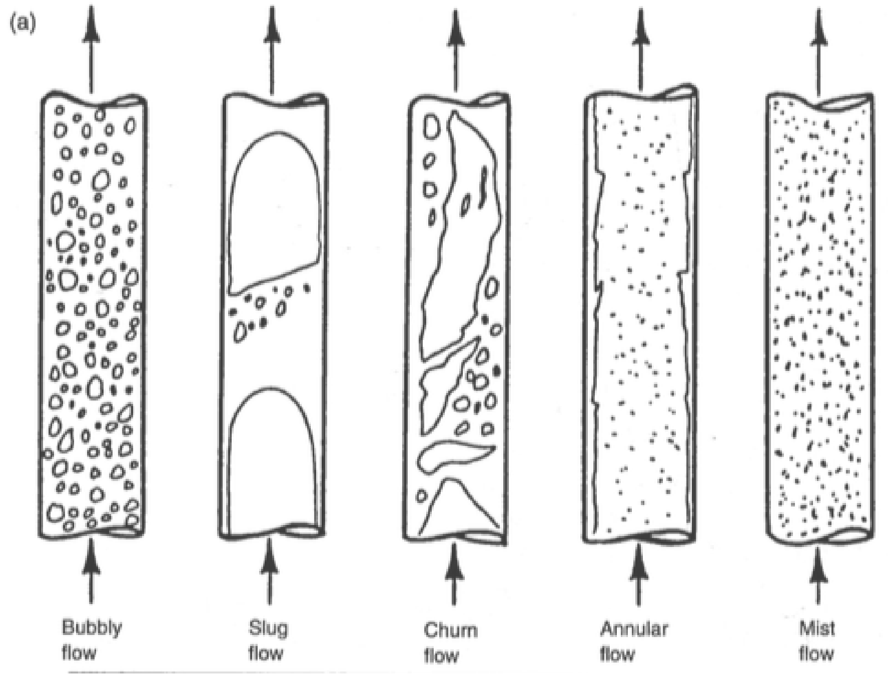

Figure 5-5: Two-phase flow regimes in vertical tubes during boiling. The amount of vapor increases from bubbly flow to mist flow, where the liquid is almost complete vaporized. Figure from pg. 403 (Serth, 2007)

Figure 5-5 illustrates how the flow in a pipe develops from a liquid to a gas during boiling. In this process of transition, the flow is in two phases and the characteristic flow regime is dependent on vapor fraction, flow rate and orientation. In bubbly flow, the vapor fraction is low and the liquid flow contains bubbles. In slug- and churn flow the vapor fraction increases and contains larger vapor pockets. From slug flow to churn flow, the instability in the flow breaks up the largest vapor pockets. In annular flow the amount of vapor is much larger than liquid, forcing the liquid out to the tube wall. When vapor fraction is very high, liquid droplets exist in the vapor flow and this kind of flow is called mist flow.

6 Simulations in Aspen HYSYS and EDR

This thesis used Aspen HYSYS and Exchanger Design & Rating (EDR) for process simulation and heat exchanger design. The HYSYS software allows a visual model construction of oil and gas processes, and is a tool for process simulation and process optimization. The EDR software is an integrated tool in HYSYS that enables thermal and hydraulic modeling of heat exchangers in simulations.

Both software are products created by the company AspenTech (Aspen Technology). The company was a result of a project at MIT that developed into something bigger. Today, AspenTech is one of the leading companies delivering software and services for the process industry in areas such as oil and gas, chemical, engineering and construction and even pharmaceutical, food, beverage and consumer packaged goods companies (AspenTech, 2015). Some of the global companies that have been and are currently using Aspen HYSYS or other software from AspenTech are BP, Technip, Shell, Statoil, ExxonMobile Chemical and Total. As Manolis Kotzabasakis, the Executive VP for products in AspenTech states: "In order for the companies to compete on a global basis, their process systems need to consume less energy and be more environmental friendly" (AspenTech, 2014). The software is created to deal with complex process manufacturing challenges, creating value and improving profitability (AspenTech, 2015).

This chapter explains the methodology used in this thesis. Aspen HYSYS and EDR version 8.4 was used.

6.1 Optimization parameters and factors in HYSYS and EDR

The objective of this thesis was to make an optimum design of a low pressure (LP) LNG fuel supply system by the use of Aspen HYSYS and EDR. First of, it was necessary to determine which process parameters were going to be fixed and which parameters that could be manipulated in the HYSYS and EDR model. This thesis focused on the vaporizer and the main gas heater (MGH) components in the fuel system, which is the regasification- and heating system. In the LP fuel supply system that was modeled in this thesis following parameters were fixed:

- The composition and the condition of the LNG feed stream
- The condition of the tank
- The flow of the natural gas (NG) stream to the gas-valve unit (GVU) (for maximum fuel consumption)

The process parameters that could be manipulated to obtain an optimum design of the system were following:

- Type and composition of heating medium
- Inlet- and outlet pressure of the heating medium
- Inlet- and outlet temperature of the heating medium
- Type of heat exchanger
- Number of heat exchangers
- Mechanical parameters such as shell inner diameter (ID), tube outer diameter (OD), tube length and baffle pitch

The optimization of heat exchangers are dependent on many factors, some of them are listed below:

- Minimum flow of heating medium
- Minimum pressure loss
- Maximum effective heat exchangers
- Minimum weight and minimum required space
- Minimum cost

Note that an optimal system design for one vessel may not be suitable for another vessel. Therefore it is important to identify factors such as space and heating sources available and evaluate the capital- and operating costs of various possible systems.

6.2 Methodology (HYSYS and EDR)

In this section, the methodology in building the models in Aspen HYSYS and EDR and the process of optimizing the process parameters will be presented. Thermal and hydraulic parameters for the heat exchanger were calculated in the external HYSYS software EDR. The sensitivity analysis was conducted based on the optimization processes in HYSYS and EDR. All the simulations models built in HYSYS were steady state models.

Three different fuel supply cases were modeled and optimized in HYSYS:

CASE 1: Supply system with two shell-and-tube exchangers

CASE 2: Supply system with one shell-and-tube exchanger

CASE 3: Supply system with one LNG exchanger and one shell-and-tube exchanger

The process system in these cases were studied, optimized and compared with each other in this thesis.

These fuel supply cases were inspired by available systems designs on the market supplied by Wärtsilä (Wärtsilä, 2012) and others. The system design in CASE 3 is directly inspired by one of Wärtsilä's new system designs for SRV-, FSRU- and JRU. The use of a compact heat exchanger to regasify the LNG instead of a shell-and-tube exchanger results in a more compact system (Wärtsilä, 2014).

6.2.1 General inputs and boundary conditions in HYSYS for CASE 1, 2 and 3

The models in Aspen HYSYS used the fluid package Peng Robinson, PR, for the simulations, which is an equation of state (EOS) often used for natural gas systems (AEA Technology plc, 2000). The components and the composition of the LNG used for all the cases are presented in Table 6-1. The components and composition data was given by Gasnor, se Appendix A for details.

The condition of the tank is presented in Table 6-2. The LNG was kept at -160°C and a pressure at 6 bar in the tank. This is common for the LP fuel system by Wärtsilä.

The tank volume was set to 200 m^3 and to a fluid level of 95%, which is a common practice at bunkering (ABS, 2015).

Table 6-1: Components and composition of LNG used in the HYSYS model .(Appendix A).

LNG/ components	Formula	Molar fraction
Methane	CH_4	0,8946
Ethane	C_2H_6	0,0665
Propane	C_3H_8	0,0141
n-Butane	C_4H_{10}	0,0021
i-Butane	C_4H_{10}	0,0090
i-Pentane	C_5H_{12}	0,0027
Nitrogen	N_2	0,0111

Table 6-2: LNG tank condition

Temperature [°C]	-160
Pressure [kPa]	600
Tank volume [m^3]	200
Level of liquid [%]	95

Table 6-3 and Table 6-4 shows the condition of the LNG feed stream and the inlet stream of NG to the GVU in the models. These specifications were set on the basis of the engine specification of a Wärtsilä dual fuel, DF, engine of the type 6L34DF with a total output power of 2610 kW (Wärtsilä, 2015). The mass flow of LNG was set to the estimated maximum fuel consumption of three engines of this type, which is 19,18 kg/h. The initial operating pressure was set to 6 bars because a GVU requires a minimum inlet pressure of 5,75 bar. Further, the GVU requires an inlet temperature between 0°C and 60°C. In the simulations, 0°C inlet temperature to the GVU was used because that will give the lowest required heat transfer. The initial pressure drops for the heat exchangers were set to 0. The pressure drop value will be calculated by the EDR software.

Table 6-3: Condition of LNG feed

Temperature [°C]	-160
Pressure [kPa]	600
Mass flow [kg/h]	19,18

Table 6-4: Condition of the inlet stream of LNG/NG to the GVU

Temperature [°C]	0 - 60
Pressure [kPa]	Min. 575
Mass flow [kg/h]	19,18

In Table 6-5, the fouling factors for the relevant streams in the fuel system are given, based on Table 5-2. The fouling factor reveals the capability of dirt accumulation that induces more resistance and affects the hydraulic calculations of the heat exchangers.

Table 6-5: Fouling factors for the streams used in EDR for hydraulic calculations

LNG, NG [m^2K/W]	0,00018
Propane [m^2K/W]	0,00019
Water-glycol solution [m^2K/W]	0,00035
Seawater [m^2K/W]	0,0009

The pressure drop in the piping system have not been included in these simulations. This is due to the pressure drop for a pipe diameter of 38,1 mm for the LNG stream being only $3{,}27 \cdot 10^{-5}$ kPa/m in HYSYS. Thus the pressure drop in pipes has a very little significance to the heat exchangers and the general supply system.

6.2.2 CASE 1: Supply system with two shell-and-tube exchangers

The CASE 1 HYSYS model is shown in Figure 6-1. The boil-off gas (BOG) from the LNG tank are let out if the pressure within the tank exceeds the design pressure with a safety margin. From the LNG tank the LNG flows to the vaporizer where a water-glycol solution is heating the LNG until it is fully vaporized to NG. The NG then flows to the next heat exchanger, the MGH, where the fuel is further heated by the water-glycol solution to the temperature that the GVU requires. Both heat exchangers use the same circuit of water-glycol solution. The adjuster, ADJ-1 in the system controls and regulates the mass flow of the water-glycol solution based on the inputted wanted temperature of the fuel outlet temperature from the vaporizer.

In this HYSYS model, the vaporizer and MGH are modeled as shell-and-tube exchangers where the LNG/ NG is flowing on the shell-side and the heating medium is flowing in the tubes. And the TEMA type chosen for these heat exchangers is based on the condition of the fluids running through the heat exchangers as discussed in section 3.3.1.

In the optimization process of the process parameters, the parameter that was manipulated in this case was the outlet temperature of the LNG/NG from the vaporizer to optimize the mass flow of the heating medium. Simulations were done from an outlet temperature of -120°C to -10°C with a step size at 10°C. Effectiveness calculations of the result of the optimization was done for both the vaporizer and the heater. See section 5.2.4 for the effectiveness equation for shell-and-tube exchangers. Further, thermal and hydraulic calculations were performed for the relevant process parameters in EDR.

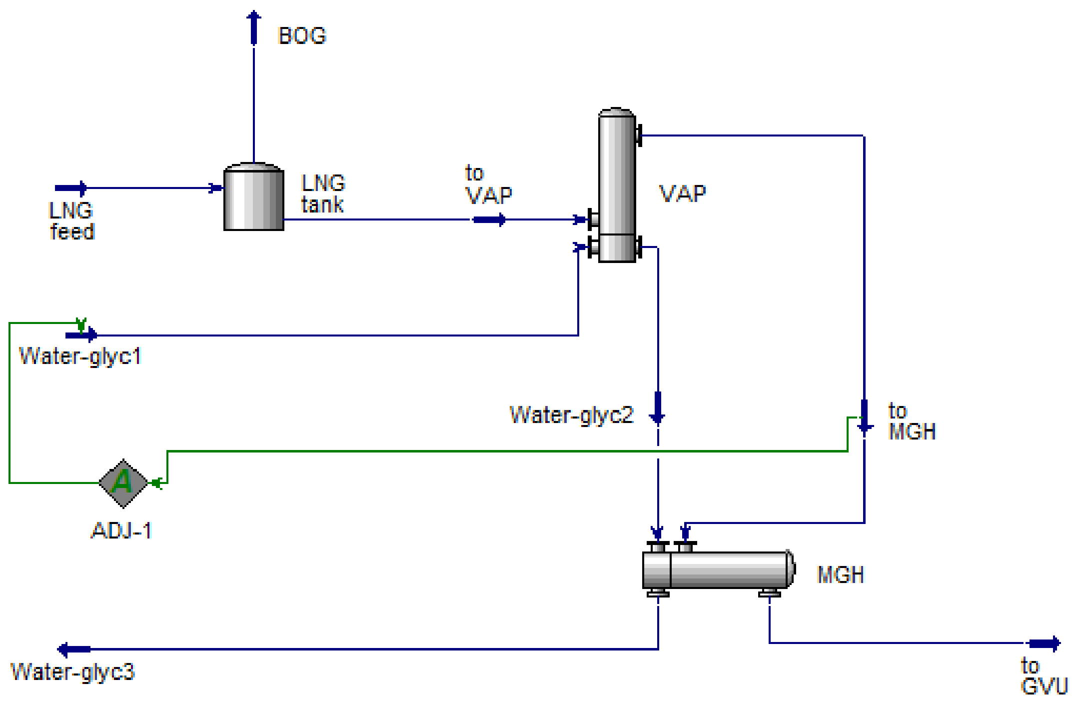

Figure 6-1: HYSYS model of the CASE 1 LP LNG fuel supply system including tank, vaporizer and MGH. A water-glycol solution is used as a heating medium for both vaporizer and MGH. The Adjuster, ADJ-1 adjusts the mass flow of the heating medium required to heat the LNG to a certain temperature.

6.2.2.1 Inputs, boundary conditions and assumptions for CASE 1

The composition of the water-glycol solution used in the simulations is presented in Table 6-6. This composition of 40-volume% of glycol will decrease the water's freezing point to -23,5°C (The Engineering Toolbox, 2015a). Further, the inlet condition of the heating medium to the vaporizer is presented in Table 6-7. The pressure of the stream is set to 7 bar to drive the stream instead of a using a pump. The inlet temperature of the heating medium is in this case fixed at 40°C. The heat exchanger condition for the vaporizer and MGH is presented in Table 6-8. For the vaporizer and the MGH, the initial pressure drop is set to zero for both the heat exchangers. HYSYS and EDR perform the pressure drop calculations. In this model the heat leak for the exchangers are assumed to be zero.

Table 6-6: Water-glycol solution composition

Component	Formula	Liquid volume fraction
Water	H_2O	0,6
Ethylene glycol	$C_2H_6O_2$	0,4

Table 6-7: Inlet condition of the water-glycol solution to the vaporizer

Inlet temperature [°C]	40
Inlet pressure [kPa]	700

Table 6-8: Heat exchanger condition for both the vaporizer and the MGH

	Vaporizer	MGH
TEMA type	B E S	A E L
Initial pressure drop [kPa]	0	0
Heat leak [kJ/h]	0	0

6.2.3 CASE 2: Supply system with one shell-and-tube exchanger

The CASE 2 HYSYS model is shown in Figure 6-2. The system in CASE 2 is similar to CASE 1 except for that the system in CASE 2 only have one heat exchanger; a vaporizer that heats the LNG to an outlet temperature of 0°C. The heating medium used in the vaporizer is a water-glycol solution. The mass flow of the heating medium is adjusted by ADJ-1 to target the outlet value of the temperature for natural gas fuel flow to the GVU.

In the optimization process of the process parameters, the parameter that was manipulated in this case was the inlet temperature of the water-glycol solution to the vaporizer to optimize the mass flow of the heating medium. Simulations were done for inlet temperatures of 40°C to 10°C with a step size at 10°C. Effectiveness calculations of the result of the optimization was done for the vaporizer. Further, thermal and hydraulic calculations were performed for the relevant process parameters in EDR.

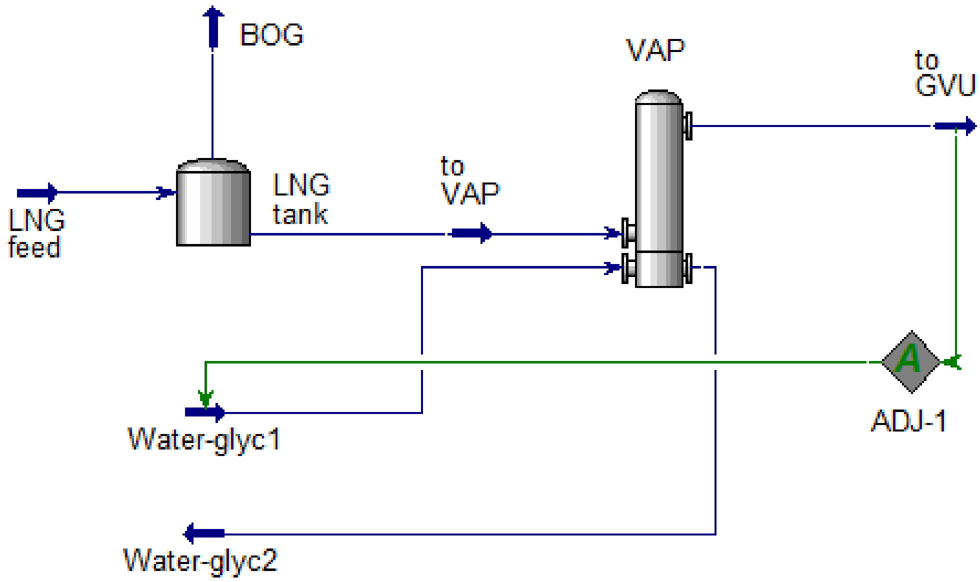

Figure 6-2: HYSYS model of the CASE 2 LP LNG fuel supply system including tank and vaporizer. Seawater is used as a heating medium for the vaporizer. The Adjuster, ADJ-1 adjusts the mass flow of the heating medium required to heat the LNG to a certain temperature.

A sensitivity analysis concerning the mechanical design was also performed for this case, manipulating the shell ID, tube OD, the tube length, L and the baffle pitch, which is the distance between two baffle centers. The values and step sizes that were used for the sensitivity analysis are listed below.

- Shell ID: 200 - 400 with step size 100 [mm]
- Tube OD: 5-25 with step size of 5 [mm]
- Tube length: 1200 – 5200 with step size of 1000 [mm]
- Baffle pitch: 80 -160 with step sizes 20 [mm]

6.2.3.1 Inputs, boundary conditions and assumptions for CASE 2

The composition of the water-glycol solution and the heat exchanger condition for the vaporizer is presented in Table 6-6 and Table 6-8, similar to CASE 1. Also in this case a water-glycol solution with 40% liquid volume of glycol. For the vaporizer it is assumed no heat leak and no pressure loss, the latter will be calculated by EDR in HYSYS when the heat exchanger is sized.

6.2.4 CASE 3: Supply system with one LNG exchanger and one shell-and-tube

In this final case, the vaporizer is modeled as a LNG exchanger, a compact heat exchanger. The LNG from the tank is heated and vaporized by the heating medium, propane. The propane enters the LNG exchanger as gas and is condensed in the process of heat transfer to the LNG/NG. The gas fuel is further heated to the required temperature in the heater, a shell-and-tube exchanger, by running seawater in the tubes while the NG is flowing in the shell.

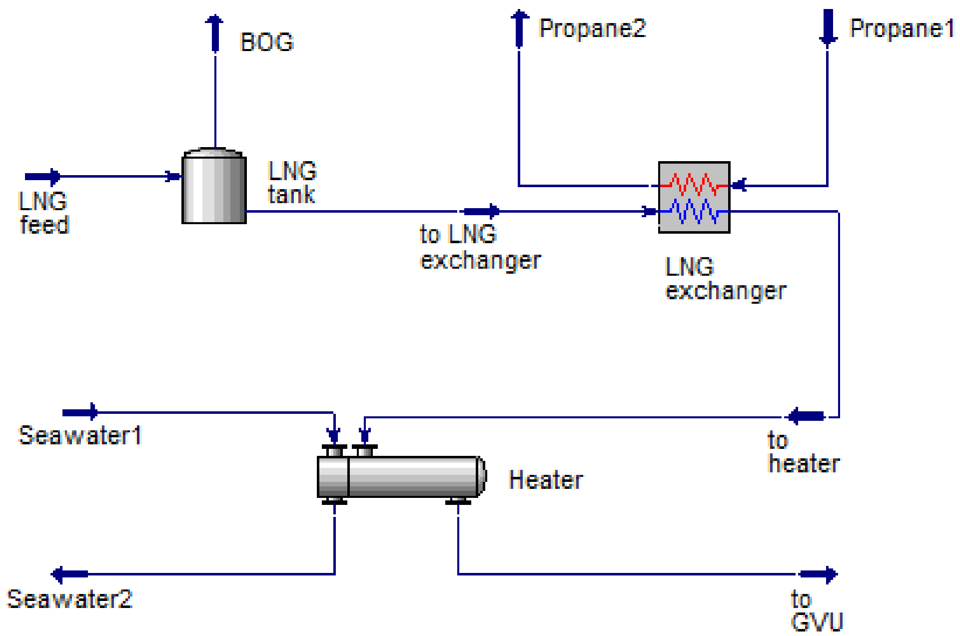

Figure 6-3: HYSYS model of the CASE 3 LP LNG fuel supply system including tank, vaporizer and a heater. Propane and seawater is used as a heating medium for the LNG exchanger and vaporizer respectively.

The mass flow of propane required in the LNG exchanger was calculated by HYSYS. For the heater, simulations of different inlet temperatures where done to determine the mass flow the seawater. Following inlet temperatures of seawater were used in the simulations; 14°C, 10°C and 6°C These seawater temperatures were chosen based on typical temperatures in the North Sea. Thermal and hydraulic calculations of the LNG exchanger and the heater were further done in EDR based for the relevant process parameters.

6.2.4.1 Inputs, boundary conditions and assumptions for CASE 3

The composition and condition of the fuel flow and the inlet flow condition for the GVU is defined in Table 6-1, Table 6-2, Table 6-3 and Table 6-4. The inlet- and outlet temperatures of the propane are set to 0°C and -5°C respectively. The conditions for the LNG exchanger and the heater are presented (Wärtsilä, 2014) where the pressure of the heating medium flow is set to 4,7 bar. The LNG exchanger is specified to heat the LNG to a temperature of -10°C. These values are set based on the specifications of a system by Wärtsilä, see Table 6-9.

Table 6-9: LNG exchanger condition. The values of the temperature and pressure are set based on specification of a system by Wärtsilä.

Exchanger/TEMA type	Compact (plate/fin)	A E L
Inlet temperature of LNG [°C]	-160	-10
Outlet temperature of NG [°C]	-10	0
Inlet temperature of Propane/seawater [°C]	0	6
Outlet temperature of Propane/seawater [°C]	-5	TBD
Inlet pressure of Propane/sewater [kPa]	470	700
Heat leak [kJ/h]	0	0
Pressure drop [kPa]	0	0

The system has been simplified by excluding the sweater circuit for the LNG exchanger to vaporize the propane. Valves, fittings and any pump have also been excluding for simplicity. The pressure of the seawater circuit is set to 7 bar to simulate a LP pump to drive the flow. Similar to the other cases it is assumed that there is no heat leak and no pressure drop, whereas the pressure drop will be calculated when the exchanger is proper sized.

7 Results and discussion

This chapter will highlight the most evident results, for full details of results on simulations from HYSYS and EDR see Appendix B and C.

7.1 CASE 1

Figure 7-1 shows the HYSYS model of the fuel supply system for CASE 1 with the process parameters that will be discussed in section 7.1.1. The figure includes the conditions for the vaporizer, main gas heater (MGH) and the streams.

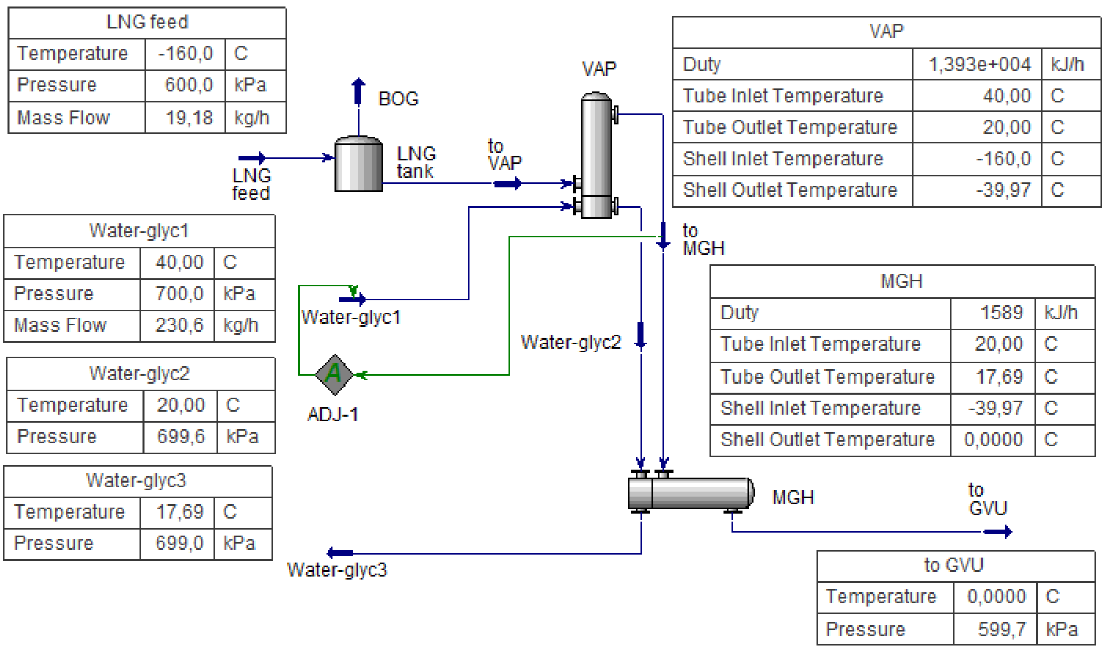

Figure 7-1: HYSYS model of the low pressure (LP) LNG fuel supply system in CASE 1. The conditions of the vaporizer and MGH are presented in the tables in the figure as well as the stream conditions for the LNG/natural gas (NG) and the water-glycol solution.

7.1.1 Process parameters optimization in HYSYS

Figure 7-2 shows the result of the optimization of outlet temperature of the LNG/NG from the vaporizer described in section 6.2.2. The result indicates that if the outlet temperature of LNG/NG from the vaporizer increases, the more mass flow of the water-glycol solution is needed. Mark that the inlet- and outlet temperature of the heating medium is fixed at 40°C and 20°C respectively in the vaporizer. However, it is not for the heater and the outlet temperature of the water-glycol solution is increasing with increasing mass flow of the heating medium.

The HYSYS simulation for case 1 showed that for the vaporizer to fully vaporize the LNG to NG, the minimum outlet temperature of the NG from the vaporizer must be about -40°C. For the heating medium to be able to heat the LNG to this temperature, Figure 7-2 shows the mass flow of the water-glycol solution need to be a minimum of 230,6 kg/h for CASE 1.

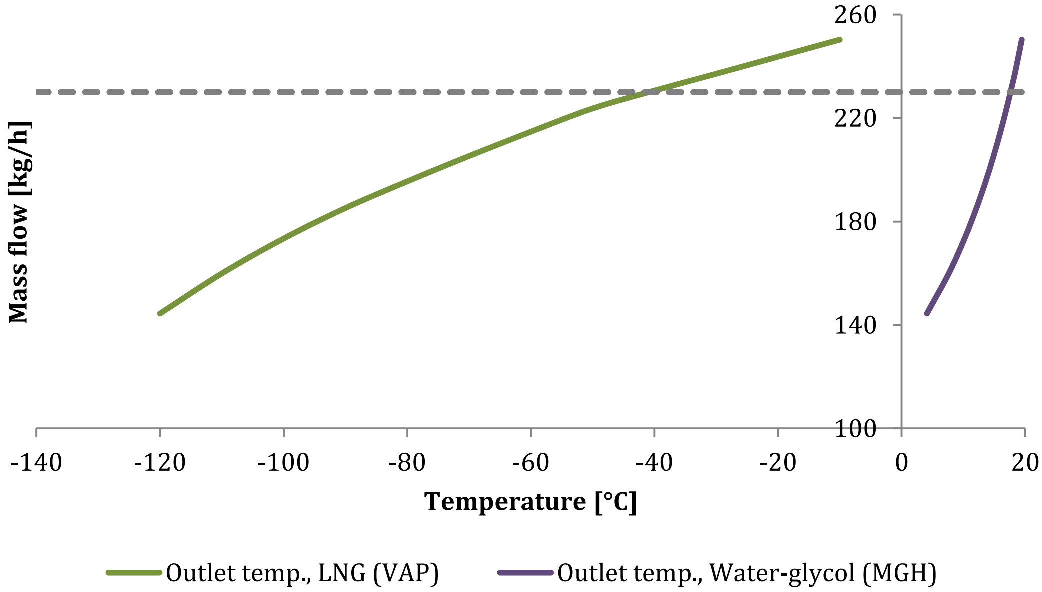

Figure 7-2: The outlet temperature of LNG from the vaporizer and the outlet temperature of the water-glycol solution from the MGH based on the mass flow of glycol solution for both of the heat exchangers. The grey stippled line marks the mass flow of the solution for the temperatures where the vaporizer fully vaporizes the LNG to NG, 230,6 kg/h.

Figure 7-3 is the result of the effectiveness calculations of the process optimization of CASE 1. The result shows the vaporizer increases effectiveness as the mass flow of the heating medium increases, and for the MGH vice versa. An expected result as more of the LNG is heated with higher mass flow in the vaporizer, consequently the heater needs to heat up less for the LNG to become 0°C. The heat exchanger system is most effective where the two lines cross at 214 kg/h. However, for the vaporizer to fully vaporize the LNG the mass flow of heating medium most be minimum of 230,6 kg/h. At this flow, the calculated effectiveness for the set of heat exchangers will be 0,71 for the vaporizer and 0,5 for the MGH as seen in Figure 7-3. This is also the most optimum design for the case indicated by the figure.

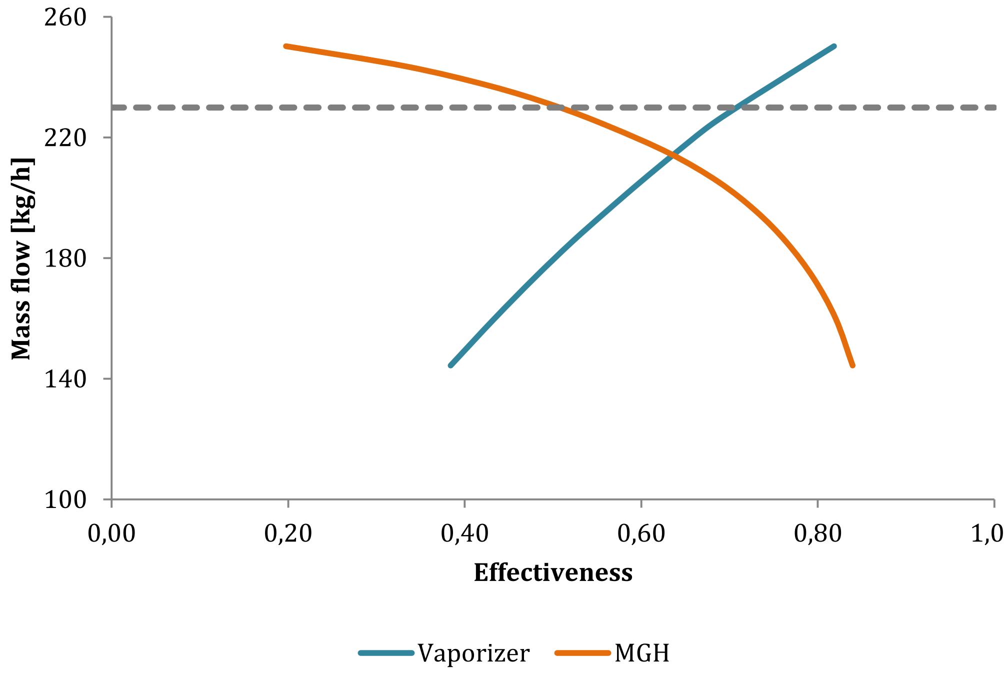

Figure 7-3: Result of effectiveness calculations for the vaporizer and MGH for varies mass flow of the heating medium. The stippled grey line marks the mass flow of water-glycol solution at 230,6 kg/h.

HYSYS provides data regarding the heat flow between the hot fluid and the cold fluid in the vaporizer and the MGH. The results of the data for the optimum design are illustrated in Figure 7-4 and Figure 7-5 . Comparing the two figures confirm that the heat transferred in the vaporizer is much higher than the heat transferred in the MGH. The hot fluid, water-glycol solution, enters the vaporizer at 40°C and exits at a temperature of 20°C, and the heat loss through the vaporizer is transferred to the cold fluid, LNG/NG. The amount of heat reduced from the water-glycol solution should by the identic amount of heat gained for the LNG/NG if no heat leak is assumed, according to the conservation of mass and energy, Eq. 5-3. However this is not supported by the values from HYSYS. This will be further discussed in section 7.4.

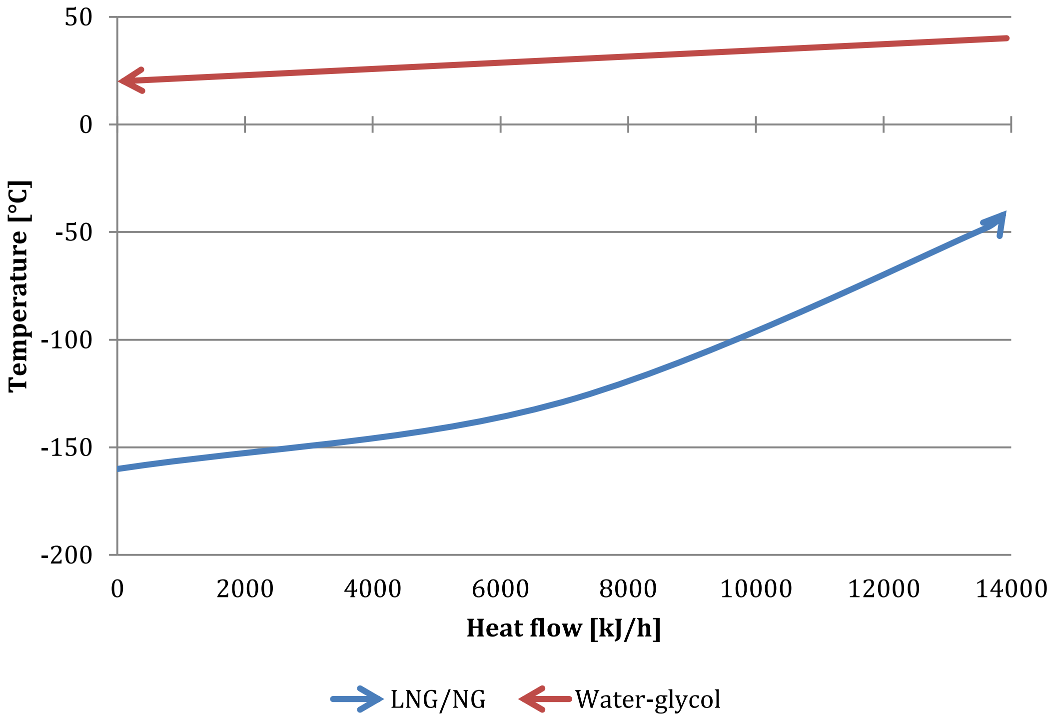

Figure 7-4: Temperature - heat flow diagram for the vaporizer

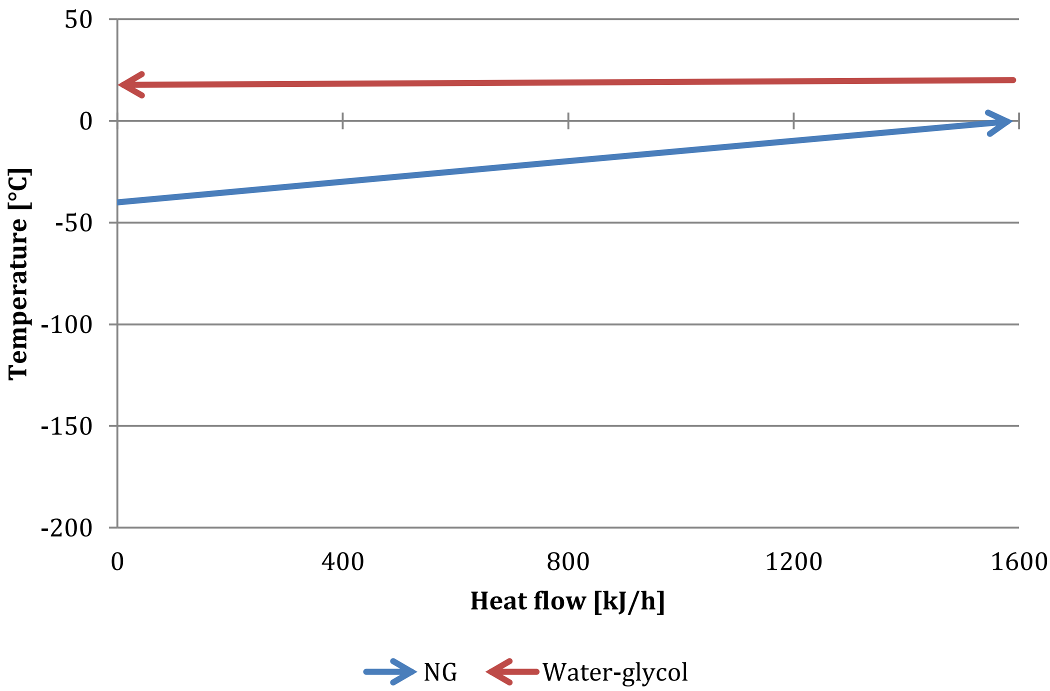

Figure 7-5: Temperature - heat flow diagram for the MGH

Table 7-1 shows the process parameters and the calculated effectiveness of the optimum design found for CASE 1 after optimization of outlet temperature of the LNG/NG from the vaporizer. The effectiveness and the effective surface area are discussed in the next section.

Table 7-1: Process parameters and the calculated effectiveness of the vaporizer and the MGH.

Process parameters	Vaporizer	MGH
Inlet temperature, LNG [°C]	-160	-40
Outlet temperature, LNG [°C]	-40	0
Inlet temperature, water-glycol [°C]	40	20
Outlet temperature, water-glycol [°C]	20	17,7
Mass flow, LNG [kg/h]	19,2	19,2
Mass flow, water-glycol [kg/h]	230,6	230,6
Effectiveness [-]	0,71	0,50

7.1.2 Mechanical design and thermal and hydraulic calculations in EDR

The thermal design parameters of the vaporizer and MGH were selected based on the optimum design found in the last section. Hence, the outlet temperature of NG in the vaporizer is set at -40°C, so the LNG is fully vaporized to NG through the vaporizer. The mechanical results of the heat exchangers for the simulations performed by EDR are presented in Table 7-2 and the thermal and hydraulic results are presented in Table 7-3.

Table 7-2: Mechanical parameters retrieved from EDR for the vaporizer and the MGH.

Mechanical parameters	Vaporizer	MGH
TEMA type	BES	BEL
Shell ID [mm]	205	205
Tube OD [mm]	19,05	19,05
Tube length [mm]	1200	1200
L/D ratio	5,9	5,9
Baffle pitch	120	135
Number of baffles	6	6
Number of tubes	23	30
Effective surface area [m^2]	1,4	2
Weight [kg]	233	219
Estimated cost [USD]	9667	9993
Oversurface [%]	149	310

The percentage of oversurface in Table 7-2 indicates that the heat transfer surface area is larger than necessary, especially for the MGH, which has a surface area that is 310% lager than necessary. In redesign of the heat exchangers, the effectiveness can be increased by decreasing the surface area. Even though liquids has a higher heat capacity than gas, meaning that it requires more energy to heat liquids than gas, the larger effective surface area for the MGH indicate that it utilizes the heat transfer area less effective.

The lower weight for the MGH compared to the vaporizer even though it has a larger effective surface area is due to the TEMA type design of the heat exchanger. The setting plans for the heat exchanger are presented in Figure 7-6 and Figure 7-7 where the difference in design is clearly visible.

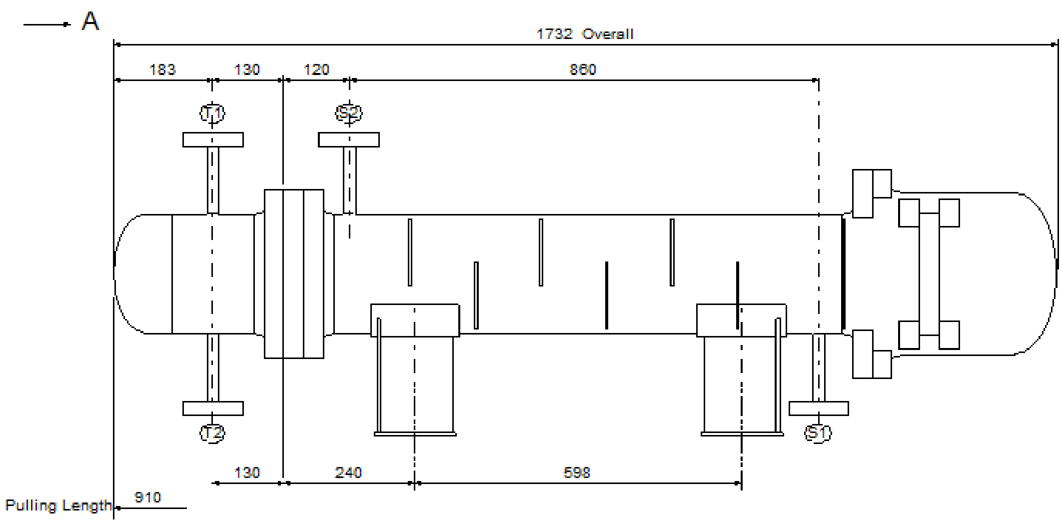

Figure 7-6: The setting plan for the vaporizer. TEMA type is BES. The drawing is retrieved from EDR for the relevant process parameters. The overall length of the heat exchanger is 1,73 m.

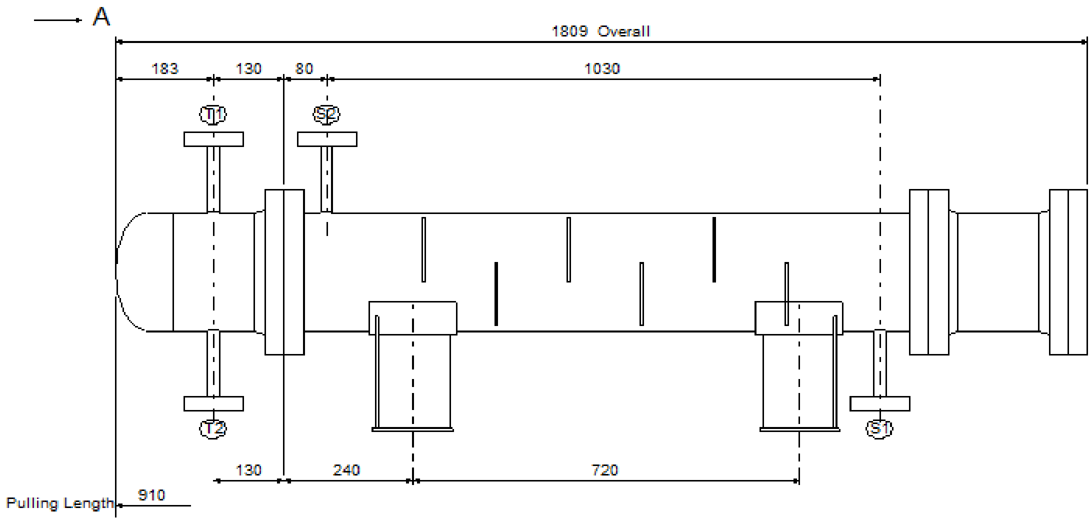

Figure 7-7: The setting plan for the MGH. TEMA type is BEL. The drawing is retrieved from EDR for the relevant process parameters. The overall length of the heat exchanger is 1,8 m.

Table 7-3 presents some of the thermal and hydraulic parameters calculated by EDR. The pressure drops on the tube-side and shell-side for both of the heat exchangers are very small. The MGH has a slightly larger pressure drop. As discussed earlier, the vaporizer is more effective than the MGH, this is further shown here where the total heat exchanged for the vaporizer is 3,9 kW, and the MGH only exchange 0,9 kW.

Table 7-3: Thermal and hydraulic parameters for the vaporizer and the MGH based on calculations in EDR

Thermal and hydraulic parameters	Vaporizer	MGH
Pressure drop, shell-side [kPa]	0,1	0,2
Pressure drop, tube-side [kPa]	0,4	0,6
Total heat exchanged [kW]	3,9	0,4
Overall heat flux [kW/m^2]	7,1	0,9
Heat duty [kJ/h]	13930	1590
LMTD [C]	123,3	35,6

7.2 CASE 2

Figure 7-8 shows the HYSYS model of the fuel supply system for CASE 2 with the process parameters that will be discussed in section 7.2.1. The figure includes the conditions for the vaporizer and the streams.

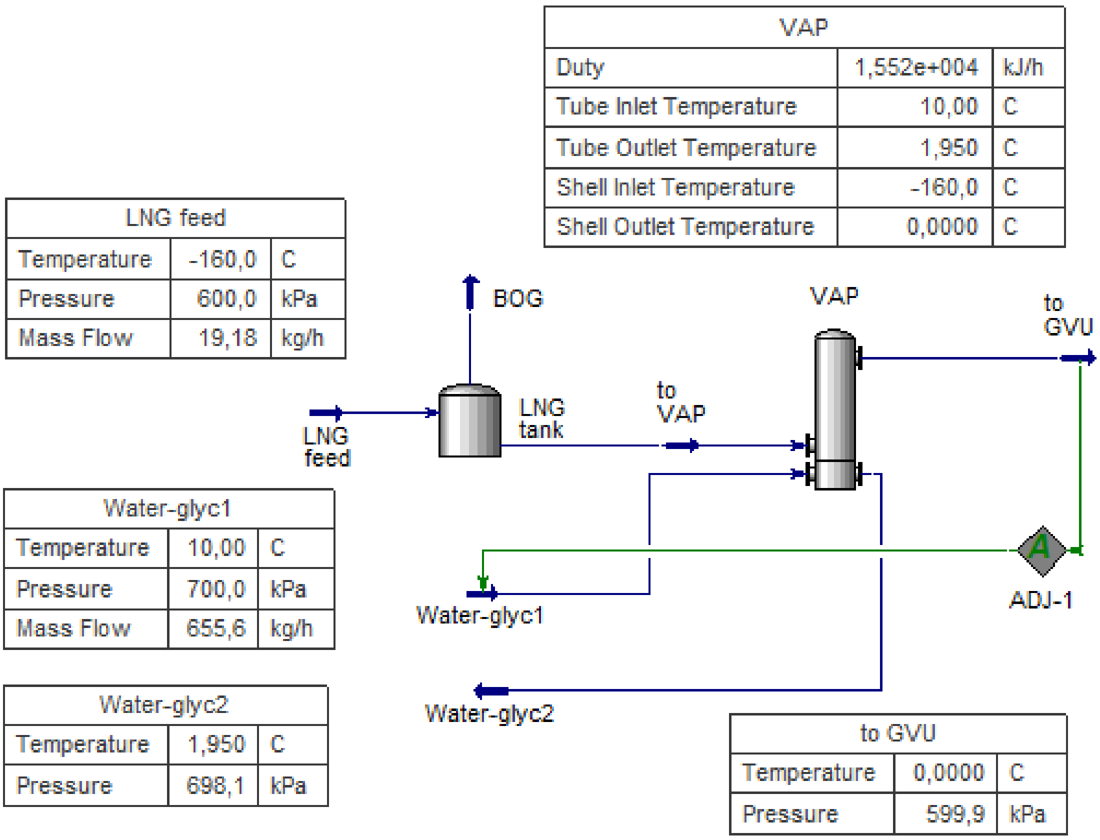

Figure 7-8: HYSYS model of the LP LNG fuel supply system in CASE 2. The conditions of the vaporizer are presented in the table in the figure as well as the stream conditions for the LNG/NG and the water-glycol solution.

7.2.1 Process parameters optimization in HYSYS

Figure 7-9 shows the result of the optimization of the inlet temperature of the water-glycol solution to the vaporizer described in section 6.2.3. The results shows that if the inlet temperature of the water-glycol solution decreases the mass flow of the solution required increases. Which means that lower inlet temperature of the water-glycol solution demands more flow to vaporize the LNG. The effectiveness of the vaporizer calculated indicate that the lower inlet temperature, hence higher mass flow, the more effective is the heat exchanger, Figure 7-10. To heat the LNG to 0°C with an inlet temperature of 10°C requires a flow of water-glycol solution at 656 kg/h, Table 7-4.

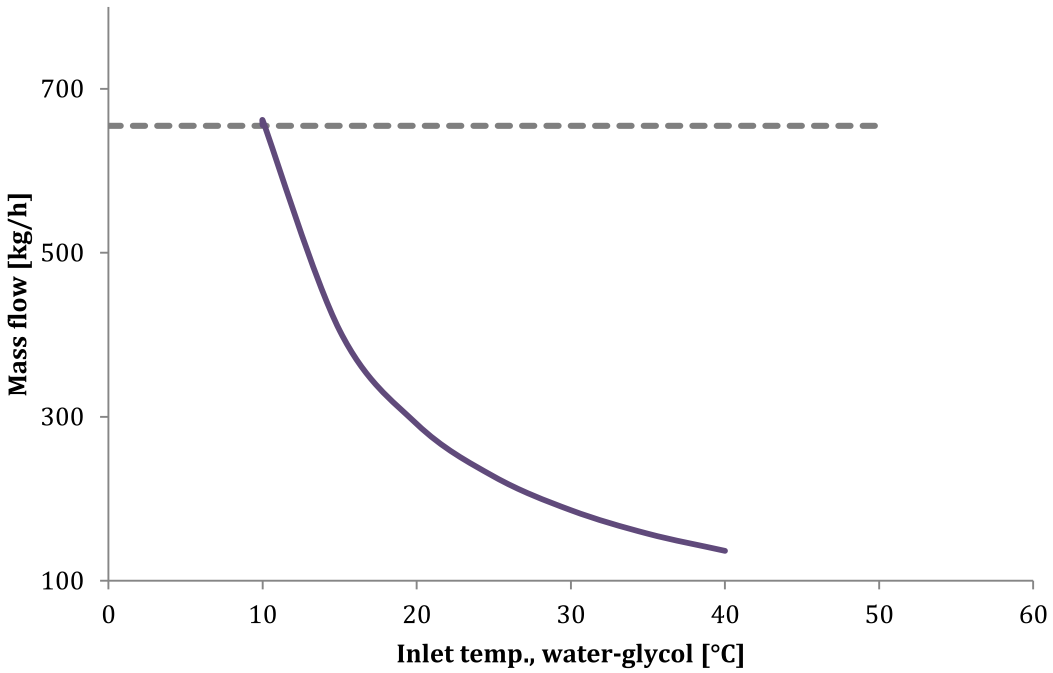

Figure 7-9: Inlet temperature of the water-glycol solution to the vaporizer based on varies mass flow of the solution. The stippled grey line marks the mass flow where the inlet temperature of the water-glycol solution is at its lowest for the simulations carried out, 656 kg/h.

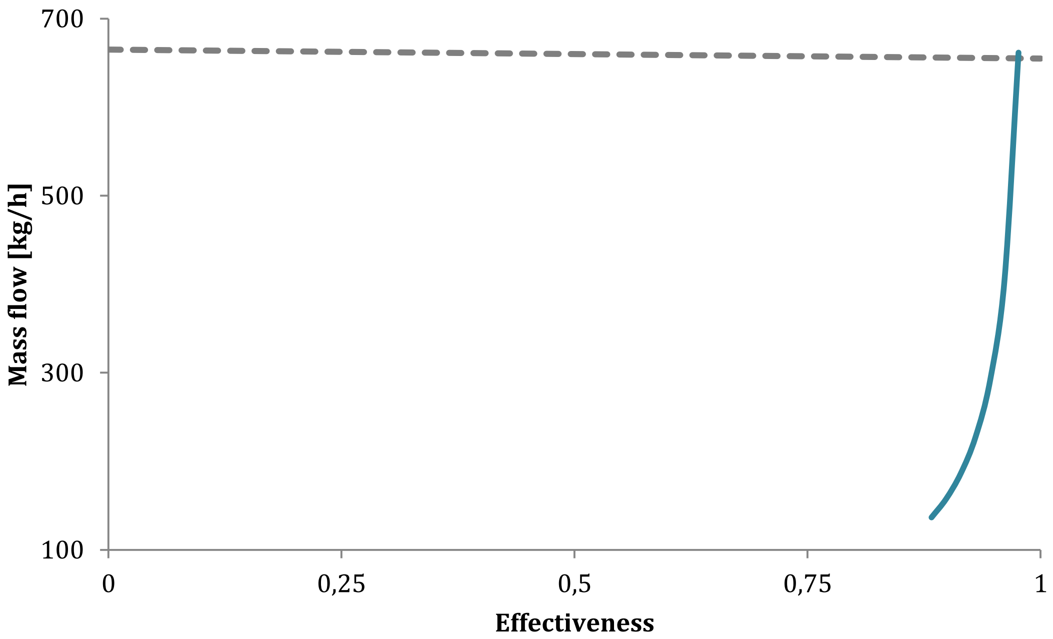

Figure 7-10: The effectiveness of the vaporizer and mass flow of the water-glycol solution. The stippled grey line marks the mass flow where the exchanger is most effective; 656 kg/h.

Figure 7-11 shows, with data from HYSYS, the heat flow between the hot fluid and the cold fluid in the vaporizer. Compared to CASE 1, the LNG/NG is absorbing more heat as the blue slope indicates, from the water-glycol solution when the solutions temperature is changing from 10°C to 2°C.

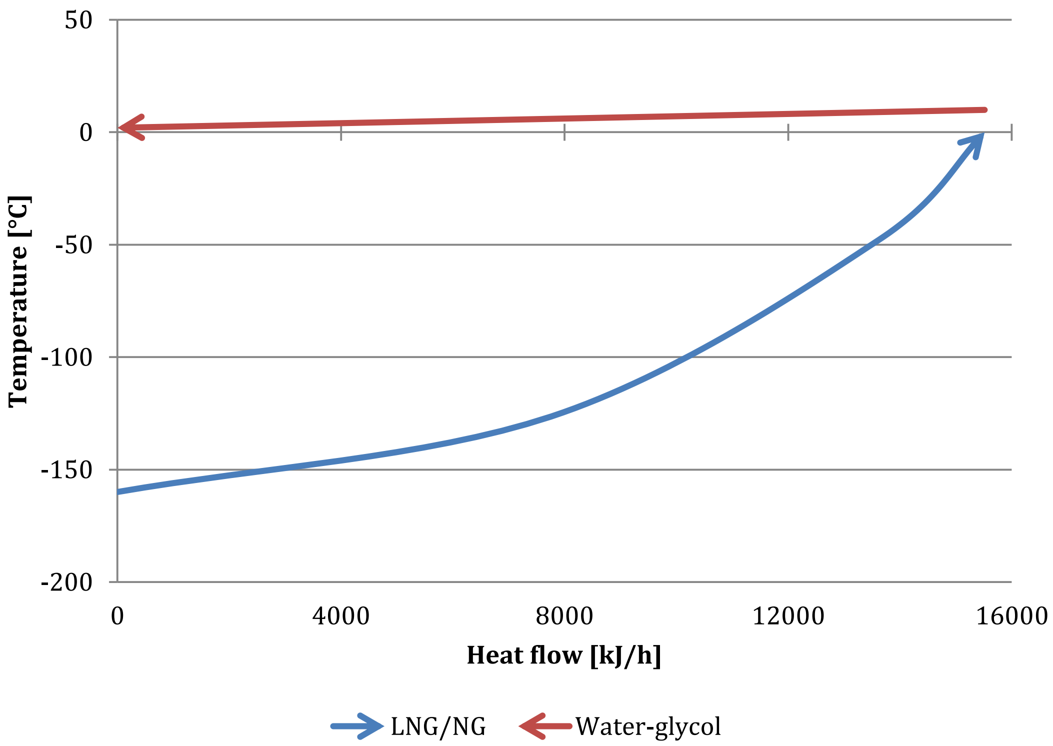

Figure 7-11: Temperature – heat flow diagram for the vaporizer.

Table 7-4 shows the process parameters the vaporizer. The table also includes the effective surface area and the effectiveness of the heat exchanger, which is 1,8 m^2 and 0,97 respectively. This very high effectiveness of the heat exchanger is caused by a low inlet temperature of the water-glycol solution, which increases the mass flow of the solution. This effect makes the heat transfer more effective because it increases the hot fluids ability to transfer heat to the cold fluid as seen in Eq. 5-3.

Table 7-4: Process parameters and the required effective surface area including the effectiveness of the vaporizer.

Process parameters	Vaporizer
Inlet temperature, LNG [°C]	-160
Outlet temperature, LNG [°C]	0
Inlet temperature, water-glycol [°C]	10
Outlet temperature, water-glycol [°C]	1,95
Mass flow, LNG [kg/h]	19,1
Mass flow, water-glycol [kg/h]	655,6
Effectiveness [-]	0,97

7.2.2 Mechanical design and thermal and hydraulic calculations in EDR

The mechanical parameters for the vaporizer are presented in Table 7-5. For this exchanger, the oversurface is only 15%, which means that it is not too large as for the heat exchangers in CASE 1. The surface area at 1,8 m^2 is also much lower than the heat exchangers in CASE 1 combined. The overall length of the exchanger as seen in the setting plan, Figure 7-12, is the same as for the vaporizer in the previous case.

Table 7-5: Mechanical parameters retrieved from EDR for the vaporizer.

Mechanical parameters	Vaporizer
TEMA type	BES
Shell ID [mm]	205
Tube OD [mm]	19,05
Tube length [mm]	1200
L/D ratio [-]	5,9
Baffle pitch [mm]	135
Number of baffles	6
Number of tubes	30
Effective surface area [m^2]	1,8
Weight [kg]	233
Estimated cost [USD]	9768
Oversurface [%]	15

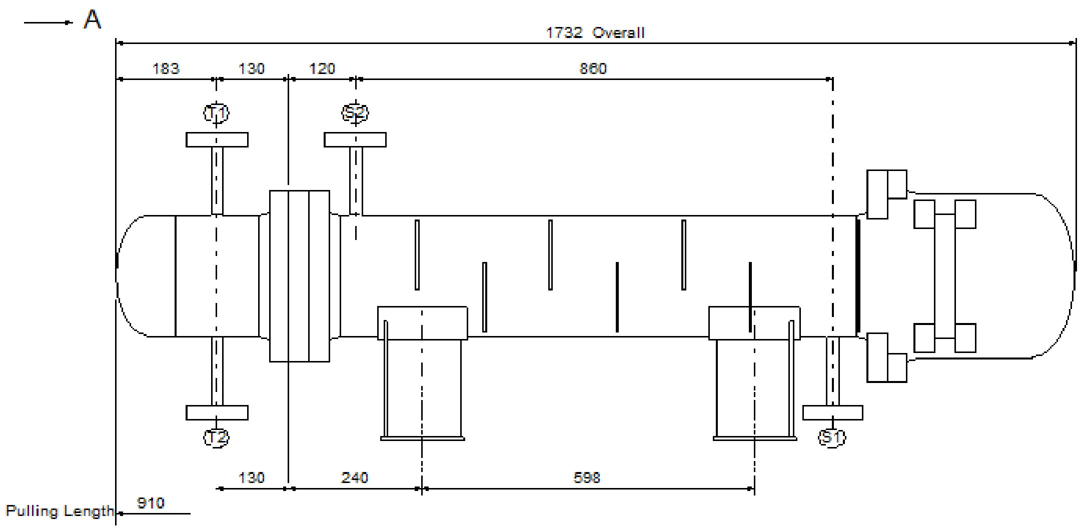

Figure 7-12: The setting plan for the vaporizer retrieved from EDR design for the relevant process parameters. The overall length of the heat exchanger is 1,73 m.

The thermal and hydraulic calculations are presented in Table 7-6. The pressure drop on the tube-side for this exchanger is somewhat larger than for the exchangers in CASE 1. However, the pressure drops are still very small and only 1,9 kPa on the tube-side. The total heat exchanged, 4,3 kW, is the same as the heat exchanged for the vaporizer and the MGH combined in previous case.

Table 7-6: Thermal and hydraulic parameters for the vaporizer based on calculations in EDR

Thermal and hydraulic parameters	Vaporizer
Pressure drop, shell-side [kPa]	0,1
Pressure drop, tube-side [kPa]	1,9
Total heat exchanged [kW]	4,3
Overall heat flux [kW/m^2]	2,8
Heat duty [kJ/h]	15520
LMTD [°C]	54,8

7.3 CASE 3

Figure 7-13 shows the HYSYS model of the fuel supply system for CASE 3 with the process parameters discussed section 7.3.1. The figure includes the conditions for the vaporizer, which is modeled as a LNG exchanger or a compact heat exchanger, heater and the streams.

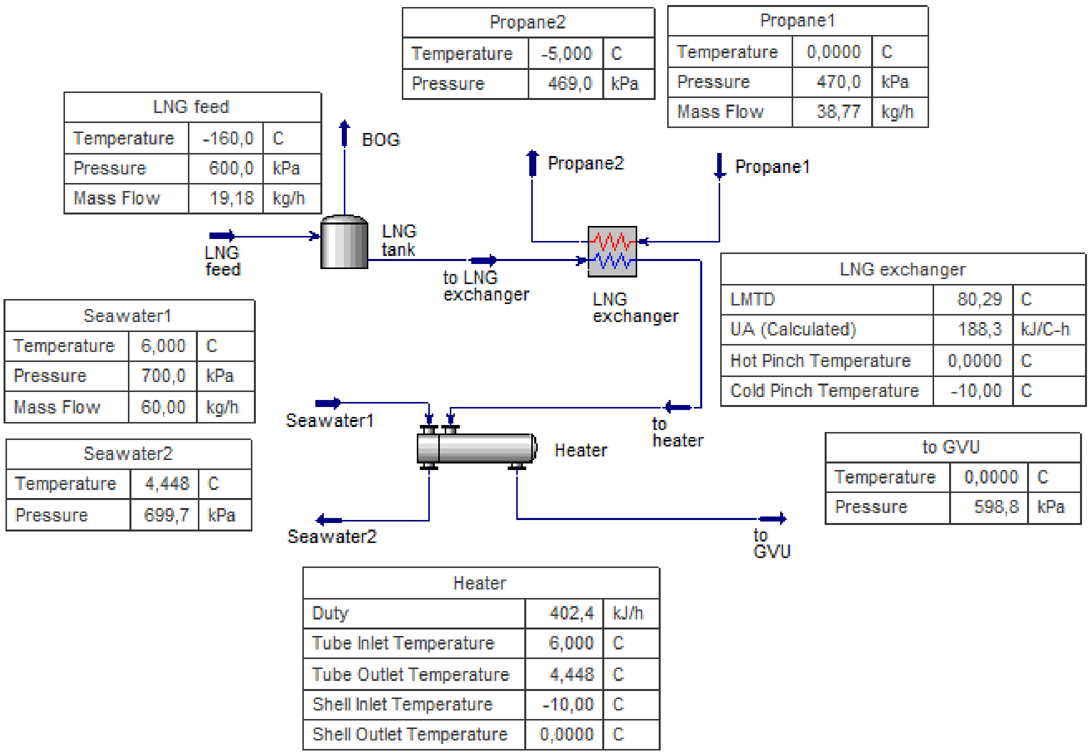

Figure 7-13: HYSYS model of the LP LNG fuel supply system in CASE 3. The conditions of the vaporizer and heater are presented in the tables in the figure as well as the stream conditions for the LNG/NG, propane and seawater.

7.3.1 Process parameters optimization in HYSYS

Most of the parameters for CASE 3 was set and the only process parameters that were manipulated was the inlet temperature of the seawater for the heater. In Figure 7-14 it is illustrated how the effectiveness of the heat exchanger increases when the inlet temperature of the seawater is decreases. The lower temperature difference between the heating- and cooling medium, the more effective is the heat transfer. The efficiency increases because the amount of heat transferred will be a larger part of the total possible heat transfer. The grey stippled line marks the inlet temperature of the seawater at the exchanger's most effective operation, which is 6°C where the effectiveness of the heat transfer rate is 0,43.

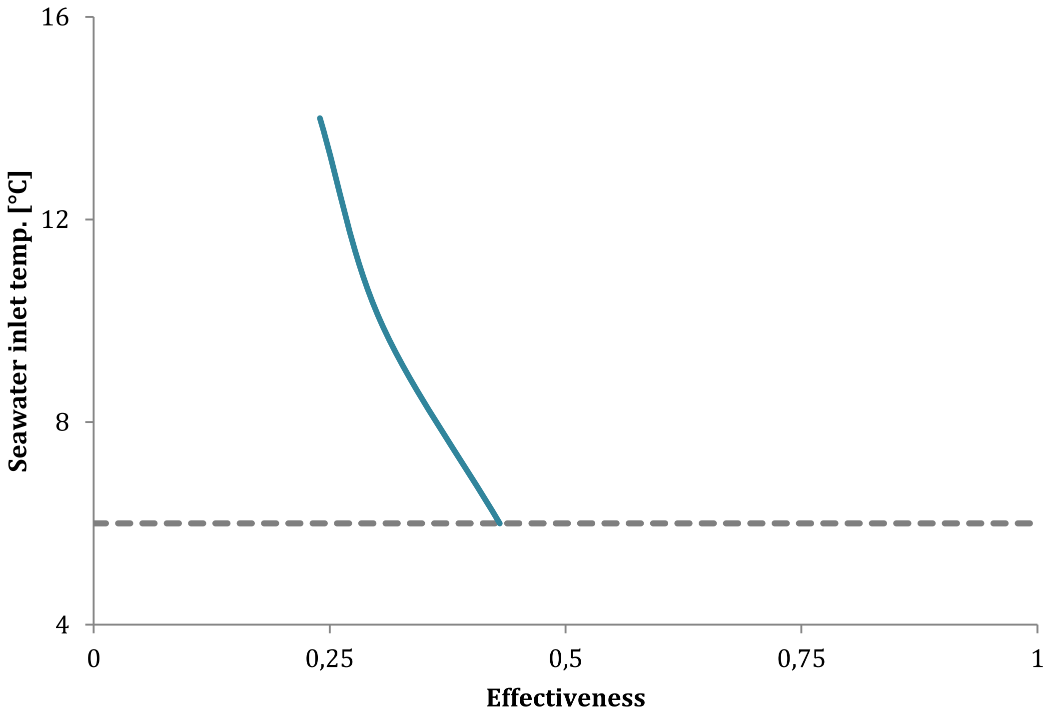

Figure 7-14: The effectiveness and inlet seawater temperature for the heater. The grey stippled line marks the temperature at which the heater is most effective, 6°C.

In Figure 7-15 the heat flow from propane to the LNG/NG is illustrated with data from HYSYS. It shows that propane is a very effective heating medium as the temperature of propane decreases only from 0°C to 5°C when it heats the LNG/NG from -160°C to -10°C. In addition the propane demands a much lower mass flow, 38,8 kg/h as seen in Table 7-7, compared to water-glycol solution discussed in CASE 1 and CASE 2.

Figure 7-16 shows the heat flow changes for the seawater and the NG in the heater. The effectiveness of this heater is somewhat lower than for the exchangers is CASE 1 and CASE 2. A larger mass flow of seawater would have increased the effectiveness of the heater.

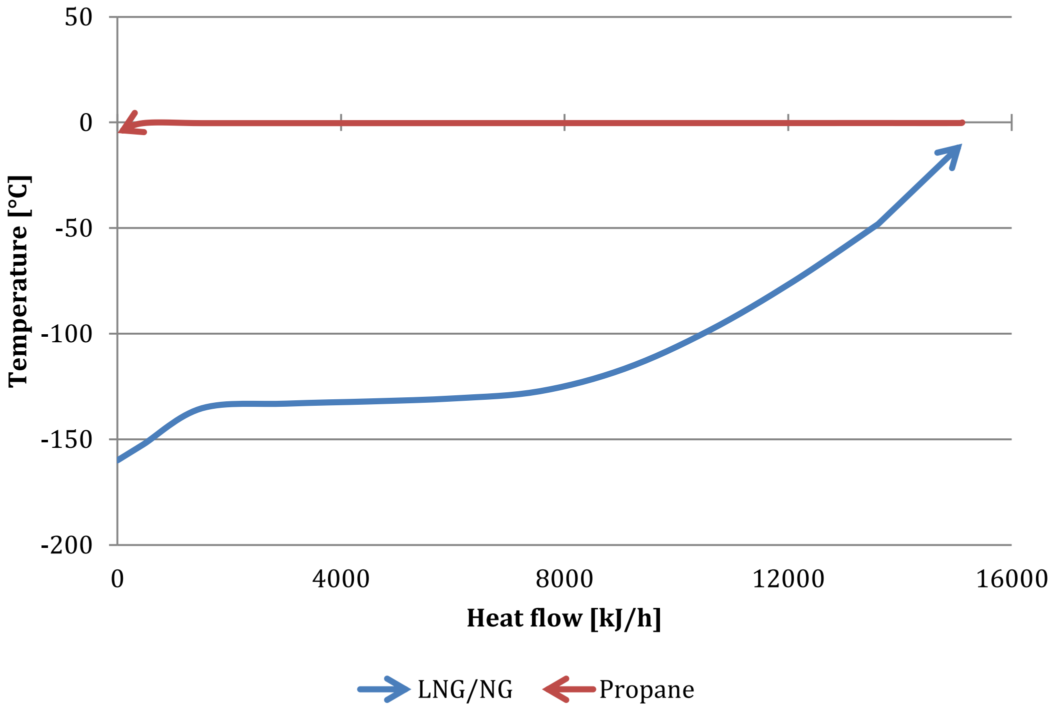

Figure 7-15: Temperature – heat flow diagram for the LNG exchanger.

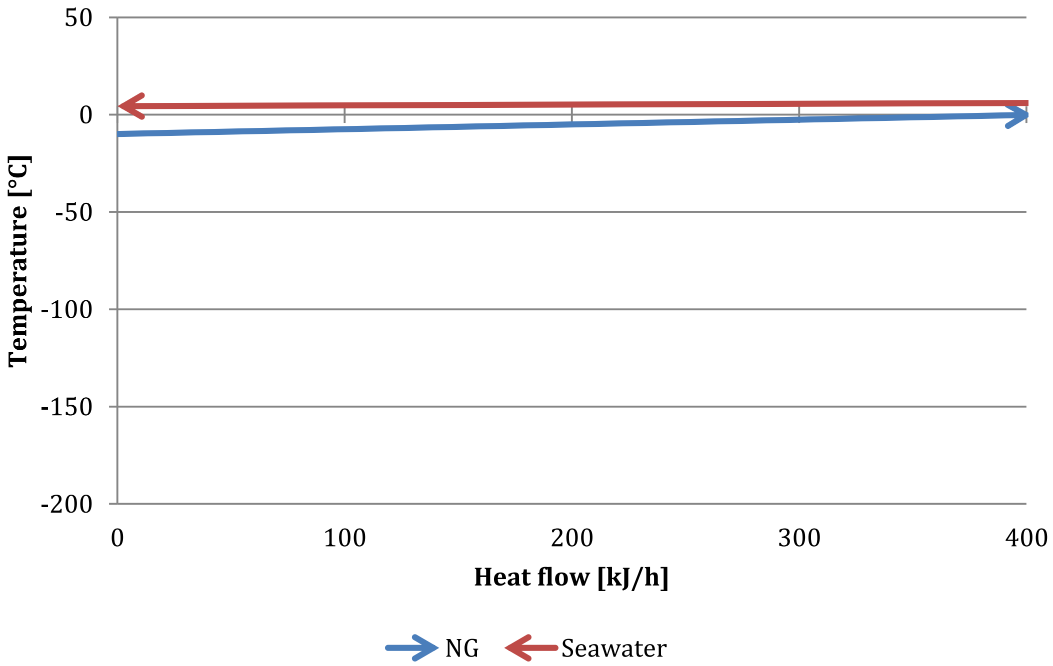

Figure 7-16: Temperature – heat flow diagram for the heater.

The process parameters for the LNG exchanger and the most effective heater are presented in Table 7-7 including the effectiveness of the exchangers, which is 0,61 for the LNG exchanger and 0,43 for the heater.

Table 7-7: The process parameters for the LNG exchanger and heater

Process parameters	LNG exchanger	Heater
Inlet temperature, LNG [°C]	-160	-10
Outlet temperature, LNG [°C]	-10	0
Inlet temperature, propane / water-glycol [°C]	0	6
Outlet temperature, propane / water-glycol [°C]	-5	4,5
Mass flow, LNG [kg/h]	19,1	19,1
Mass flow, propane / water-glycol [kg/h]	38,8	60
Effectiveness [-]	0,61	0,43

7.3.2 Mechanical design and thermal and hydraulic calculations in EDR

The EDR for a compact heat exchanger that was supposed to size the LNG exchanger including performing the thermal and hydraulic calculations did not work. Therefore, the mechanical design and the thermal and hydraulic calculations are only presented for the heater, which is a shell-and-tube exchanger. The mechanical parameters concerning the design of the heater are presented in Table 7-8. For this exchanger, the oversurface is 359%, which can explain the low effectiveness of 0,43. The setting plan of the heater with an overall length of 1, 79 m is shown in Figure 7-17.

Table 7-8: Mechanical parameters retrieved from EDR for the heater.

Mechanical parameters	Heater
TEMA type	AEL
Shell ID [mm]	205
Tube OD [mm]	19,05
Tube length [mm]	1200
L/D ratio [-]	5,9
Baffle pitch [mm]	135
Number of baffles	6
Number of tubes	30
Effective surface area [m^2]	2
Weight [kg]	231
Estimated cost [USD]	10834
Oversurface [%]	359

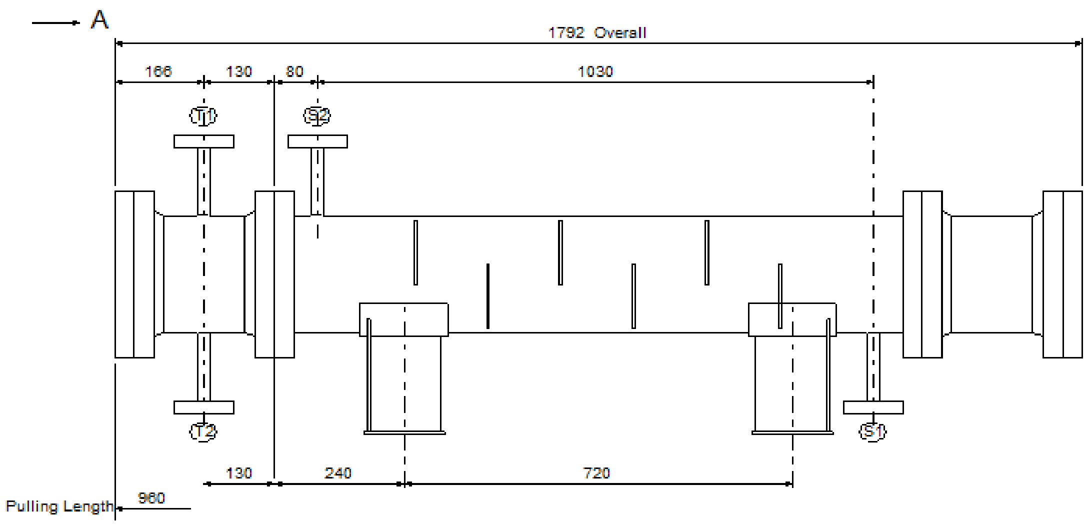

Figure 7-17: The setting plan for the heater retrieved from EDR design for the relevant process parameters. The overall length of the heat exchanger is 1,79 m.

The thermal and hydraulic calculations for the heater are presented in Table 7-9. The pressure drops at 0,2 kPa for the shell-side and 0,04 for the tube-side are very small and similar for the other cases. The calculations show that the total heat exchanged in the heater at 0,1 kW is very small, and the exchanger should be resized to better fit its purpose.

Table 7-9: Thermal and hydraulic parameters for the heater based on calculations in EDR

Thermal and hydraulic parameters	**Heater**
Pressure drop, shell-side [kPa]	0,2
Pressure drop, tube-side [kPa]	0,04
Total heat exchanged [kW]	0,1
Overall heat flux [kW/m^2]	0,3
Heat duty [kJ/h]	402,4
LMTD [°C]	18

7.4 Remarks on HYSYS simulations

This section will discuss the different flaws met when using HYSYS and a weakness in the methodology used for CASE 1.

It was experienced that the heat exchanger components in HYSYS does not conserve the energy and mass as expected. Based on calculation of the equation of conservation of energy Eq. 5-3. To illustrate how much calculations of mass flow of the heating medium in HYSYS deviates from the energy balance, an overview of values are presented in Table 7-10. The calculations are made for an inlet- and outlet temperature of LNG at -160°C and 0°C respectively, and an outlet temperature of the water-glycol solution at 10°C with the inlet temperatures of 40°C, 30°C and 20°C. The mass flow of LNG is set to 19,18 kg/h. Comparing the calculation between the energy balance and HYSYS shows that HYSYS is about 1,65 times larger than the calculations by the energy balance for all the three cases.

Table 7-10: Comparison of the required mass flow of water-glycol solution calculated in HYSYS and with the energy balance

Mass flow of water-glycol	HYSYS	Energy balance
Mass flow at 40 °C [kg/h]	146,4	88,7
Mass flow at 30 °C [kg/h]	220,2	133,9
Mass flow at 20 °C [kg/h]	441	269,4

Another thing to note is that the adjuster utilized to estimate the flow required to heat the LNG/NG to certain temperatures is affected by the temperature set for the vaporizer and the MGH. In CASE 1, the temperatures that are set may not be beneficial for the overall design of the system though the flow estimated is larger than necessary. The fuel supply system can operate with a lower flow of water-glycol solution, which will affect the outlet temperature of the heating medium from the MGH and as long as this is not causing a temperature cross and that the inlet temperature of the NG to the GVU is in the range of 0°C - 60°C, the heat exchanger will become more effective. Considering that these are steady-state simulations, the heat exchanger will be able to operate with these temperatures and flow, this may not be the case in dynamic mode. The heat exchanger should be sized so it can operate in the most efficient way including a control system that will adjust the flow of heating medium according to the fuel consumption in a way that temperature crosses in the heat exchangers are avoid and the requirements still comply.

In CASE 1, the certain temperature where the LNG is fully vaporized to NG should have been detected first, and then the amount of water-glycol solution should have been estimated for the vaporizer. Further, the MGH should have been sized and adjusted based on the process parameters from the vaporizer. This problem only affects CASE 1, because there are two heat exchangers with one joint circuit of hot fluid.

7.5 Evaluation of the system design for CASE 1, 2 and 3

A brief comparison of the three system designs is done in Table 7-11 between the total weight, and investment cost. It is important to note that the system in CASE 3 is not comparable though the lack of calculations and estimates for the compact exchanger. The weight and costs for the compact exchanger are therefore not included in Table 7-11.

Table 7-11: Comparison of the fuel supply system comprising of heat exchangers for each system design. Note that the costs are investments costs for the heat exchangers only.

	CASE 1	**CASE 2**	**CASE 3***
Weight [kg]	452	233	231
Costs [USD]	19660	9768	10834

The operating costs concerning the amount glycol or propane, pump duty and external heat for the water-glycol solution circuits have not been considered. An optimal system for one vessel type is not necessary the optimal system for another vessel type. Factors such as space available, heat sources available and flexibility must also be considered before designing the system. Heat such as waste heat from the engines or from other sources may be available for heating the heating medium. In the three different cases, water-glycol solution, propane and seawater are used as heating medium in heat exchangers.

The final heat exchanger design in CASE 1 and CASE 3 should have been resized to make them more effective. As seen, the exchangers in these cases are clearly oversized and resizing these can reduce costs, weight and space required.

The heat exchanger in CASE 2 proves to be very effective. The little disadvantage with this system is that it does not allow the utility of BOG as fuel without an external heater connected to the fuel system.

A compact heat exchanger is smaller in size than a shell-and-tube exchanger but in the other end, the investments costs for these types of exchangers are higher than for a shell-and-tube exchanger as discussed in 3.3. A more compact system is to prefer when the spaces available for a fuel system is narrow.

7.6 Sensitivity analysis for shell-and-tube exchanger

A sensitivity analysis for the shell-and-tube exchangers were performed to give an overview of how process parameters affect other process parameters in the system. Further, to understand how the mechanical design can affect the thermal and hydraulic parameters such as the heat transfer rate and the pressure drop. By manipulating the mechanical design in EDR such as the inner diameter, ID, of the shell, outer diameter, OD, of the tube, tube length, L, and the baffle pitch, which is the spacing between the baffle center, it is possible to see how these parameters affect the overall design of the heat exchanger. The sensitivity analysis was performed on CASE 2. Although this case only had one heat exchanger in the system, the relations between the parameters and the design will be universal for shell-and-tube exchangers.

7.6.1 Effect of process parameters

In this section, the effect of the process parameters on the effectiveness and other process parameters will be summed up based on the discussion of the optimizations performed in HYSYS for each case. The following process parameters will be discussed:

- Effect of inlet temperature of the heating medium
- Effect of mass flow of the heating medium
- Effect of the outlet temperature of LNG from the vaporizer

As seen in Figure 7-10 and Figure 7-14 the inlet temperature of the heating medium such as the water-glycol solution and seawater has a major impact on the effectiveness of the heat exchanger. The lower inlet temperature decreases the temperature difference between the fluids, which make the heat transfer more effective than a larger temperature difference as discussed earlier.

Figure 7-9 illustrates how the mass flow of the heating medium varies with the inlet temperature of the heating medium. The lower inlet temperature, the more flow is required to heat the LNG to a certain temperature. As mentioned, a lower inlet temperature of water-glycol solution or seawater will induce a better heat transfer, and therefore a more effective heat exchanger.

The outlet temperature of LNG from the vaporizer sets the limit of mass flow of the heating medium required to heat the LNG illustrated in Figure 7-2. Higher outlet temperature means more energy required to heat the LNG from -160°C to the preferred temperature. In CASE 1 it also affect the MGH in size and performance. Higher outlet temperature from the vaporizer leads to a higher inlet temperature to the MGH that results in a smaller heat exchanger needed.

7.6.2 Effect of increasing shell ID

When the shell inner diameter (ID) increases, the total volume of the heat exchanger increases as well as the number of tubes, which means that the effective surface area increases as well. As the tube length is constant, the L/D ratio will decrease with larger shell ID. The larger exchanger with more tubes will result in a heavier and more expensive device. Table 7-12 shows the pressure drop on the shell-side is very small and stable around 0,1 kPa. The pressure drop on the tube-side increases for then to decrease.

Table 7-12: Heat exchanger parameters from the sensitivity simulations retrieved from EDR where the shell ID increases.

	EDR 1	EDR 2	EDR 3	EDR 4	EDR 5
Shell ID [mm]	205	250	300	350	400
Tube OD [mm]	19,1	19,1	19,1	19,1	19,1
Tube length [mm]	1200	1200	1200	1200	1200
L/D ratio [-]	5,9	4,8	4	3,4	3
Baffle pitch [mm]	135	135	135	135	135
Number of baffles	6	6	6	6	6
Number of tubes	30	38	70	108	150
Effective surface area [m^2]	1,8	2,3	4,1	6,4	8,9
Weight [kg]	233	303	406	541	671
Pressure drop, shell-side [kPa]	0,1	0,1	0,1	0,1	0,1
Pressure drop, tube-side [kPa]	1,9	7,2	3,7	2,5	1,9
Estimated cost [USD]	9768	11376	13591	16044	18661

7.6.3 Effect of increasing tube OD

With an increasing outer diameter (OD) of the tube, where the ratio of the tube length L and the shell ID is constant as well as the spacing between baffles, the fewer tubes will fit within the shell as seen in Table 7-13. The fewer tubes mean smaller effective surface area, which affect the cost of the heat exchanger. The lesser material, the cheaper heat exchanger. However, the shell ID is constant, which accounts for most of the weight so the total weight will not be much affected. The pressure drop on the shell-side is very small and is stable around 0,1 kPa for the varying sizes of tube OD. On the tube-side the pressure drop is 99 kPa for the smallest tube OD on 5 mm, but the pressure drop decreases with a factor of 2 for the three first stages. The larger tube OD, the lesser pressure drop on the tube-side as seen from Eq. 5-10.

Table 7-13: Heat exchanger parameters from the sensitivity simulations retrieved from EDR where the tube OD increases.

	EDR 1	EDR 2	EDR 3	EDR 4	EDR 5
Shell ID [mm]	205	205	205	205	205
Tube OD [mm]	5	10	15	20	25
Tube length [mm]	1200	1200	1200	1200	1200
L/D ratio [-]	5,9	5,9	5,9	5,9	5,9
Baffle pitch [mm]	135	135	135	135	135
Number of baffles	6	6	6	6	6
Number of tubes	290	68	32	26	18
Effective surface area [m^2]	4,6	2,2	1,5	1,6	1,4
Weight [kg]	226	216	227	232	237
Pressure drop, shell-side [kPa]	0,1	0,1	0,1	0,1	0,1
Pressure drop, tube-side [kPa]	99,1	40,5	19,5	1,8	1,2
Estimated cost [USD]	13133	10198	9758	9766	9895

7.6.4 Effect of increasing tube length

When the tube length increases, the effective surface area increases. Longer tube length causes longer shell length as well, contributing to a larger volume and weight for the heat exchanger. As the length increases, there is more room for baffles, which increases the turbulence of the heat exchanger. When the shell ID is constant, the L/D ratio increases with the increasing tube length. As discussed will a larger and heavier exchanger cost more than a smaller one.

As seen in the Table 7-14 the pressure drop on shell-side is constant and similar to the shell-side pressure drop for the two previous cases with increasing shell ID and tube OD. The small increase on 0,001 kPa for every additional meters of the tube length is not of a significant magnitude. However, the pressure drop on the tube-side will increase with longer tubes due to longer flow path with friction Eq. 5-10.

Table 7-14: Heat exchanger parameters from the sensitivity simulations retrieved from EDR where the tube length increases.

	EDR 1	EDR 2	EDR 3	EDR 4	EDR 5
Shell ID [mm]	250	250	250	250	250
Tube OD [mm]	19,1	19,1	19,1	19,1	19,1
Tube length [mm]	1200	2200	3200	4200	5200
L/D ratio [-]	4,8	4,8	12,8	16,8	20,8
Baffle pitch [mm]	135	135	135	135	135
Number of baffles	6	12	20	28	34
Number of tubes	38	38	38	38	38
Effective surface area [m^2]	2,3	4,5	6,8	9,1	11,3
Weight [kg]	303	394	487	580	672
Pressure drop, shell-side [kPa]	0,1	0,1	0,1	0,1	0,1
Pressure drop, tube-side [kPa]	7,2	11	14,4	18	21
Estimated cost [USD]	11376	12094	12879	13665	16244

7.6.5 Effect of increasing baffle pitch

The baffle pitch is the distance between the centers of the baffles. The larger pitch, the fewer baffles. The baffles have no effect on the effective surface area of the device though it is not involved in the surface area that heat transfer occurs. However, as mentioned earlier, the baffles induce turbulence on the shell-side flow and increase the heat transfer surface area of the fluid. From the equation of Nusselt number, a larger *Re* will contribute to a larger heat transfer coefficient, which will increase the heat transfer.

As seen in table Figure 7-15 the number of baffles does not have a significance effect on the heat exchangers thermal and mechanical design except from the number of baffles itself. The pressure drops on both the shell-side and tube-side are stable as well as the weight and cost.

Table 7-15: Heat exchanger parameters from the sensitivity simulations retrieved from EDR where the baffle pitch increases.

	EDR 1	**EDR 2**	**EDR 3**	**EDR 4**	**EDR 5**
Shell ID [mm]	250	250	250	250	250
Tube OD [mm]	19,1	19,1	19,1	19,1	19,1
Tube length [mm]	1200	1200	1200	1200	1200
L/D ratio [-]	4,8	4,8	4,8	4,8	4,8
Baffle pitch [mm]	80	100	120	140	160
Number of baffles	10	8	6	6	4
Number of tubes	38	38	38	38	38
Effective surface area [m^2]	2,3	2,3	2,3	2,3	2,3
Weight [kg]	306	304	303	303	302
Pressure drop, shell-side [kPa]	0,1	0,1	0,1	0,1	0,1
Pressure drop, tube-side [kPa]	7,4	7,1	7	7,1	6,5
Estimated cost [USD]	11510	11443	11376	11376	11309

8 Conclusions and proposal for further work

8.1 Conclusions

This thesis has given a better understanding how process parameters affect other process parameters. Further, how mechanical parameters affect thermal and hydraulic parameters for LP LNG fuel supply systems.

- When the outlet temperature for the cold fluid from a heat exchanger increased, the required mass flow of the hot fluid increased as well. Larger temperature difference of the inlet- and outlet temperature of cold fluid, the more heat is required.
- A high mass flow of the heating medium causes a higher outlet temperature of the heating medium.
- The smaller inlet temperature difference of the hot fluid and cold fluid, the more effective is the heat transfer, which means that more heat is absorbed by the cold fluid. Note that a larger mass flow of the hot fluid is required for lower temperature of the cold fluid.
- Propane is more effective as a heating medium than water-glycol solution and seawater because a greater heat capacity.
- Increased shell ID increased the shell volume, which makes room for more tubes. The effective surface area increases with the increasing tubes, which increases the total weight and costs.
- Increased tube OD results in fewer tubes and a smaller effective surface area. Further, larger tubes affected the tube-side pressure – it decreased when the tube OD increased.
- Increased tube length leads to a longer shell and larger effective surface area. A larger device means higher weight and costs. A longer flow path for the fluids will increase the accumulated friction and the tube-side pressure drop.
- Increased baffle pitch decreases the number of baffles. Fewer baffles induced less turbulence. It had no significant effects on either the pressure drop, weight and costs.
- The more effective heat exchanger, the smaller device is needed.
- The pressure drop on the shell-side is not affected significantly by other mechanical parameters.

- HYSYS do not conserve the energy with respect to the energy equation. The simulation tool estimated a higher required flow of the heating medium than necessary.

8.2 Proposal for further work

One must evaluate the different fuel system design with respect to the vessel type. There are many factors to include to reach the optimal design of the fuel system such as heat sources, space, weight and of course costs, both investment costs and operation costs. This study focuses on the heat exchangers, the vaporizer and the MGH only, and factors such as the pressure loss in pipes and the pressure loss through valves and fittings are not included. To get an overall understanding of the complete fuel system including tank, PBU, piping system and other devices concerning regasification, heating or re-liquefaction, a further study can be conducted with a system including these components. It is also interesting to investigate how an irregular fuel consumption will affect the components and the process parameters in the system and especially how a transient behavior of the system affect the pressure in the tank.

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Appendix

Appendix A - Safety data sheet for LNG

RENT BRENSEL LNG - LIQUEFIED NATURAL GAS - KOLLSNES2

Dok.ansv.	Ameln
Fagansv.:	Ameln
ISO-6976 1983 ed.	
ISO-6577 1991 ed.	
AVL, DGC og Kl. Mc Kinley	
Gasnor AS	
Dato	02.01.2008

Tetthet [kg/Nm3]:		0,7577
Tetthet [kg/Sm3]:		0,7182
Relativ tetthet:		0,5853
Metantall:		83,7
Motoroktantall:		131,0
Støkiometri:	mol/mol	kg/kg
Oksygen:	2,06	3,75
Luft:	9,85	16,21
Utslippsfaktor:	mol/mol	kg/kg
	1,05	2,73

Tetthet flytende fase	
gr.C	kg/m3
-162	442,0
-160	439,2
-158	436,3

Prøve I.D:	KOLLSNES2: 5 analyser i desember 2007				
Gass komposisjon		Brennverdi			
		Brutto	Netto		
Produkt:	Mol %	MJ/kg	MJ/kg	Vol %	Masse %
Nitrogen	0,675	0,000	0,000	0,677	1,114
Karbondioksyd	0,000	0,000	0,000	0,000	0,000
Metan	94,647	55,500	50,013	94,697	89,462
Etan	3,751	51,876	47,485	3,724	6,645
Propan	0,541	50,345	46,352	0,531	1,406
i-butan	0,262	49,356	45,573	0,253	0,896
n-butan	0,060	49,500	45,715	0,058	0,205
i-pentan	0,064	48,900	45,239	0,060	0,272
n-pentan	0,000	49,011	45,353	0,000	0,000
hexan	0,000	48,678	45,105	0,000	0,000
heptan	0,000	48,437	44,896	0,000	0,000
SUM:	100,000			100,000	100,000

Gass komposisjon		Brennverdi pr.element:		Molvekt	Molfraksjon
		Brutto	Netto		
Produkt:	Mol %	MJ/kg	MJ/kg	kg/Kmol	
Nitrogen	0,68	0,00	0,00	28,013	0,189
Karbondioksyd	0,00	0,00	0,00	44,010	0,000
Metan	94,65	49,65	44,74	16,043	15,184
Etan	3,75	3,45	3,16	30,069	1,128
Propan	0,54	0,71	0,65	44,096	0,239
i-butan	0,26	0,44	0,41	58,123	0,152
n-butan	0,06	0,10	0,09	58,123	0,035
i-pentan	0,06	0,13	0,12	72,150	0,046
n-pentan	0,00	0,00	0,00	72,150	0,000
hexan	0,00	0,00	0,00	86,177	0,000
heptan	0,00	0,00	0,00	100,205	0,000
SUM:	100,00	54,48	49,17		16,973

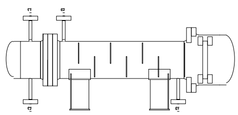
	BRUTTO		NETTO	
Enhet:	Brennverdi:	Wobbe Indeks:	Brennverdi:	Wobbe Indeks:
MJ/kg	54,483	71,217	49,174	64,278
MJ/Sm3	39,129	51,147	35,316	46,164
MJ/Nm3	41,280	53,960	37,258	48,702
kcal/kg	13012,663	17009,516	11744,837	15352,276
kcal/Sm3	9345,491	12215,968	8434,958	11025,764
kcal/Nm3	9859,420	12887,751	8898,815	11632,095
Btu/lb	23424,608	30619,501	21142,344	27636,237
kWh/kg	15,134	19,783	13,660	17,855
kWh/Sm3	10,869	14,208	9,810	12,823
kWh/Nm3	11,467	14,989	10,350	13,528

Appendix B

B-1 – CASE 1 – Vaporizer - Heat exchanger specification

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 205 / 1200mm		Type BES	Hor	Connected in 1 parallel	1 series
7	Surf/unit(eff.) 1,4 m2		Shells/unit 1		Surf/shell (eff.) 1,4	m2
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name		to EVAP->to MGH		Water-glyc1->Water-glyc2	
11	Fluid quantity, Total	kg/h	19		231	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	231	231
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	-40	40	20
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1077,24	/ 1093,2
19	Viscosity	cp	/ 0,1465	/	/ 3,0277	/ 5,3529
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 3,056	/ 2,985
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3394	/ 0,3299
24	Latent heat	kJ/kg	423,8	383,5		
25	Pressure (abs)	kPa	600	599,922	700	699,629
26	Velocity	m/s	0,1		0,09	
27	Pressure drop, allow./calc.	kPa	50	0,078	50	0,371
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045	Ao based
29	Heat exchanged 3,9 kW				MTD corrected 138,59	C
30	Transfer rate, Service 20,5		Dirty 51,1		Clean 52,8	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**					**Sketch**
32				**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g		kPa	700 / /	800 / /	
34	Design temperature		C	35	75	
35	Number passes per shell			1	4	
36	Corrosion allowance		mm	3,18	3,18	
37	Connections	In	mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out		1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate		/ -	/ -	
40	Tube No. 23	OD 19,05		Tks- Avg 2,11 mm	Length 1200 mm	Pitch23,81 mm

41	Tube type	Plain			Material		Carbon Steel	Tube pattern	30	
42	Shell	Carbon Steel	ID	205	OD	219,08	mm	Shell cover	Carbon Steel	
43	Channel or bonnet		Carbon Steel					Channel cover	Carbon Steel	
44	Tubesheet-stationary		Carbon Steel					Tubesheet-floating	Carbon Steel	
45	Floating head cover		Carbon Steel					Impingement protection	None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	43,02	H	Spacing: c/c	120	mm
47	Baffle-long	-		Seal type				Inlet	195,21	mm
48	Supports-tube			U-bend			Type			
49	Bypass seal				Tube-tubesheet joint		Exp. 2 grv			
50	Expansion joint	-			Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe			Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe								
54	Code requirements	ASME Code Sec VIII Div 1					TEMA class	R - refinery service		
55	Weight/Shell	233,3	Filled with water		281,9			Bundle	50,6	kg
56	Remarks									
57										
58										

B-2 – CASE 1 – Vaporizer - Thermal parameters

Overall Performance

Design (Sizing)		Shell Side		Tube Side	
Total mass flow rate	kg/h	19		231	
Vapor mass flow rate (In/Out)	kg/h	0	19	0	0
s flow rate	kg/h	19	0	231	231
Vapor mass quallity		0	1	0	0
Temperatures	C	-160	-40	40	20
Dew point / Bubble point	C	-48,08			
Operating Pressures	kPa	600	599,922	700	699,629
Film coefficient	W/(m2*K)	71,1		206,7	
Fouling resistance	m2*K/W	0,00018		0,00045	
Velocity (highest)	m/s	0,1		0,09	
Pressure drop (allow./calc.)	kPa	50 / 0,078		50 / 0,371	

Total heat exchanged	kW	3,9			Unit	BES	4	pass	1	ser	1 par
Overall clean coeff. (plain/finned)	W/(m2*K)	52,8	/		Shell size	205	-	1200	mm		Hor
Overall dirty coeff. (plain/finned)	W/(m2*K)	51,1	/		Tubes	Plain					
Effective area (plain/finned)	m2	1,4	/		No. 23	OD	19,05	Tks		2,11	mm
Effective MTD	C	138,59			Pattern	30		Pitch		23,81	mm
Actual/Required area ratio (dirty/clean)		2,49	/	2,58	Baffles	Single segmental			Cut(%d)	43,02	
Vibration problem (Tasc/TEMA)		No	/	No							
RhoV2 problem				No	Total cost	9667	Dollar(US)				

Resistance Distribution

Overall Coefficient / Resistance Summary				Clean	Dirty	Max Dirty
Area required			m2	0,5	0,5	1,4
Area ratio: actual/required				2,58	2,49	1
Overall coefficient			W/(m2*K)	52,8	51,1	20,5
Overall resistance			m2*K/W	0,01895	0,01958	0,04884
Shell side fouling			m2*K/W	0.0	0,00018	0,00855
Tube side fouling				0.0	0,00045	0,02135
Resistance Distribution		**W/(m2*K)**	**m2*K/W**	%	%	%
	Shell side film	21750,7	0,01406	74,22	71,83	28,79
	Shell side fouling	5555,6	0,00018		0,92	17,5
	Tube wall	21750,7	0,00005	0,24	0,23	0,09
	Tube side fouling *	2224,8	0,00045		2,3	43,7
	Tube side film *	206,7	0,00484	25,54	24,72	9,91

* Based on outside surface - Area ratio: Ao/Ai = 1,28

Heat Transfer Coefficients

Film coefficients		W/(m2*K)	Shell Side		Tube Side	
			Bare area (OD) / Finned area		Bare area (OD) / ID area	
Overall film coefficients			71,1 /		206,7 /	265,4
	Vapor sensible		34,7 /		/	
	Two phase		70,7 /		/	
	Liquid sensible		762,3 /		206,7 /	265,4
Heat Transfer Parameters			**In**	**Out**	**In**	**Out**
Prandtl numbers	Vapor			0,78		
	Liquid		2,35		27,26	48,43
Reynolds numbers	Vapor Nominal			1190,94		
	Liquid Nominal		73,64		453,98	256,77

MTD and Flux

Temperature Difference	C	Heat Flux (based on tube O.D)		kW/m2
Overall effective MTD	138,59	Overall actual flux		7,1
One pass counterflow MTD	140,95	Critical flux		60
LMTD based on end points	123,32	Highest actual flux	29,6	
Effective MTD correction factor	1,12	Highest actual/critical flux	0,45	
Wall Temperatures	**C**			
Shell mean metal temperature	-92,32			
Tube mean metal temperature	-6,52			
Tube wall temperatures (highest/lowest)	28,17 / -121,32			

Duty Distribution

Heat Load Summary	Shell Side		Tube Side	
	kW	% total	kW	% total
Vapor only	0,1	2	0	0
2-Phase vapor	0,8	21	0	0
Latent heat	2,3	61	0	0
2-Phase liquid	0,2	5	0	0
Liquid only	0,4	11	-3,9	100
Total	3,9	100	-3,9	100

Pressure Drop

Pressure Drop kPa	Shell Side	Tube Side
Maximum allowed	50	50
Total calculated	0,078	0,371
Gravitational	0	0

Frictional	0,078			0,371		
Momentum change	0			0		
Pressure drop distribution	**m/s**	**kPa**	**%dp**	**m/s**	**kPa**	**%dp**
Inlet nozzle	0,08	0,001	1,74	0,39	0,09	24,41
Entering bundle	0			0,09	0,005	1,34
Inside tubes				0,09 0,04	0,228	61,52
Inlet space Xflow	0	0	0,08			
Baffle Xflow	0 0,1	0,001	1,02			
Baffle windows	0 0,08	0	0,09			
Outlet space Xflow	0,06	0	0,26			
Exiting bundle	0,06			0,04	0,008	2,08
Outlet nozzle	6,07	0,076	96,82	0,39	0,039	10,66
Intermediate nozzles						

Thermosiphon Piping

Piping reference points	**m**	**Pressure points**	**kPa**	**C**	**Quality**
Height of liquid in column		Liquid level in column			
Height of heat transfer region inlet		Inlet to heated section			
Height of heat transfer region outlet		Boiling boundary position			
Height of column return line		Outlet of heated section		-40	1
		Exit of outlet piping			

Pressure changes (-loss/+gain)	**kPa**	**Inlet circuit**	**Exchanger**	**Outlet circuit**
Frictional				
Gravitational				
Momentum				
Flashing				
Nozzles				
Total				

Inlet Circuit Piping											
Inlet circuit element											
Frictional pressure drop at entry	kPa										
Frictional pressure drop at element	kPa										
Outlet Circuit Piping											
Outlet circuit element											
Frictional pressure drop at entry	kPa										
Frictional pressure drop at element	kPa										
Void fraction											
Two phase flow pattern											

Flow Analysis

Shell Side Flow Fractions	Inlet	Middle	Outlet	Diameter Clearance	mm
Crossflow	0,31	0,3	0,39		
Window	0,87	0,67	0,85		
Baffle hole - tube OD	0,01	0,06	0,03	0,79	
Baffle OD - shell ID	0,12	0,28	0,12	3,18	
Shell ID - bundle OTL	0,44	0,29	0,36	44,45	
Pass lanes	0,12	0,08	0,1		
Rho*V2 Analysis	**Flow area**	**Velocity**	**Density**	**Rho*V2**	**TEMA limit**
	mm2	**m/s**	**kg/m3**	**kg/(m*s2)**	**kg/(m*s2)**
Shell inlet nozzle	151	0,08	462	3	2232
Shell entrance	1656	0,01	462	0	5953
Bundle entrance	14630	0	462	0	5953
Bundle exit	14630	0,06	5,81	0	5953
Shell exit	1656	0,55	5,81	2	5953
Shell outlet nozzle	151	6,07	5,81	214	

	mm2	m/s	kg/m3		
Tube inlet nozzle	151	0,39	1077,24	167	8928
Tube inlet	691	0,09	1077,24	8	
Tube outlet	691	0,04	1093,2	2	
Tube outlet nozzle	151	0,39	1093,2	164	

Thermosiphons and Kettles

Thermosiphons	
Thermosiphon stability	
Vertical tube side thermosiphons	
Flow reversal criterion - top of the tubes (should be >0.5)	
Flooding criterion - top of the tubes (should be >1.0)	
Fraction of the tube length flooded	
Kutateladze Number in axial nozzle	
Kettles	
Recirculation ratio	
Quality at top of bundle	
Entrainment fraction	

Methods Summary

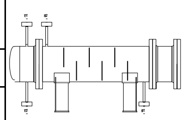
	Hot side	Cold side
Heat Transfer Coefficient multiplier	No	No
Heat Transfer Coefficient specified	No	No
Pressure drop multiplier	No	No
Pressure drop calculation option	friction only	friction only
Calculation method	Advanced method	
Desuperheating heat transfer method	Wet wall	

Multicomponent condensing heat transfer method	HTFS - Silver-Bell
Vapor shear enhanced condensation	Yes
Liquid subcooling heat transfer (vertical shell)	Not Used
Subcooled boiling accounted for in	Heat transfer & pressure drop
Post dryout heat transfer accounted for in	Yes
Correction to user-supplied boiling curve	Boiling curve not used
Falling film evaporation method	HTFS recommended method
Single phase tube side heat transfer method	HTFS recommended method
Lowfin Calculation method	HTFS / ESDU

B-3 - CASE 1 – MGH - Heat exchanger specification

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 205 / 1200mm	Type BEL	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 2 m2	Shells/unit 1		Surf/shell (eff.)	2	m2
8			**PERFORMANCE OF ONE UNIT**			
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		231	
12	Vapor (In/Out)	kg/h	19	19	0	0
13	Liquid	kg/h	0	0	231	231
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-40	0	20	17,69
17	Dew / Bubble point	C				
18	Density (Vap / Liq)	kg/m3	5,81 /	/	/ 1093,2	/ 1095,02
19	Viscosity	cp	0,0091 /	/	/ 5,353	/ 5,7693
20	Molecular wt, Vap		18,12	18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	2,043 /	/	/ 2,985	/ 2,977
23	Thermal conductivity	W/(m*K)	0,0237 /	/	/ 0,3299	/ 0,3287
24	Latent heat	kJ/kg				
25	Pressure (abs)	kPa	599,9	599,697	699,6	698,989
26	Velocity	m/s	0,15		0,11	
27	Pressure drop, allow./calc.	kPa	50	0,203	50	0,611
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based
29	Heat exchanged 0,4	kW		MTD corrected	35,03	C
30	Transfer rate, Service 6,2		Dirty 25,6		Clean 26	W/(m2*K)
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**	



32				**Shell Side**	**Tube Side**
33	Design/vac/test pressure:g		kPa	700 / /	800 / /
34	Design temperature		C	35	55
35	Number passes per shell			1	6
36	Corrosion allowance		mm	3,18	3,18
37	Connections	In	mm	1 12,7 / -	1 12,7 / -
38	Size/rating	Out		1 12,7 / -	1 12,7 / -
39	Nominal	Intermediate		/ -	/ -
40	Tube No. 30	OD 19,05	Tks- Avg 2,11 mm	Length 1200 mm	Pitch23,81 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel	ID 205	OD 219,08 mm	Shell cover	-
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel

44	Tubesheet-stationary	Carbon Steel				Tubesheet-floating	-		
45	Floating head cover	-				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	44,56 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	223,98	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	214	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	-		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	-							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	218,9	Filled with water		255,6		Bundle	68,4	kg
56	Remarks								
57									
58									

B-4 – CASE 1 – MGH - Thermal parameters

Overall Performance

Design (Sizing)		Shell Side		Tube Side	
Total mass flow rate	kg/h	19		231	
Vapor mass flow rate (In/Out)	kg/h	19	19	0	0
Liquid mass flow rate	kg/h	0	0	231	231
Vapor mass quallity		1	1	0	0
Temperatures	C	-40	0	20	17,69
Dew point / Bubble point	C				
Operating Pressures	kPa	599,9	599,697	699,6	698,989
Film coefficient	W/(m2*K)	30,5		178,5	
Fouling resistance	m2*K/W	0,00018		0,00045	
Velocity (highest)	m/s	0,15		0,11	
Pressure drop (allow./calc.)	kPa	50 / 0,203		50 / 0,611	

Total heat exchanged	kW	0,4			Unit	BEL	6	pass	1	ser	1 par
Overall clean coeff. (plain/finned)	W/(m2*K)	26	/		Shell size	205	-	1200	mm		Hor
Overall dirty coeff. (plain/finned)	W/(m2*K)	25,6	/		Tubes	Plain					
Effective area (plain/finned)	m2	2	/		No.	30	OD	19,05	Tks	2,11	mm
Effective MTD	C	35,03			Pattern	30		Pitch		23,81	mm
Actual/Required area ratio (dirty/clean)		4,1	/	4,16	Baffles	Single segmental			Cut(%d)	44,56	
Vibration problem (Tasc/TEMA)		No	/	No							
RhoV2 problem				No	Total cost	9993	Dollar(US)				

Resistance Distribution

Overall Coefficient / Resistance Summary	Clean	Dirty		Max Dirty

Area required		m2	0,5	0,5	2
Area ratio: actual/required			4,16	4,1	1
Overall coefficient		W/(m2*K)	26	25,6	6,2
Overall resistance		m2*K/W	0,03843	0,03906	0,16003
Shell side fouling		m2*K/W	0.0	0,00018	0,03477
Tube side fouling			0.0	0,00045	0,08682
Resistance Distribution	**W/(m2*K)**	**m2*K/W**	%	%	%
Shell side film	21789,5	0,03279	85,31	83,93	20,49
Shell side fouling	5555,6	0,00018		0,46	21,73
Tube wall	21789,5	0,00005	0,12	0,12	0,03
Tube side fouling *	2224,8	0,00045		1,15	54,26
Tube side film *	178,5	0,0056	14,57	14,34	3,5

* Based on outside surface - Area ratio: Ao/Ai = 1,28

Heat Transfer Coefficients

Film coefficients	**W/(m2*K)**	**Shell Side**		**Tube Side**	
		Bare area (OD) / Finned area		**Bare area (OD) / ID area**	
Overall film coefficients		30,5 /		178,5 / 229,3	
Vapor sensible		30,5 /		/	
Two phase		/		/	
Liquid sensible		/		178,5 / 229,3	
Heat Transfer Parameters		**In**	**Out**	**In**	**Out**
Prandtl numbers	Vapor	0,78	0,76		
	Liquid			48,43	52,25
Reynolds numbers	Vapor Nominal	1535,03	1329,63		
	Liquid Nominal			342,36	317,65

MTD and Flux

Temperature Difference	C	Heat Flux (based on tube O.D)		kW/m2
Overall effective MTD	35,03	Overall actual flux		0,9
One pass counterflow MTD	35,48	Critical flux		
LMTD based on end points	35,58	Highest actual flux	1,5	
Effective MTD correction factor	0,98	Highest actual/critical flux		
Wall Temperatures	**C**			
Shell mean metal temperature	-16,45			
Tube mean metal temperature	13,49			
Tube wall temperatures (highest/lowest)	17,04 / 9,53			

Duty Distribution

Heat Load Summary	Shell Side		Tube Side	
	kW	% total	kW	% total
Vapor only	0,4	100	0	0
2-Phase vapor	0	0	0	0
Latent heat	0	0	0	0
2-Phase liquid	0	0	0	0
Liquid only	0	0	-0,4	100
Total	0,4	100	-0,4	100

Pressure Drop

Pressure Drop kPa	Shell Side	Tube Side
Maximum allowed	50	50
Total calculated	0,203	0,611
Gravitational	0	0
Frictional	0,203	0,611
Momentum change	0	0

Pressure drop distribution	m/s	kPa	%dp	m/s	kPa	%dp
Inlet nozzle	6,07	0,11	54,24	0,39	0,088	14,48
Entering bundle	0,08			0,11	0,011	1,73
Inside tubes				0,11 0,11	0,456	74,66
Inlet space Xflow	0,08	0	0,08			
Baffle Xflow	0,13 0,15	0,001	0,37			
Baffle windows	0,08 0,1	0	0,1			
Outlet space Xflow	0,09	0	0,1			
Exiting bundle	0,1			0,11	0,016	2,67
Outlet nozzle	7,21	0,092	45,11	0,39	0,039	6,46
Intermediate nozzles						

Thermosiphon Piping

Piping reference points	m	Pressure points	kPa	C	Quality
Height of liquid in column		Liquid level in column			
Height of heat transfer region inlet		Inlet to heated section			
Height of heat transfer region outlet		Boiling boundary position			
Height of column return line		Outlet of heated section		0	1
		Exit of outlet piping			

Pressure changes (-loss/+gain)	kPa	Inlet circuit	Exchanger	Outlet circuit
Frictional				
Gravitational				
Momentum				
Flashing				
Nozzles				
Total				

Inlet Circuit Piping										
Inlet circuit element										

Frictional pressure drop at entry	**kPa**										
Frictional pressure drop at element	**kPa**										
Outlet Circuit Piping											
Outlet circuit element											
Frictional pressure drop at entry	**kPa**										
Frictional pressure drop at element	**kPa**										
Void fraction											
Two phase flow pattern											

Flow Analysis

Shell Side Flow Fractions	**Inlet**	**Middle**	**Outlet**	**Diameter Clearance**	**mm**
Crossflow	0,65	0,51	0,65		
Window	0,85	0,66	0,85		
Baffle hole - tube OD	0,04	0,09	0,04	0,79	
Baffle OD - shell ID	0,11	0,24	0,11	3,18	
Shell ID - bundle OTL	0,08	0,06	0,08	12,7	
Pass lanes	0,11	0,09	0,12		
Rho*V2 Analysis	**Flow area**	**Velocity**	**Density**	**Rho*V2**	**TEMA limit**
	mm2	**m/s**	**kg/m3**	**kg/(m*s2)**	**kg/(m*s2)**
Shell inlet nozzle	151	6,07	5,81	214	2232
Shell entrance	895	1,02	5,81	6	5953
Bundle entrance	10929	0,08	5,81	0	5953
Bundle exit	10929	0,1	4,89	0	5953
Shell exit	895	1,22	4,89	7	5953
Shell outlet nozzle	151	7,21	4,89	254	
	mm2	**m/s**	**kg/m3**		
Tube inlet nozzle	151	0,39	1093,2	164	8928

Tube inlet	518	0,11	1093,2	14	
Tube outlet	518	0,11	1095,02	14	
Tube outlet nozzle	151	0,39	1095,02	164	

Thermosiphons and Kettles

Thermosiphons	
Thermosiphon stability	
Vertical tube side thermosiphons	
Flow reversal criterion - top of the tubes (should be >0.5)	
Flooding criterion - top of the tubes (should be >1.0)	
Fraction of the tube length flooded	
Kutateladze Number in axial nozzle	
Kettles	
Recirculation ratio	
Quality at top of bundle	
Entrainment fraction	

Methods Summary

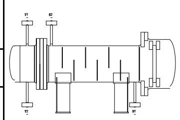
	Hot side	**Cold side**
Heat Transfer Coefficient multiplier	No	No
Heat Transfer Coefficient specified	No	No
Pressure drop multiplier	No	No
Pressure drop calculation option	friction only	friction only
Calculation method	Advanced method	
Desuperheating heat transfer method	Wet wall	
Multicomponent condensing heat transfer method	HTFS - Silver-Bell	
Vapor shear enhanced condensation	Yes	

Liquid subcooling heat transfer (vertical shell)	Not Used
Subcooled boiling accounted for in	Heat transfer & pressure drop
Post dryout heat transfer accounted for in	No
Correction to user-supplied boiling curve	Boiling curve not used
Falling film evaporation method	HTFS recommended method
Single phase tube side heat transfer method	HTFS recommended method
Lowfin Calculation method	HTFS / ESDU

B-5 - CASE 2 – Vaporizer - Heat exchanger specification

Heat Exchanger Specification Sheet

1												
2												
3												
4												
5												
6	Size	205	/	1200mm	Type	BES	Hor	Connected in	1	parallel	1	series
7	Surf/unit(eff.)	1,8	m2	Shells/unit	1	Surf/shell (eff.)	1,8	m2				
8	**PERFORMANCE OF ONE UNIT**											
9	Fluid allocation		**Shell Side**		**Tube Side**							
10	Fluid name											
11	Fluid quantity, Total	kg/h	19		656							
12	Vapor (In/Out)	kg/h	0	19	0	0						
13	Liquid	kg/h	19	0	656	656						
14	Noncondensable	kg/h	0		0							
15												
16	Temperature (In/Out)	C	-160	0	10	1,95						
17	Dew / Bubble point	C	-48,08	-135,23								
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3						
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8215						
20	Molecular wt, Vap			18,12								
21	Molecular wt, NC											
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928						
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199						
24	Latent heat	kJ/kg	436,2	379,8								
25	Pressure (abs)	kPa	600	599,908	700	698,076						
26	Velocity	m/s	0,13		0,06							
27	Pressure drop, allow./calc.	kPa	50	0,093	50	1,924						
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based					
29	Heat exchanged	4,3	kW	MTD corrected	75,99	C						
30	Transfer rate, Service	31,9	Dirty	36,8	Clean	37,7	W/(m2*K)					
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**							
32			**Shell Side**	**Tube Side**								
33	Design/vac/test pressure:g	kPa	700 / /	800 / /								
34	Design temperature	C	35	45								
35	Number passes per shell		1	2								
36	Corrosion allowance	mm	3,18	3,18								
37	Connections	In	mm	1	12,7	/	-	1	12,7	/	-	
38	Size/rating	Out		1	12,7	/	-	1	12,7	/	-	
39	Nominal	Intermediate				/	-			/	-	
40	Tube No.	30	OD	19,05	Tks- Avg	2,11	mm	Length	1,2	m	Pitch23,81	mm
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30						
42	Shell	Carbon Steel	ID	205	OD	219,08	mm	Shell cover	Carbon Steel			
43	Channel or bonnet	Carbon Steel	Channel cover	Carbon Steel								



44	Tubesheet-stationary		Carbon Steel				Tubesheet-floating		Carbon Steel		
45	Floating head cover		Carbon Steel				Impingement protection			None	
46	Baffle-cross	Carbon Steel		Type	Single segmental	Cut(%d)	41,48 H	Spacing: c/c		120	mm
47	Baffle-long	-			Seal type				Inlet	195,21	mm
48	Supports-tube				U-bend			Type			
49	Bypass seal				Tube-tubesheet joint			Exp. 2 grv			
50	Expansion joint	-			Type						
51	RhoV2-Inlet nozzle	3		Bundle entrance		0		Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe			Tube Side		Flat Metal Jacket Fibe				
53	Floating head	Flat Metal Jacket Fibe									
54	Code requirements	ASME Code Sec VIII Div 1					TEMA class		R - refinery service		
55	Weight/Shell	233,1		Filled with water		281,6			Bundle	56,7	kg
56	Remarks										
57											
58											

B-4 – CASE 2 – Vaporizer - Thermal parameters

Overall Performance

Design (Sizing)		Shell Side		Tube Side	
Total mass flow rate	**kg/h**	19		656	
Vapor mass flow rate (In/Out)	**kg/h**	0	19	0	0
Liquid mass flow rate	**kg/h**	19	0	656	656
Vapor mass quallity		0	1	0	0
Temperatures	**C**	-160	0	10	1,95
Dew point / Bubble point	**C**	-48,08			
Operating Pressures	**kPa**	600	599,908	700	698,076
Film coefficient	**W/(m2*K)**	52,4		135,3	
Fouling resistance	**m2*K/W**	0,00018		0,00045	
Velocity (highest)	**m/s**	0,13		0,06	
Pressure drop (allow./calc.)	**kPa**	50 / 0,093		50 / 1,924	

Total heat exchanged	**kW**	4,3		Unit BES 2 pass 1 ser 1 par
Overall clean coeff. (plain/finned)	**W/(m2*K)**	37,7 /		Shell size 205 - 1200 **mm** Hor
Overall dirty coeff. (plain/finned)	**W/(m2*K)**	36,8 /		Tubes Plain
Effective area (plain/finned)	**m2**	1,8 /		No. 30 OD 19,05 Tks 2,11 mm
Effective MTD	**C**	75,99		Pattern 30 Pitch 23,81 **mm**
Actual/Required area ratio (dirty/clean)		1,15 /	1,18	Baffles Single segmental Cut(%d) 41,48
Vibration problem (Tasc/TEMA)		No /	No	
RhoV2 problem			No	Total cost 9768 Dollar(US)

Resistance Distribution

Overall Coefficient / Resistance Summary	Clean	Dirty	Max Dirty

Area required		m2	1,5	1,5	1,8
Area ratio: actual/required			1,18	1,15	1
Overall coefficient		W/(m2*K)	37,7	36,8	31,9
Overall resistance		m2*K/W	0,02652	0,02715	0,03135
Shell side fouling		m2*K/W	0.0	0,00018	0,00138
Tube side fouling			0.0	0,00045	0,00345
Resistance Distribution	**W/(m2*K)**	**m2*K/W**	%	%	%
Shell side film	21789,5	0,01909	71,97	70,3	60,88
Shell side fouling	5555,6	0,00018		0,66	4,4
Tube wall	21789,5	0,00005	0,17	0,17	0,15
Tube side fouling *	2224,8	0,00045		1,66	11
Tube side film *	135,3	0,00739	27,86	27,21	23,57

* Based on outside surface - Area ratio: Ao/Ai = 1,28

Heat Transfer Coefficients

Film coefficients	**W/(m2*K)**	**Shell Side**		**Tube Side**	
		Bare area (OD) / Finned area		**Bare area (OD) / ID area**	
Overall film coefficients		52,4 /		135,3 / 173,8	
Vapor sensible		37,5 /		/	
Two phase		70,5 /		/	
Liquid sensible		312 /		135,3 / 173,8	
Heat Transfer Parameters		**In**	**Out**	**In**	**Out**
Prandtl numbers	Vapor		0,76		
	Liquid	2,35		68,48	89,88
Reynolds numbers	Vapor Nominal		1163,94		
	Liquid Nominal	83,08		138,45	106,11

MTD and Flux

Temperature Difference	C	Heat Flux (based on tube O.D)		kW/m2
Overall effective MTD	75,99	Overall actual flux		2,8
One pass counterflow MTD	82,34	Critical flux		46,9
LMTD based on end points	54,57	Highest actual flux	10,5	
Effective MTD correction factor	1,39	Highest actual/critical flux	0,21	

Wall Temperatures		C		
Shell mean metal temperature		-54,81		
Tube mean metal temperature		-19,46		
Tube wall temperatures (highest/lowest)	8,05	/	-125,22	

Duty Distribution

Heat Load Summary	Shell Side		Tube Side	
	kW	% total	kW	% total
Vapor only	0,5	12	0	0
2-Phase vapor	0,8	19	0	0
Latent heat	2,4	55	0	0
2-Phase liquid	0,2	4	0	0
Liquid only	0,4	10	-4,3	100
Total	4,3	100	-4,3	100

Pressure Drop

Pressure Drop	kPa	Shell Side	Tube Side
Maximum allowed		50	50
Total calculated		0,093	1,924
Gravitational		0	0
Frictional		0,093	1,924
Momentum change		0	0

Pressure drop distribution	m/s	kPa	%dp	m/s	kPa	%dp
Inlet nozzle	0,08	0,001	1,47	1,09	0,721	37,46
Entering bundle	0			0,06	0,002	0,12
Inside tubes				0,06 0,06	0,882	45,86
Inlet space Xflow	0	0	0,06			
Baffle Xflow	0 0,13	0,001	0,93			
Baffle windows	0 0,11	0	0,14			
Outlet space Xflow	0,08	0	0,24			
Exiting bundle	0,08			0,06	0,003	0,18
Outlet nozzle	7,21	0,09	97,16	1,09	0,315	16,38
Intermediate nozzles						

Thermosiphon Piping

Piping reference points	m	Pressure points	kPa	C	Quality
Height of liquid in column		Liquid level in column			
Height of heat transfer region inlet		Inlet to heated section			
Height of heat transfer region outlet		Boiling boundary position			
Height of column return line		Outlet of heated section		0	1
		Exit of outlet piping			

Pressure changes (-loss/+gain)	kPa	Inlet circuit	Exchanger	Outlet circuit
Frictional				
Gravitational				
Momentum				
Flashing				
Nozzles				
Total				

Inlet Circuit Piping											
Inlet circuit element											

Frictional pressure drop at entry **kPa**										
Frictional pressure drop at element **kPa**										
Outlet Circuit Piping										
Outlet circuit element										
Frictional pressure drop at entry **kPa**										
Frictional pressure drop at element **kPa**										
Void fraction										
Two phase flow pattern										

Flow Analysis

Shell Side Flow Fractions	**Inlet**	**Middle**	**Outlet**	**Diameter Clearance mm**	
Crossflow	0,31	0,29	0,38		
Window	0,87	0,64	0,84		
Baffle hole - tube OD	0,02	0,09	0,04	0,79	
Baffle OD - shell ID	0,11	0,26	0,12	3,18	
Shell ID - bundle OTL	0,56	0,35	0,46	44,45	
Pass lanes	0	0	0		
Rho*V2 Analysis	**Flow area**	**Velocity**	**Density**	**Rho*V2**	**TEMA limit**
	mm2	**m/s**	**kg/m3**	**kg/(m*s2)**	**kg/(m*s2)**
Shell inlet nozzle	151	0,08	462	3	2232
Shell entrance	1518	0,01	462	0	5953
Bundle entrance	12938	0	462	0	5953
Bundle exit	12938	0,08	4,89	0	5953
Shell exit	1518	0,72	4,89	3	5953
Shell outlet nozzle	151	7,21	4,89	254	
	mm2	**m/s**	**kg/m3**		
Tube inlet nozzle	151	1,09	1101,04	1320	8928

Tube inlet	2592	0,06	1101,04	4	
Tube outlet	2592	0,06	1107,3	4	
Tube outlet nozzle	151	1,09	1107,3	1313	

Thermosiphons and Kettles

Thermosiphons	
Thermosiphon stability	
Vertical tube side thermosiphons	
Flow reversal criterion - top of the tubes (should be >0.5)	
Flooding criterion - top of the tubes (should be >1.0)	
Fraction of the tube length flooded	
Kutateladze Number in axial nozzle	
Kettles	
Recirculation ratio	
Quality at top of bundle	
Entrainment fraction	

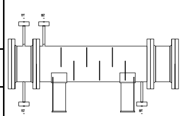
Methods Summary

	Hot side	Cold side
Heat Transfer Coefficient multiplier	No	No
Heat Transfer Coefficient specified	No	No
Pressure drop multiplier	No	No
Pressure drop calculation option	friction only	friction only
Calculation method	Advanced method	
Desuperheating heat transfer method	Wet wall	
Multicomponent condensing heat transfer method	HTFS - Silver-Bell	
Vapor shear enhanced condensation	Yes	

Liquid subcooling heat transfer (vertical shell)	Not Used
Subcooled boiling accounted for in	Heat transfer & pressure drop
Post dryout heat transfer accounted for in	Yes
Correction to user-supplied boiling curve	Boiling curve not used
Falling film evaporation method	HTFS recommended method
Single phase tube side heat transfer method	HTFS recommended method
Lowfin Calculation method	HTFS / ESDU

B-5 - CASE 3 – Heater - Heat exchanger specification

Heat Exchanger Specification Sheet

1							
2							
3							
4							
5							
6	Size 205 / 1200mm		Type AEL	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.)	2 m2	Shells/unit 1		Surf/shell (eff.)	2	m2
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		**Shell Side**		**Tube Side**		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		60		
12	Vapor (In/Out)	kg/h	19	19	0	0	
13	Liquid	kg/h	0	0	60	60	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-10	0	6	4,45	
17	Dew / Bubble point	C					
18	Density (Vap / Liq)	kg/m3	5,08 /	/	/ 1021,63	/ 1022,77	
19	Viscosity	cp	0,0101 /	/	/ 1,4574	/ 1,5263	
20	Molecular wt, Vap		18,12	18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	2,088 /	/	/ 4,32	/ 4,321	
23	Thermal conductivity	W/(m*K)	0,0276 /	/	/ 0,5798	/ 0,577	
24	Latent heat	kJ/kg					
25	Pressure (abs)	kPa	599	598,781	700	699,956	
26	Velocity	m/s	0,15		0,03		
27	Pressure drop, allow./calc.	kPa	50	0,219	50	0,044	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based
29	Heat exchanged	0,1 kW			MTD corrected	9,32	C
30	Transfer rate, Service	5,9	Dirty 27,3		Clean	27,7	W/(m2*K)
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**		
32			**Shell Side**	**Tube Side**			
33	Design/vac/test pressure:g	kPa	700 / /	800 / /			
34	Design temperature	C	35	45			
35	Number passes per shell		1	6			
36	Corrosion allowance	mm	3,18	3,18			
37	Connections	In mm	1 12,7 / -	1 12,7 / -			
38	Size/rating	Out	1 12,7 / -	1 12,7 / -			
39	Nominal	Intermediate	/ -	/ -			
40	Tube No. 30	OD 19,05	Tks- Avg 2,11 mm	Length 1200 mm	Pitch23,81 mm		
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30		
42	Shell Carbon Steel	ID 205	OD 219,08 mm	Shell cover	-		
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel		

44	Tubesheet-stationary	Carbon Steel				Tubesheet-floating		-	
45	Floating head cover	-				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	44,56 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	223,98	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	245	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	-		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	-							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	230,9	Filled with water		267,6		Bundle	68,4	kg
56	Remarks								
57									
58									

B-4 – CASE 3 – Heater - Thermal parameters

Overall Performance

Design (Sizing)		Shell Side		Tube Side	
Total mass flow rate	kg/h	19		60	
Vapor mass flow rate (In/Out)	kg/h	19	19	0	0
Liquid mass flow rate	kg/h	0	0	60	60
Vapor mass quallity		1	1	0	0
Temperatures	C	-10	0	6	4,45
Dew point / Bubble point	C				
Operating Pressures	kPa	599	598,781	700	699,956
Film coefficient	W/(m2*K)	31,4		241,9	
Fouling resistance	m2*K/W	0,00018		0,00045	
Velocity (highest)	m/s	0,15		0,03	
Pressure drop (allow./calc.)	kPa	50 / 0,219		50 / 0,044	

Total heat exchanged	kW	0,1		Unit AEL 6 pass 1 ser 1 par
Overall clean coeff. (plain/finned)	W/(m2*K)	27,7 /		Shell size 205 - 1200 mm Hor
Overall dirty coeff. (plain/finned)	W/(m2*K)	27,3 /		Tubes Plain
Effective area (plain/finned)	m2	2 /		No. 30 OD 19,05 Tks 2,11 mm
Effective MTD	C	9,32		Pattern 30 Pitch 23,81 mm
Actual/Required area ratio (dirty/clean)		4,59 /	4,67	Baffles Single segmental Cut(%d) 44,56
Vibration problem (Tasc/TEMA)		No /	No	
RhoV2 problem			No	Total cost 10834 Dollar(US)

Resistance Distribution

Overall Coefficient / Resistance Summary	Clean	Dirty	Max Dirty

Area required		m2	0,4	0,4	2
Area ratio: actual/required			4,67	4,59	1
Overall coefficient		W/(m2*K)	27,7	27,3	5,9
Overall resistance		m2*K/W	0,03605	0,03668	0,16821
Shell side fouling		m2*K/W	0.0	0,00018	0,03779
Tube side fouling			0.0	0,00045	0,09437
Resistance Distribution	**W/(m2*K)**	**m2*K/W**	%	%	%
Shell side film	21789,5	0,03187	88,4	86,89	18,94
Shell side fouling	5555,6	0,00018		0,49	22,47
Tube wall	21789,5	0,00005	0,13	0,13	0,03
Tube side fouling *	2224,8	0,00045		1,23	56,1
Tube side film *	241,9	0,00413	11,47	11,27	2,46

* Based on outside surface - Area ratio: Ao/Ai = 1,28

Heat Transfer Coefficients

Film coefficients		**W/(m2*K)**	**Shell Side**		**Tube Side**	
			Bare area (OD) / Finned area		**Bare area (OD) / ID area**	
Overall film coefficients			31,4 /		241,9 /	310,6
	Vapor sensible		31,4 /		/	
	Two phase		/		/	
	Liquid sensible		/		241,9 /	310,6
Heat Transfer Parameters			**In**	**Out**	**In**	**Out**
Prandtl numbers	Vapor		0,77	0,76		
	Liquid				10,86	11,43
Reynolds numbers	Vapor Nominal		1375,22	1329,66		
	Liquid Nominal				327,21	312,42

MTD and Flux

Temperature Difference	C	Heat Flux (based on tube O.D)		kW/m2
Overall effective MTD	9,32	Overall actual flux		0,3
One pass counterflow MTD	9,61	Critical flux		
LMTD based on end points	9,61	Highest actual flux	0,4	
Effective MTD correction factor	0,97	Highest actual/critical flux		

Wall Temperatures	C
Shell mean metal temperature	-4,13
Tube mean metal temperature	4,05
Tube wall temperatures (highest/lowest)	5,26 / 2,87

Duty Distribution

Heat Load Summary	Shell Side		Tube Side	
	kW	% total	kW	% total
Vapor only	0,1	100	0	0
2-Phase vapor	0	0	0	0
Latent heat	0	0	0	0
2-Phase liquid	0	0	0	0
Liquid only	0	0	-0,1	100
Total	0,1	100	-0,1	100

Pressure Drop

Pressure Drop kPa	Shell Side	Tube Side
Maximum allowed	50	50
Total calculated	0,219	0,044
Gravitational	0	0
Frictional	0,219	0,044
Momentum change	0	0

Pressure drop distribution	m/s	kPa	%dp	m/s	kPa	%dp
Inlet nozzle	6,94	0,126	57,47	0,11	0,006	14,49
Entering bundle	0,1			0,03	0,001	1,73
Inside tubes				0,03 0,03	0,033	74,64
Inlet space Xflow	0,09	0	0,09			
Baffle Xflow	0,14 0,15	0,001	0,36			
Baffle windows	0,1 0,1	0	0,1			
Outlet space Xflow	0,09	0	0,09			
Exiting bundle	0,1			0,03	0,001	2,67
Outlet nozzle	7,22	0,092	41,89	0,11	0,003	6,47
Intermediate nozzles						

Thermosiphon Piping

Piping reference points	m	Pressure points	kPa	C	Quality
Height of liquid in column		Liquid level in column			
Height of heat transfer region inlet		Inlet to heated section			
Height of heat transfer region outlet		Boiling boundary position			
Height of column return line		Outlet of heated section		0	1
		Exit of outlet piping			

Pressure changes (-loss/+gain)	kPa	Inlet circuit	Exchanger	Outlet circuit
Frictional				
Gravitational				
Momentum				
Flashing				
Nozzles				
Total				
Inlet Circuit Piping				
Inlet circuit element				

Frictional pressure drop at entry **kPa**										
Frictional pressure drop at element **kPa**										
Outlet Circuit Piping										
Outlet circuit element										
Frictional pressure drop at entry **kPa**										
Frictional pressure drop at element **kPa**										
Void fraction										
Two phase flow pattern										

Flow Analysis

Shell Side Flow Fractions	**Inlet**	**Middle**	**Outlet**	**Diameter Clearance**	**mm**
Crossflow	0,65	0,51	0,65		
Window	0,85	0,67	0,85		
Baffle hole - tube OD	0,04	0,09	0,04	0,79	
Baffle OD - shell ID	0,11	0,24	0,11	3,18	
Shell ID - bundle OTL	0,08	0,06	0,08	12,7	
Pass lanes	0,12	0,09	0,12		
Rho*V2 Analysis	**Flow area**	**Velocity**	**Density**	**Rho*V2**	**TEMA limit**
	mm2	**m/s**	**kg/m3**	**kg/(m*s2)**	**kg/(m*s2)**
Shell inlet nozzle	151	6,94	5,08	245	2232
Shell entrance	895	1,17	5,08	7	5953
Bundle entrance	10929	0,1	5,08	0	5953
Bundle exit	10929	0,1	4,88	0	5953
Shell exit	895	1,22	4,88	7	5953
Shell outlet nozzle	151	7,22	4,88	255	
	mm2	**m/s**	**kg/m3**		
Tube inlet nozzle	151	0,11	1021,63	12	8928

Tube inlet	518	0,03	1021,63	1	
Tube outlet	518	0,03	1022,77	1	
Tube outlet nozzle	151	0,11	1022,77	12	

Thermosiphons and Kettles

Thermosiphons	
Thermosiphon stability	
Vertical tube side thermosiphons	
Flow reversal criterion - top of the tubes (should be >0.5)	
Flooding criterion - top of the tubes (should be >1.0)	
Fraction of the tube length flooded	
Kutateladze Number in axial nozzle	
Kettles	
Recirculation ratio	
Quality at top of bundle	
Entrainment fraction	

Methods Summary

	Hot side	**Cold side**
Heat Transfer Coefficient multiplier	No	No
Heat Transfer Coefficient specified	No	No
Pressure drop multiplier	No	No
Pressure drop calculation option	friction only	friction only
Calculation method	Advanced method	
Desuperheating heat transfer method	Wet wall	
Multicomponent condensing heat transfer method	HTFS - Silver-Bell	
Vapor shear enhanced condensation	Yes	

Liquid subcooling heat transfer (vertical shell)	Not Used
Subcooled boiling accounted for in	Heat transfer & pressure drop
Post dryout heat transfer accounted for in	No
Correction to user-supplied boiling curve	Boiling curve not used
Falling film evaporation method	HTFS recommended method
Single phase tube side heat transfer method	HTFS recommended method
Lowfin Calculation method	HTFS / ESDU

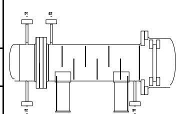
Appendix C – Sensitivity analysis for shell-and-tube exchanger

C-1 – Shell ID 1

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 205 / 1200mm	Type BES	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 1,8 m2	Shells/unit 1		Surf/shell (eff.)	1,8	m2
8	PERFORMANCE OF ONE UNIT					
9	Fluid allocation		Shell Side		Tube Side	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8215
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	698,076
26	Velocity	m/s	0,13		0,06	
27	Pressure drop, allow./calc.	kPa	50	0,093	50	1,924
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045	Ao based
29	Heat exchanged 4,3 kW			MTD corrected	75,99	C
30	Transfer rate, Service 31,9	Dirty	36,8	Clean	37,7	W/(m2*K)

31	CONSTRUCTION OF ONE SHELL				Sketch
32			Shell Side	Tube Side	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	2	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 30	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm



41	Tube type	Plain			Material		Carbon Steel		Tube pattern	30
42	Shell	Carbon Steel	ID	205	OD 219,08	mm	Shell cover		Carbon Steel	
43	Channel or bonnet		Carbon Steel				Channel cover		Carbon Steel	
44	Tubesheet-stationary		Carbon Steel				Tubesheet-floating		Carbon Steel	
45	Floating head cover		Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel		Type	Single segmental	Cut(%d)	41,48 H	Spacing: c/c	120	mm
47	Baffle-long	-			Seal type				Inlet 195,21	mm
48	Supports-tube				U-bend		Type			
49	Bypass seal				Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-			Type					
51	RhoV2-Inlet nozzle		3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side		Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head		Flat Metal Jacket Fibe							
54	Code requirements		ASME Code Sec VIII Div 1				TEMA class		R - refinery service	
55	Weight/Shell	233,1		Filled with water		281,6			Bundle 56,7	kg
56	Remarks									
57										
58										

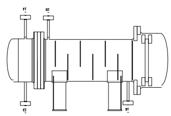
C-2 – Shell ID 2

Heat Exchanger Specification Sheet

1												
2												
3												
4												
5												
6	Size	250	/	1200mm	Type	BES	Hor	Connected in	1	parallel	1	series
7	Surf/unit(eff.)	2,3	m2	Shells/unit	1	Surf/shell (eff.)	2,3	m2				

8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	692,767
26	Velocity	m/s	0,09		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,092	50	7,233
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based
29	Heat exchanged	4,3	kW		MTD corrected 76,38	C
30	Transfer rate, Service	25,1	Dirty 32,7		Clean 33,4	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**							**Sketch**	
32				**Shell Side**		**Tube Side**			
33	Design/vac/test pressure:g		kPa	700 / /		800 / /			
34	Design temperature		C	35		45			
35	Number passes per shell			1		6			
36	Corrosion allowance		mm	3,18		3,18			
37	Connections	In	mm	1 12,7 / -		1 12,7 / -			
38	Size/rating	Out		1 12,7 / -		1 12,7 / -			
39	Nominal	Intermediate		/ -		/ -			
40	Tube No. 38	OD	19,05	Tks- Avg 2,11	mm	Length 1,2	m	Pitch23,81	mm
41	Tube type	Plain		Material		Carbon Steel		Tube pattern	30
42	Shell	Carbon Steel	ID 250	OD 266	mm	Shell cover		Carbon Steel	
43	Channel or bonnet		Carbon Steel			Channel cover		Carbon Steel	
44	Tubesheet-stationary		Carbon Steel			Tubesheet-floating		Carbon Steel	

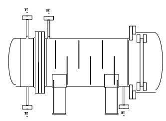


45	Floating head cover	Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	157,71	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	302,7	Filled with water		376,8		Bundle	78,9	kg
56	Remarks								
57									
58									

C-3 – Shell ID 3

Heat Exchanger Specification Sheet

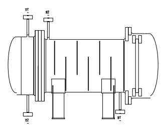
1						
2						
3						
4						
5						
6	Size 300 / 1200mm	Type BES Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 4,1 m2	Shells/unit 1	Surf/shell (eff.)	4,1	m2	
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8213
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,907	700	696,272
26	Velocity	m/s	0,08		0,11	
27	Pressure drop, allow./calc.	kPa	50	0,093	50	3,728
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045 Ao based	
29	Heat exchanged 4,3 kW		MTD corrected		76,72	C
30	Transfer rate, Service 13,5		Dirty 28,5		Clean 29	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	6	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 70	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel	ID 300 OD 318 mm		Shell cover	Carbon Steel
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel

45	Floating head cover		Carbon Steel			Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	39,41 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	157,71	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	406,1		Filled with water	515,6		Bundle	125,2	kg
56	Remarks								
57									
58									

C-4 – Shell ID 4

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 350 / 1200mm	Type BES Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 6,4 m2	Shells/unit 1	Surf/shell (eff.)	6,4	m2	
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8215
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	697,49
26	Velocity	m/s	0,08		0,04	
27	Pressure drop, allow./calc.	kPa	50	0,092	50	2,51
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045 Ao based	
29	Heat exchanged 4,3 kW			MTD corrected	77,08	C
30	Transfer rate, Service 8,7		Dirty 25,3	Clean 25,7		W/(m2*K)
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**	
32			**Shell Side**	**Tube Side**		
33	Design/vac/test pressure:g	kPa	700 / /	800 / /		
34	Design temperature	C	35	45		
35	Number passes per shell		1	6		
36	Corrosion allowance	mm	3,18	3,18		
37	Connections	In mm	1 12,7 / -	1 12,7 / -		
38	Size/rating	Out	1 12,7 / -	1 12,7 / -		
39	Nominal	Intermediate	/ -	/ -		
40	Tube No. 108	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81	mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern	30
42	Shell Carbon Steel	ID 350	OD 370 mm	Shell cover	Carbon Steel	
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel	
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel	

45	Floating head cover		Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	35,03 H	Spacing: c/c		135	mm
47	Baffle-long	-		Seal type				Inlet	157,71	mm
48	Supports-tube			U-bend			Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv			
50	Expansion joint	-		Type						
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side			Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe								
54	Code requirements	ASME Code Sec VIII Div 1					TEMA class	R - refinery service		
55	Weight/Shell	540,6		Filled with water	693,3			Bundle	180,1	kg
56	Remarks									
57										
58										

C-5 – Shell ID 5

Heat Exchanger Specification Sheet

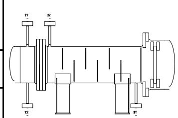
1							
2							
3							
4							
5							
6	Size 400 / 1200mm		Type BES	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 8,9 m2		Shells/unit 1		Surf/shell (eff.)	8,9	m2
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		**Shell Side**		**Tube Side**		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		656		
12	Vapor (In/Out)	kg/h	0	19	0	0	
13	Liquid	kg/h	19	0	656	656	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-160	0	10	1,95	
17	Dew / Bubble point	C	-48,08	-135,23			
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3	
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8215	
20	Molecular wt, Vap			18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928	
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199	
24	Latent heat	kJ/kg	436,2	379,8			
25	Pressure (abs)	kPa	600	599,908	700	698,053	
26	Velocity	m/s	0,07		0,04		
27	Pressure drop, allow./calc.	kPa	50	0,092	50	1,947	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based
29	Heat exchanged 4,3 kW				MTD corrected	76,77	C
30	Transfer rate, Service 6,3		Dirty 23,6		Clean	24	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	6	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 150	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel	ID 400	OD 420 mm	Shell cover	Carbon Steel
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel
45	Floating head cover	Carbon Steel		Impingement protection	None

46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	31,75 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	156,21	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1			TEMA class		R - refinery service		
55	Weight/Shell	670,6	Filled with water		876		Bundle	247,7	kg
56	Remarks								
57									
58									

C-6 – Tube OD 1

Heat Exchanger Specification Sheet

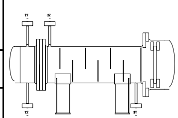
No.							
1							
2							
3							
4							
5							
6	Size 205 / 1200mm		Type BES Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 4,6 m2		Shells/unit 1	Surf/shell (eff.)	4,6	m2	
8	PERFORMANCE OF ONE UNIT						
9	Fluid allocation		Shell Side		Tube Side		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		656		
12	Vapor (In/Out)	kg/h	0	19	0	0	
13	Liquid	kg/h	19	0	656	656	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-160	0	10	1,95	
17	Dew / Bubble point	C	-48,08	-135,23			
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29	
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201	
20	Molecular wt, Vap			18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928	
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199	
24	Latent heat	kJ/kg	436,2	379,8			
25	Pressure (abs)	kPa	600	599,906	700	600,885	
26	Velocity	m/s	0,11		0,42		
27	Pressure drop, allow./calc.	kPa	25	0,094	100	99,115	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based
29	Heat exchanged 4,3 kW			MTD corrected	77,09	C	
30	Transfer rate, Service 12,2		Dirty 62,1	Clean 64,6		W/(m2*K)	
31	CONSTRUCTION OF ONE SHELL				Sketch		
32			Shell Side	Tube Side			
33	Design/vac/test pressure:g	kPa	700 / /	800 / /			
34	Design temperature	C	35	45			
35	Number passes per shell		1	8			
36	Corrosion allowance	mm	3,18	3,18			
37	Connections	In mm	1 12,7 / -	1 12,7 / -			
38	Size/rating	Out	1 12,7 / -	1 12,7 / -			
39	Nominal	Intermediate	/ -	/ -			
40	Tube No. 290	OD 5	Tks- Avg 0,56 mm	Length 1,2 m	Pitch6,25 mm		
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30		
42	Shell Carbon Steel	ID 205	OD 219,08 mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel		

45	Floating head cover	Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	40,2 H	Spacing: c/c	120	mm
47	Baffle-long	-		Seal type			Inlet	204,21	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint		Exp. 2 grv			
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0	Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	226,2		Filled with water	275,5		Bundle	44,4	kg
56	Remarks								
57									
58									

C-7 - Tube OD 2

Heat Exchanger Specification Sheet

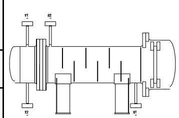
1								
2								
3								
4								
5								
6	Size 205 / 1200mm		Type BES	Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 2,2 m2		Shells/unit 1		Surf/shell (eff.)	2,2	m2	
8	**PERFORMANCE OF ONE UNIT**							
9	Fluid allocation		**Shell Side**		**Tube Side**			
10	Fluid name							
11	Fluid quantity, Total	kg/h	19		656			
12	Vapor (In/Out)	kg/h	0	19	0	0		
13	Liquid	kg/h	19	0	656	656		
14	Noncondensable	kg/h	0		0			
15								
16	Temperature (In/Out)	C	-160	0	10	1,95		
17	Dew / Bubble point	C	-48,08	-135,23				
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29		
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201		
20	Molecular wt, Vap			18,12				
21	Molecular wt, NC							
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928		
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199		
24	Latent heat	kJ/kg	436,2	379,8				
25	Pressure (abs)	kPa	600	599,906	700	659,462		
26	Velocity	m/s	0,1		1,33			
27	Pressure drop, allow./calc.	kPa	25	0,094	100	40,538		
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00039	Ao based	
29	Heat exchanged 4,3	kW			MTD corrected	77,1	C	
30	Transfer rate, Service 25,9		Dirty	52,1	Clean	53,8	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**			
32			**Shell Side**		**Tube Side**			
33	Design/vac/test pressure:g	kPa	700 / /		800 / /			
34	Design temperature	C	35		45			
35	Number passes per shell		1		8			
36	Corrosion allowance	mm	3,18		3,18			
37	Connections	In mm	1 12,7 / -		1 12,7 / -			
38	Size/rating	Out	1 12,7 / -		1 12,7 / -			
39	Nominal	Intermediate	/ -		/ -			
40	Tube No. 68	OD 10	Tks- Avg 0,56	mm	Length 1,2 m	Pitch12,5	mm	
41	Tube type Plain		Material		Carbon Steel	Tube pattern	30	
42	Shell Carbon Steel	ID 205	OD 219,08	mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel		



45	Floating head cover	Carbon Steel					Impingement protection	None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	40,6 H	Spacing: c/c	130	mm
47	Baffle-long	-		Seal type			Inlet	179,21	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	215,7	Filled with water		266,4		Bundle	33,9	kg
56	Remarks								
57									
58									

C-8 – Tube OD 3

Heat Exchanger Specification Sheet

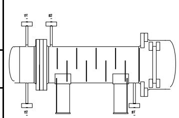
1								
2								
3								
4								
5								
6	Size 205 / 1200mm	Type BES	Hor	Connected in	1	parallel	1	series
7	Surf/unit(eff.)	1,5	m2	Shells/unit 1	Surf/shell (eff.)	1,5		m2
8	**PERFORMANCE OF ONE UNIT**							
9	Fluid allocation		**Shell Side**		**Tube Side**			
10	Fluid name							
11	Fluid quantity, Total	kg/h	19		656			
12	Vapor (In/Out)	kg/h	0	19	0	0		
13	Liquid	kg/h	19	0	656	656		
14	Noncondensable	kg/h	0		0			
15								
16	Temperature (In/Out)	C	-160	0	10	1,95		
17	Dew / Bubble point	C	-48,08	-135,23				
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29		
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201		
20	Molecular wt, Vap			18,12				
21	Molecular wt, NC							
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928		
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199		
24	Latent heat	kJ/kg	436,2	379,8				
25	Pressure (abs)	kPa	600	599,907	700	680,511		
26	Velocity	m/s	0,11		0,38			
27	Pressure drop, allow./calc.	kPa	25	0,093	100	19,489		
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based	
29	Heat exchanged	4,3	kW		MTD corrected	76,45	C	
30	Transfer rate, Service	37,3	Dirty	43,3	Clean	44,5	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**			
32			**Shell Side**	**Tube Side**				
33	Design/vac/test pressure:g	kPa	700 / /	800 / /				
34	Design temperature	C	35	45				
35	Number passes per shell		1	6				
36	Corrosion allowance	mm	3,18	3,18				
37	Connections	In mm	1 12,7 / -	1 12,7 / -				
38	Size/rating	Out	1 12,7 / -	1 12,7 / -				
39	Nominal	Intermediate	/ -	/ -				
40	Tube No. 32	OD 15	Tks- Avg 1,65	mm	Length 1,2 m	Pitch18,75	mm	
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30		
42	Shell Carbon Steel	ID 205	OD 219,08	mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel		

45	Floating head cover		Carbon Steel			Impingement protection	None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	43,25 H	Spacing: c/c	120 mm
47	Baffle-long	-		Seal type			Inlet	201,21 mm
48	Supports-tube			U-bend		Type		
49	Bypass seal			Tube-tubesheet joint		Exp. 2 grv		
50	Expansion joint	-		Type				
51	RhoV2-Inlet nozzle	3	Bundle entrance		0	Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe		
53	Floating head	Flat Metal Jacket Fibe						
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service	
55	Weight/Shell	227,4		Filled with water	276,6		Bundle	45,2 kg
56	Remarks							
57								
58								

C-9 – Tube OD 4

Heat Exchanger Specification Sheet

1							
2							
3							
4							
5							
6	Size 205 / 1200mm	Type BES	Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 1,6 m2	Shells/unit 1		Surf/shell (eff.)	1,6	m2	
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		**Shell Side**		**Tube Side**		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		656		
12	Vapor (In/Out)	kg/h	0	19	0	0	
13	Liquid	kg/h	19	0	656	656	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-160	0	10	1,95	
17	Dew / Bubble point	C	-48,08	-135,23			
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3	
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8216	
20	Molecular wt, Vap			18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928	
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199	
24	Latent heat	kJ/kg	436,2	379,8			
25	Pressure (abs)	kPa	600	599,907	700	698,184	
26	Velocity	m/s	0,16		0,07		
27	Pressure drop, allow./calc.	kPa	25	0,093	100	1,816	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00044	Ao based
29	Heat exchanged 4,3	kW		MTD corrected	75,7	C	
30	Transfer rate, Service 35,3		Dirty 36,2	Clean	37	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**		
32			**Shell Side**	**Tube Side**			
33	Design/vac/test pressure:g	kPa	700 / /	800 / /			
34	Design temperature	C	35	45			
35	Number passes per shell		1	2			
36	Corrosion allowance	mm	3,18	3,18			
37	Connections	In mm	1 12,7 / -	1 12,7 / -			
38	Size/rating	Out	1 12,7 / -	1 12,7 / -			
39	Nominal	Intermediate	/ -	/ -			
40	Tube No. 26	OD 20	Tks- Avg 2,11 mm	Length 1,2 m	Pitch25	mm	
41	Tube type Plain		Material	Carbon Steel	Tube pattern	30	
42	Shell Carbon Steel	ID 205	OD 219,08 mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel		

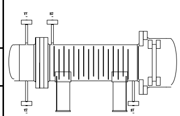


45	Floating head cover		Carbon Steel			Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	41,25 H	Spacing: c/c	100	mm
47	Baffle-long	-		Seal type			Inlet	143,71	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	232,2		Filled with water	280,9		Bundle	55,6	kg
56	Remarks								
57									
58									

C-10 – Tube OD 5

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 205 / 1200mm		Type BES Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 1,4 m2		Shells/unit 1	Surf/shell (eff.)	1,4	m2
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8216
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,9	700	698,518
26	Velocity	m/s	0,4		0,05	
27	Pressure drop, allow./calc.	kPa	25	0,1	100	1,482
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00042	Ao based
29	Heat exchanged 4,3	kW		MTD corrected	75,77	C
30	Transfer rate, Service 41,4		Dirty 42,1	Clean	43,2	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	2	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 18	OD 25	Tks- Avg 2,11 mm	Length 1,2 m	Pitch31,25 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel	ID 205	OD 219,08 mm	Shell cover	Carbon Steel
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel

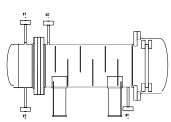
45	Floating head cover	Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	13,63 H	Spacing: c/c	50,8	mm
47	Baffle-long	-		Seal type			Inlet	105,21	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	237,3	Filled with water		285,4		Bundle	60	kg
56	Remarks								
57									
58									

C-11 – Tube length 1

Heat Exchanger Specification Sheet

1									
2									
3									
4									
5									
6	Size	250 / 1200mm	Type	BES	Hor	Connected in	1	parallel	1 series
7	Surf/unit(eff.)	2,3 m2	Shells/unit	1		Surf/shell (eff.)	2,3		m2

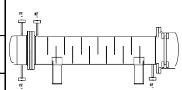
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	692,767
26	Velocity	m/s	0,09		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,092	50	7,233
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045	Ao based
29	Heat exchanged	4,3 kW		MTD corrected	76,38	C
30	Transfer rate, Service	25,1	Dirty 32,7	Clean	33,4	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	6	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 38	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm
41	Tube type	Plain	Material	Carbon Steel	Tube pattern 30
42	Shell	Carbon Steel	ID 250 OD 266 mm	Shell cover	Carbon Steel
43	Channel or bonnet		Carbon Steel	Channel cover	Carbon Steel
44	Tubesheet-stationary		Carbon Steel	Tubesheet-floating	Carbon Steel

45	Floating head cover		Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c		135	mm
47	Baffle-long	-		Seal type				Inlet	157,71	mm
48	Supports-tube			U-bend		Type				
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv			
50	Expansion joint	-		Type						
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe				
53	Floating head	Flat Metal Jacket Fibe								
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class		R - refinery service		
55	Weight/Shell	302,7	Filled with water		376,8			Bundle	78,9	kg
56	Remarks									
57										
58										

C-12 – Tube length 2

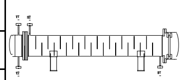
Heat Exchanger Specification Sheet

#								
1								
2								
3								
4								
5								
6	Size	250 / 2200mm	Type	BES	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.)	4,5 m2	Shells/unit	1		Surf/shell (eff.)	4,5	m2
8	**PERFORMANCE OF ONE UNIT**							
9	Fluid allocation		**Shell Side**		**Tube Side**			
10	Fluid name							
11	Fluid quantity, Total	kg/h	19		656			
12	Vapor (In/Out)	kg/h	0	19	0	0		
13	Liquid	kg/h	19	0	656	656		
14	Noncondensable	kg/h	0		0			
15								
16	Temperature (In/Out)	C	-160	0	10	1,95		
17	Dew / Bubble point	C	-48,08	-135,23				
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29		
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8205		
20	Molecular wt, Vap			18,12				
21	Molecular wt, NC							
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928		
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199		
24	Latent heat	kJ/kg	436,2	379,8				
25	Pressure (abs)	kPa	600	599,907	700	689,022		
26	Velocity	m/s	0,09		0,19			
27	Pressure drop, allow./calc.	kPa	50	0,093	50	10,978		
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based		
29	Heat exchanged	4,3	kW		MTD corrected	76,41	C	
30	Transfer rate, Service	12,5	Dirty	30,5	Clean	31,1	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**			
32			**Shell Side**	**Tube Side**				
33	Design/vac/test pressure:g	kPa	700 / /	800 / /				
34	Design temperature	C	35	45				
35	Number passes per shell		1	6				
36	Corrosion allowance	mm	3,18	3,18				
37	Connections	In mm	1 12,7 / -	1 12,7 / -				
38	Size/rating	Out	1 12,7 / -	1 12,7 / -				
39	Nominal	Intermediate	/ -	/ -				
40	Tube No. 38	OD 19,05	Tks- Avg 2,11	mm	Length 2,2 m	Pitch23,81	mm	
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30		
42	Shell	Carbon Steel	ID 250	OD 266 mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel		

45	Floating head cover	Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	252,71	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	394,4	Filled with water		512,6		Bundle	118,2	kg
56	Remarks								
57									
58									

C-13 – Tube length 3

Heat Exchanger Specification Sheet

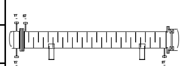
1								
2								
3								
4								
5								
6	Size 250 / 3200mm		Type BES	Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.)	6,8 m2	Shells/unit 1		Surf/shell (eff.)	6,8	m2	
8	**PERFORMANCE OF ONE UNIT**							
9	Fluid allocation		**Shell Side**		**Tube Side**			
10	Fluid name							
11	Fluid quantity, Total	kg/h	19		656			
12	Vapor (In/Out)	kg/h	0	19	0	0		
13	Liquid	kg/h	19	0	656	656		
14	Noncondensable	kg/h	0		0			
15								
16	Temperature (In/Out)	C	-160	0	10	1,95		
17	Dew / Bubble point	C	-48,08	-135,23				
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29		
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201		
20	Molecular wt, Vap			18,12				
21	Molecular wt, NC							
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928		
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199		
24	Latent heat	kJ/kg	436,2	379,8				
25	Pressure (abs)	kPa	600	599,906	700	685,614		
26	Velocity	m/s	0,09		0,19			
27	Pressure drop, allow./calc.	kPa	50	0,094	50	14,386		
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based		
29	Heat exchanged	4,3	kW		MTD corrected	76,36	C	
30	Transfer rate, Service	8,3	Dirty	30,5	Clean	31,1	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**					**Sketch**		
32			**Shell Side**		**Tube Side**			
33	Design/vac/test pressure:g	kPa	700 / /		800 / /			
34	Design temperature	C	35		45			
35	Number passes per shell		1		6			
36	Corrosion allowance	mm	3,18		3,18			
37	Connections	In mm	1 12,7 / -		1 12,7 / -			
38	Size/rating	Out	1 12,7 / -		1 12,7 / -			
39	Nominal	Intermediate	/ -		/ -			
40	Tube No. 38	OD 19,05	Tks- Avg 2,11	mm	Length 3,2 m	Pitch23,81	mm	
41	Tube type	Plain	Material		Carbon Steel	Tube pattern	30	
42	Shell	Carbon Steel	ID 250	OD 266 mm	Shell cover	Carbon Steel		
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel		
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel		

45	Floating head cover	Carbon Steel			Impingement protection					None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29	H	Spacing: c/c		135	mm
47	Baffle-long	-		Seal type					Inlet	212,71	mm
48	Supports-tube			U-bend			Type				
49	Bypass seal			Tube-tubesheet joint				Exp. 2 grv			
50	Expansion joint	-		Type							
51	RhoV2-Inlet nozzle	3	Bundle entrance		0			Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side			Flat Metal Jacket Fibe				
53	Floating head	Flat Metal Jacket Fibe									
54	Code requirements	ASME Code Sec VIII Div 1			TEMA class				R - refinery service		
55	Weight/Shell	487,3	Filled with water		649,4				Bundle	158,7	kg
56	Remarks										
57											
58											

C-14 - Tube length 4

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 250 / 4200mm	Type BES	Hor	Connected in 1 parallel	1 series	
7	Surf/unit(eff.) 9,1 m2	Shells/unit 1		Surf/shell (eff.)	9,1	m2
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,905	700	682,05
26	Velocity	m/s	0,09		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,095	50	17,95
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045 Ao based	
29	Heat exchanged 4,3	kW		MTD corrected	76,36	C
30	Transfer rate, Service 6,2		Dirty 30,8	Clean	31,4	W/(m2*K)

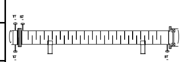
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	6	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 38	OD	19,05 Tks- Avg 2,11 mm	Length 4,2 m	Pitch23,81 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel		ID 250 OD 266 mm	Shell cover	Carbon Steel
43	Channel or bonnet		Carbon Steel	Channel cover	Carbon Steel
44	Tubesheet-stationary		Carbon Steel	Tubesheet-floating	Carbon Steel

45	Floating head cover	Carbon Steel				Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c	135	mm
47	Baffle-long	-		Seal type			Inlet	172,71	mm
48	Supports-tube			U-bend		Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	580,2		Filled with water	786,2		Bundle	199,2	kg
56	Remarks								
57									
58									

C-15 – Tube length 5

Heat Exchanger Specification Sheet

1							
2							
3							
4							
5							
6	Size 250 / 5200mm	Type BES	Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.) 11,3 m2	Shells/unit 1		Surf/shell (eff.)	11,3	m2	
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		**Shell Side**		**Tube Side**		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		656		
12	Vapor (In/Out)	kg/h	0	19	0	0	
13	Liquid	kg/h	19	0	656	656	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-160	0	10	1,95	
17	Dew / Bubble point	C	-48,08	-135,23			
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,29	
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8201	
20	Molecular wt, Vap			18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928	
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199	
24	Latent heat	kJ/kg	436,2	379,8			
25	Pressure (abs)	kPa	600	599,904	700	679,02	
26	Velocity	m/s	0,09		0,19		
27	Pressure drop, allow./calc.	kPa	50	0,096	50	20,98	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based	
29	Heat exchanged 4,3	kW		MTD corrected	76,42	C	
30	Transfer rate, Service 5	Dirty	30,4	Clean	31	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**		
32			**Shell Side**	**Tube Side**			
33	Design/vac/test pressure:g	kPa	700 / /	800 / /			
34	Design temperature	C	35	45			
35	Number passes per shell		1	6			
36	Corrosion allowance	mm	3,18	3,18			
37	Connections	In mm	1 12,7 / -	1 12,7 / -			
38	Size/rating	Out	1 12,7 / -	1 12,7 / -			
39	Nominal	Intermediate	/ -	/ -			
40	Tube No. 38	OD 19,05	Tks- Avg 2,11	mm	Length 5,2 m	Pitch23,81	mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern	30	
42	Shell Carbon Steel	ID 250	OD 266	mm	Shell cover	Carbon Steel	
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel	
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel	

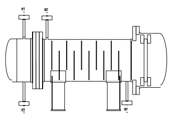


45	Floating head cover		Carbon Steel			Impingement protection	None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c 135	mm
47	Baffle-long	-		Seal type			Inlet 267,71	mm
48	Supports-tube			U-bend		Type		
49	Bypass seal			Tube-tubesheet joint		Exp. 2 grv		
50	Expansion joint	-		Type				
51	RhoV2-Inlet nozzle	3	Bundle entrance		0	Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe		
53	Floating head	Flat Metal Jacket Fibe						
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service	
55	Weight/Shell	671,9		Filled with water	922		Bundle 238,5	kg
56	Remarks							
57								
58								

C-16 – Baffle pitch 1

Heat Exchanger Specification Sheet

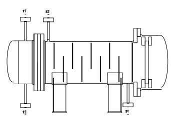
1						
2						
3						
4						
5						
6	Size 250 / 1200mm	Type BES	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 2,3 m2	Shells/unit 1		Surf/shell (eff.)	2,3	m2
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,906	700	692,614
26	Velocity	m/s	0,16		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,094	50	7,386
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045	Ao based
29	Heat exchanged 4,3	kW		MTD corrected	76,28	C
30	Transfer rate, Service 25,1	Dirty	36,5	Clean	37,3	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**				**Sketch**
32			**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g	kPa	700 / /	800 / /	
34	Design temperature	C	35	45	
35	Number passes per shell		1	6	
36	Corrosion allowance	mm	3,18	3,18	
37	Connections	In mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out	1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate	/ -	/ -	
40	Tube No. 38	OD 19,05	Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm
41	Tube type Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel	ID 250	OD 266 mm	Shell cover	Carbon Steel
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel

45	Floating head cover	Carbon Steel					Impingement protection	None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	29,04 H	Spacing: c/c	80	mm
47	Baffle-long	-		Seal type			Inlet	135,21	mm
48	Supports-tube			U-bend			Type		
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance			0	Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side			Flat Metal Jacket Fibe		
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1					TEMA class	R - refinery service	
55	Weight/Shell	306,2		Filled with water		379,9	Bundle	82,4	kg
56	Remarks								
57									
58									

C-17 – Baffle pitch 2

Heat Exchanger Specification Sheet

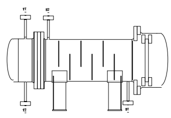
1								
2								
3								
4								
5								
6	Size 250 / 1200mm		Type BES	Hor	Connected in	1 parallel	1 series	
7	Surf/unit(eff.)	2,3 m2	Shells/unit 1		Surf/shell (eff.)	2,3	m2	
8	**PERFORMANCE OF ONE UNIT**							
9	Fluid allocation		**Shell Side**		**Tube Side**			
10	Fluid name							
11	Fluid quantity, Total	kg/h	19		656			
12	Vapor (In/Out)	kg/h	0	19	0	0		
13	Liquid	kg/h	19	0	656	656		
14	Noncondensable	kg/h	0		0			
15								
16	Temperature (In/Out)	C	-160	0	10	1,95		
17	Dew / Bubble point	C	-48,08	-135,23				
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3		
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209		
20	Molecular wt, Vap			18,12				
21	Molecular wt, NC							
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928		
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199		
24	Latent heat	kJ/kg	436,2	379,8				
25	Pressure (abs)	kPa	600	599,907	700	692,908		
26	Velocity	m/s	0,13		0,19			
27	Pressure drop, allow./calc.	kPa	50	0,093	50	7,092		
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based	
29	Heat exchanged	4,3	kW		MTD corrected	76,4	C	
30	Transfer rate, Service	25	Dirty	32,8	Clean	33,5	W/(m2*K)	
31	**CONSTRUCTION OF ONE SHELL**				**Sketch**			
32			**Shell Side**	**Tube Side**				
33	Design/vac/test pressure:g	kPa	700 / /	800 / /				
34	Design temperature	C	35	45				
35	Number passes per shell		1	6				
36	Corrosion allowance	mm	3,18	3,18				
37	Connections	In mm	1 12,7 / -	1 12,7 / -				
38	Size/rating	Out	1 12,7 / -	1 12,7 / -				
39	Nominal	Intermediate	/ -	/ -				
40	Tube No. 38	OD 19,05	Tks- Avg 2,11	mm	Length 1,2 m	Pitch23,81	mm	
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30		
42	Shell	Carbon Steel	ID 250	OD 266 mm	Shell cover	Carbon Steel		
43	Channel or bonnet		Carbon Steel		Channel cover	Carbon Steel		
44	Tubesheet-stationary		Carbon Steel		Tubesheet-floating	Carbon Steel		

45	Floating head cover		Carbon Steel			Impingement protection			None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c		100	mm
47	Baffle-long	-		Seal type				Inlet	145,21	mm
48	Supports-tube			U-bend			Type			
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv			
50	Expansion joint	-		Type						
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit		0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side			Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe								
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class		R - refinery service		
55	Weight/Shell	304,4		Filled with water	378,2			Bundle	80,6	kg
56	Remarks									
57										
58										

C-18 – Baffle pitch 3

Heat Exchanger Specification Sheet

1						
2						
3						
4						
5						
6	Size 250 / 1200mm		Type BES Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 2,3 m2		Shells/unit 1	Surf/shell (eff.)	2,3	m2
8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	693,015
26	Velocity	m/s	0,11		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,092	50	6,985
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035 0,00045	Ao based
29	Heat exchanged 4,3	kW		MTD corrected	76,45	C
30	Transfer rate, Service 25		Dirty 33,9	Clean	34,6	W/(m2*K)

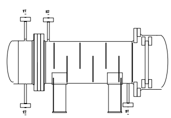
31	**CONSTRUCTION OF ONE SHELL**					**Sketch**
32				**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g		kPa	700 / /	800 / /	
34	Design temperature		C	35	45	
35	Number passes per shell			1	6	
36	Corrosion allowance		mm	3,18	3,18	
37	Connections	In	mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out		1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate		/ -	/ -	
40	Tube No. 38		OD	19,05 Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81 mm
41	Tube type	Plain		Material	Carbon Steel	Tube pattern 30
42	Shell Carbon Steel			ID 250 OD 266 mm	Shell cover	Carbon Steel
43	Channel or bonnet			Carbon Steel	Channel cover	Carbon Steel
44	Tubesheet-stationary			Carbon Steel	Tubesheet-floating	Carbon Steel

45	Floating head cover	Carbon Steel			Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d) 37,29 H	Spacing: c/c	120	mm
47	Baffle-long	-		Seal type		Inlet	195,21	mm
48	Supports-tube			U-bend	Type			
49	Bypass seal			Tube-tubesheet joint		Exp. 2 grv		
50	Expansion joint	-		Type				
51	RhoV2-Inlet nozzle	3	Bundle entrance	0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side	Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe						
54	Code requirements	ASME Code Sec VIII Div 1			TEMA class	R - refinery service		
55	Weight/Shell	302,7	Filled with water	376,8		Bundle	78,9	kg
56	Remarks							
57								
58								

C-19 – Baffle pitch 4

Heat Exchanger Specification Sheet

1							
2							
3							
4							
5							
6	Size 250 / 1200mm		Type BES	Hor	Connected in	1 parallel	1 series
7	Surf/unit(eff.) 2,3 m2		Shells/unit 1		Surf/shell (eff.)	2,3	m2
8	**PERFORMANCE OF ONE UNIT**						
9	Fluid allocation		**Shell Side**		**Tube Side**		
10	Fluid name						
11	Fluid quantity, Total	kg/h	19		656		
12	Vapor (In/Out)	kg/h	0	19	0	0	
13	Liquid	kg/h	19	0	656	656	
14	Noncondensable	kg/h	0		0		
15							
16	Temperature (In/Out)	C	-160	0	10	1,95	
17	Dew / Bubble point	C	-48,08	-135,23			
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3	
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,8209	
20	Molecular wt, Vap			18,12			
21	Molecular wt, NC						
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928	
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199	
24	Latent heat	kJ/kg	436,2	379,8			
25	Pressure (abs)	kPa	600	599,908	700	692,913	
26	Velocity	m/s	0,09		0,19		
27	Pressure drop, allow./calc.	kPa	50	0,092	50	7,087	
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045	Ao based
29	Heat exchanged 4,3	kW			MTD corrected	76,4	C
30	Transfer rate, Service 25		Dirty	32,4	Clean	33	W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**					**Sketch**
32			**Shell Side**	**Tube Side**		
33	Design/vac/test pressure:g	kPa	700 / /	800 / /		
34	Design temperature	C	35	45		
35	Number passes per shell		1	6		
36	Corrosion allowance	mm	3,18	3,18		
37	Connections	In mm	1 12,7 / -	1 12,7 / -		
38	Size/rating	Out	1 12,7 / -	1 12,7 / -		
39	Nominal	Intermediate	/ -	/ -		
40	Tube No. 38	OD	19,05 Tks- Avg 2,11 mm	Length 1,2 m	Pitch23,81	mm
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30
42	Shell Carbon Steel	ID 250	OD 266 mm	Shell cover	Carbon Steel	
43	Channel or bonnet	Carbon Steel		Channel cover	Carbon Steel	
44	Tubesheet-stationary	Carbon Steel		Tubesheet-floating	Carbon Steel	

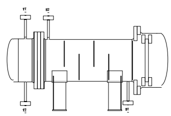
45	Floating head cover		Carbon Steel		Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c	140 mm
47	Baffle-long	-		Seal type			Inlet	145,21 mm
48	Supports-tube			U-bend		Type		
49	Bypass seal			Tube-tubesheet joint		Exp. 2 grv		
50	Expansion joint	-		Type				
51	RhoV2-Inlet nozzle	3	Bundle entrance	0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe		
53	Floating head	Flat Metal Jacket Fibe						
54	Code requirements	ASME Code Sec VIII Div 1			TEMA class	R - refinery service		
55	Weight/Shell	302,7	Filled with water	376,8		Bundle	78,9	kg
56	Remarks							
57								
58								

C-20 – Baffle pitch 5

Heat Exchanger Specification Sheet

1									
2									
3									
4									
5									
6	Size	250 / 1200mm	Type	BES	Hor	Connected in	1	parallel	1 series
7	Surf/unit(eff.)	2,3	m2	Shells/unit	1	Surf/shell (eff.)	2,3		m2

8	**PERFORMANCE OF ONE UNIT**					
9	Fluid allocation		**Shell Side**		**Tube Side**	
10	Fluid name					
11	Fluid quantity, Total	kg/h	19		656	
12	Vapor (In/Out)	kg/h	0	19	0	0
13	Liquid	kg/h	19	0	656	656
14	Noncondensable	kg/h	0		0	
15						
16	Temperature (In/Out)	C	-160	0	10	1,95
17	Dew / Bubble point	C	-48,08	-135,23		
18	Density (Vap / Liq)	kg/m3	/ 462	/	/ 1101,04	/ 1107,3
19	Viscosity	cp	/ 0,1465	/	/ 7,5273	/ 9,821
20	Molecular wt, Vap			18,12		
21	Molecular wt, NC					
22	Specific heat	kJ/(kg*K)	/ 3,088	/	/ 2,952	/ 2,928
23	Thermal conductivity	W/(m*K)	/ 0,1928	/	/ 0,3246	/ 0,3199
24	Latent heat	kJ/kg	436,2	379,8		
25	Pressure (abs)	kPa	600	599,908	700	693,502
26	Velocity	m/s	0,08		0,19	
27	Pressure drop, allow./calc.	kPa	50	0,092	50	6,498
28	Fouling resistance (min)	m2*K/W	0,00018		0,00035	0,00045 Ao based
29	Heat exchanged	4,3	kW		MTD corrected	76,54 C
30	Transfer rate, Service	25	Dirty	32,4	Clean	33,1 W/(m2*K)

31	**CONSTRUCTION OF ONE SHELL**					**Sketch**
32				**Shell Side**	**Tube Side**	
33	Design/vac/test pressure:g		kPa	700 / /	800 / /	
34	Design temperature		C	35	45	
35	Number passes per shell			1	6	
36	Corrosion allowance		mm	3,18	3,18	
37	Connections	In	mm	1 12,7 / -	1 12,7 / -	
38	Size/rating	Out		1 12,7 / -	1 12,7 / -	
39	Nominal	Intermediate		/ -	/ -	

40	Tube No. 38	OD 19,05	Tks- Avg 2,11	mm	Length 1,2 m	Pitch23,81	mm
41	Tube type	Plain	Material	Carbon Steel	Tube pattern	30	
42	Shell	Carbon Steel	ID 250 OD 266	mm	Shell cover	Carbon Steel	
43	Channel or bonnet	Carbon Steel			Channel cover	Carbon Steel	
44	Tubesheet-stationary	Carbon Steel			Tubesheet-floating	Carbon Steel	

45	Floating head cover		Carbon Steel			Impingement protection		None	
46	Baffle-cross	Carbon Steel	Type	Single segmental	Cut(%d)	37,29 H	Spacing: c/c	160	mm
47	Baffle-long	-		Seal type			Inlet	255,21	mm
48	Supports-tube			U-bend			Type		
49	Bypass seal			Tube-tubesheet joint			Exp. 2 grv		
50	Expansion joint	-		Type					
51	RhoV2-Inlet nozzle	3	Bundle entrance		0		Bundle exit	0	kg/(m*s2)
52	Gaskets - Shell side	Flat Metal Jacket Fibe		Tube Side		Flat Metal Jacket Fibe			
53	Floating head	Flat Metal Jacket Fibe							
54	Code requirements	ASME Code Sec VIII Div 1				TEMA class	R - refinery service		
55	Weight/Shell	301,5	Filled with water		375,8		Bundle	77,8	kg
56	Remarks								
57									
58									