NTNU
Det skapende universitet

Preliminary experimental investigation on a multi-stage cryogenic heat pipe heat exchanger

Jakob Aljoscha Mann

Natural Gas Technology
Innlevert: desember 2014
Hovedveileder: Trygve Magne Eikevik, EPT

Norges teknisk-naturvitenskapelige universitet
Institutt for energi- og prosessteknikk

Abstract

Natural gas is a versatile energy resource. The demand for natural gas is constantly rising and new sources for natural gas are needed. Some of these sources can be very remote and conventional production methods cannot be applied. Small remote natural gas reservoirs might hold valuable resources but the construction of a pipeline from these reservoirs is not feasible. In these cases small scale liquefaction plants can help to transform the gas into a liquid and thus open up new ways of storing, transporting and using the resource. These small scale liquefaction plants have to consider the investment cost, efficiency and running costs of a project.

Heat pipes are closed systems that can achieve high heat transfer rates from one end of the heat pipe to the other. These properties can be used in a heat exchanger to create a compact heat exchanger design. Such heat exchangers have already been used for example in waste heat recovery from factory exhaust. In this work the application of a heat pipe heat exchanger for the use in a natural gas liquefaction heat exchanger is investigated.

An overview over the natural gas market in China is given to highlight the roll that liquefied natural gas will play in China's future. Conventional natural gas liquefaction methods are explained and the equipment commonly used in these methods is listed. An explanation of the heat pipe is given and the limitations during the operation of such a system are explained.

Finally the experimental setup that was constructed is explained and the experimental procedures are highlighted. The results of the experiments with the heat pipes are presented with a conclusion. In addition some thoughts on further work in this field of research are presented.

Contents

Table of Figures

Bibliography

Amode, J. (1936). *Analysis of a heat pipe heat exchanger*.

Bahadori, A. (2014). *Natural Gas Processing.*

Charlstons Solicitors. (2013). *Natural Ressources.*

Chen, Y. (2013). *Development Strategies of the Chinese Natural Gas Market.*

Fahghri, A. (1995). *Heat Pipe Science and Technology.*

Higashi, N. (2009). *Natural Gas in China: Market Evaluation and Strategy.*

Hong, Z. (2013). *The China-Myanmar Energy Pipelines: Risks and Benefits.*

Hydrocarbon Asia. (2012). *LNG Terminals in China and Related Developments.*

ILEX Energy Consulting. (2003). *Importing Gas into the UK - Gas Quality Issues.*

International Energy Agency. (2012). *Oil and Gas Security, People's Republic of China.*

Kohler, T., & Bruentrup, M. (2014). *Choose the best refrigeration technology for small-scale LNG production.*

Lee, Y., & Bedrossian, A. (1978). *The characteristics of heat exchangers using heat pipes or thermosyphons.*

Mokhatab, S. (2014). *Handbook of Liquefied Natural Gas.*

Noie, S. (2005). *Heat transfer characteristics of a two-phase closed thermosyphon.*

PetroMin. (2013). *China's Natural Gas Market.*

Platinum Broking. (2013). *China Natural Gas Industry Guide.*

Tan, J., & Liu, C. (1990). *Predicting te preformance of a heat-pipe heat exchanger, using the effectiveness-NTU method.*

U.S. Energy Information Administration. (2014). *China.*

Vasiliev, L. (2005). *Heat pipes in modern heat exchangers.*

W., L. S. (2013). *The Geopolitics of Natural Gas: Natural Gas in the People's Republic of China*.

1 Introduction

1.1 Natural Gas

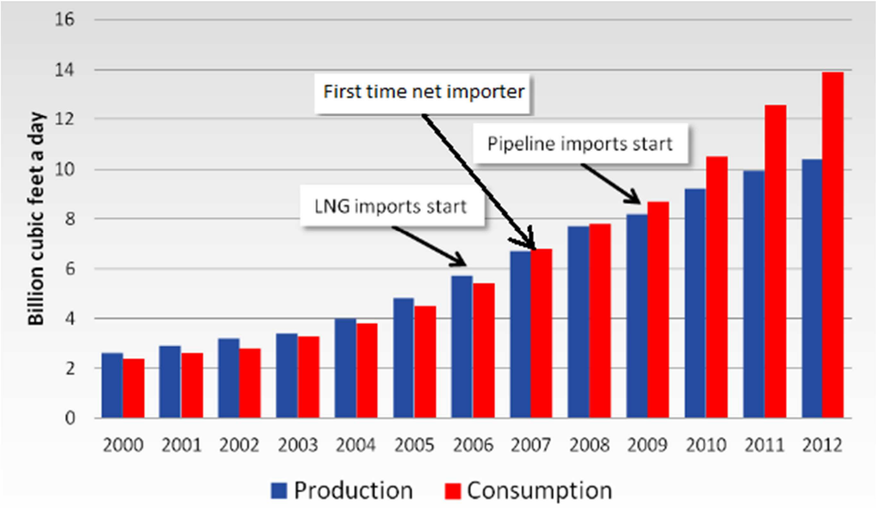
Natural gas is a fossil fuel which is primarily composed of methane. Depending on its origin it can contain a number of other components such as Ethan, Propane and higher hydrocarbons, water, carbon dioxide, hydrogen sulphide, helium and nitrogen. Conventional natural gas resources can be found in underground deposits as associated gas when found in the same deposit as crude oil or as non-associated gas when found in a well producing only gas. After the gas is produced from the well it is treated to remove unwanted components such as heavier hydrocarbons, water and hydrogen sulphide. Other sources for natural gas include tight gas, shale gas, coal bed methane and methane hydrates. These resources are called unconventional gas resources. Unconventional natural gas resources are in different stages of development. Much of the future growth of the natural gas sector is expected to come from unconventional natural gas resources.

The typical mode of transportation for natural gas is pipelines. These pipelines are expansive to construct. Especially to transport natural gas across long distances is an expansive endeavour. This has led to the natural gas market to be limited by geographical restrictions. Other than oil which can be transported across oceans and has an established global market, natural gas has established local markets in the Americas, Europe and Asia (with a link to the Middle East) with limited interaction between the markets.

High price differences between these regions and natural gas resources located far from potential markets have led to the development of liquefied natural gas as an alternative mode of transportation. When natural gas is cooled to -159°C to -162°C, depending on exact composition of the gas, it condenses to a liquid. This liquid takes up only 1/600 of the volume of the same amount of natural gas in gaseous form. This makes it economically feasible to transport the liquefied natural gas in special ships. This natural gas from a remote source or bought in a market with a low price can later be regasified and distributed through an existing pipeline network.

1.2 Natural gas in China

Natural gas has been advocated by the Chinese government for a number of years as part of the solution to the countries environmental problems. In the past the growth of the natural gas industry in China was hindered by a lack of infrastructure and investments. Some milestones in the past to overcome these problems were the completion of the first West-East pipeline in 2004, the country’s first natural gas import (in the form of LNG) in 2006. In the following year China became a net importer of natural gas.



Sources: U.S. Energy Information Administration, BP Statistical Review of World Energy

Figure 1-1 Recent development in Chinas natural gas production and consumption (U.S. Energy Information Administration, 2014)

In 2007 China joined the top 10 most gas consuming nations in the world. Starting in 2009, China imported natural gas by pipeline from Turkmenistan in addition to the increasing LNG imports. Thanks to these early efforts to expand the usage of natural gas, the consumption has seen a rapid increase in more recent years. In 2010 China consumed as much gas as Germany one of Europe’s biggest natural gas users.

In total numbers China has reached a natural gas consumption of 147 bcm in 2012 which is fivefold the 2010 level (Chen, 2013). Other energy sectors however where not stagnant either and their total numbers still exceed that of gas many times. Therefore the percentage of natural gas in Chinas total energy consumption still remains low. It is only because of the efforts of the Chinese government to promote natural gas that this comparably expensive resource has not been replaced with cheaper coal or electricity from coal in more cases. In this way the natural gas share in the total energy consumption saw a slow but steady growth in the past.

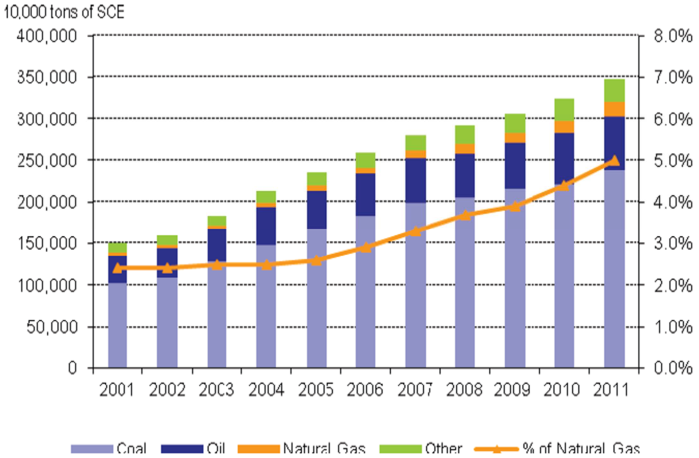


Figure 1-2 Primary energy consumption (in SCE) and natural gas share (in %) (Platinum Broking, 2013)

Despite equally impressive growth in the natural gas production in that time span, China is no longer able to sustain its own consumption. They began importing natural gas in 2006 in the country’s first LNG receiving terminal in Guangdong. They became a net importer of natural gas in 2007 (International Energy Agency, 2012). Domestic production in 2012 was 108 bcm amounting for 73% of Chinas natural gas needs. The Chinese government wants to prevent an increasing dependency on foreign natural gas resources and strongly advocates domestic production. Through their 10th and 11th five year plans production in 2012 has already reached four times the level of 2000. Chinas natural gas industry is still fairly new in comparison with other country’s which has often limited the possibilities for growth in the past. This means that in the future growth rates could increase even further. To support this ambition, in 2011 the Chinese government released the 12th Five Year Plan which includes many new policies to help the growth of natural gas.

Chinas natural gas production is centered in three major basins: the Tarim basin, the Ordos basin and the Sichuan basin. These basins contain more than half of Chinas proven natural gas resources (Higashi, 2009). All three of them are located inland away from the countries east coast where the urban centers are located that consume most of the country’s energy. In many cases missing infrastructure was a limiting factor for production and natural gas was only extracted for local use. This however was only seldom possible where local residents had the initiative and financial means to make the transition and to buy an expensive energy source such as natural gas.

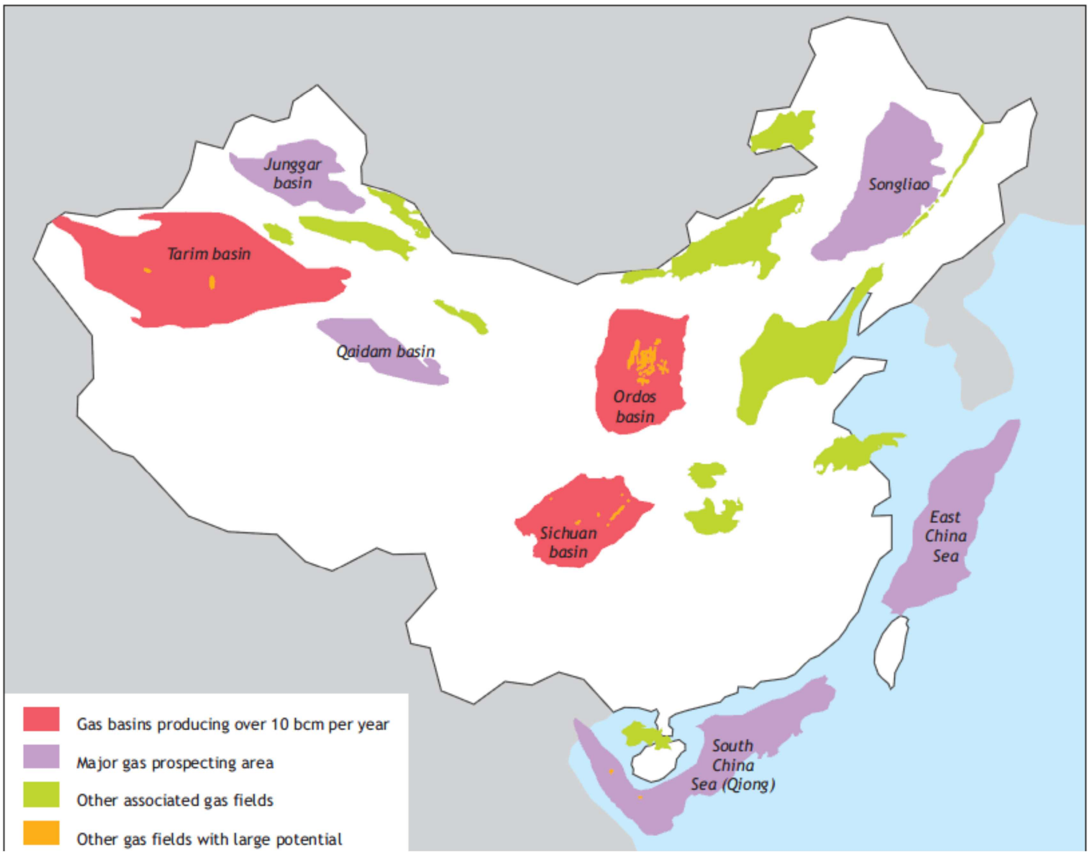


Figure 1-3 Established and suspected natural gas deposits in China (Higashi, 2009)

In the upstream market 3 national oil companies (NOC’s) control the market: CNOOC, Sinopec and CNPC. These three companies were founded during China’s economic reforms in the 1980s. Each of the three companies was established with a specific goal in mind. CNOOC was supposed to enable cooperation with foreign partners in offshore development in 1982, Sinopec was found as a downstream company in 1983 and CNPC was responsible for the countries onshore oil and gas business starting 1988. Today all 3 of the companies have investments along the value chain in both the oil and gas business and have outgrown their original purposes.

Table 1-1 Key facts on China's national oil companies (Higashi, 2009)

	CNPC/Petro China	**Sinopec**	**CNOOC**
Government share	90%	77.42%	70.6%
Total assets (USD million)	139 848	96 658	23 657
Income (USD million)	110 154	158 938	11 892
Net profit (USD million)	20 477	7 458	4 079
Net profit/income ratio	18.6%	4.7%	34.3%
Oil production (1 000 b/d)	2 298	96 658	23 657
Gas production (mcf/d)	4 458	774	560

With most of Chinas natural gas resources located inland it is not surprising that CNPC, the first company to hold responsibility for these locations, is still the most dominant player in Chinas natural gas industry. The company holds the 75% of the domestic gas resources and 80% of the infrastructure. The other 2 NOCs however are looking for opportunities to gain a stronger foothold in this field. CNOOC is trying to do so with numerous LNG projects. This focus has made them the market leader in Chinas emerging LNG market. China United Coalbed Methane Corporation CUCBM was found in 1996 as a fourth player with focus on Coalbed methane resources. Process was slow however and the company does not possess the importance of its 3 predecessors. In 2007 the CBM market was opened to foreign investors to speed up the process.

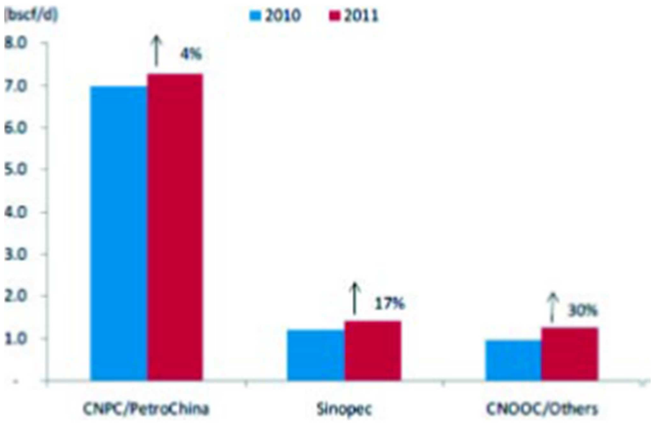


Figure 1-4 Growth in production of the different NOCs (PetroMin, 2013)

Smaller companies exist between these major players but their activities are mostly limited to small regions and to single steps in the value chain.

Natural gas production in China consists largely of onshore gas production. Offshore gas production does exist and more resources for both oil and gas would be in within technological and financial reach in front of the coasts. Despite the close proximity to the urban coastal area where these energy products would be desperately needed the exploration in these areas is unlikely in the near future. Many of the resources lay in areas over which China and other countries both claim ownership. Individual cases include disputes over islands or territories between China and Japan, the Republic of Taiwan, Vietnam, Malaysia, Brunei and the Philippines respectively (W., 2013).

In 2010 total domestic gas production in China was 94.8 bcm of which 9 bcm came from coal-bed methane. [2] Other unconventional natural gas resources form an even smaller part of domestic production at the moment. At the same time the Chinese government had the goal to increase conventional production to 138.5 bcm annually until 2015.

Regionally the Sichuan Basin is the most important production area followed by the Ordos and Tarim Basin. Their 2007 production levels ranked at 17 bcm in the Sichuan Basin, 15.5 cm in the Ordos Basin and 12 bcm in the Tarim Basin (Higashi, 2009).

China has two standing import agreements. The first is the natural gas supply to the west east pipeline project which is imported from Turkmenistan. The second and more recently commenced project is the import of natural gas from Myanmar through the Myanmar pipeline. Current negotiations between Russia and China are dealing with 2 possible import routes for a Sino Russian import project.

The first interregional pipeline project in China was the first west-east pipeline. The need for a west east pipeline existed independently of Chinas plans to import natural gas and this pipeline did not cross Chinese borders. The goal of this project was to transport the gas reserves of the Tarim-Basin in Xinjiang province to Shanghai. In 2004 it was completed with an annual capacity of 12 bcm. This project was later expanded to include a second phase with the goal of delivering natural gas from Turkmenistan to the city of Guangzhou on Chinas lower east coast. A 30 year agreement for 30 bcm annually was made in 2009. After 3 years of construction this second phase of the project began operation in 2012. A third phase of the project aims to increase the capacity for imported gas from Central Asian countries to China. When completed, it will increase the capacity by another 30 bcm annually in bring the combined capacity of the west-east pipeline project up to 72 bcm annually of both imported and domestically produced gas (Chen, 2013).

Source: China National Petroleum Corp

Figure 1-5 China's west to east pipeline projects (Chen, 2013)

Another import pipeline began operation only recently. The China-Myanmar pipeline project includes a crude oil and a natural gas pipeline. China has been strongly pushing for the completion of this project despite civil unrests and changes in government in Myanmar during the years of construction. Aside from financial interests and resources, China’s motivation for this project was also geopolitical. With this pipeline China has created a way for crude oil and gas to enter their country without passing through Strait of Malacca. This path is open not only for resources of Myanmar. While rich in natural resources the country is small and its contribution to Chinas total energy needs in absolute numbers is modest. In the future however this pipeline could also be supplied with oil from Africa or the Middle East.

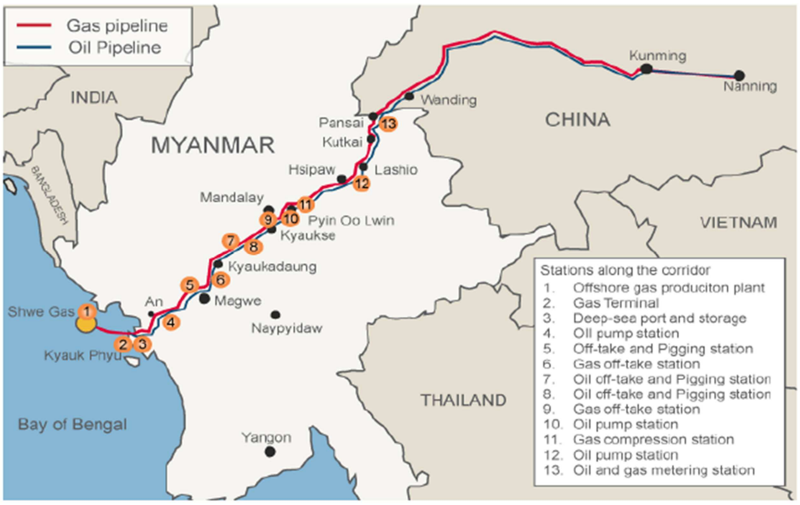


Figure 1-6 Sino Myanmar pipeline project (Chen, 2013)

On the 23 October 2013 the gas pipeline has commenced operation. It is now planned to deliver 12 bcm annually to China (Charlstons Solicitors, 2013). This gas will be used in the Chinese city Kunming. Plans to expand the capacity of the natural gas pipeline already exist. China plans to increase it to 24 bcm by 2019 which would make it Myanmar’s largest natural gas customer (Hong, 2013).

Lastly China and Russia have been in discussions over natural gas import pipelines for many years. China and Russia share borders east and west of Mongolia which is enveloped between the two countries. There are two possible options for a gas import pipeline: One directly from

Russia to China over the eastern border and the second west of Mongolia not through the countries narrow shared border at that place but through already existing pipeline infrastructure through Kazakhstan (Chen, 2013).

Both projects were hold back many times by disagreements about the pricing of the natural gas. This was the initial reason for China to seek natural gas imports elsewhere and finally signing a deal with Turkmenistan in the past (Higashi, 2009). However pricing reforms towards a more liberal natural gas market in China are expected in the future. In 2011 Guangdong and Guangxi began to set their gas prices based on imported fuel oil and LPG prices. It is unknown how fast these reforms will be allowed to develop and if they will be extended to other regions (International Energy Agency, 2012). It is however a first step towards the liberation of the gas market in China and every step in this direction will also mean a step towards higher gas prices and easier conditions for Russian gas import projects.

In 2013 the Chinese President Xi visited Russia. He chose the country as his first foreign destinations to visit in is presidency. During his visit a Memorandum of Understanding was signed between China’s CNPC and Russia’s Gazprom concerning a 30 year and 38 bcm annual pipeline project through the eastern border of the two countries (Chen, 2013). This shows that China’s interest in the realization of gas imports from Russia remains high and a deal in one or maybe both of the possible locations will most likely be realized in the near future (with the eastern project being the more promising candidate at this time).

China started importing natural gas as LNG in 2006 when the countries first LNG receiving terminal in Guangdong Dapeng began operation. This option of importing natural gas is especially appealing to China because most of its natural gas consumption centers are located along the coast. Despite this late emergence on the international LNG market China has become the third largest LNG importer worldwide by absolute volume in 2012. The country imported a total of 19.5 bcm of LNG in that year. With this import volume, China has surpassed UK and Spain and only the two countries with long LNG import history Japan and Korea still import a higher volume than China (Chen, 2013).

Figure 1-7 LNG terminals on China's coast (Hydrocarbon Asia, 2012)

After the first project started operation in 2006 in Guangdong, a second project was built in Fujian and a third one was built in Shanghai. The terminal in Fujian received its first shipment in 2009 while the start of operation in the Shanghai terminal was planned for the same year but delayed to 2010. All three of these projects were initiated by CNOOC. This National Oil Company has secured the market leaders position in the Chinese LNG trade but its 2 competitors have launched LNG projects as well. CNPC build two receiving terminals in Dalin and Jiangsu respectively which started operation in 2011 (Chen, 2013).

In the race to secure market shares in the emerging LNG trade all three NOC’s have many projects planned for the future. Even if import volume will rise as expected, the available receiving capacity is still likely to exceed actually imported volume in the future. This will however give china a unique position in the spot market. Should the price for LNG drop unexpectedly the country could easily increase their LNG import volume faster than any other buyer (Hydrocarbon Asia, 2012).

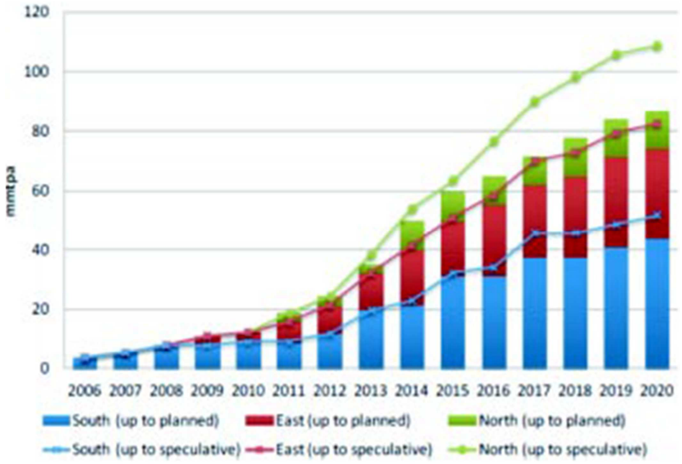


Figure 1-8 Future development of LNG import capacity (Hydrocarbon Asia, 2012)

The supply structure to the Chinese LNG market is very diverse. Many terminals receive shipments from more than one contractor and often from more than one country. Qatar and Australia are the biggest suppliers. In 2012 Qatar has surpassed Australia which was the biggest supplier of LNG to China until then.

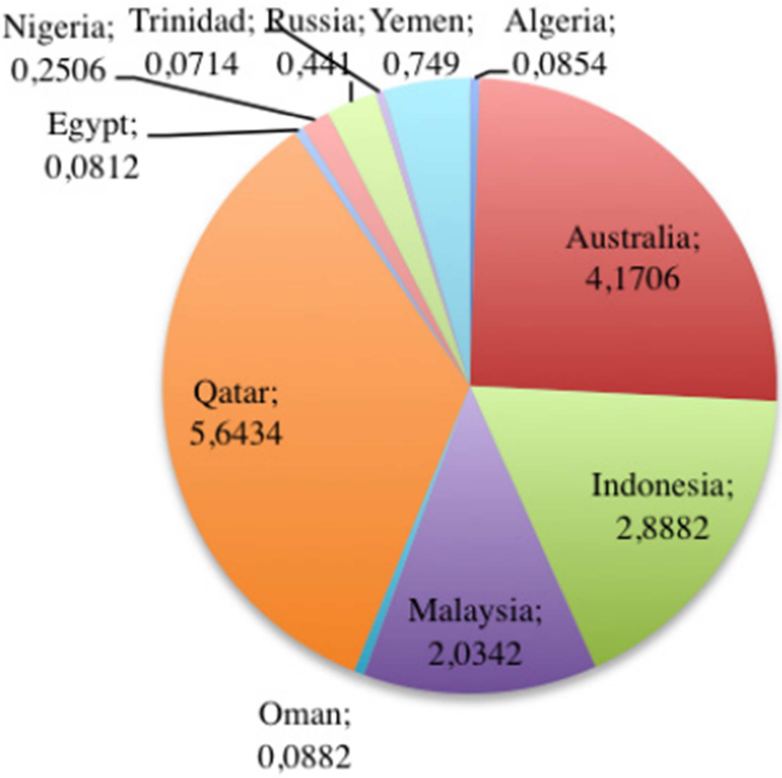


Figure 1-9 Natural Gas imports (in bcm) January to October 2012 (volume per exporting country) (Chen, 2013)

1.3 Liquefied Natural Gas

When natural gas is cooled to between -159°C and -162°C it condenses and becomes a liquid. In this form it takes up only 1/600th of the volume of gaseous natural gas. Depending on the composition of the feed gas and the exact pre-treatment and liquefaction process the composition if liquefied natural gas can vary between different liquefied natural gas plants.

Table 1-2 Typical Composition of LNG from Various Liquefaction Plants (ILEX Energy Consulting, 2003)

Component, mole %	Nigeria LNG	Arun LNG	Brunei LNG	Oman LNG	Atlantic LNG	Kenai LNG
Methane	87.9	88.48	89.4	90	95	99.8
Ethane	5.5	8.36	6.3	6.35	4.6	0.1
Propane	4	1.56	2.8	0.15	0.38	0
Butane	2.5	1.56	1.3	2.5	0	0
Nitrogen	0.1	0.04	0.2	1	0.02	0.1

Liquefied natural gas can be transported by ship. Energy is required for the liquefaction of the natural gas but the transportation of natural gas in liquid form becomes economical if the distance between source and consumption becomes too long for a pipeline to be feasible. When transported more than 1500 km by sea or 5000 km over land the transportation of natural gas is as liquefied natural gas is economic (Bahadori, 2014). Natural gas resources that were believed too remote and too far away from a potential market can become monetized if liquefied natural gas is considered as a mode of transportation.

When natural gas is processed from the well in normally undergoes pre-treatment before it is distributed by pipeline. During these pre-treatment steps a number of unwanted components is removed from the natural gas. Water is removed from the gas to prevent the formation of hydrates that can cause severe blockages in pipelines. Higher carbon hydrates are removed to regulate the heating value of the natural gas and to sell the higher hydrocarbons as separate products. Acidic gases are removed to prevent corrosion in the treatment equipment and transportation pipelines.

In much the same way natural gas has to be pre-treated before it can be liquefied. Water and higher hydrocarbons form solids at the low temperatures encountered during the liquefaction process and would cause blockages in the equipment. Acidic gases could lead to corrosion in the equipment. In contrast to normal natural gas pre-treatment nitrogen does not have to be removed separately. It condenses at -196°C, a lower temperature than methane, and thus is still present as a gas after the methane has been liquefied. It can then be removed relatively simple in an end flash.

After the liquefaction natural gas also becomes an alternative to other fuels because of the reduced volume. Its storage density is higher than that of other diesel fuel alternatives such as compressed natural gas. In addition the NO_X emissions of LNG are significantly lower than those of diesel fuel (Bahadori, 2014).

2 Liquefaction processes

After the natural gas has been pre-treated to remove water and acidic gasses it can be liquefied. For this phase change it has to be cooled to approximately -160°C depending on the exact composition of Methane, Ethane and other components. This low temperature is usually reached in 3 steps: precooling, liquefaction and sub cooling.

2.1 Cascade liquefaction process

In the cascade process several different refrigerants are used in separate cycles. Each refrigerant evaporates at an individual constant temperature. More equipment is needed to manage multiple refrigerant cycles. However with the use of individual refrigerants for each of the liquefaction processes steps the heat exchange area can be kept small and the process remains economical for large scale operations.

2.2 Mixed refrigerant process

In contrast the mixed refrigerant process utilizes only one refrigerant. This refrigerant is usually a mixture of different hydrocarbons and nitrogen as they are readily available at liquefaction plant sides. The purpose of the mixed refrigerant is to match the cooling curve of the natural gas as closely as possible. Since only one cooling cycle with the mixed refrigerant is used, this process uses a lower number of compressors and heat exchangers than the cascade process and thus has lower investment cost. The thermal efficiency achieved with this process however is lower than that of a cascade process because a single mixed refrigerant cannot match the cooling curve of the natural gas as closely as 3 separate cooling cycles. Also the start-up and line-up of this process takes longer since time is needed to precisely adjust the refrigerant bland every time (Mokhatab, 2014).

2.3 Expander based process

Expansion based liquefaction cycles use an expander instead of a valve to provide a refrigerant to cool the natural gas. The refrigerant in this process is in its gaseous state during the entire cycle. This leads to a simpler process with a wider temperature gap between the natural gas and the refrigerant especially in the beginning of the cooling process with the warm natural gas. The overall efficiency that is achieved with this method is low. This disadvantage however is offset in some scenarios by the simplicity of the process when stability is a major concern or different liquefaction conditions are expected.

2.4 APCI C3-MR process

In reality many liquefaction plants use a combination of the 3 processes to combine their advantages or offset their disadvantages. The most used process in base load liquefied natural gas production and contributing for 75% of the total natural gas market is the C3-MR process. This combination of a precooling with propane and liquefaction with a mixed refrigerant can be seen as a combination of a 2 step cascade and a mixed refrigerant process. The process was developed by Air Products and Chemicals Inc. and has dominated the LNG technology since the 1970 with single train productions as large as 5 MTPA (Mokhatab, 2014).

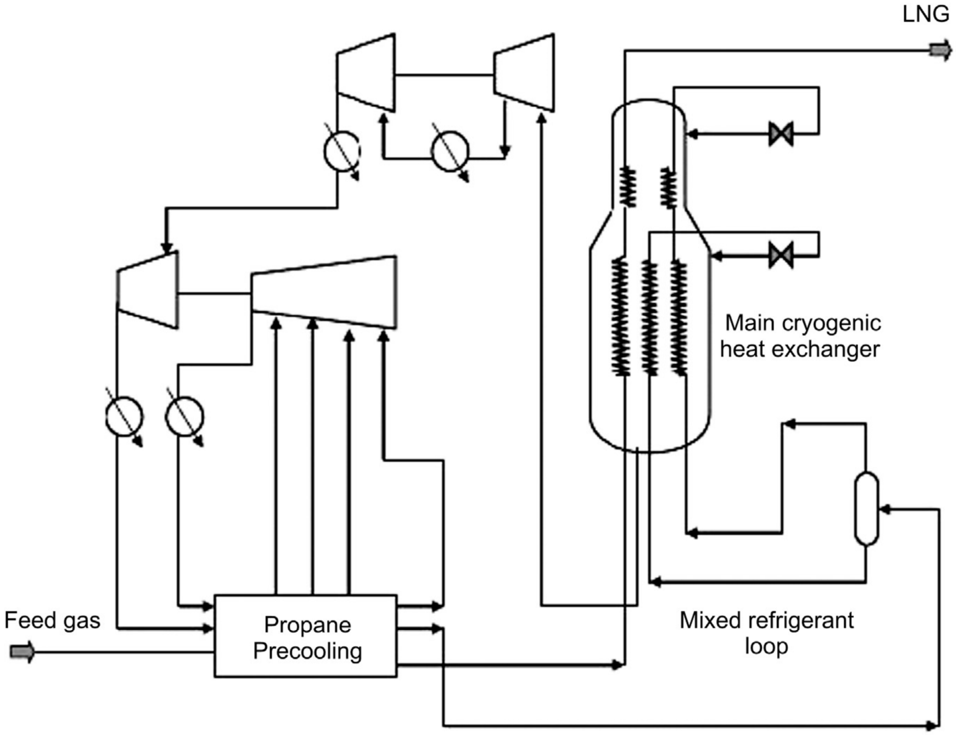


Figure 2-1 C3-MR process schematic (Mokhatab, 2014)

During the precooling with propane a temperature of approximately -35°C is reached. The precooling stage is simultaneously used to partly condense the mixed refrigerant used in the later step. The gas is then liquefied in a spiral wound heat exchanger by the mixed refrigerant composed of nitrogen, methane, ethane and butane. To provide the final cooling for the natural gas liquefaction the mixed refrigerant is being flashed across Joule-Thompson valves in the upper part of the spiral wound heat exchanger. The design has been modified in the past in order to achieve larger plant sizes. For example the addition of a third cooling cycle to achieve the final liquefaction and sub cooling with nitrogen as a refrigerant has been used to make productions of 10 MTPA possible without the development of larger spiral wound heat exchangers.

2.5 Phillips optimized cascade

The Phillips optimized cascade process uses 3 refrigerants: propane, ethylene and methane. Philips Petroleum Company developed the process in the 1960 and it was first used in 1969 in ConocoPhillips’ Kenai LNG plant in Alaska (Mokhatab, 2014).

The process uses a separate cooling cycle for each refrigerant and multiple steps of methane refrigeration are possible. Air or cooling water are used to condense propane and remove compression heat and the propane is used to condense ethylene and remove heat from the natural gas. After several steps of cooling the heavier hydrocarbons are removed from the feed gas stream and the remaining methane rich stream is further cooled with methane as a refrigerant.

Plate and fin heat exchangers or core in kettle heat exchangers can be used in this process. These are less complicated in design than the spiral wound heat exchangers used in some other processes and can be supplied by several manufactures.

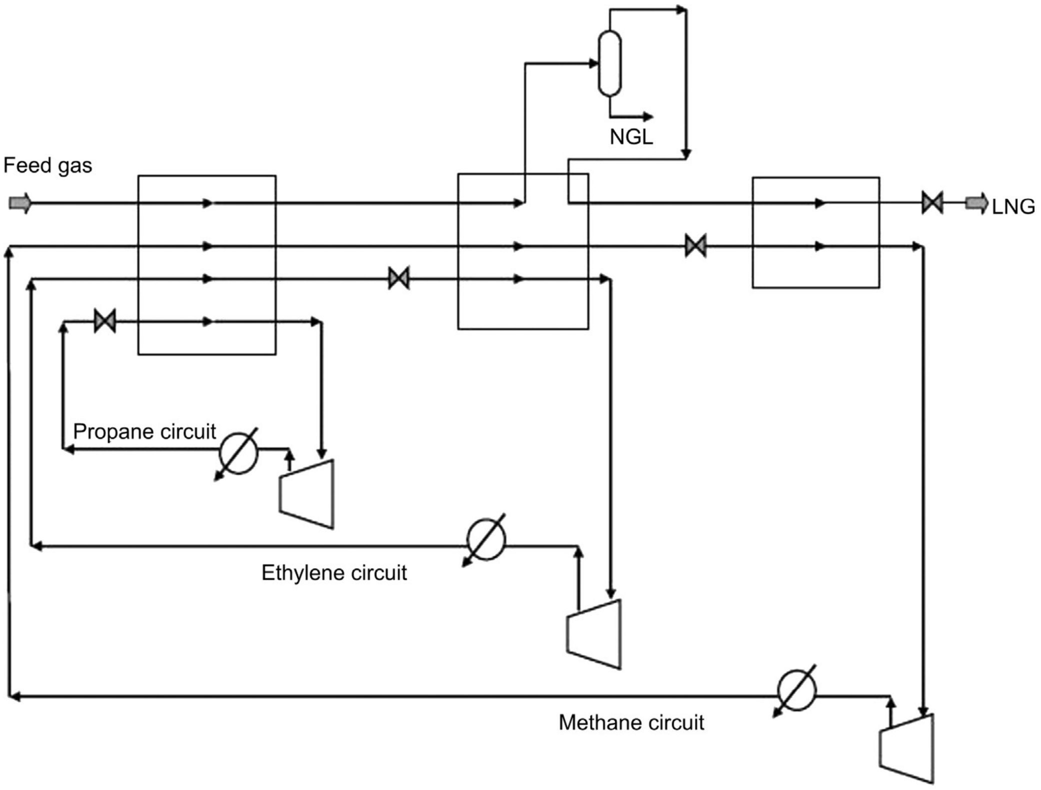


Figure 2-2 Phillips optimized cascade process schematics (Mokhatab, 2014)

2.6 Black and Veatch Pritchard Prico process

The Prico process was developed by Black and Veatch Pritchard and is a simple mixed refrigerant process. The mixed refrigerant is made up of nitrogen, methane, ethane, propane and iso-pentane to match the cooling curve of the natural gas. This process has a low equipment course but the efficiencies achieved are not as high as in multi cycle processes.

The mixed refrigerant goes through a compression expansion cycle. After the compression however the refrigerant stream is cooled in two steps. It is first partly condensed with a suitable air or water stream and then led through the cold box where it completely condenses. Afterwards it is flashed across an expansion valve to further drop the temperature so that it can be used in the cold box to cool down the natural gas. The feed gas stream in this process is also cooled down in to steps. After being first cooled down to about 35°C most of the heavy hydrocarbons in the feed gas stream are condensed and can be removed. The remaining gas stream is then lead back into the cold box to cool it down further and condense it to liquefied natural gas. This also avoids the freeze out problem in the later stages of the liquefaction problem that can occur if heavier hydrocarbons are still present at the very low temperatures that are reached near the end of the process. If heavier hydrocarbons were still present in the heat exchanger at this point they could form solids and lead to a blockage in the system.

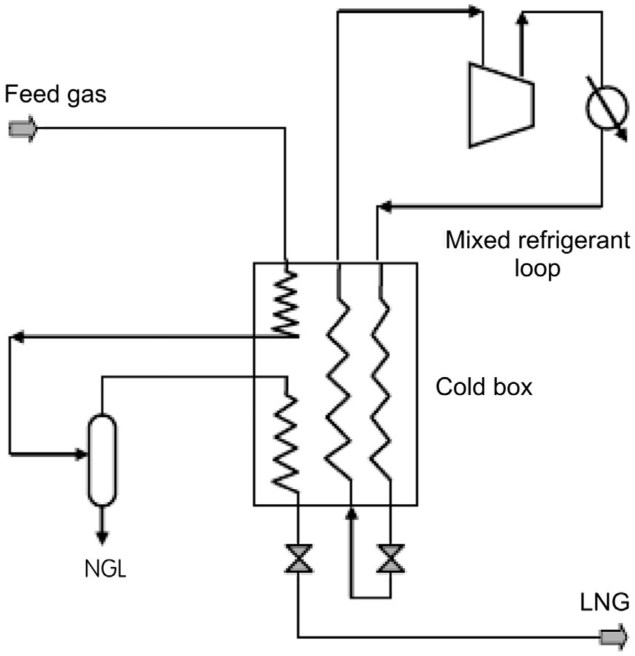


Figure 2-3 Prico process schematic (Mokhatab, 2014)

2.7 Statoil/Linde mixed fluid cascade process

This process design was develop especially for the use in base load liquefied natural gas plants in harsher environments such as the Snøhvit plant in Norway where it is in use today.

It combines mixed refrigerants with a cascade design in an effort to improve the thermodynamic efficiency and create a flexible plant. Plate and fin heat exchangers are used in the precooling step while the liquefaction and sub cooling are achieved in spiral wound heat exchangers. The refrigerants are made up of methane, ethane, propane and nitrogen. The only plant build with this design so far has a capacity of 4.3 MTPA.

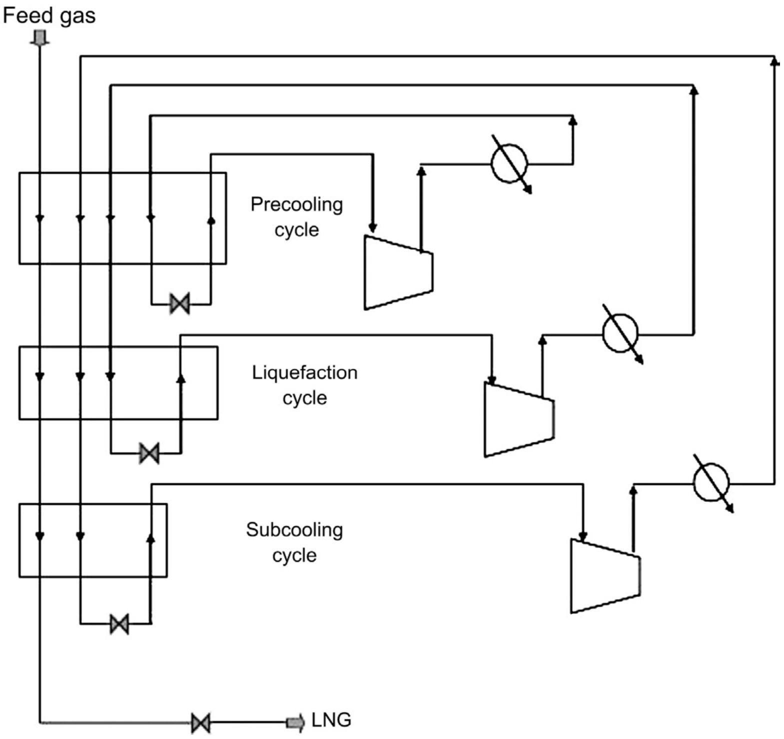


Figure 2-4 Statoil/Linde mixed fluid cascade process schematic (Mokhatab, 2014)

2.8 IFP/Axens Liquefin process

The IFP and Axens Liquefin process uses two mixed refrigerants in a process that was design to achieve high capacities with simple equipment. First the feed gas is precooled to -50°C to -80°C in plate and fin heat exchangers with the first mixed refrigerant. In this step the refrigerant is used at 3 different pressure levels. After this step the heavier hydrocarbons can be removed from the precooled gas. The second mixed refrigerant is completely condensed in the precooling and cryogenic section of the process. Afterwards it is expanded and then sent back through the cryogenic section to liquefy and sub cool the natural gas.

In this process a large part of the condensation duty is shifted to the precooling stage allowing a single refrigerant composition and simpler heat exchanger design. Even though the process is claimed to have lower cost due to simple heat exchanger design and high possible capacities no base load plants using this design have been built yet (Mokhatab, 2014).

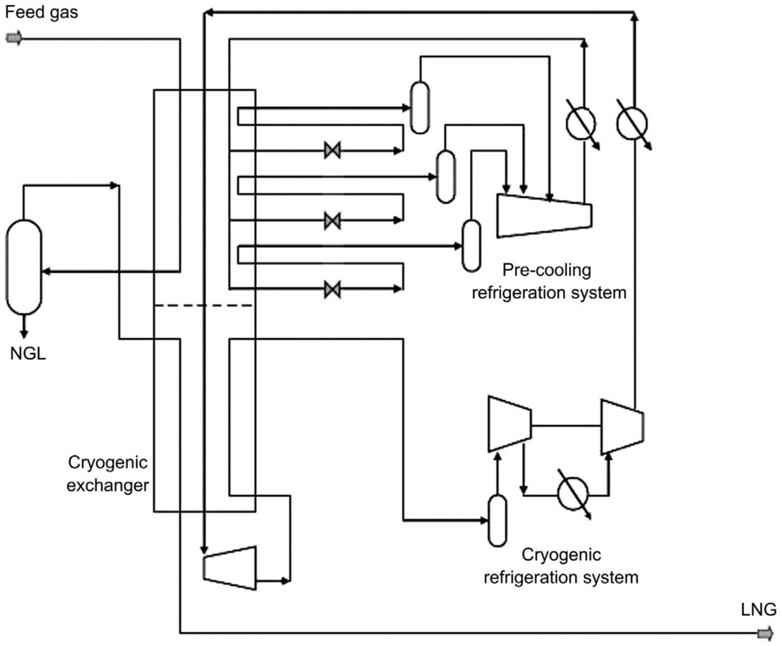


Figure 2-5 IFP/Axens Liquefin process schematics (Mokhatab, 2014)

2.9 Shell dual mixed refrigerant process

The dual mixed refrigerant process was developed by Shell to utilize 2 separate mixed refrigerant cycles to liquefy natural gas. In the precooling cycle the gas is cooled to -50°C before it is liquefied by the second mixed refrigerant cycle. The technology was first used in a base load plant in Russia with a capacity of 4.8 MTPA in 2009.

The process differs from other processes described in that it uses spiral wound heat exchanger in the precooling step where many other processes use plate and fin heat exchangers or core in kettle heat exchangers. Also instead of using a pure component for the precooling like propane in the C3MR process, this process uses a mixed refrigerant mainly composed of propane end ethylene for the precooling. The second mixed refrigerant used for the cooling of the gas in this process is composed of nitrogen, methane, ethane and propane. After the mixed refrigerant has cooled the natural gas in the cryogenic heat exchanger it leaves the heat exchanger as vapour. It is compressed in an axial compressor and further compressed in a two-stage centrifugal compressor. Afterwards it is cooled by air and further cooled and partly condensed by the precooling mixed refrigerant cycle.

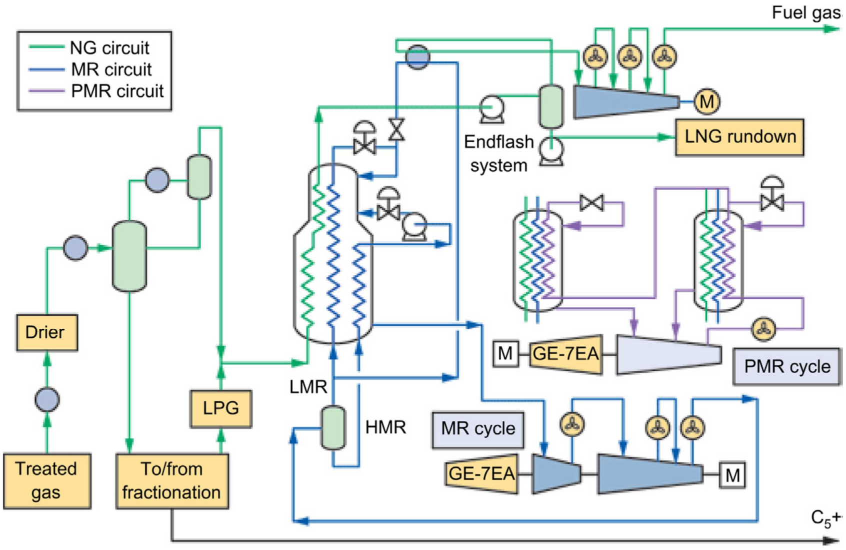


Figure 2-6 Shell dual mixed refrigerant process schematics (Mokhatab, 2014)

3 Heat exchangers

Different kind of heat exchangers can be used in the liquefaction process of natural gas. Natural gas is an energy resources and the cooling of the gas requires a large amount of energy. The liquefaction process can therefore be seen as a perceptual loss of the product. With a higher energy efficiency of the process less of the product is lost in the liquefaction process.

Plate and fin heat exchangers offer a high heat transfer area. The production process of these heat exchangers is simple and well established so that production costs are low. However the internal geometry of these heat exchangers is not optimal for liquefaction processes. The temperature distribution inside of a plate and fin heat exchanger is not perfect. This can lead to hot and cold spots inside of certain areas of the heat exchanger. The cold spots can be problematic in the application of natural gas liquefaction. If an area of the heat exchanger is cooled more than other areas of the heat exchanger it can lead to the natural gas in this area not just being liquefied but also freezing into a solid. If this happens an area of the heat exchanger is blocked and the liquefaction process cannot function in an optimal way. Typically plate and fin heat exchangers are used both in cascade and mixed refrigerant liquefaction plants.

Spiral wound heat exchangers are often used in high volume natural gas liquefaction plants. In this type of heat exchangers higher surface areas can be achieved. In this way the spiral wound heat exchanger can operate with a larger temperature gradient. They are however less flexible than plate and fin heat exchangers. Also since this type of heat exchanger is more difficult to construct and there are fewer suppliers. As a result, the costs for them are higher. Typically spiral wound heat exchangers are used in mixed refrigerant liquefaction plants.

Heat pipe heat exchangers use heat pipes with an intermediate working fluid to transfer the heat of the natural gas to the coolant. Their main advantage over other heat exchangers is their mean temperature distribution. Inside other heat exchangers hot or cold spots can lead to problems during the operation. In the application of liquefying natural gas a cold spot can cause components of the natural gas to freeze out and create a blockage in the heat exchanger. Heavier hydro carbons as well as water and CO_2 can cause these problems even in small amounts. In that case the flow inside of some parts of the heat exchanger can be completely blocked and this would lead to the loss of that area inside of the heat exchanger for heat transfer and a significant loss in the heat exchangers efficiency. In a heat pipe heat exchanger the temperature distribution does not produce any spots cold enough for a freeze out.

3.1 Efficiency

To methods can be used when evaluating a heat pipe heat exchangers overall efficiency, the lo-mean temperature difference method and the effectiveness-number of transfer units method (Fahghri, 1995).

A heat transfer balance of the heat exchanger gives:

$$Q = UA(T_h - T_c)$$

Where Q is the heat transferred, U is the overall heat transfer coefficient, A is the heat transfer area and T_h and T_c are the temperatures of the high and low temperature streams.

The overall heat transfer coefficient can be calculated by modelling the heat exchanger and its inner components as a number of thermal resistances. For a heat pipe heat exchanger with fined heat pipes with finned surfaces the equation looks as follows.

$$\frac{1}{UA} = \frac{1}{U_c A_c} = \frac{1}{U_h A_h} = \frac{1}{(\eta_0 hA)_c} + R_{f,c} + R_{hp} + R_{f,h} + \frac{1}{(\eta_0 hA)_h}$$

η_0 is the fin efficiency, h is the heat transfer coefficient, $R_{f,c}$ and $R_{f,h}$ thermal resistances in the cold and hot side of the heat exchanger due to fouling and R_{hp} is the thermal resistance of the heat pipe.

3.1.1 LMTD method

For the log mean temperature difference method an assumption of no heat losses has to be made. This means that all the heat removed from the hot temperature stream is transported to the cold temperature stream. This assumption is reasonable only for steady state applications.

In this case the heat balance of the heat exchanger can be written as

$$Q = U_t A_t \Delta T_{LM}$$

where U_t is the total heat transfer coefficient and A_t the total heat transfer area. ΔT_{LM} is the logarithmic mean temperature difference and can be calculated with the following formula

$$\Delta T_{LM} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)}$$

where $T_{h,in}$ and $T_{h,out}$ are the inlet and outlet temperatures of the hot stream and $T_{c,in}$ and $T_{c,out}$ are the inlet and outlet temperatures of the cold stream.

Because we assume that there are no heat losses the heat removed from the hot stream and the heat gained by the cold stream are assumed to be equal.

$$Q = Q_h = Q_c$$

with

$$Q_h = \dot{m}_h c_{p,h}(T_{h,in} - T_{h,out})$$

and

$$Q_c = \dot{m}_c c_{p,c}(T_{c,in} - T_{c,out})$$

where $\dot{m}_h$ and $\dot{m}_c$ are the mass flow rates of the hot and the cold stream and $c_{p,h}$ and $c_{p,c}$ are the specific heat capacities of the hot and cold flow.

Assuming that the inlet temperatures and mass flow rates are know this gives us a set of 3 equations with the 3 unknown Q, $T_{h,out}$ and $T_{c,out}$.

The most difficult part of this method is to determine the heat transfer coefficient of the heat pipe which is a combination of many components individual heat resistances.

3.1.2 Ɛ-NTU method

The effectiveness of a heat exchanger can be described as the rate of the actual heat transferred in the heat exchanger to the maximum possible heat transferred between the two streams in an infinite long heat exchanger.

This effectiveness can be calculated by

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,in})} = \frac{C_c(T_{c,out} - T_{c,in})}{C_{min}(T_{h,in} - T_{c,in})}$$

where C is the product of the mass flow rate and the specific heat capacity of a stream and C_{min} is the smaller of those values from the hot and the cold stream.

For a counter flow heat exchanger the effectiveness can be calculated with the exponential equation

$$\varepsilon = \frac{1 - exp\left[-\frac{U_t A_t}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \frac{C_{min}}{C_{max}} exp\left[-\frac{U_t A_t}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}$$

The number of transfer units is defined as

$$NTU = \frac{U_t A_t}{C_{min}}$$

3.1.3 Heat transfer coefficient

In a heat exchanger the overall heat transfer coefficient of a single heat transfer element is defined as

$$U_p = \frac{1}{R_p}$$

where R_p is the overall thermal resistance. R_p is defined as the sum of all individual resistances.

In the case of a heat pipe heat exchanger

$$R_p = R_{eva} + R_{f,eva} + R_{w,eva} + R_{wi,eva} + R_{i,eva} + R_v + R_{i,cond} + R_{wi,cond} + R_{w,cond} + R_{f,cond} + R_{cond}$$

where

R_{eva} and R_{cond} are the convective resistances on the outer surfaces of the evaporator and condenser end of the heat pipe,

$R_{f,eva}$ and $R_{f,cond}$ are fouling resistances on the outside surfaces of the evaporator and condenser end of the heat pipe,

$R_{w,eva}$ and $R_{w,cond}$ are the resistances of the heat pipe walls of the evaporator and condenser end of the heat pipe,

$R_{wi,eva}$ and $R_{wi,cond}$ are the resistances of the heat pipe wick inside of evaporator and condenser end of the heat pipe,

$R_{i,eva}$ and $R_{i,cond}$ are the resistances that occur due to the phase changes of the intermediate working fluid inside of the heat pipes evaporator and condenser section,

and R_v is the resistance that occurs due to the temperature drop in the vapour flow from hot to cold side inside of the heat pipe (Fahghri, 1995).

In a heat pipe heat exchanger for natural gas liquefaction we can assume that pre-treatment has taken place and fouling will not likely be a problem. Therefore $R_{f,eva}$ and $R_{f,cond}$ can be neglected. In a thermosyphon heat exchanger there is no additional heat transfer resistance introduced by a wick so that $R_{wi,eva}$ and $R_{wi,cond}$ can be neglected. Also we can assume if the heat pipe operates with an optimal working fluid and charge amount the internal resistances

$R_{i,eva}$, $Ri_{,cond}$ and R_v will only be a minimal contribution to the overall heat transfer coefficient. Finally the wall of the heat pipes are thin enough so that the thermal resistance they introduce can be neglected in comparison to the convective resistances. Therefore the only terms that have a significant contribution to the overall heat transfer resistance are

$$R_p = R_{eva} + R_{cond}$$

4 Heat pipes

A heat pipe is a closed system using an internal working fluid to transport heat from one end of the heat pipe to the other end. The part of the heat pipe that absorbs heat is called the evaporator section while the part of the heat pipe that gives of heat is called the condenser section. Between these two sections is a third one called the adiabatic or transport section. The working fluid inside of the heat pipe is evaporated in the evaporator section. The vapour then travels through the adiabatic section and transports the absorbed heat to the other end. In the condenser section the vapour condenses and gives of the latent heat to the environment outside of the heat pipe. Finally the working fluid is transported back to the evaporator section. The method of transport for the liquid from the condenser to the evaporator section can differ in different kinds of heat pipes.

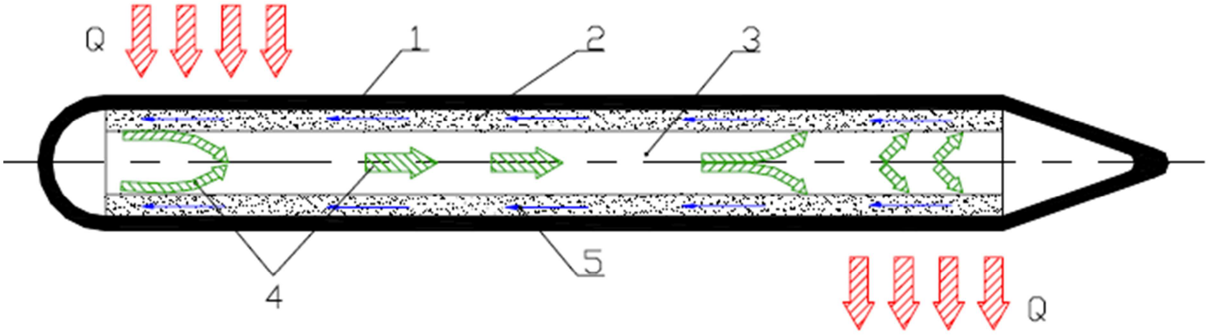


Figure 4-1 Heat pipe schematic with (1) heat pipe case, (2) wick, (3) vapour flow channel, (4) vapour and (5) liquid (Vasiliev, 2005)

Two phase closed thermo siphons simply consist of a closed cylinder containing the working fluid. In this type of heat pipe the evaporator section is always located below the condenser section. In this way the transport of the condensed working fluid in the top part of the heat pipe is initiated by gravity. This allows for a fast transportation of liquid working fluid from the condenser to the evaporator section which is one of the advantages of this heat pipe. The simple and cheap construction process is another advantage. The major disadvantage are constrains in the geometrical orientation. The evaporator section always needs to be located below the condenser section. (Fahghri, 1995)

Capillary driven heat pipes use a wick structure along the inner wall of the heat pipe to pump the liquid working fluid from evaporator to condenser section. An advantage of this method is that the transport of the liquid working fluid is independent of gravity and the heat pipes evaporator section can be at the same height or higher than the condenser section. A disadvantage is the limited speed at which the liquid working fluid is transported back. (Fahghri, 1995)

Other types of heat pipes exist. Axial groove heat pipes use grooves in the inner wall as a transportation mechanism instead of a wick structure. Flat plate heat pipes are a kind of capillary driven heat pipes used in the cooling of electronics. Rotating heat pipes use centrifugal force to return the working fluid to the evaporator. (Fahghri, 1995)

The working fluid inside the heat pipe must be chosen according to the operating temperatures of the heat pipe application. By adjusting the internal pressure of the heat pipe fluids can be used from the point where there saturation pressure is higher than 0.1 atm and lower than 20 atm. Higher internal pressures prove difficult do to the high necessary wall thickness which inhibits heat transport into and out of the heat pipe. Lower pressures can cause the vapour pressure limit to be encountered (see below). The working fluid must also be compatible with the container material. The parts of a heat pipe should not be capable of any chemical reaction with each other or form a galvanic cell together. Heat pipes are closed systems and do not have any moving parts. If all chosen components are capable with each other a heat pipe can be operated for a long time without any maintenance. Below is a list of working fluids with their useful temperature ranges. (Fahghri, 1995)

Table 4-1 - List of possible working fluids with temperature ranges Faghri 1995

Working Fluid	Melting Point in K at 1 atm	Boiling Point in K at 1 atm	Useful Range in K
Helium	1.0	4.21	2-4
Hydrogen	13.8	20.38	14-31
Neon	24.4	27.09	27-37
Nitrogen	63.1	77.35	70-103
Argon	83.9	87.29	84-116
Oxygen	54.7	90.18	73-119
Methane	90.6	111.4	91-150
Krypton	115.8	119.7	116-160
Ethane	89.9	184.6	150-240
Freon 22	113.1	232.2	193-297
Ammonia	195.5	239.9	213-373
Freon 21	138.1	282.0	233-360
Freon 11	162.1	296.8	233-393
Pentane	143.1	309.2	253-393
Freon 113	236.5	320.8	263-373
Acetone	180.0	329.4	273-393
Methanol	175.1	337.8	283-403
Flutec PP2	223.1	349.1	283-433
Ethanol	158.7	351.5	273-403
Heptane	182.5	371.5	273-423
Water	273.1	373.1	303-473
Toluene	178.1	383.7	323-473
Flutec PP9	203.1	433.1	273-498
Naphtalene	353.4	490	408-478
Dowtherm	285.1	527.0	423-668
Mercury	234.2	630.1	523-923
Sulphur	385.9	717.8	530-947
Cesium	301.6	943.0	723-1173
Rubidium	312.7	959.2	800-1275

Potassium	336.4	1032	773-1273
Sodium	371.0	1151	873-1473
Lithium	453.7	1615	1273-2073
Calcium	1112	1762	1400-2100
Lead	600.6	2013	1670-2220
Inidium	429.7	2353	2000-3000
Silver	1234	2485	2073-2573

During the design and operation of a heat pipe some limitations are important to keep in mind. The following is a list of operational limitations and their characteristics (Fahghri, 1995):

The capillary limit is encountered when the wick structure is not able to transport enough working fluid to the evaporator section. This state is called dry out and can be recognized by a sudden increase in the evaporator wall temperature. This is the most often encountered limitation in low temperature heat pipes that rely on a wick structure for working fluid transportation.

The sonic limit describes the limitation of the speed of the vaporized working fluid inside of the heat pipe. This limitation is most commonly encountered in heat pipes using liquid metal as a working fluid. The sonic limitation can be raised by increasing the evaporator temperature. When this limitation is encountered the axial temperature of the heat pipe will drop significantly.

The boiling limit is associated with the radial heat flow into the evaporator section of a heat pipe. This differs from other heat pipe limitations which usually limit the axial heat flux. However the maximum radial heat flux also limits the maximum axial heat transport. When too much heat enters the evaporator section it causes the working fluid to boil and the wall temperature to increase. This can lead to vaporized working fluid inside of the wick structure up to the point where the wick completely dries out. In wickless heat pipes the boiling limit can occur due to film boiling instead.

The entertainment limit describes forces between the liquid working fluid moving from condenser to evaporator section and the vaporized working fluid moving from evaporator to condenser section. Since these two flows move in opposite directions shear forces occur between them. This can cause droplets of liquid working fluid to become entrained into the vapour flow and be torn back to the condenser section from which it came. If this happens to a significant part of the liquid stream traveling from the condenser section it can cause the evaporator section to dry out. This limitation can be detected by sounds made by the entrained droplets hitting the condenser end.

The condenser limit is encountered when the heat cannot be removed from the condenser and of the heat pipe at a fast enough rate. The condenser end might also limit the heat transfer of which the heat pipe is capable if non condensable gases are present in the heat pipe and accumulate in the condenser end. This reduces the effective end of the condenser section and reduces the heat transfer area available.

The vapour flow inside the heat pipe might be limited by the vapour pressure limit at lower then operating temperatures if viscous forces become dominant in the vapour flow. This can reduce the heat transport through the heat pipe during start up or after operation from a frozen state.

5 Experimental apparatus

5.1 Vacuum hull

The vacuum hull is a stainless steel (SUS304) cylinder with a diameter of 500mm. It has a flat bottom and is closed on the top with a 580mm diameter steel lid. A 560 mm diameter rubber O-ring is used between the hull and the lid and 24 screws with a 12 mm diameter are used to fasten the lid. The vacuum hull can be evacuated by a molecular vacuum pump.

5.2 Hot and cold channel chamber

Two chambers have been constructed from stainless steel (SUS304). Each chamber measures 300 mm in height and 230 mm in width and depth. The hot chamber is located below the cold chamber. Both are separated by a 8 mm thick stainless steel (SUS304) plate.

5.3 Refrigerant and heating elements

Nitrogen is used as a refrigerant in all experiments. It is supplied from a mixed storage gas and liquid nitrogen tank. A gas stream is extracted from the top of the tank and a liquid stream from the bottom. Both are mixed to gain a nitrogen gas stream at the desired temperature for the experiment. Nitrogen is non-poisonous and not explosive. After it has passed through the heat exchanger it can be vented to the atmosphere. These were important considerations when choosing a refrigerant for the experiments.

Using natural gas in the hot chamber would be a considerable safety risk. Natural gas is primarily composed of methane and can be explosive when mixed with air. While running an experiment with the heat pipe heat exchanger the disposal or circulation of the natural gas stream would present a problem. Therefore heating elements were used on the evaporator end of the heat pipes instead to simulate the heat intake from a natural gas stream. 10 polyimide heating elements were used for this purpose with a power 24 watt each.

They are attached directly to the outer wall of the heat pipes evaporator ends. The power to the heating elements is supplied by a GWINSTEK GPS-2303C DC power supply.

5.4 Heat pipes

The heat pipes are constructed from closed copper cylinders with an outer diameter of 16mm and a wall thickness of 1.2 mm. There are a total of 30 heat pipes inside of the heat pipe heat exchanger. They are arranged in 6 rows of 5. The vertical displacement is 38.1 mm from the centre of one heat pipe to the next while the distance between 2 rows is 33 mm. 2 heat pipes of one row and one heat pipe of the next row always form a triangle with 3 equal sides of 38.1 mm with each centre of a heat pipe in one corner.

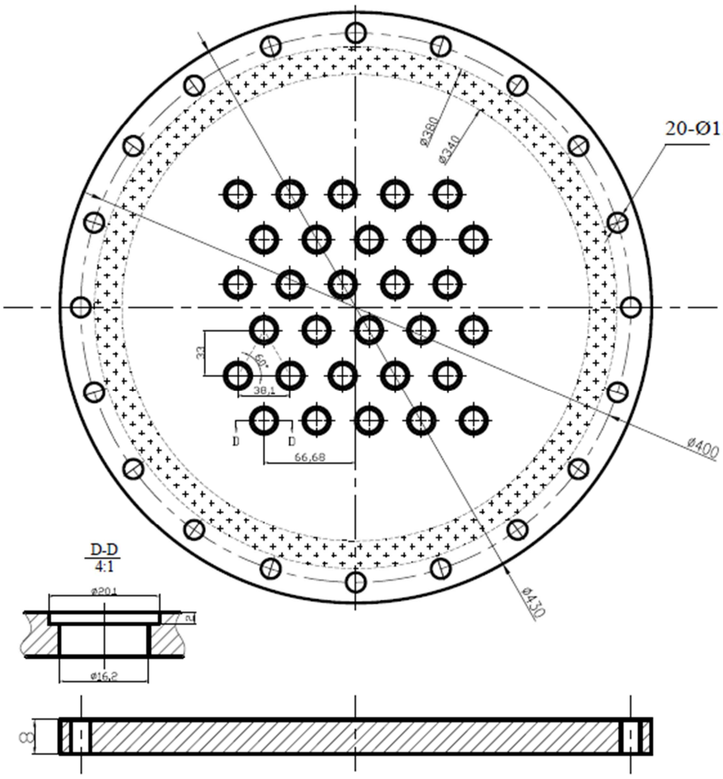


Figure 5-1 heat pipe arrangement as seen from the top down view (flow direction is bottom to top)

All heat pipes are connected to each other on the evaporator and on the condenser end. The working medium can be charged into all heat pipes simultaneously. During operation the working medium can flow from on heat pipe to another.

Copper plates are installed vertically in the hot and cold chamber to act as fins on the heat pipes outer surfaces. The plates are 0.3 mm thick and placed with a distance of 2,2 mm between and cover the entire chamber.

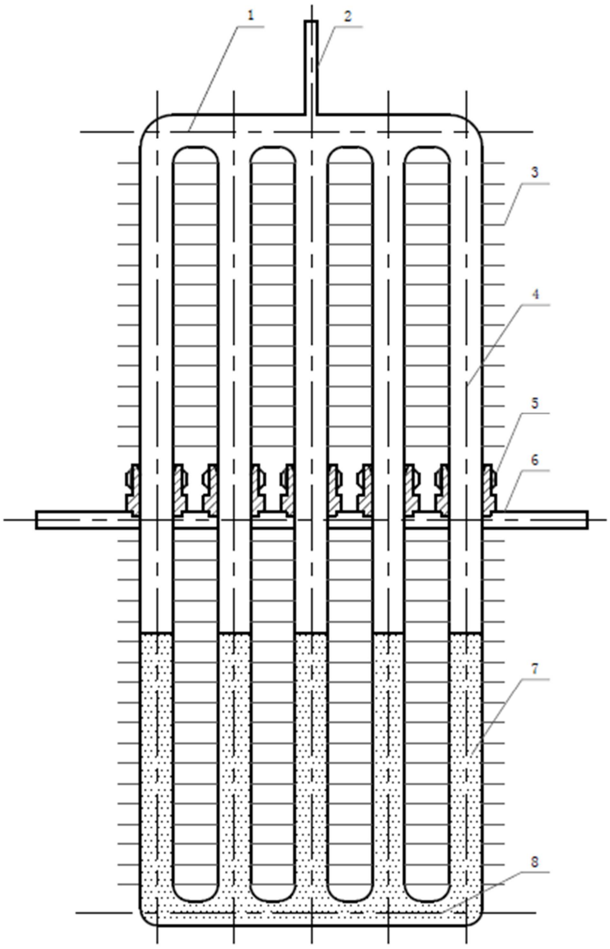


Figure 5-2 heat pipes arrangement as seen in flow direction with an arbitrary amount of working fluid (1) connection on condenser end, (2) charging pipe, (3) fins, (4) heat pipe, (5) sealing structure, (6) separation plate, (7) working medium, (8) connection on evaporator end

5.5 Data acquisition

Pt100 heat resistances are installed in various positions to measure temperature levels inside of the heat exchanger during operation.

The heat pipes wall temperature is measured in 3 locations: At the condenser end wall, at the adiabatic section wall and at the evaporator end wall.

The nitrogen refrigerant temperatures are measured at the inlet and outlet by 4 heating resistances at each location.

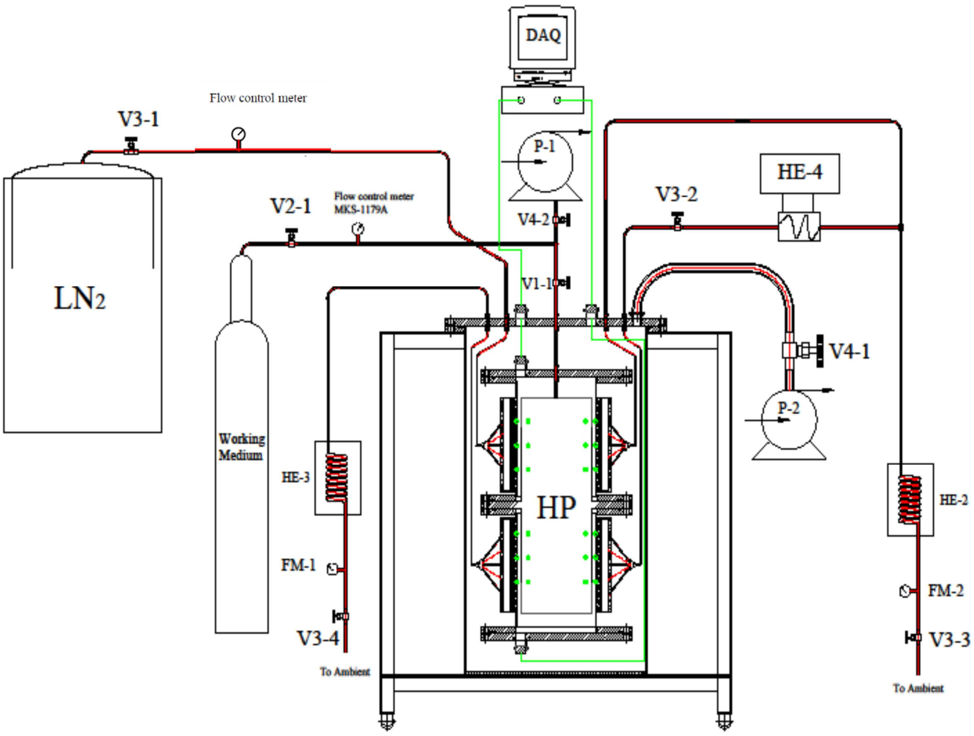


Figure 5-3 working schematic of the heat pipe heat exchanger with auxiliary systems

Figure 5-4 heat pipe heat exchanger (from left to right: heat pipes and vertical distribution plates, distribution caps, heat exchanger in vacuum hull)

6 Experiments

Table 6-1 Experiment parameters

Experiment number	Charging volume $[V_l/V_t]$	Design operating temperature [K]	Design working pressure [MPa]	Filling amount [kg]	Refrigerant (N_2) flow [SLPM]	Heating power [W]	Heat loss rate	Date
1	20%	300	0.033259	0.42	100	70	<5%	23.7.
						128		
						92		
						75		
2	20%	290	0.021373	0.42	200	70	<5%	24.7.
						128		
						92		
3	20%	290	0.021373	0.42	250	70	<5%	25.7.
						128		
						92		
4	20%	280	0.013230	0.42	200	70	<5%	30.7.
						128		
						92		

6.1 Preparation

Before an experiment the desired temperature and pressure inside of the heat pipe during operation are chosen. With these values as well as the heat pipes internal volume, the density of the working medium at these parameters and the ideal gas law the necessary mass of working fluid is calculated.

6.2 Charging process

For the experiments acetone was charged into the heat pipes as a working medium. The appropriated amount which was calculated before was measured by weighting the heat exchanger during the charging procedure and observing the added weight.

After the heat pipes were charged with the working medium the heat exchanger was placed in the vacuum hull and the top lid of the vacuum hull was closed.

6.3 Performance testing

To start the experiment the vacuum hull is evacuated. The nitrogen flow is introduced to cool down the entire heat exchanger to the design temperature of the heat pipe.

The heating value is set at the power supply of the heating elements. The initial value of the experiment is chosen first. Now the temperature measurements are monitored until a stable working point is reached. If a stable working point is recognized or a stable operation is not possible under these conditions the heating value is changed to the next step.

7 Experimental results

During the experiments temperature measurements were carried out at a number of spots. The following graphs show the temperature curves of the experiments. The first graph for each experiment shows the refrigerant inlet and outlet temperatures of the nitrogen gas that is used as a coolant. The second graph of each experiment shows the temperatures of the three sections of the heat pipe the condenser section, adiabatic section and the evaporator section.

In the first experiment a low heating value of 70 watt did not lead to steady state working conditions. After 4000 second the heating value was increased to 92 watt. At these operating conditions the evaporator end temperature can be seen to increase suddenly in sharp spikes. This is an indication of a heat pipe dry out. All the intermediate working fluid is in vapours state at the same time and there is no liquid intermediate working fluid present at the evaporator end to absorb the heat. Under these conditions a steady operation is also not possible. After about 6000 seconds of total elapsed time the heating value was reduced to 75 watt. Under these conditions the heat pipe operates with a refrigerant inlet temperature of 23°C and a refrigerant outlet temperature of 39°C. The temperatures of the different sections of the heat pipe at these conditions are 38°C at the condenser and adiabatic section measurement points and 53°C at the evaporator measurement point.

During the second experiment a steady operation could again not be established with the lowest heating value of 70 watt in a reasonable amount of time. The heating value was increased after 2100 seconds. The highest heating value of 128 watt lead to the appearance of heat spikes in the evaporator temperature curve again. The dry out limitation occurs again. After 4500 seconds the heating value was decreased to 92 watt. A steady operation of the heat pipe with the following temperatures takes place: nitrogen inlet temperature of 19°C, nitrogen outlet temperature of 35°C, condenser and adiabatic section temperature of 33°C and evaporator section temperature of 47°C.

The third experiment has the same characteristics of non-steady behaviour for the lowest heating value (70 watt until 4700 seconds past) and encountering the dry out limitation at the highest heating value (128 watt until 6600 seconds past. A stable working point is found with a heating value of 92 watt, a nitrogen inlet temperature of 20°C, a nitrogen outlet temperature of 34°C, condenser and adiabatic section temperature of 32°C and an evaporator section temperature of 45°C.

These observations are repeated in the fourth experiment with a non-steady operation at the lowest heating value (70 watt until 3400 seconds) and encountering the dry out limit with the highest heating value (128 watt until 4500 seconds past). Finally a stable working point is found with a heating value of 92 watt and with the resulting temperatures of 16°C at nitrogen inlet, 32°C at nitrogen outlet, 32°C at condenser and evaporator section and 43°C at the evaporator section.

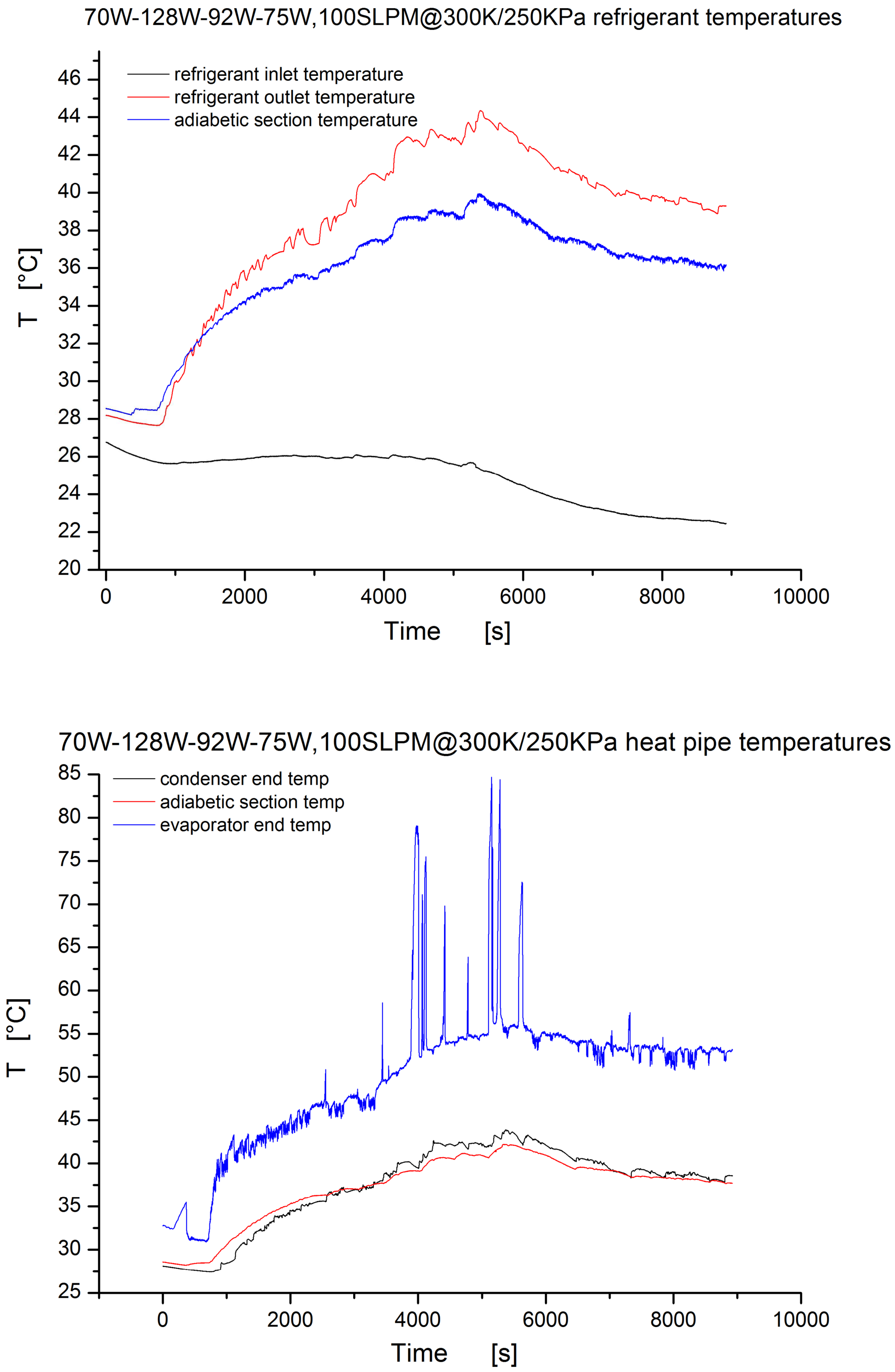


Figure 7-1 Experiment 1 data

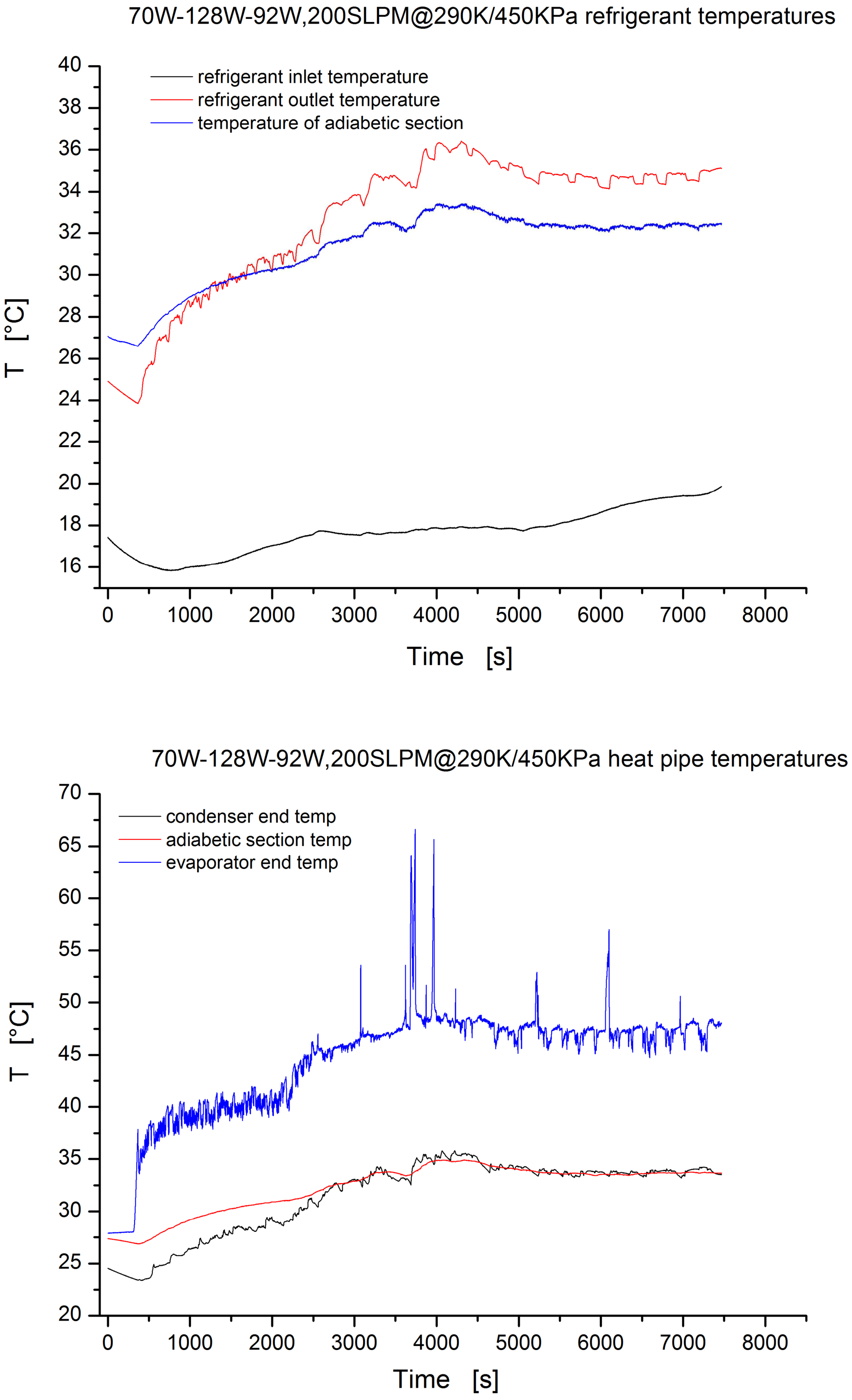


Figure 7-2 Experiment 2 data

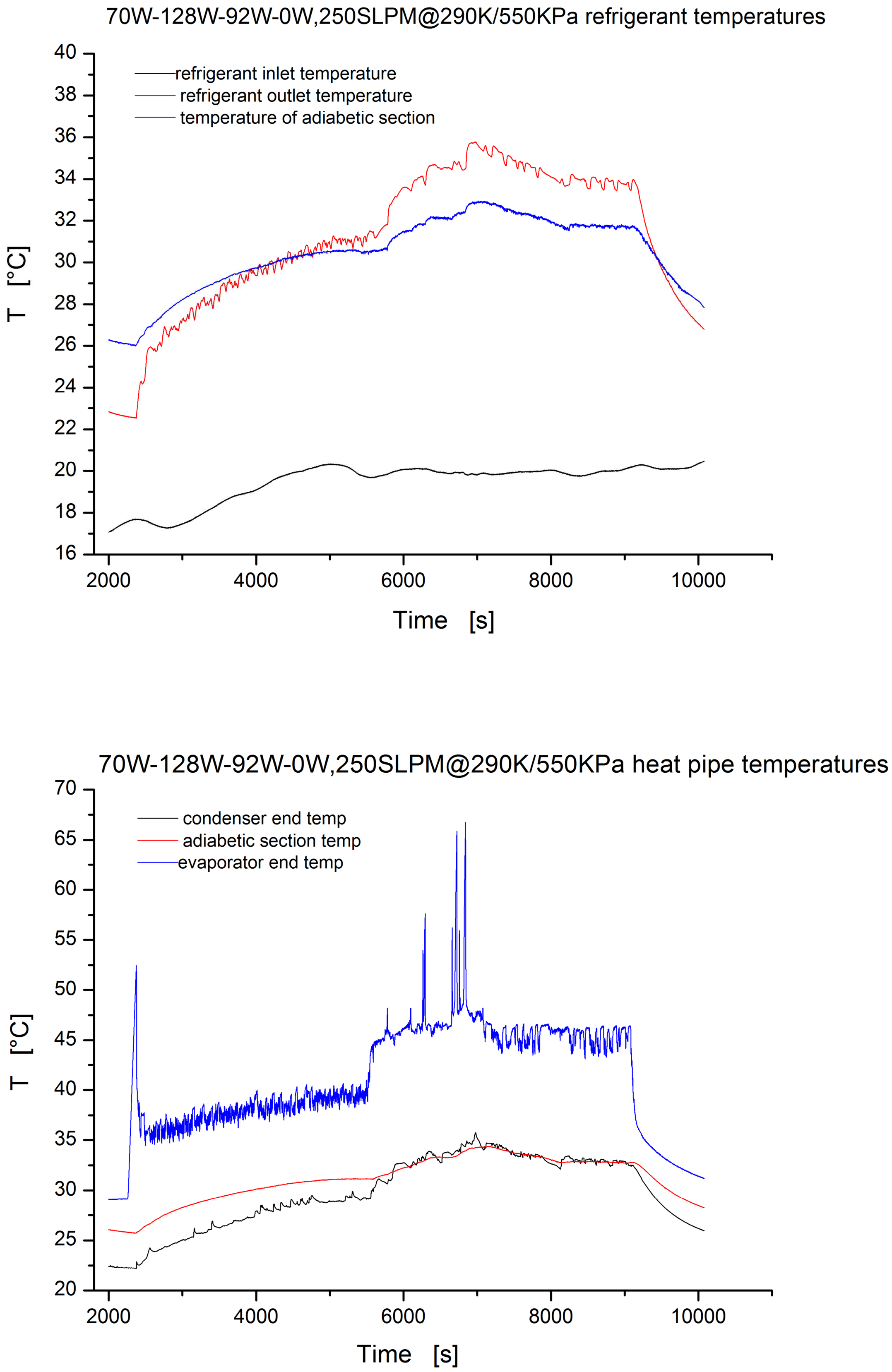
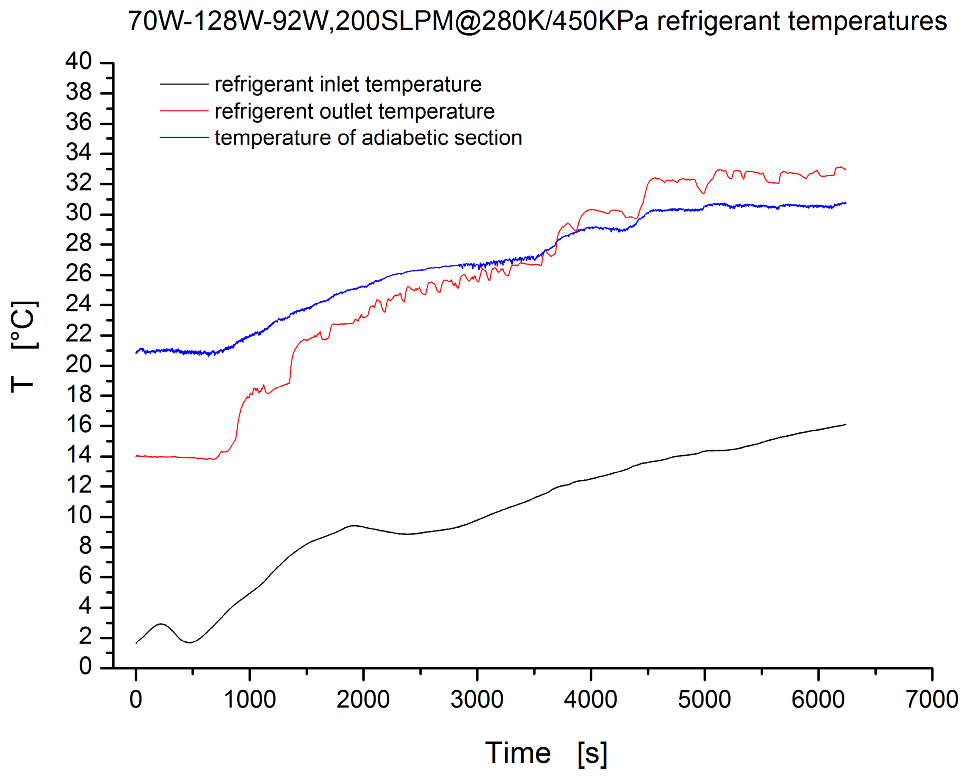
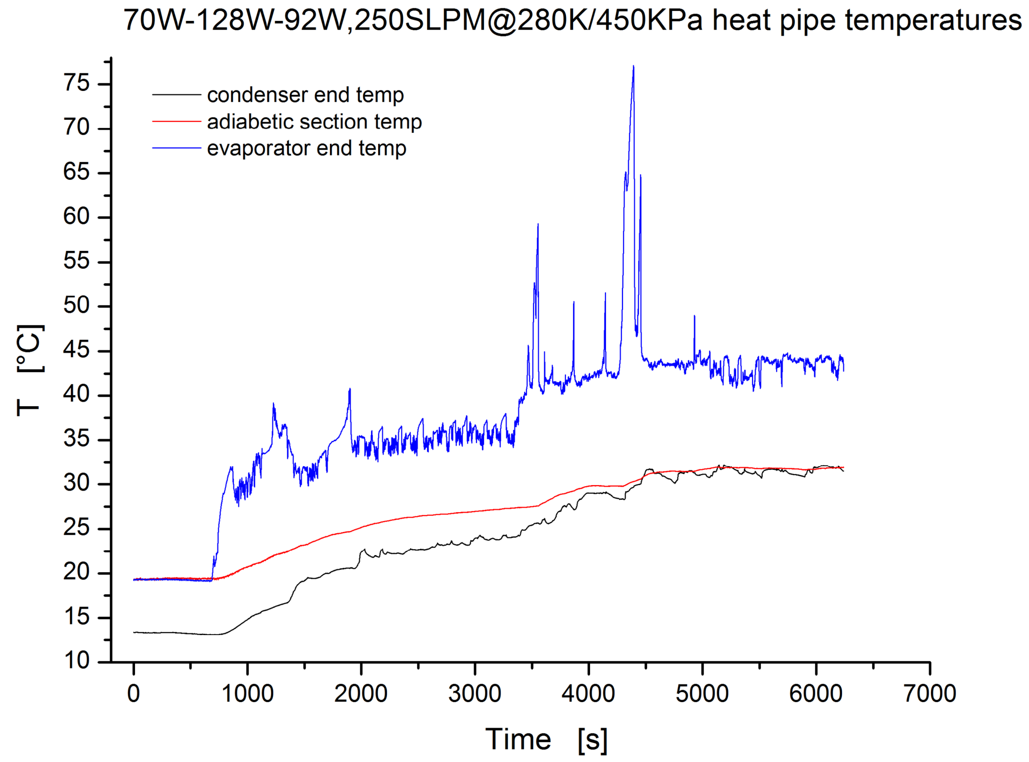


Figure 7-3 Experiment 3 data



70W-128W-92W,200SLPM@280K/450KPa refrigerant temperatures



70W-128W-92W,250SLPM@280K/450KPa heat pipe temperatures

Figure 7-4 Experiment 4 data

8 Analysis

8.1 Charging ratio

When the heat pipe is charged with a curtained amount of working fluid, a drop in temperature will cause a drop in the volume filling ratio. However in the range of 30K and with an initial volume charging ratio of 20% the charge ratio was between 19% and 21% at all times. Experiments show that the influence of this fluctuation on the performance of the heat pipe is small.

8.2 Heating power and refrigerant flow rate

If the heat input into the evaporator section is too small the liquid working medium in the evaporator end cannot be evaporated at a fast enough rate for efficient heat transfer inside of the heat pipe. This is the case at the 75 W heat input step.

With a heat input of 128 W the heat pipe can no longer transport the heat efficiently because the internal working fluid is completely evaporated. The refrigerant flow is no longer sufficient condense the vapour at a fast enough rate. The evaporator section of the heat pipe dries out. Because the heat is continuously added to the evaporator section high increases in the evaporator section wall temperature can be seen. In this case the heat transfer behaviour becomes very unstable.

At the 92 W heat input step the heat pipe operates at a stable working condition. The heat input is sufficient for evaporation to take place in the evaporator section of the heat pipe. Also the refrigerant flow is high enough to condense the internal working fluid at a fast enough rate so that enough working fluid can flow back to the evaporator to absorb more heat.

8.3 Condenser section efficiency

The condenser section efficiency is calculated as the ratio of cooling capacity and electric heating supplied. The cooling capacity is calculated from the mass flow supplied and the inlet and outlet temperatures measured during the experiment and the mass flow of refrigerant provided. The electric heating is set at the beginning of an experiment and raised in intervals to simulate different operating conditions. The refrigerant heat capacity is estimated as a constant between the inlet and outlet temperature.

$$\varepsilon = \frac{\dot{m}_{N_2} \times \Delta t \times c_p}{Q_{ele}}$$

$$\Delta t = t_{out} - t_{in}$$

Table 8-1 Results overview

Experiment number	Temperature [K]	Pressure [kPa]	Flow rate [SLPM]	Enthalpy difference [kJ/kg]	Heating power [W]	Heat transfer efficiency (condensing section)
1	300	250	100	14.8	75	40.6%
2	290	450	200	17.2	92	77%
3	290	550	250	15.1	92	84.5%
4	280	450	200	20.3	92	91%

9 Conclusion

In the heat pipe heat exchanger two constrains are most limiting to operation. First the heat pipe must operate under stable conditions. Secondly the convective heat transfer that occurs outside of the evaporator and condenser section of the heat pipe introduces the most significant thermal resistance that occurs in the heat exchanger. An increase in the refrigerant flow rate can significantly improve the heat transfer into the evaporator section and out of the condenser section. This means that at a constant pressure a higher flow rate will lead to a higher overall efficiency. However the heat pipe still has to operate at a stable working point for this advantage to be exploited.

10 Future work

In an effort to increase the understanding of the heat pipe heat exchanger at low temperature levels the experiments should be extended to the entire temperature range that will have to be covered by a heat exchanger for natural gas liquefaction. To preform experiments at lower temperatures new working fluids for the heat pipe will have to be selected and tested.

In order to overcome some of the limitations of the heat pipe some auxiliary systems could be introduced to the heat pipe heat exchanger. In order to overcome the boiling limit which was sometimes encountered a loop for the working fluid could be added to the heat pipes. Such a system could extract vaporous working fluid from the condenser of the heat pipes and use additional cooling capacity to condense it before reintroducing it into the evaporator end of the heat pipe. In this way at times of very high heat transfer through the heat pipe the system would not become unstable or experience a complete failure because of the lack of liquid working fluid in the evaporator section but would only require additional cooling.

A further improvement could be made with a vortex tube. Such a tube is known to separate a gas stream into a hotter and a colder stream. The vaporous working fluid extracted from the condenser end of the heat pipe could be separated into two streams and afterwards only the hot stream would be cooled down with a low grade source of cooling. Afterwards both streams would be mixed again and reintroduced into the evaporator end of the heat pipe.