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Conceptual Process Design

Process Systems Engineering
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Subject :

Design of A District Heating System Including The Upgrading of Residual Industrial Waste Heat

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Summary

This study was aimed to evaluate the feasibility of using a waste heat stream from DSM for a District Heating System. A conceptual design was carried out with emphasis on the unit for upgrading the residual waste heat.

Having reviewed heat pump technology, mechanical heat pump was found to be the best option for recovering the heat from the residual waste water. This heat pump (ammonia loop) combined with a natural gas fired heater will provide the districts with the required heat. The energy that can be extracted from the waste heat from DSM (100 TJ/annum) with a mechanical heat pump represents approximately 8% of the required energy input of the total Upgrading Unit. Therefore, additional energy input will be provided by a gas fired heater as combustion, which will also serve as a back up system**.**

Heat pump technology is a sustainable method of heating and district heating system using heat pumps is a proven and well-known technology. It is widely used in Europe, Canada and the USA. Currently there is a district heating system in Norway using ammonia in a mechanical heat pump, which includes one airport and adjacent residential buildings and the design of another facility in the future has been considered.

Moreover, nowadays environmental concerns require that the flue-gas emissions resulting from the supply of energy to processes should be minimized. By using the mechanical heat pump for upgrading the heat in this specific design, 4% emissions reduction will be achieved.

The outputs of the design were established based on the required plant capacity and the results of the evaluation of the demand fluctuations and heat losses in the distribution system. Thus, the Upgrading Unit was designed for 923 Tera Joules per annum in order to compensate the heat losses in the pipeline network and supply the districts with a maximum annual demand of 780 Tera Joules. This capacity corresponds to approximately 11,000 houses being supplied with heat.

The Upgrading Unit will be located as close as possible to DSM site in Delft, The Netherlands. The configuration of the distribution system and pipeline lengths were estimated based on the districts listed in a previous feasibility study.

The economical plant life was assumed to be at least 15 years, considering continuous operation 24 h/day, 365 days/year**.** The total investment is MUS$29.3 and the production costs are MUS$20.3. They were calculated with Lang's factored estimation method aiming to achieve ± 25% accuracy. As a result of the economic evaluation of this design, a negative cash flow of MU$6.1 was found. The Upgrading Unit might be profitable if one considers the possibility of using it as a power station in addition to the normal function as a heat producing plant. Decreasing the number of connected districts or collecting more waste water or waste water with a higher energetic value would be another option.

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Chapter 1 Introduction

Currently, enormous amounts of waste heat are generated on a daily basis by a wide variety of not only industrial processes but also commercial units. Typically the waste heat is low grade. Its temperature is around 100ºC and most frequently below this value, which is so low that conventional heat recovery systems do not operate with sufficient efficiency to make the recovery of energy from such sources economical. As a result, vast quantities of waste heat are simply dumped to the atmosphere, ground or water thereby contributing to the overall greenhouse effect and effectively raising the operational costs.

According to Baker and Sherif [55], the first DHS was built at Lockport, New York State, USA in 1877. Considering the large number of facilities currently existing in some regions of the world, District Heating Systems (DHS) using industrial residual wastes heat has proven to be an efficient and reliable method for house and space heating. Such a method is totally in compliance with the new tendency of reducing the costs of fossil based fuels, decreasing the emission of greenhouse gases, and using more innovative, sustainable and cost-effective sources of primary heating energy. DHS using heat pumps is a proven and well-known technology. It is widely used in Europe, Canada and the USA. For instance, in Germany there are 43 DHS (combined heat and power plants) and in The Netherlands there are 33 district heating systems [60], not to mention other several facilities in many countries in Europe. It is important to emphasize that there is a district heating system in Norway using ammonia as the working fluid in a MHP, which includes one airport and adjacent residential buildings. Building a similar heating system is under consideration for the future in another facility [23].

A technical, economic and environmental feasibility study with respect to a DSH to be built in Delft was performed in 2004 [22], which points out the use of a Chemical Heat Pump (CHP) as being a potential alternative for the upgrading method. Thus, the objective of this design is to validate the previous feasibility study and carry out a conceptual design of the DHS, with emphasis on the upgrading unit(s) for the upgrading of the residual waste-heat.

The project consists in designing a DHS using a residual waste heat stream produced by a pharmaceutical industry (DSM). This company is located nearby Delft, The Netherlands, and generates approximately 100 Tera Joules per annum (TJ/annum) residual industrial waste heat at a temperature of 25-35ºC, which is currently cooled down and further disposed to the North Sea. The scope of the project comprises the design of a Heat Upgrading Unit to supply 780 TJ/annum to the neighborhood, which is its annual heating demand.

In order to design the Upgrading Unit a thorough study was performed in order to evaluate not only the CHP alternative, but also the Mechanical Heat Pump (MHP). The aim was to find out the option that fits best into this specific situation. Based on the economical and technical criteria, a MHP using ammonia was found to be the best alternative for this design. Even though the waste water from DSM supplies approximately 100 TJ/annum, it is important to mention that the plant capacity can only be achieved with additional energy input from a natural gas fired heater, which will also serve as a back up system**.** The MHP designed in this CPD accounts for 15 % of the total heat generated in the Upgrading Unit. Further details on this evaluation can be found in Chapters 2, 3 and 5 of this report.

Due to the fact that the previous feasibility study has carried out a comprehensive technical and economic evaluation of the pipeline including heat losses through it, the CPD team was instructed by the Principal to focus mostly on the design of the Upgrading Unit as far as the validation of the previous study is concerned. Nevertheless, the pipeline network configuration was reviewed and based on estimated lengths, the pipelines were designed and heat and exergy losses calculations were performed.

Likewise in the previous study, DSM was also considered as being a black box in this design and the main target was the output of the process. The heating infrastructure in the individual buildings and/or houses in the districts were not included in the design as well.

Although the specification of the waste water stream from DSM was not given due to confidentiality reasons, it is mentioned in the previous feasibility study [22] that its quality is too low. Therefore this stream cannot be reused. The owner(s) of the Upgrading Unit (the party that extracts the heat) will be responsible for disposing the waste water. At present, the waste water flows to the North Sea in a pipeline from the Hoogheemraadschap Delfland and DSM has a contract to utilize the pipeline owned by the Water Board. This CPD considers that the waste water from DSM will be supplied to the Upgrading Unit and after releasing heat to the heat pump it will be disposed to the North Sea using the current pipeline. DSM is supposed to finish its contract with the Water Board, which will then be the responsibility of the owner(s) of the Upgrading Unit.

In order to design this heating system technical, environmental and economic constraints and requirements must be fulfilled so that at the end of the project the design satisfies the needs for which it was thought. The technical requirements are thoroughly discussed in the previous feasibility study [22]. The most important are the following:

- Reliability of the supply;
  It implies in satisfying the required demand and having minimum number of outages (e.g. for electricity it cannot exceed 4 hours).
- Energy efficiency of the infrastructure;
  High energy efficiency of the Upgrading Unit and minor energy losses in the network should be provided.
- Low CO_2 emissions;
  One of the major advantages of this project is the reduction of the carbon dioxide emissions in Delft. Values of CO_2 released to the atmosphere should

be in agreement with the Climate Plan, which sets the reduction of this emission at least 16,800 ton/year in Delft [56].

- Safety;
  It must be as high as possible
- Operability and maintainability;
  Easy start-up and shut-down should be provided. Maintenance should be as practical and easy as possible.
- Backup system;
  This design has to provide backup for the heating system in case DSM shuts down for maintenance or even moves or closes down its facility.
- Costs
  A major economic requirement is minimum costs for the various parts involved.

Any economic advantage of the heat pump technology employed as heating device depends enormously on the cost of electricity compared with the cost of a fuel such as natural gas.

Chapter 2 Process Options and Selection

2.1 Process Concept Chosen

Large amounts of energy are usually released to the environment in the form of low-temperature waste heat and it is almost impossible to improve the energy efficiency of most processes without waste-heat recovery. Based on the heat content of the residual waste water stream available and the energy to be generated by the system, two alternatives were analyzed for the design of the heat Upgrading Unit: mechanical heat pump and chemical heat pump. Both methods are based on the concept that low temperature sources coupled to a suitable heat pump can be utilized to upgrade heat to a higher temperature. The source of low temperature is usually industrial waste heat, but solar thermal collectors are also mentioned in the literature.

The following criteria were taken into consideration for choosing the type of heat pump to be used :

- Technical feasibility: as high as possible
- Costs: as low as possible
- Efficiency (COP): as high as possible
- Reliability: low number of failures, few shut-down periods
- Safety: as high as possible
- Sustainability: as high as possible
- Impact on the environment: as low as possible

A heat pump is a device that transfers heat from a colder to a warmer reservoir, expending energy which is given off to the warmer reservoir along with energy extracted from the colder reservoir. The principle of the heat pump is the same as for the refrigerating machines. The difference from a refrigeration plant is that the main purpose is to heat up the warm reservoir other than the cold one.

Heat pump technology is a sustainable method of heating. It reduces the consumption of oil and gas, decreasing air pollution. It is important to have a continuous heat source in order to achieve the best performance. In this Conceptual Process Design the heat pump is going to be used for heat recovery from the residual waste water from DSM. Based on this concept, the temperature in a liquid can be increased by adding high-quality energy (electricity, in the case of mechanical heat pump) in small amounts and low-quality energy, in the form of waste water in large amounts. In general, approximately one part electricity is added to three parts low-quality energy [23].

2.2 Choosing between Mechanical and Chemical Heat Pumps

Mechanical heat pumps have been widely used to recover and upgrade waste heat especially in industry. They use the latent heat of the working fluid and shaft work to recover and upgrade low-level thermal energy (e.g. waste heat streams). The so called working fluid, a volatile liquid, circulates through the four main equipment of the system undergoing a cycle. Although technical development of high-efficiency mechanical heat pumps has been achieved, the maximum temperature that can be reached by the most advanced systems that deliver thermal energy is around 110ºC, according to several heat pump experts [10], [28]. Such a temperature is rather low for practical utilization as heating medium in the industry. This factor along with high operational costs and low efficiency limits the use of MHP to small scales. However, the level of temperature of MHP is suitable for the Upgrading Unit, since the required temperature at the DHS is 90ºC.

The four main components of a MHP are the evaporator, the compressor, the condenser and the expansion valve, which are connected to a closed loop. The evaporator is where the working liquid boils and evaporates under low pressure. Low-temperature energy, in this present case in the form of waste water, is added. The gas is compressed to a higher pressure and temperature in the compressor. The hot gas is then sent to the condenser where it releases heat to the water that is the heating medium to be delivered to the DHS. Figure 2.1 shows a simplified scheme of a MHP.

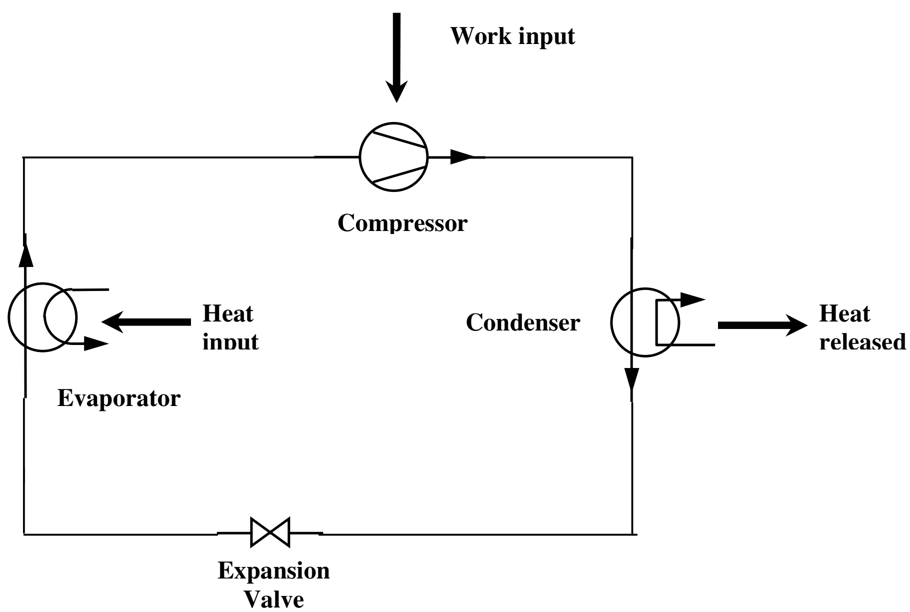

Figure 2.1 Simplified scheme of a MHP

Sorption heat pumps, which use heat of absorption and adsorption, deliver thermal energy at relatively high temperatures and can be designed at large scales for industrial applications, but they require large pressure shifts among system components, which result in increasing operating costs and maintenance problems. The advantages and disadvantages of MHP are shown in Table 2.1.

Table 2.1 Pros and Cons of MHP

Pros	Cons
Environmentally benign	It can only deliver thermal energy at a maximum temperature of 110^{o}C (quite low for practical heating source)
It can deliver thermal energy at a relatively high temperatures (absorption heat pump)	It is limited to small scales (mechanical heat pump)
It can be designed at large scales for industrial applications (absorption heat pump)	High operating cost
	Low efficiency
	Maintenance problems

A CHP, as the second alternative to be investigated for this design, is a system that makes use of a reversible chemical reaction to change the temperature level of the thermal energy that is stored by chemical substances. Therefore, in this type of systems chemical reactions play a major role in absorbing and releasing heat. This type of heat pump was chosen as the best option to be used to design the Upgrading Unit in the previous feasibility study [22]. That is why the CPD team evaluated it in order to validate the technical and economic feasibility of applying such a method to make proper use of the residual waste heat from DSM.

The hydrogenation and dehydrogenation of the continuous liquid-gas isopropanol-acetone system was the system investigated. The choice of this working pair was based on the availability of information in the literature taking also into consideration the range of applicability for continuous process, which mostly requires liquid-gas types. The working pair mentioned in the previous feasibility study [22], i.e., salt impregnated carbon fibers (the system NH_3-$CoCl_2$), is part of the solid-gas type of reactive medium in CHP and it is mostly applicable for batch systems. Moreover, its temperature range does not fit to the waste water from DSM, since in most salt-ammonia vapor heat pumps the residual waste heat temperature is within the range 80-150ºC. Another drawback of the salt-ammonia pairs is the absence of reliable thermodynamic data in the available literature. The data are rare and, when available, inconsistent between different experiments [83].

Heat pumps based on acetone/hydrogen/isopropanol have been extensively studied and besides the fact of being well-known it can absorb heat at a relatively low temperature and has fewer hazards than other systems [21]. Other working pairs were also sorted out, but they are applicable either for cooling (13-40ºC) or for high-temperatures (400-800ºC). The reversible chemical reaction and the respective decoupled forward and backward reactions are the following:

$$(CH_3)_2\,CO[g] \;+\; H_2[g] \;\leftrightarrow\; (CH_3)_2\,CHOH[l] \qquad (2.1)$$

$$(CH_3)_2\,CHOH[l] \;\rightarrow\; (CH_3)_2\,CHOH[g] \;+\; H_2[g] \qquad (2.1.a)$$

ΔH = 100.4 kJ/mol

$$(CH_3)_2\,CO[g] \;+\; H_2[g] \;\rightarrow\; (CH_3)_2\,CHOH[g] \qquad (2.1.b)$$

ΔH = -55.0 kJ/mol

In the backward reaction, isopropanol is decomposed into acetone and hydrogen by an endothermic dehydrogenation reaction. For this reaction to occur at lower temperature, heat can be supplied from a lower temperature source. The hydrogenation reaction is exothermic and heat is released causing upgrading of heat for any suitable application. In other words, the useful heat is obtained from the exothermic reaction and the heat supplied (at lower temperature) is used for endothermic reaction. For the dehydrogenation reaction usually homogeneous catalysis is applied whereas the exothermic reaction is carried out in gas phase mostly in packed bed reactors. Under normal conditions the conversion for both reactions is incomplete and, therefore, a distillation column is necessary to separate the products. A heat exchanger is introduced to balance those points of the system where heating or cooling is necessary. Unfortunately, in this process, a fraction of the recovered waste heat is lost at a temperature lower than the temperature at which the endothermic reaction is carried out since heat is released to the environment by means of cooling water or air in the condenser of the distillation column.

Many researches have been performed for different catalysts enhancing the endothermic/exothermic reactions of the working pair isopropanol-acetone. Some references in literature [11] mention that low-grade heat from solar thermal collectors could supply heat to an endothermic reactor of a liquid-gas system. Taneda et al. [17] noted that heat at 80ºC could be upgrade by liquid-gas CHP to 150-200ºC. The endothermic liquid-phase dehydrogenation of isopropanol produces acetone and hydrogen with appropriate catalysts (e.g. Ru/C, Ru-Pt/C, Ru-Pd/C, Raney nickel, etc) at 80ºC [21]. In the exothermic reaction gas-phase hydrogenation of acetone into isopropanol takes place at 200ºC with another catalyst (e.g. Ni-Cu/alumina, Raney nickel). Data regarding deactivation is rare, but one of the references [21] states that researches conducted at the Institut Francais du Petrole with respect to the dehydrogenation reaction at 150ºC, the Raney nickel catalysts is reported to remain active for more than two years of continuous on-stream service. Figure 2.2 shows a simplified flow diagram for this process.

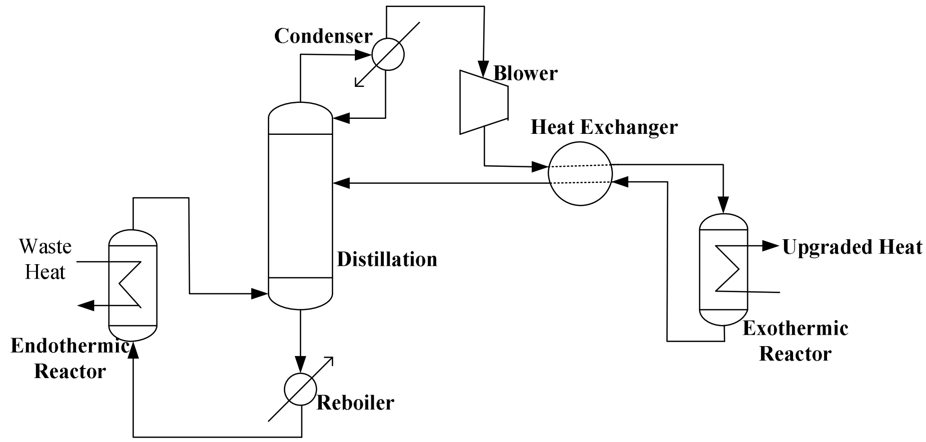

Figure 2.2 Simplified scheme of the isopropanol/acetone CHP

The endothermic reaction, eq.2.1.a, occurs at 80ºC and, according to the concept of a heat pump, the heat should be supplied from a source that is available at a temperature at least 5ºC higher than that if one considers a minimum approach temperature of 5.6ºC, which is usually estimated by standard rules of thumb. The heat required for the reaction is mostly supplied by a heat medium, which is directly employed in the reactor vessel heating system. In addition, part of the total heat for the endothermic reactor is provided by the steam used in the reboiler of the distillation column. However, the waste heat stream from DSM has temperature within the range 25-30ºC and cannot be used as a heat medium in the heating system of the endothermic reactor. It means that the best way to apply a CHP concept to the waste water from the DSM would be to use a working pair that fits to the range of 20-25ºC. Unfortunately, working pairs are not available for continuous applications as required in a DHS.

In order to validate the feasibility of this system for the Upgrading Unit and estimate costs to compare with the figures presented in the previous feasibility study [22], a rigorous steady-state simulation of this CHP was performed with AspenPlus based on the assumption that a heat medium (e.g. steam) would provide the endothermic reactor with 100 TJ/annum, which is the same amount of energy that would be supplied by the waste water from DSM. The rest of the heat would be provided by the low-pressure steam to be used in the reboiler of the distillation column. The heat released by the exothermic reaction would be transferred to the tap water coming from the districts to heat it up from 66.5ºC to 70.9ºC. Higher temperature would be achieved by using high flow rate in the exothermic reactor, which would require high duty in the reboiler, condenser, bigger equipment and, as a consequence, higher costs. Therefore, likewise in the mechanical heat pump with ammonia, the desired temperature of the tap water (137.2ºC), which would be sent to the district buildings would finally be obtained after heating it up in a fired heater using natural gas. Hot flue gases from the fired heater would pre-heat the cold tap water as it was mentioned in the mechanical heat pump. The description of the CHP simulation is presented in Appendix 30. The Aspen file containing this simulation is available in the CD-ROM.

Based on the results of the simulation, the expected energy parameter coefficient of performance (COP) for the chemical heat pump system is 0.174. Calculations are presented in Appendix 22.

It is important to remark two additional problems in the case of using a CHP as described above. Firstly, low-pressure steam should be available to run the exothermic reaction and the distillation and usage for the released condensate should be provided. Secondly, cooling water for the condenser of the distillation column would be required. These two utilities would require external supply or even self generation.

The possibility of upgrading the waste heat water from DSM to a temperature higher than 100ºC so that it could be used with the isopropanol-acetone working pair was evaluated, but this option is also not feasible due to the following reasons:

- Additional heat (e.g. low pressure steam) would be required to heat up the waste water stream from 25ºC to at least 105ºC;
- The only way to cool down the waste water prior to the final disposal would be via heat exchange with cooling water. The lowest temperature in the CHP system is the in the condenser of the distillation column (~18-20ºC). Thus, additional cooling water would be required and the final temperature of the disposed waste water would be around 25ºC, which is practically the same temperature of the stream produced by the DSM

Table 2.2 Pros and Cons of Chemical Heat Pump

Pros	**Cons**
It does not need electrical power	The operation is a bit more complex (involves chemical reaction and more complex unit operation)
Generally it has higher thermodynamic efficiencies	A bit more expensive in terms of equipment cost
Possibility of energy storage	By-products formation
It provides ability to capture the rejected low-grade heat and to reuse it at increased temperature levels in various industrial processes	Catalyst deactivation
It has fewer hazards	

Table 2.2 shows advantages and disadvantages of CHP. One remarkable issue regarding the CHP is the formation of by products, which may hamper the

functionality and the capacity of the system. The selectivity's of the endothermic and exothermic reactions are not 100% and, as a result, by-products will be formed and will accumulate in the system, which will lead to the following issues:

- Decreasing in the Coefficient of Performance (COP). By-products form a vapor stream that circulates through the system without contributing to the heat pump process;
- Presence of a higher circulating mass flow to obtain the same capacity;
- Poisoning the catalysts;
- Affecting the performance of the distillation column (e.g. higher reboiler temperature, inefficient separation) and heat exchangers.

Purging, compensating by make-up, might be a solution to the by-products problem. However, the complexity and operating costs of the system will increase. Additionally, emissions will be regularly generated.

The purchase costs of equipments in a CHP were calculated by following the method of the book by Seider, Seader and Lewin [1] based on the results obtained from the simulation with AspenPlus. The results are shown in Table 2.3 and the detailed calculation is attached in Appendix 31.

Table 2.3 List of main equipment – Chemical Heat Pump

	Main Specification (Characteristics)	**Material of Construction**	**Cost (f.o.b. 2004) (US\$ x 10^3)**
Endothermic Reactor	44 m^2	Carbon Steel	21
Exothermic Reactor	487 m^2	Carbon Steel	62
Distillation Column	H=14 m, D=4 m	Carbon Steel	236
Condenser	753 m^2	Carbon Steel	84
Reboiler	191 m^2	Carbon Steel	36
Blower	374 kW	Cast Iron	119
Heat Exchanger	107 m^2	Carbon Steel	27
Air Blower	1921 kW	Cast Iron	433
Fired Heater	136899 kW	Carbon Steel	6421
Pumps	418 kW	Cast Iron	63
Spare Pump	418 kW	Cast Iron	63

The total equipment purchase costs of the CHP is approximately US\$8.34 million and the total investment is approximately US\$28.6 million, which is higher than the value US\$12.5 million (€10.4 million) as reported in the previous feasibility study [22]. Total investment calculation is shown in Appendix 32.

Besides the technical unfeasibility of using the waste heat from the DSM for a chemical heat pump, the choice of a MHP can also be justified based on the annual costs of energy to generate 780 TJ/annum as it is shown in Table 2.4. The bottom row of this table shows the figures regarding heating the districts using only regular fired heater. The prices for electrical energy are approximately 3 times as much as the price for the same amount of energy in the form of natural gas. That explains the fact that although the energy input for “just fired heater” is higher, the costs are lower since mechanical heat pump uses electrical driven compressors.

Table 2.4 Energy input per annum by gas and electricity to supply 780 TJ/annum to the districts

	Energy Costs (Million US$)	Input to get 780 TJ/annum (TJ/annum)
Chemical heat pump	13.69	1519
Mechanical heat pump	12.31	1196
Just fired heater	10.95	1242

Based on the explanations given above, a MHP was chosen as the best alternative for the Upgrading Unit.

As it is shown in Table 2.4, based on the results of the simulation for this MHP, an input of 1196 TJ/annum (natural gas + electricity) is required to supply 780 TJ/annum to the districts. Results of calculations reveal that in order to generate the same amount of energy via regular fired heaters (i.e. burning natural gas without the heat of the waste water) the energy required is 1242 TJ/annum. Taking these figures into account, the reduction of emissions due to the use of this mechanical heat pump was estimated to be approximately 4%. Figure 2.3 shows a simplified block diagram regarding energy input/output and losses in TJ/annum for this MHP.

A summary of the main regular MHP equipments is shown in Table 2.5 and the detailed calculations are attached in Appendices 3. The purchase costs of equipments in this MHP were calculated by following the method of the book by Seider, Seader and Lewin [1] based on the results obtained from a rigorous simulation, which was performed using AspenPlus.

2.3 Choosing The Working Fluid of Mechanical Heat Pump

Ammonia was selected to be used as working fluid in the mechanical heat pump. Naturally existing in the biosphere, ammonia is a natural working fluid, has negligible global environmental disadvantages since it does not damage the ozone layer or harm the environment in any way. Actually, natural working fluids are long-term alternatives to the chlorofluorocarbon-based refrigerants (CFCs). Thermodynamically ammonia has very good properties and although it is very well-known in refrigerant plants, it had not been previously used in heat pumps. Nowadays it is leading as the working fluid in some countries. There is a large district heating system in Norway using ammonia, which includes an airport and residential buildings and a similar heating system is under consideration for the future in another facility [23]. Ammonia is not yet used in high-temperature industrial heat pumps due to the fact that there are currently no suitable high-pressure compressors available (40 bar, maximum) [28]. Codes, regulations and legislation have been developed aiming to cope with the toxic and, to some extent, the flammable characteristics of ammonia. Once efficient high-pressure compressors are developed, ammonia will be an excellent high-temperature working fluid.

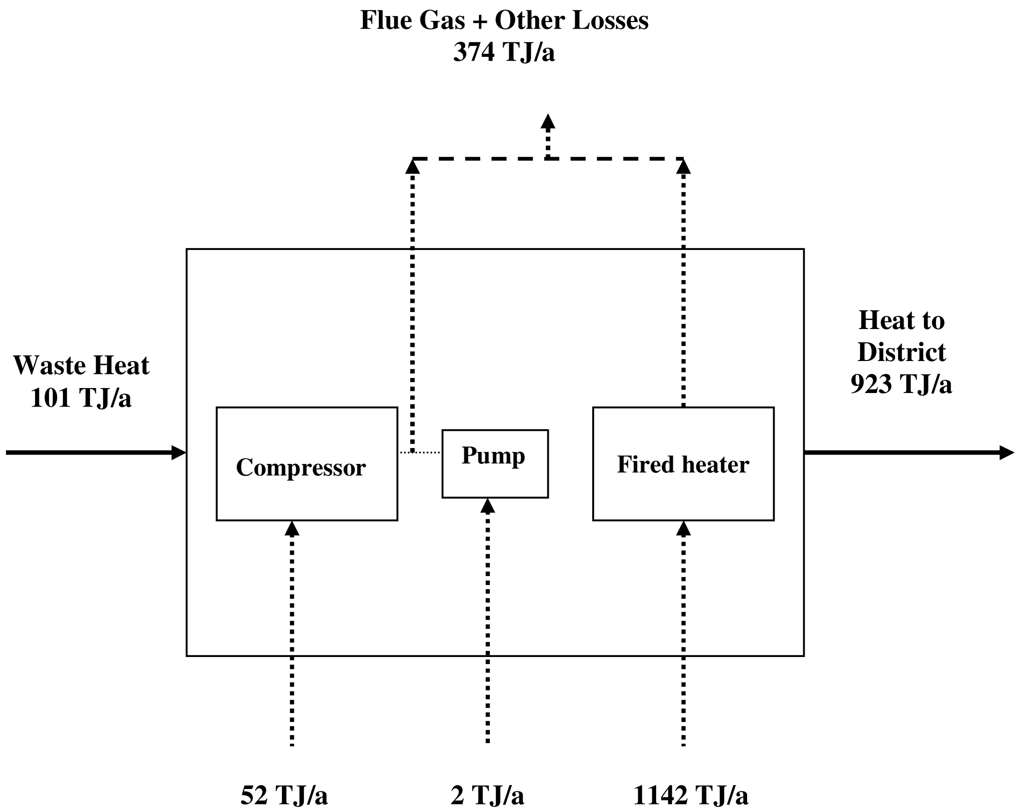

Figure 2.3 Simplified energy scheme of the MHP

2.4 Choosing The Heat Transfer Medium to The Districts

An ideal heat source for heat pumps in residential buildings must have a high and stable temperature during the heating season, be abundantly available, not be corrosive or polluted, have favorable thermophysical properties and require low investment and operational costs. Additionally, the availability is a key factor to determine its use. The residual waste water fulfills most of these requirements, but, according to the feasibility study [22], its quality is poor. In order to use the industrial waste water from DSM for DHS safety is a major aspect to be taken into consideration. The safety could be affected by the quality of the water that runs through the city and perhaps also throughout the buildings. Tap water was chosen as the heat source to be sent to the districts per pipeline involving a closed water loop, which transfers heat to the internal water-closed loop in the district and returns to the Upgrading Unit.

Table 2.5 List of main equipment (Mechanical Heat Pump)

	Main specification (characteristics)	**Material of Construction**	**Cost (f.o.b. 2004) (US$ x 10^3)**
Ammonia Evaporator	737 m^2	Carbon Steel	113
Compressor 1	690 kW	Carbon Steel	536
Compressor 2	946 kW	Carbon Steel	690
Intercooler	263 m^2	Carbon Steel	44
Ammonia Condenser	220 m^2	Carbon Steel	59
Ammonia Collector Vessel	1.6 m^3	Carbon Steel	18
Air Blower	2025 kW	Cast Iron	295
Fired Heater	136899 kW	Carbon Steel	6421
Pump	418 kW	Cast Iron	63
Spare Pump	418 kW	Cast Iron	63
Heat Exchanger (Total of 5, one in each Heat Station)	207 m^2	Carbon Steel	155

Chapter 3 Basis of Design

This CPD consists of:

- Designing a mechanical heat pump, which uses ammonia as working fluid, consisting of one evaporator, two compressors, one intercooler, one condenser, one ammonia collector vessel, one expansion valve and normal auxiliary equipment associated with the compressors (e.g. lubrication system, knock out drum, etc). This system will transfer the heat of the DSM waste water to tap water that circulates throughout the Heat Stations, located nearby the district buildings, and returns to the Upgrading Unit via a pipeline network. The maximum temperature that the tap water stream will achieve with the ammonia thermal cycle is 77.8ºC;
- Designing a fired heater in order to heat the tap water downstream the heat pump up to 137.2ºC, which is the temperature required to supply a heating demand equivalent to 780 TJ/annum (923 TJ/annum in order to compensate heat losses in the distribution system). The fired heater will be designed so that its capacity allows it to serve as a back up for the heating system of the district;
- Designing a piping delivery system consisting of: two pumps (one spare), valves, instrumentation and pre-fabricated steel pipes insulated with Micro-PUR and protected by a polyethylene shield. The pipeline infrastructure will be provided next to the municipal road in accordance with the conditions mentioned in the previous feasibility study;
- Validating the chemical heat pump (CHP) proposed by the previous feasibility study.

3.1 The Block Scheme of The Chosen Process

The block scheme of the mechanical heat pump system using ammonia cycle is presented in Figure 3.1

3.2 Thermodynamic Properties

For MHP simulation Peng-Robinson Equation of State was chosen to be used for the ammonia, water, natural gas and air for the entire range of pressure and temperature. These components are very well known and all required data for the steady-state simulation and other calculations are available either in the databanks or literature. For pipeline network simulation, Steam-Tables model from AspenPlus was applied because the whole system involves liquid water at pressure above 5 bar.

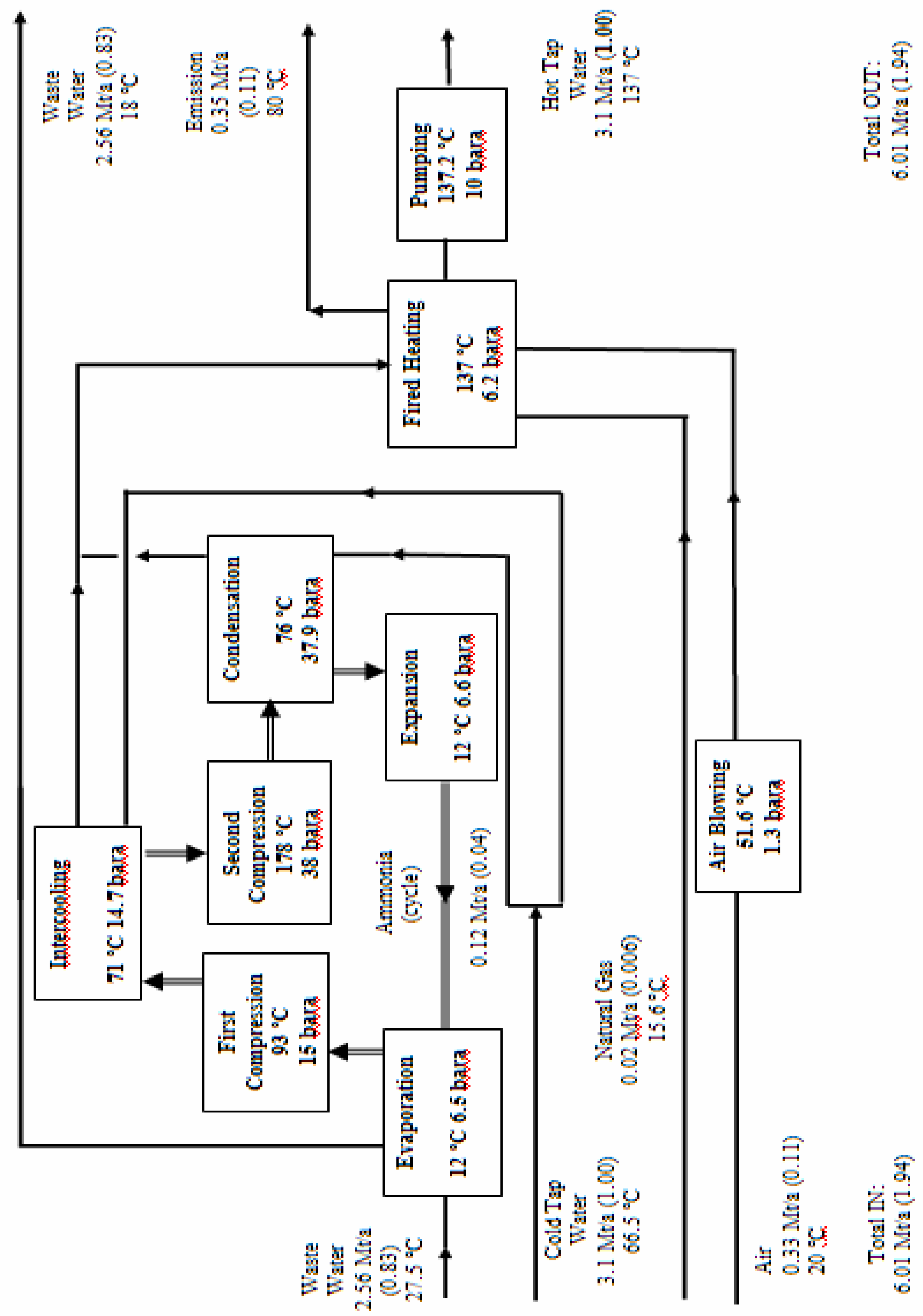

Figure 3.1 The block scheme of the Mechanical Heat Pump

3.2.1 Thermodynamic Methods

Literature: AspenPlus

Binary parameters: - from databanks available in AspenPlus

Input:
- note units
- test for known situation
  e.g., with AspenPlus through the proper generator:
  tables and graphs of L + G equilibrium,
  phase envelope: γ, x; P, x; T, x

Extra in ASPENPLUS: None

3.2.2 Selection of Thermodynamic Models (See item 2.3.5 for abbreviations)

PVT L and V : PR, Steam-Tables
fugacity : PR
liquid density : PR

3.2.3 Liquid/Vapor High Pressure

PR: k_{ij} required and available for the components/mixtures involved (especially ammonia)

3.2.4 γ-models

Wilson model with Henry's law was employed for the simulation of CHP alternative

3.2.5 Abbreviation and Its Meaning

PR : Peng Robinson equation

3.3 List of Pure Component Properties

3.3.1 Ammonia

Table 3.1 Ammonia properties

Name		Ammonia
Structural Formula		NH_3
Molecular Weight	kg/kgmol	17.032

Table 3.1 Ammonia properties (*continued*)

Phase	S/L/V	V
Boiling Point	^oC	-33.33
Melting Point	^oC	-77.72
Flash Point	^oC	N/A
Ignition Temperature	^oC	850
Auto-ignition Temperature	^oC	850
Flammable Limits in Air	% vol	16 - 25 %
Lower Explosion Limit (LEL)	% vol	15
Upper Explosion Limit (UEL)	% vol	28
Liquid Density	kg/m^3	681.91
Vapor Density	kg/m^3	0.8898
Chemical Reactivity		Avoid using copper as material of construction
MAC		20
LD_{50}	mg/kg	3500
LC_{50}	ppm rat (1 hour)	7650

3.3.2 Water

Table 3.2 Water properties

Name		Water
Structural Formula		H_2O
Molecular Weight	kg/kgmol	18
Phase	S/L/V	L
Boiling Point at 1 atm	^oC	100
Melting Point at 1 atm	^oC	0
Flash Point	^oC	N/A
Ignition Temperature	^oC	N/A
Auto-ignition Temperature	^oC	N/A
Flammable Limits in Air	% vol	N/A

Table 3.2 Water properties (*continued*)

Lower Explosion Limit (LEL)	% vol	N/A
Upper Explosion Limit (UEL)	% vol	N/A
Liquid Density, 25 ^oC, 1 atm	kg/m^3	997.08
Vapor Density, 100 ^oC, 1 atm	kg/m^3	0.5980
Chemical Reactivity		N/A
MAC		N/A
LD_{50}		N/A
LC_{50}		N/A

3.3.3 Combustion Air

Table 3.3 Combustion air properties

Name		Air
Composition		
N_2	%-vol	79
O_2	%-vol	21
Molecular Weight	kg/kgmol	28.84
Phase	S/L/V	V
Boiling Point (incipient)	^oC	-194.5
Melting Point (incipient)	^oC	-213.4
Flash Point	^oC	N/A
Ignition Temperature	^oC	N/A
Auto-ignition Temperature	^oC	N/A
Flammable Limits in Air	% vol	N/A
Lower Explosion Limit (LEL)	% vol	N/A
Upper Explosion Limit (UEL)	% vol	N/A
Liquid Density, 1 atm at boiling point	kg/m^3	875
Vapor Density 25 ^oC, 1 atm	kg/m^3	1.1850
Chemical Reactivity		N/A
MAC		N/A
LD_{50}		N/A
LC_{50}		N/A

3.3.4 Natural Gas

Table 3.4 Natural gas properties

Name		Natural Gas
Composition		
CO_2	% vol	0.9
N_2	% vol	14.4
Methane	% vol	81.4
Ethane	% vol	2.7
Propane	% vol	0.4
Butane	% vol	0.2
Molecular Weight	kg/kgmol	18.554
Phase	S/L/V	V
Boiling Point at 1 atm	^oC	-183
Melting Point at 1 atm	^oC	-
Flash Point	^oC	-
Ignition Temperature	^oC	670
Auto-ignition Temperature	^oC	540
Flammable Limits in Air	% vol	5-15
Lower Explosion Limit (LEL)	% vol	4.0
Upper Explosion Limit (UEL)	% vol	94
Liquid Density, 25 ^oC, 1 atm	kg/m^3	0.833
Vapor Density, 0 ^oC, 1 atm	kg/m^3	0.644
Chemical Reactivity		-
MAC		N/A
LD_{50}		N/A
LC_{50}		N/A

3.4 Basic Assumption

3.4.1 Plant Capacity.

The plant capacity is 923 TJ/annum.

3.4.2 Economical Plant Life.

The economical plant life was assumed to be at least 15 years, considering continuous operation 24 h/day, 365 days/year**.**

3.4.3 Location

The Upgrading Unit will be located as close as possible to DSM site in Delft, The Netherlands in order to facilitate the connections with utilities, the usage of the infrastructure, and the connections with the existing pipeline for final disposal of the waste water to the North Sea.

3.4.4 Battery Limit

The battery limits stop at the Heat Stations to be located near the districts**.** Figure 3.2 presents a block diagram of the battery limit, which consists of:

- the Upgrading Unit;
- the pipeline network including the hot and cold tap water streams that circulate within the Upgrading Unit and the Heat Stations;
- the heat exchangers in the Heat Stations;
- part of the residual waste water pipeline entering the Upgrading Unit;
- part of the existing waste water pipeline leading to the North Sea;
- part of the pipeline belonging to each district still inside the Heat Station (district hot water pipe exiting the heat exchanger until the temperature controller)

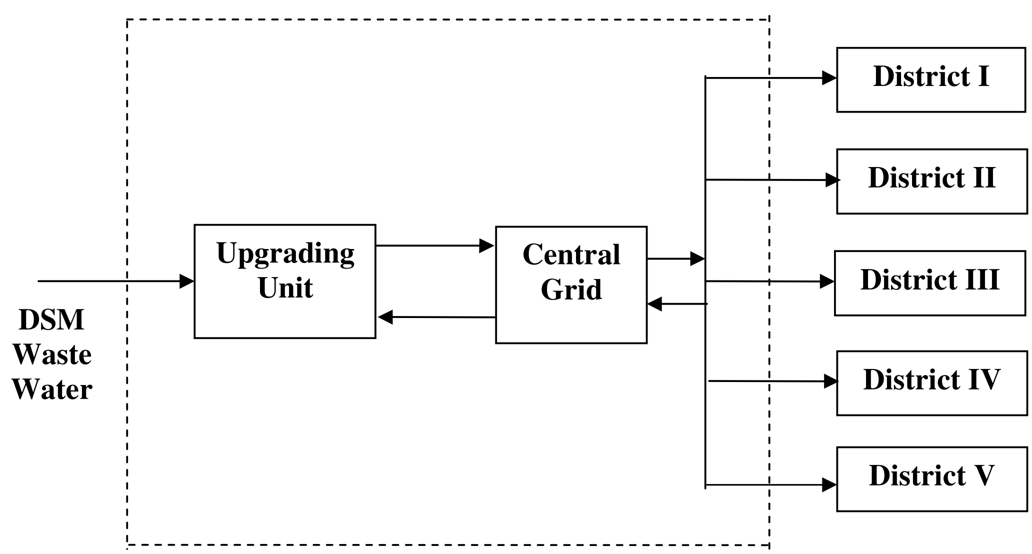

Figure 3.2. Battery limit block diagram

3.5 All Streams Passing through The Battery Limit

3.5.1 Feedstock

1. Waste Water inlet

Table 3.5 Waste water inlet stream

Stream Name : **WWIN**					
Comps	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**		
Water	%wt	100	100	(1)	(1) Composition was not given due to confidentiality reasons. Contaminants not harmful for the process
Total		100	100		(2) Values taken in consultation with the Principal
Process Condition and Price					(3) Delivery per pipeline
Temp	°C	27.5		(2)	(4) Taken as free of charge as agreed with the Principal during the BOD review.
Press.	Bara	3.0		(3)	
Phase	V/L/S	L			
Price	US$	0		(4)	

2. Cold Tap Water Inlet

Table 3.6 Cold tap water inlet

Stream Name : **UUIN**					
Comps	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**		
Water	%wt	100	100	(1)	(1) Contaminants not harmful for the process.
Total		100	100		(2) Delivery per pipeline
Process Condition and Price					(3) As per consultation with the Principal, the price was taken based on the value of energy paid by the public (including 6% BTW tax + 14.6 Euro cent/m^3 Environmental tax) [79]
Temp	°C	60			
Press.	Bara	9.0		(2)	
Phase	V/L/S	L			
Price	€/m^3	1.503		(3)	

3.5.2 Chemical

Ammonia

Table 3.7 Ammonia stream

Stream Table : **Ammonia**					
Comp	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**		
Ammonia	% wt	99.5 - 99.8	100	(1)	(1) For the simulation, 100 % wt ammonia was assumed
Moisture	% wt	0.2 - 0.5	0	(2)	(2) Contaminants not harmful for the process
Oil	ppm	< 5	0	(2)	(3) Delivery per tank truck. For the start up the amount of ammonia was estimated considering the inventory of the system (pipes + equipment), which corresponds to 700 kg. Calculations are present in Appendix 1. The required amount is dictated by the compressor detailed design and the most accurate value is provided by the compressor manufacturer. Refill must be periodically provided to compensate any losses. Pressurized liquid at room temperature
Total			100		
Process Condition and Price					(4) Source [30]
Temp	oC	20		(3)	
Press	Bara	10		(3)	
Phase	V/L/S	L		(3)	
Price	US$/ton	350 - 375		(4)	

3.5.3 Product

Hot Tap Water Outlet

Table 3.8 Hot tap water outlet stream

Stream Name : UUOUT					
Comps	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**		
Water	%wt	100	100	(1)	(1) Tap water quality standards
Total		100	100		(2) Delivery per pipeline

Table 3.8 Hot tap water outlet stream (*continued*)

Process Condition and Price				(3) As per consultation with the Principal, the price was taken based on the value of energy paid by the public. (including 6% BTW tax + 14.6 Euro cent/m^3 Environmental tax)
Temp	^oC	97.0		
Press.	Bara	10.0		
Phase	V/L/S	L	(2)	
Price	€/ m^3	1.503	(3)	

3.5.4 Wastes

1. Waste Water Outlet

Table 3.9 Waste water outlet stream

Stream Name : WWOUT					
Comps	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**		
Water	%wt	100	100	(1)	(1) Composition was not given due to confidentiality reasons. Contaminants not harmful for the process.
Total		100	100		(2) Value established by the CPD team as design basis.
Process Condition and Price					(3) Delivered for disposal per pipeline, pressure should be enough to compensate pressure drop
Temp	^oC	18		(2)	(4) Gathered from the previous feasibility study [22]. It was agreed during the BOD review that the disposal the North Sea will be performed using the existing pipeline. The current contract between DSM and the Water Board will be transferred to the owner(s) of the Upgrading Unit.
Press.	Bara	2.5		(3)	
Phase	V/L/S	L			
Price	€/year	200,000		(4)	

2. Off-gas

Table 3.1 Off-gas stream

Stream Name : EMISSIONS					
Comps	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**	(1)	(1) Estimated with AspenPlus (RSTOIC)
Methane	%wt	0	0		
Ethane	%wt	0	0		
Propane	%wt	0	0		
n-Butane	%wt	0	0		
n-Pentane	%wt	0	0		
Carbon Dioxide	%wt	13.73	13.73		
Carbon Monoxide	%wt	0	0		
Nitrogen Dioxide	ppm	11.9	11.9		
Nitrogen Monoxide	ppm	73.8	73.8		
Oxygen	%wt	2.27	2.27		
Nitrogen	%wt	73.15	73.15		
Water	%wt	10.85	10.85		
Total		100	100		
Process Condition and Price					
Temp	^oC	79.85			
Press.	Bara	1.30			
Phase	V/L/S	V			
Price		-			

3.5.5 Utility

Natural Gas

Table 3.11. Natural gas properties

Stream Table : **NGSTD**					
Comp	**Units**	**Specification**		**Notes**	**Additional Information**
		Available	**Design**	(1),(2)	(1) Contaminants not harmful for the
Methane	%-vol	81.3	81.3		(2) Delivery per pipeline
Ethane	%-vol	2.9	2.9		(3) Source [48]
Propane	%-vol	0.4	0.4		(4) Actually, natural gas should be considered as a feedstock rather than utility

Table 3.11 Natural gas properties (*continued*)

n-butane	%-vol	0.1	0.1		
C5+	%-vol	0.1	0.1		
Nitrogen	%-vol	14.3	14.3		
CO_2	%-vol	0.9	0.9		
Process Condition and Price					
Temp	oC	15.55			
Press	Bara	0.986923			
Phase	V/L/S	V			
Price	€/m^3	0.18 (ex. taxes)		(3),(4)	

3.6 Margin

Margin was calculated for the MHP, which is the option chosen for the design and also for CHP alternative in order to validate the figures presented in the previous feasibility study [22]. The margin is relatively high due to the fact that the ratio gas price for domestic use to gas price for industrial use is approximately 3. Figure 3.3 shows the gas prices for domestic and industrial use in The Netherlands [48]. Table 3.12 shows the results of the calculations.

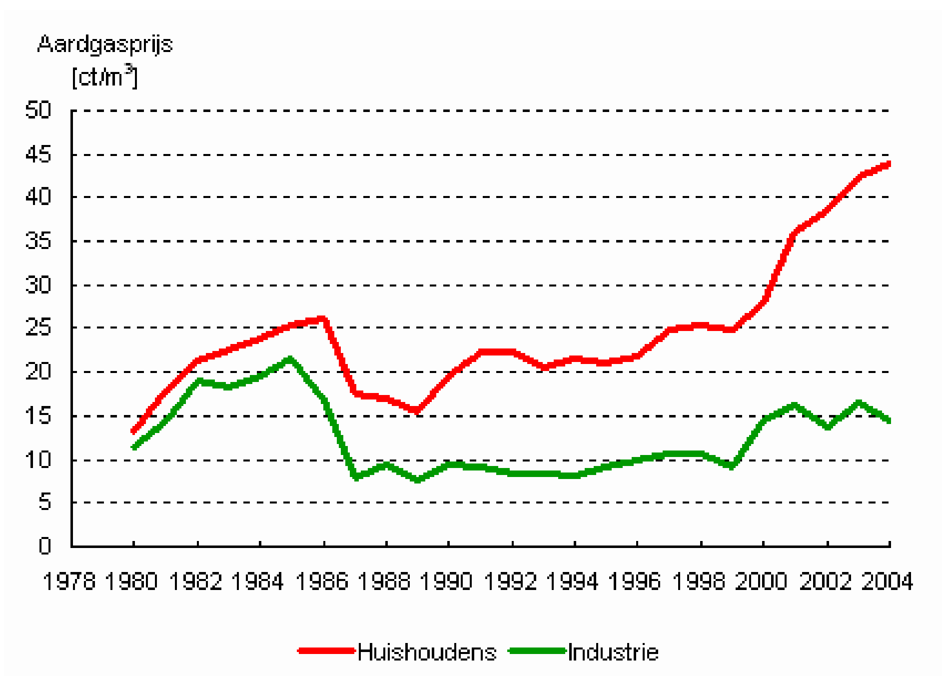

Figure 3.3. Prices for gas domestic (upper curve) and industrial usage

Table 3.12 Margin based on energy costs and revenues

		Chemical Heat Pump		Mechanical Heat Pump
In				
	gas	€	10,565,166	€ 7,994,508
	electricity	€	900,280	€ 2,314,534
Out	heat	€	11,923,148	€ 11,923,148
Margin		€	457,701	€ 1,614,106
		$	**546,379**	**$ 1,926,830**

Chapter 4 Thermodynamic Properties

Thermodynamic properties such as specific heats, enthalpies, vapor pressures, activity coefficients, reaction equilibrium constants, etc, are required in the simulation.

For the MHP simulation, Peng-Robinson Equation of State was chosen to be used for the ammonia, water, natural gas and air for the entire range of pressure and temperature. These components are very well known and all required data for steady-state simulation and other calculations are available either in the databanks, including Aspen databanks, or literature. Binary parameters are not required for most of the blocks used in the simulation because in the MHP the working fluid (ammonia) and water circulate in independent closed loops and they only exchange heat; i.e. these chemicals never get mixed. The only cases of mixing occur in the fired heater and the blower where the components of natural gas and air are mixed.

Regarding the Pipeline network simulation, the Steam-Tables thermodynamic model was selected to model the liquid water system, which operates at moderate pressure (>5 bar).

As for the CHP alternative, which had to be evaluated in order to validate the previous feasibility study [22], a simulation of this process was performed with AspenPlus. As far as thermodynamic models are concerned, the Wilson-Henry approach was selected for the vapor-liquid equilibrium and physical properties calculation in the simulation due to the low pressure of the system (1.0-1.5 bar), the polarity of acetone and isopropanol, and the supercritical behavior of hydrogen. Systems with low-molecular-weight alcohols are represented best by Wilson activity coefficient model, but with carbon atoms above 3 the superiority is less remarked [57]. Additionally, one of the articles that were taken as reference mentions the use of Wilson model for the simulation. The binary parameters for Wilson are available in the databank of AspenPlus. The Henry binary parameters for hydrogen-isopropanol were not available in the databank. In order to overcome this problem the same values for hydrogen-acetone were used since they exist in the databank. The impact of this approximation in the calculation was not evaluated, but it should not be high because acetone and isopropanol are both organic solvents, polar and do not differ too much in terms of molecular size. Figure 4.1 shows a P-xy diagram for the, isopropanol-acetone binary system in which values calculated with AspenPlus using the same thermodynamic model utilized in the simulation of the CHP process (Wilson model) are plotted along with experimental data retrieved from Dechema Series [58]. According to this plot a good agreement was obtained between both sorts of data. The experimental data from Dechema can be found in Appendix 34.

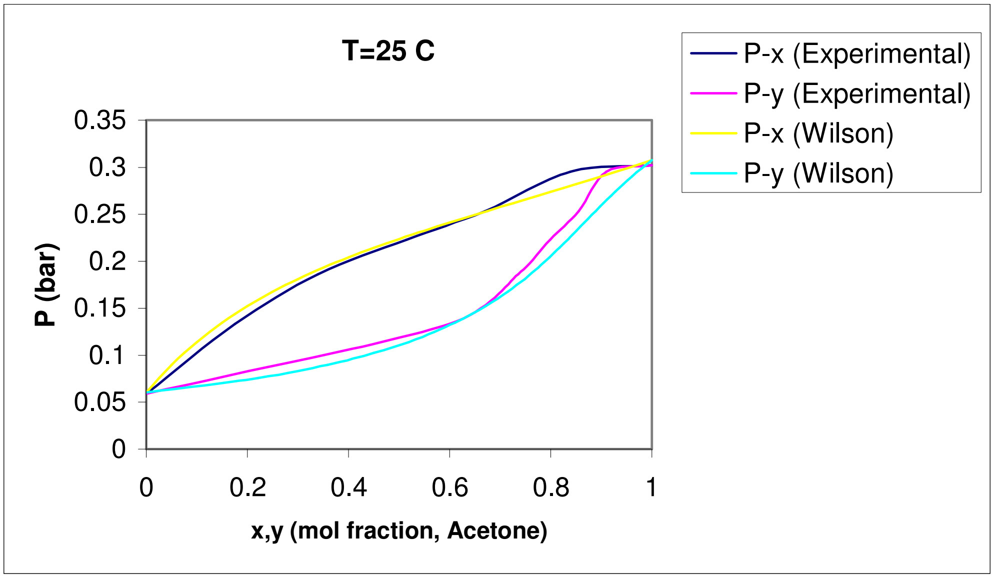

Figure 4.1 P-xy diagram for Isopropanol-Acetone at 25ºC

For additional calculations carried out in Mathcad, temperature-dependent correlation such as ammonia vapor pressure as a function of the temperature and water liquid specific heat as a function the temperature were used and their range of validity were properly taken into consideration (as can be seen in Appendix 2).

Chapter 5 Process Structure and Description

The process structure is comprised by the Upgrading Unit, where the heat is produced and transferred to the cold tap water, and the pipeline network, which is used for heat transportation to the districts.

5.1 District Size

Proceeding with the design of the mechanical heat pump to be used in the Upgrading Unit, some additional evaluations were carried out based on the annual demand/consumption of energy in The Netherlands. Data regarding the demand are shown in Appendix 14. The demand data was also employed as basis for the design of the delivery system and the operability and the controllability of the Upgrading Unit as well, because the design should cope with the fluctuations in the energy demand in the districts [60].

Just to get an idea of the district size and of what a demand for a district of 780 TJ/annum means, some calculations were performed in order to estimate the number of houses or buildings that correspond to the average energy demand in a typical Dutch DHS. Taking into consideration that the annual heat consumption of hot water and heating in a Dutch house is around 2200m^3 of natural gas, and such an amount is equivalent to 70 GJ [60], approximately 11000 houses can be supplied by the heating produced in a 780 TJ/annum capacity.

5.2 The Temperature of The Cold Tap Water Returning from The Districts

The temperature of the cold tap water returning from the districts is a function of the hot tap water temperature exiting the Upgrading Unit as well as of the heat losses in the pipeline network (both ways). It was preliminary assumed to be 60ºC. As it was stated in the BOD Report, the preliminary value (60ºC) had to be reviewed taking into consideration the design of the pipeline network, since the main focus at that phase had been the Upgrading Unit other than the pipeline network, which was thoroughly designed in the previous feasibility study [22]. The assumption before the BOD review had considered the heat losses in the pipeline network to be around 10% of the energy provided by the Upgrading Unit [26]. Some technical articles mention this percentage as being 8.6% [18].

However, a further evaluation based on the historical data regarding the daily and monthly demand fluctuations of DHS in The Netherlands [60] led to a more accurate estimation of this temperature. Moreover, the calculations of the heat losses in the pipelines revealed that the temperature of the cold water entering the Upgrading Unit is approximately 67ºC.

Based on the temperature profile, the mass flow rate of tap water in the MHP was calculated and it was used as the basis for the steady-state simulations. Actually, these calculations are done iteratively, i.e. a temperature profile is set, the heat losses are estimated and a new temperature profile is generated. This procedure is repeated until the last and the former values of temperature differ within an acceptable tolerance. Detailed calculations are present in the Appendices 8 and 19.

5.3 The Temperature of The Hot Tap Water Leaving The Upgrading Unit

The pipeline network has to be calculated taking into consideration the maximum and minimum capacities of the distribution system. In the preliminarily calculations, the tap water flow rate was calculated based on inlet and outlet streams temperatures of 68ºC and 92ºC, respectively. However, the results of the pipeline preliminary calculations (Appendix 6) revealed that the diameter of the main pipe (the header between the Upgrading Unit and the districts) is higher than 914mm (36inch), which is the maximum commercial diameter available in the pipeline data bank such as Crane. It is important to point out that the insulation thickness was not taken into consideration yet. It sounds very strange to design a DHS pipeline network like that. Additionally, as it was stated in the previous feasibility study, the pipeline infrastructure should be located next to the municipal road because the municipality owns the road and it wants to protect the green area. In addition to that, the glass fiber infrastructure, which is expected to be implemented in the next few years [22], is located in the same area and maybe they can be combined. In this case, it would be practically impossible to use an infrastructure such as the one that is presented in the Figure 5.1., in which a typical pipeline arrangement of a Dutch DHS is shown [85].

The results of the demand fluctuations and operability revealed that transporting heat by means of hot tap water to districts keeping low temperature difference between the Upgrading Unit and the districts (92ºC-90ºC) would lead to extremely high costs (huge pipeline diameters), besides the other disadvantages mentioned above. Hence, it was decided to transport lower water flow rate with higher energy content (higher temperature) other than high flow rate with low energy content.

The best way to accomplish this goal is to transport hot tap water (~350 m^3/h) at approximately 137ºC through the pipelines to heat exchangers located nearby each district in the so called Heat Stations. In these heat exchangers heat is removed from the hot tap water stream to a tap water stream that runs internally in the specific district and is totally independent on the pipeline network designed in this CPD, the so called district tap water. The tap water cycle in each district is outside the battery limit specified for this project. The heat exchangers are designed so that the temperature of the district tap water exiting the Heat Station is 90ºC, which is a requirement established by the Principal. The final simulation of the pipeline network and the heat losses was conducted based on this concept; that is, transporting water at higher energy content. Calculations can be found in Appendix 8.

Figure 5.1 Pipeline infrastructure in a DHS in Witbrand, Tilburg, The Netherlands

5.4 Fluctuations and Operability

The waste water is continuously supplied by DSM at temperatures between 25ºC and 30ºC with a flow rate of approximately 2.5 million m^3 per annum (~291.8 t/h). Considering an average input temperature of 27.5ºC, in order to achieve the energy content of approximately 100 TJ/annum, the outlet temperature should be 18ºC.

The Upgrading Unit has to supply the districts with 780 TJ/annum. The desired temperature of the hot tap water delivered at the districts should be 90ºC according to information provided by the Principal. The tap water flow rate was calculated in order to satisfy this amount of energy along with the tap water inlet and outlet streams temperatures, which were taken as 67ºC and 137ºC respectively. In this case, the Upgrading Unit would be designed to generate 923 TJ/annum so that heat losses in the pipeline network would be compensated and the districts would be supplied with 780 TJ/annum.

Although a maximum annual demand of 780 TJ at the districts was assumed, it is important to take into consideration that this demand is not constant and obviously depends very much on the season and the time of the day. Thus, there are big fluctuations in both monthly and hourly demand and they should be taken into consideration. The design was made for the average heat consumption, i.e. assuming a constant heat demand (approximately 371 m^3/h of hot tap water or 0.09 TJ/h) and adjusting the design afterwards with an over design factor to properly satisfy the monthly and hourly peaks. Figure 5.2 presents the monthly heat consumption pattern in The Netherlands, whereas the daily demand distribution is illustrated by Figure 5.3. Appendix 14 presents the detailed calculations.

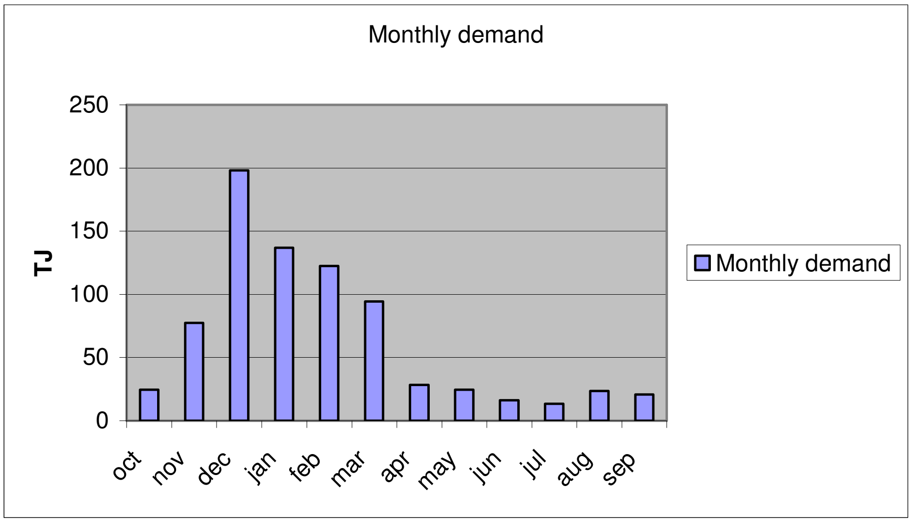

Figure 5.2. Monthly heat consumption pattern

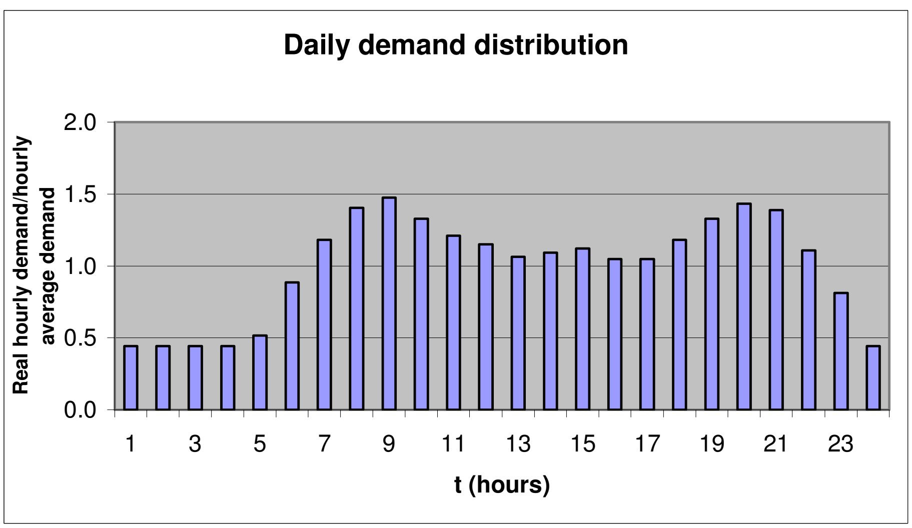

Figure 5.3. Daily demand distribution

In order to satisfy the hourly and monthly peaks, two options were considered as follows:

Option 1. Buffer tanks

The first option to overcome these fluctuations was by means of two buffer tanks, in addition to an over designed fired heater. Based on Figure 5.1, during the month with the peak consumption, the heat demand is as 3 times as much as the average monthly demand. Therefore, the fired heater should be over designed by a factor of 3.
Moreover, the volume of buffer tanks should be enough to store the energy of usage above the average demand, which is approximately the accumulation from 07:00 till 22:00 o'clock. According to figures presented by Figure 5.2, two buffer storage tanks with capacity equivalent to 3.57 hours each are needed. It means a volume of 3.57 times the hourly average flow rate of a certain day a year. In the worst case, the monthly demand is the highest. Therefore, it is 3 times as much as the average monthly demand, and this all leads to a total volume of approximately 4200 m^3. Designing a tank for 4200 m^3 and assuming H/D ratio of 2.5 yields a tank with a height of 32 m and a diameter of 13 meter. The use of buffer tanks will also provide the system with flexibility, especially in case of maintenance. The maximum time available for maintenance (tank volume over the lowest flow rate) is approximately 50 hours. Appendix 3 shows the tank dimensions calculations.

Option 2. Fired Heater

Another option, which is likely a better way to overcome the above average usage, is to increase the capacity of the fired heater by a higher factor, but now with respect to the hourly fluctuations, as well as the daily fluctuations. The hourly peaks are not that big relatively to monthly peak; only a maximum factor of 1.48. Therefore, the fired heater capacity should be increased in order to overcome both monthly and daily fluctuations. It was found that, this is the best option due to the fact that the buffer tanks are big as shown before. This is a problem because in this area space is not widely available, which makes it an expensive solution. Other big issue is related to foundation for the tanks.

Therefore, it was decided to use a fired heater, with an over capacity with a factor of 5, to compensate monthly fluctuations, hourly fluctuations and serve as a back-up system for the ammonia loop. It should be mentioned that fired heater are flexible equipments, in the sense that fired hater are provided with turndown ratio up to 50:1. Furthermore, if the demand of the district is lower than the capacity of the MHP, the unloading/loading capacity control of the compressors will decrease the ammonia flow in the MHP.

The disadvantage of this option is the presence of a fired heater that is not efficiently used; only 20% of its capacity is utilized over the year. The idleness of the fired heater is higher during the summer time, when only approximately 5% of the capacity is used. That is why the majority of DHS's in Europe are combined with a power station, the so called combined heat and power (CombHP).

Since the average heating demand is approximately 9 times as much as the average energy input from DSM waste heat stream and the peak demand is approximately 25 time as much as this value, it is impossible to avoid the usage of a gas-fired heater to overcome this situation.

Given the required plant capacity and the results from the evaluation of the demand fluctuations and studies on heat losses in the pipelines, the outputs of the design were mostly established. As a consequence of the heat losses calculations in the pipeline network, the Upgrading Unit has to be designed for 923 TJ/annum in order to compensate the heat losses on the network. The tap water flow rate was calculated in order to satisfy this amount of energy along with the temperatures the hot and cold tap water, 67ºC and 137ºC respectively.

5.5 The Upgrading Unit

The Upgrading Unit is a room equipped with the MHP equipment (ammonia thermal cycle), the atmospheric gas-fired heater, the air blower, the hot tap water pump and all associated equipment, utilities and infrastructure.

High-pressure compressors raise the maximum achievable condensing temperature of ammonia from 58ºC to 78ºC [28]. The maximum discharge pressure of a compressor corresponds to a maximum temperature of 190ºC according to heuristics presented in [1]. In order to avoid reaching the maximum allowable discharge temperature, the two-stage compressor design with intermediate cooling system is used. Appendix 4 shows the calculation of the compression ratio and the number of compressors required. Two-stage compressor is preferred due to the fact that it needs less power than a one-stage compressor and, as a result, it performs better. Two-stage compression combined with intermediate gas cooling lowers discharge temperature and the pressure ratio in the cylinders and, consequently, oil carry-over is reduced and the overall efficiency is improved. Thus, the intercooler between the two-stage compressors reduces the gas temperature between the low-pressure and high pressure compressors. Figure 5.4 presents a simplified scheme of a two-stage compressor with intermediate cooling.

In Appendix 5 an illustration of a two-stage compressor unit with intermediate gas cooling system is shown. In the gas cooling system the hot gases from the discharge of the first compressor get cooled by the working fluid down stream the condenser in a heat exchanger. The main advantage of this concept is that the whole cycle is arranged in a very compact unit. However, such a concept is not useful in this CPD due to the fact that heat is removed from the compressor discharge to heat up part of the cold tap water as a matter of heat integration.

An alternative to the use of an intercooler is the presence of a receiver vessel, which would be placed in the system between the compressors to reduce gas temperature between the low-pressure and high-pressure compressors by means of and expansion valve. Such a vessel separates liquid and vapor phases of a two-phase working fluid flow. This equipment has a controlled temperature/pressure as well as liquid level.

Although the increase in temperature of the cold tap water in the intercooler is only 0.5ºC, the COP of the cycle with the intercooler is around 1.2 as much as in the cycle with a receiver. The heat released with the intercooler is higher than with receive, since in the receiver heat is lost to get two-phase flow.

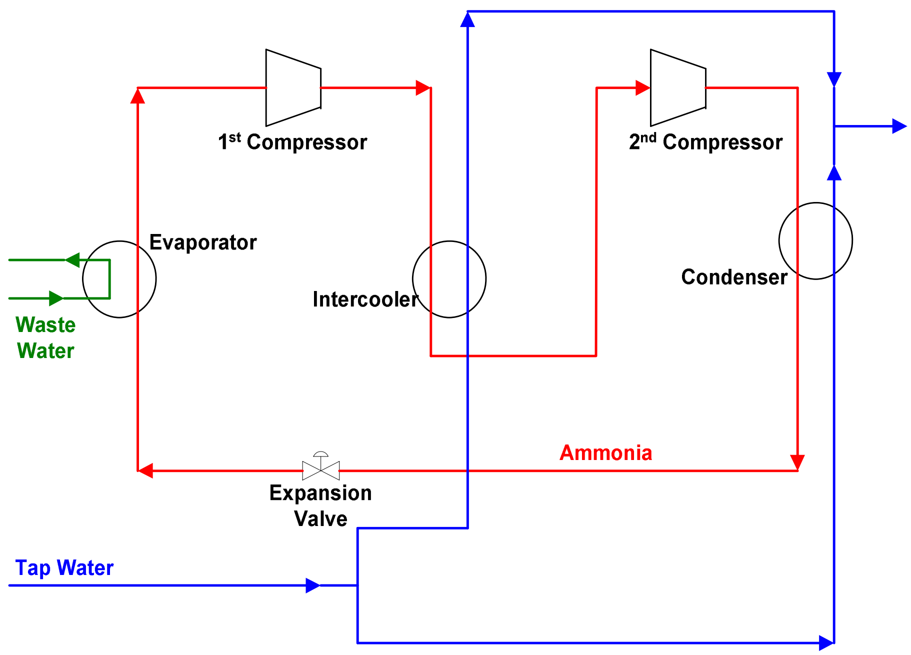

Figure 5.4 MHP with two-stage compressor with intermediate cooling

5.6 Process Description

The mechanical heat pump designed by this CPD team follows a conventional concept of thermal cycle using ammonia and consists of the following main pieces of equipment: one evaporator, two reciprocating compressors, one intercooler, one condenser and one expansion valve.

The process flow scheme (PFS) and the stream summary tables for this process are shown in the Appendices 11 and 12, respectively. The residual waste water, WWIN<1>, enters the Upgrading Unit at 27.5ºC. It represents an energy input of 100 TJ/annum to the cycle. This stream releases heat to evaporate ammonia in the ammonia evaporator (E01). The cold waste water outlet stream from this evaporator, WWOUT<2>, at 18ºC is further sent to final disposal per pipeline in the North Sea. The boiled ammonia, COMP1IN<3>, at 6.5 bar and 12ºC, is then compressed to a higher pressure and temperature in two stages. In the low-pressure compressor (K01) ammonia vapor is compressed from 6.5 to 15 bar, and the outlet gas, COMP1OUT<4>, enters the intercooler (E02) at 93.2ºC and transfer heat to a cold tap water stream, WCOOLERIN<10>. In the high-pressure compressor (K02) ammonia downstream the intercooler (E02), COMP2IN<5>, is compressed to 38 bar. This hot gas at 179.3ºC, COMP2OUT<6>, enters the ammonia condenser (E03) where it is cooled down to 75.6ºC and the condensation heating is used to preheat the cold tap water, WCONDIN<11>, which leaves this equipment, WCONDOUT<13>, at 77.6ºC. The liquid ammonia, ACONDOUT<7>, is then sent back to the ammonia evaporator (E01) through an expansion valve, which regulates between high and low pressure. The expansion valve outlet stream, EVAPIN<8>, at 6.6 bar and 12ºC enters the ammonia evaporator (E01) and closes the cycle. A collector vessel (V01) located downstream the ammonia condenser (E03) is employed to collect liquid ammonia only in case of shut-down.

The cold tap water that returns from the districts, UUIN<9>, at 66.5ºC and 9 bar is split into two streams. One of them is sent to the intercooler (E02), WCOOLERIN<10>, whereas the other one, WCONDIN<11>, flows to the ammonia condenser (E03). Afterwards, the preheated tap water outlet streams from the intercooler (E02) and ammonia condenser (E03) are mixed and create a preheated cold tap water stream, FURNACEIN<14>, at 77.8ºC. Thus, the ammonia cycle adds only 11.3ºC to the temperature of the cold tap water that enters the Upgrading Unit.

The desired temperature (137.2ºC) of the hot tap water, UUOUT<20>, which is sent to the districts through the pipeline network is finally achieved after heating up the cold tap water stream, FURNACEIN<14>, in an atmospheric fired heater F(01) using natural gas. Natural gas, NGSTD<16>, is fed into the fired heater (F01) to be burnt with air generating the required heat to heat up the preheated cold tap water, FURNACEIN<14>, from 77.8ºC to 137.2ºC. Combustion air, AIRIN<17>, is sucked from the atmosphere and feeds a blower (K03). The discharge of the blower, AIROUT<18>, is fed into the fired heater (F01). The off-gas, EMISSIONS<19>, from the fired heater (F01) is released to the atmosphere through the stack. The hot tap water, FURNACEOUT<15>, is then sent to a pump (P01) that delivers hot tap water (UUOUT<20>), at 10 bar, to the distribution system.

The hot tap water exiting the Upgrading Unit at 137.2ºC and 10 bar, UUOUT<20>, is able to provide the district with 780 TJ/annum and the fired heater (F01) has the capacity to be used also as a backup system (over design: 500%). The simulation of this system was carried out with AspenPlus. Other additional calculations were performed with Mathcad and are present in the Appendices 2, 4, 8 and 10.

5.7 Pipeline network

The heat transportation to the houses will be done by means of hot tap water through pipes. This network will be made of pre-fabricated carbon steel pipes insulated and protected by a polyethylene shield. Carbon steel pipes were chosen due to the large range of applications regarding temperature and pressure levels. The pipes will be under grounded (80cm).

Five district areas among those listed as likely to be connected in the previous feasibility study [22] were chosen in order to make a draft configuration of the distribution system and the tap water pipeline network. Based on the map of the municipality of Delft, five major branches of the main pipeline were identified and their respective lengths were defined. Figure 5.5 shows the estimated perimeter covered by the pipelines [74].

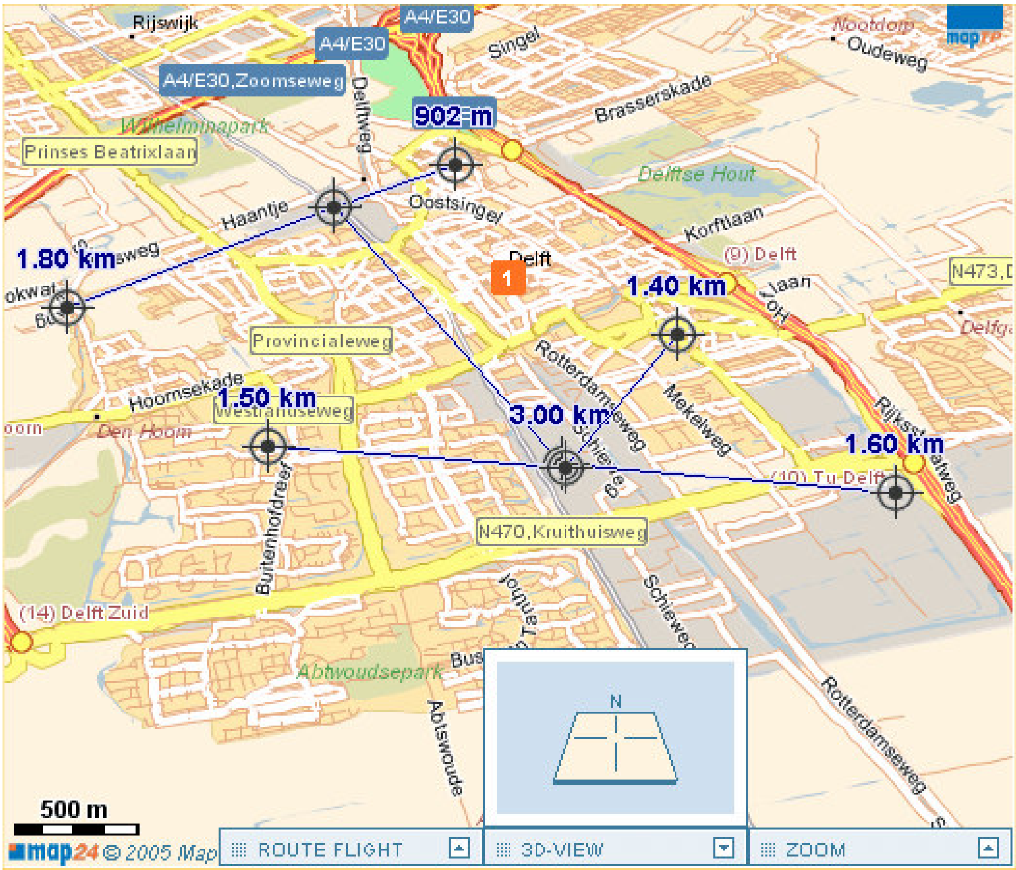

Figure 5.5 Estimated perimeter covered by the pipelines.

The heat distribution system comprises a branched pipeline network for distributing the heat generated in the Upgrading Unit to the consumers in the districts and Heat Stations (HS) where the heat is transferred, by means of shell-tube heat exchangers, from the hot tap water to the cold water, which is part of the internal water cycle of each specific district. The Heat Stations are located as closed as possible to the district and consist of a room with a heat exchanger, valves, fittings and instrumentation items. The water cycle in the districts is out of the scope of this CPD.

The hot water stream exiting the Upgrading Unit will be split into three branches in the Central Grid One (CG1), which is located at the exit of the Upgrading Unit. The highest flow rate is sent to the Central Grid Two (CG2), while the other two sub-streams are sent to the heat exchangers HEX1 and HEX2. In CG2, which is located at approximately 3000m from the Upgrading Unit, the hot water flow rate is equally split into three additional branches which supply hot water to heat exchangers HEX3, HEX4 and HEX5. These five heat exchangers are located in the heat stations mentioned above. In such configuration, the total hot water flow rate is divided by five in order to supply each heat exchanger with the same flow rate of hot water. The cold water pipeline returning from the heat station to the Upgrading Unit is located in parallel with the hot water pipeline and the cold water flows inversely compared to the hot water flow. Both cold and hot water pipelines form the whole pipeline network which is used in this DHS. The total length of the pipeline network is approximately 10,200 m. Figure 5.6 shows a simplified block diagram of the pipeline network.

Given the length and the flow rates of both hot and cold tap water corresponding to each branch, the diameter of each pipe was calculated with AspenPlus by means of the block "Pipe". The diameters of the pipes are 333 mm and 381 mm. The calculated pressure drop and average velocity in the pipes were compared with the values established be rules of thumb [62] and they were found to be in within the acceptable ranges. Based on the pressure drop in the pipelines the hot water pump was designed. The pump and pipes were designed for the maximum water flow rate, which takes into consideration demand fluctuations.

The hot tap water exits the Upgrading Unit at 137.2 ºC and 10 bar (13 bar, peak value), UUOUT<20>, and the cold tap water enters the Upgrading Unit at 66.5ºC and 9 bar (6 bar peak value) UUIN<9>. The results of the pipeline calculations are shown in Table 5.1. The values of average velocity and pressure drop presented in this table are small because they were calculated taking nominal flow rates (steady-state) circulating through pipes whose diameters were designed for the peak demand. The flowsheet diagram from AspenPlus simulation is presented in Appendix 24 and the electronic version of the simulation is available in a CD rom.

During the detailed engineering design phase a comprehensive study should be carried out in order to evaluate the possibility of operating the pump with different impellers according to the peaks in the monthly demands. It will save electrical energy, which is spent in the motor of the pump. Such a study can only be performed with more accurate information regarding the length of the pipes, which is directly linked with the pump capacity.

As for insulation material, polyurethane is largely used in districts heating systems particularly for reliability, efficiency and durability reasons. Actually, most pre-insulated systems for low temperature (below 120ºC) employ polyurethane (PUR) insulation. However, its thermal conductivity (0.36 Watt/(m.K)) is pretty high compared to other materials such as glass wool (0.041 Watt/(m.K)) and Micro-PUR (0.031 Watt/(m.K)). Micro-PUR is a type of polyurethane foam with the pores filled with cyclopentane gas, which has been used for insulating pipeline network in DHS's in The Netherlands [63]. This material is capable of preventing heat loss and, as a result, it keeps the temperature in the required levels and is applicable for several pipe sizes. The maximum temperature for employing Micro-PUR is the same as for PUR, i.e. ~135ºC [84], but it is 33% more expansive than PUR. Shield made of plastic material like polyethylene is commonly used as casing for polyurethane foam insulations.

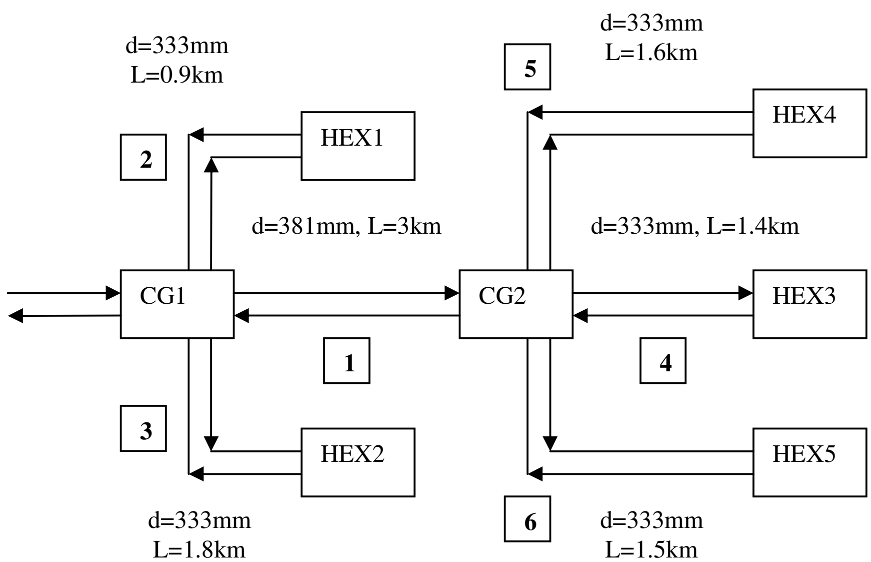

Figure 5.6 Simplified block diagram of the pipeline network

Table 5.1 Pipeline network calculation results

Branch	Diameter (mm)	Length (km)	Average Velocity (m/sec)	Pressure Drop (bar)
To HS 1	333 (14”)	0.9	0.24	0.01
From HS 1	333 (14”)	0.9	0.23	0.02
To CG 2	381 (16”)	3.0	0.55	0.15
From CG 2	381 (16”)	3.0	0.52	0.15
To HS 2	333 (14”)	1.4	0.24	0.02
From HS 2	333 (14”)	1.4	0.23	0.02
To HS 3	333 (14”)	1.6	0.24	0.02
From HS 3	333 (14”)	1.6	0.23	0.02
To HS 4	333 (14”)	1.8	0.24	0.02
From HS 4	333 (14”)	1.8	0.23	0.02
To HS 5	333 (14”)	1.5	0.24	0.02
From HS 5	333 (14”)	1.5	0.23	0.02

As it is stated in the previous feasibility study [22] the pipeline infrastructure should be located next to the municipal road because the municipality owns the road and it wants to protect the green area. In addition to that, a glass fiber infrastructure will be located in the same area and maybe they can be combined.

Heat losses will take place throughout the pipeline. Thus, a trade-off between the temperature of water and the level of insulation has to be established. Since the exact number of districts that will be connected is currently unknown, the material, length and the insulation thickness have to be more accurately studied and specified in a further detailed engineering phase of the project. Moreover, pipes and pumps inside the districts as well as insulation in the individual houses are not included in this design.

Heat Losses in Pipeline Network

Inefficiency in water heating systems is primarily caused by inefficiency of the heat equipment and by heat losses in the distribution pipes. Therefore, while transporting the energy no major losses should occur. There is a trade-off between costs and insulation of the pipeline, the temperature of the water delivered to the districts and the energy losses.

The heat losses of both hot tap water and cold tap water were estimated via heat transfer laws, which comprise conduction and convection heat transfer methods. The insulation and the soil resistances to heat transfer were taken into consideration, whereas the resistance to heat transfer in the film as well as in the pipe wall were considered to be negligible. The calculations are given in Appendix 19.

The pipeline network was assumed to have a total length of 10,200m. Micro-PUR was found to be the best insulation material because it has low thermal conductivity, it is suitable for this service and it is available in the market. The thickness is assumed to be 5cm and the pipes will be 80cm under grounded. Figure 5.7 shows the heat losses per length unit for different types of insulation taken the hot tap water conditions as basis. According to this plot, the heat losses for polyurethane are approximately 5 times as much as the ones calculated for materials such us Micro-PUR and glass wool. In terms of temperature difference between hot tap water leaving the Upgrading Unit and the temperature of hot tap water at the inlet of each Heat Stations, polyurethane leads to approximately 5ºC whereas Micro-PUR and glass wool leads to 1ºC, as it is shown in Figure 5.8.

Figure 5.9 illustrates the influence of insulation thickness in the heat losses in the pipelines. It shows that in order to achieve the same level of temperature difference as Micro-PUR, insulation with PUR should have thickness higher than 30 cm. It means that costs of the insulation would be higher and higher depth would be required to lie down the pipeline, which would lead to higher construction costs.

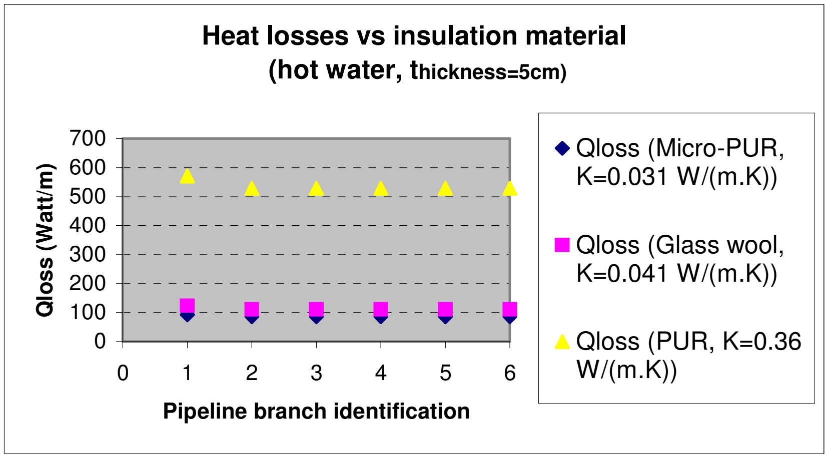

Figure 5.7 Heat losses per length unit for different types of insulation
(Pipeline branch identification refers to different pipelines in Figure 5.6)

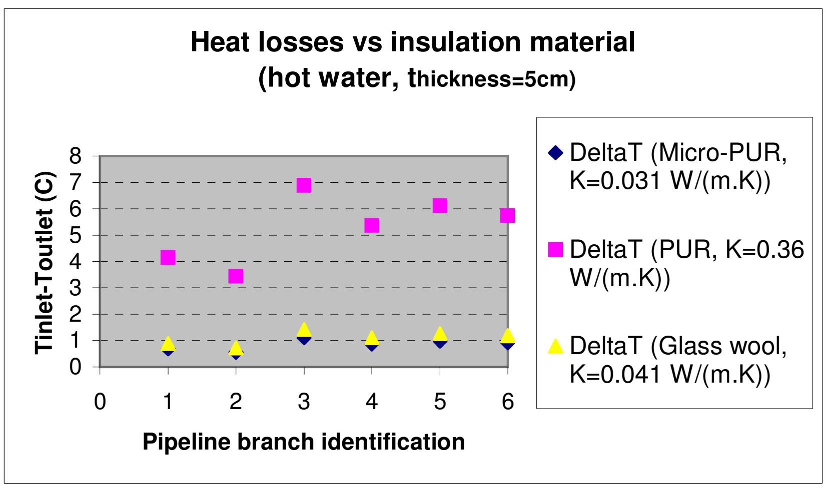

Figure 5.8 Temperature difference due to heat losses for
different types of insulation
(Pipeline branch identification refers to different pipelines in Figure 5.6)

Depending on the insulation material and thickness, and also on the pipeline length, the energy losses in the pipeline network usually vary within the range 8-10% of the energy provided by the Upgrading Unit [18],[26]. Table 5.2 presents the results of heat losses calculation for the hot tap water pipes in the pipeline network.

A summary of insulation material and energy savings is shown in Table 5.3. Different insulation materials are shown in the table in order to illustrate the effect of the thermal conductivity in the energy savings. According to this table, the Upgrading Unit can be designed for 813 TJ/annum if Micro-PUR is used, whereas with PUR the capacity is 923 TJ/annum. Therefore, when Micro PUR is used instead of PUR one will safe 110 TJ on a yearly basis, this equals to 12 %.

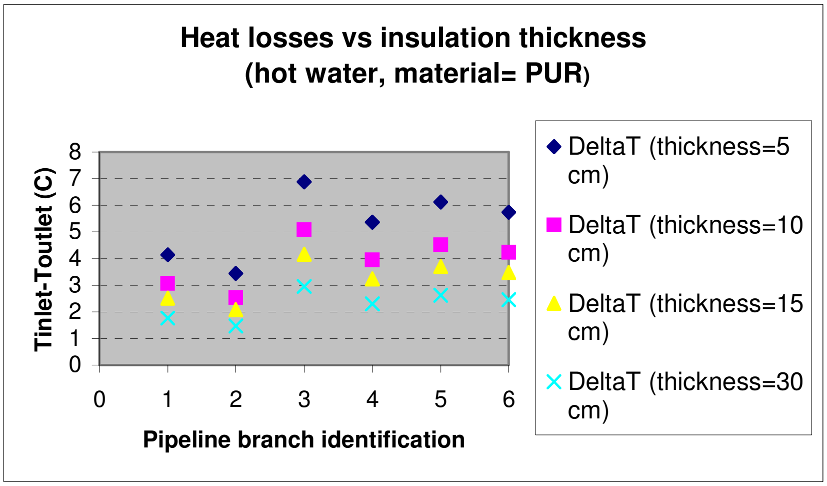

Figure 5.9 Temperature difference due to heat losses for
different thickness of polyurethane insulation
(Pipeline branch identification refers to different pipelines in Figure 5.6)

Since the lengths in the pipeline network were roughly estimated, the heat losses calculations were performed using Micro-PUR, but assuming 923 TJ/annum as the capacity so that inaccuracies with respect to distances between the Upgrading Unit and the districts are somehow compensated. Even though the maximum temperature for Micro-PUR application is approximately 135ºC, there is no technical constraint in using it for this design because the hot water temperature will be ~132ºC as shown in Table 5.3, instead of 137ºC, which was used as basis for the design. Moreover, as the design capacity was taken as 923 TJ/annum, the flow rate of hot tap water can be adjusted so that temperature lower than 137ºC is reached. Detailed calculations can be found in Appendix 15. A block diagram containing the temperature profiles in the pipeline network is presented in Appendix 16.

5.8 Exergy Losses

Exergy (availability or available energy) is a property used to determine the useful work potential of a given amount of energy at some specific state. It is important to realize that exergy does not represent the amount of work that a work-producing device will actually deliver upon installation. It represents the upper limit on the amount of work that a device can deliver without violating any thermodynamics law.

Table 5.2 Heat and exergy losses of pipes insulated with Micro-PUR (k=0.031 Watt/(m.K)), 5cm thick, 80cm under grounded, ground surface temperature 15ºC.

Pipe diameter (mm)	Pipe length (m)	Heat Loss Hot water stream (kWatt)	Exergy Losses Hot + Cold water streams (kWatt)
381	3000	285.8	85.9
333	900	766.2	22.5
333	1800	153.2	45.1
333	1400	119.2	35.1
333	1600	179.0	40.1
333	1500	167.9	37.6
Total		1671.3	266.3

Table 5.3 Summary of insulation material and energy savings

	Units	PUR	Micro-PUR
Input from Upgrading Unit	TJ/annum	923	813
Hot tap water temp. exiting the Upgrading Unit	ºC	137	132
Cold tap water temp. entering the Upgrading Unit	ºC	67	69
Delta T (in-out)	ºC	71	63
Percentage heat losses	%	15	4

Exergy balance can be stated as the exergy change of a system during a process. It is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of entropy generation (irreversibility). Exergy destroyed is proportional to the entropy generated. Irreversibility such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, non-quasi-equilibrium compression, or expansion always generate entropy, and anything that generates entropy always destroys exergy. Since chemical reaction is a significant source of thermodynamic inefficiency, it is advisable to minimize the use of combustion where combustion is necessary, the use of excess air should be minimized, and the reactants should be pre-heated. Irreversibility due to friction or unrestrained expansion of gases is generally secondary in importance to those in combustion and heat transfer.

In the entire DHS process (Upgrading Unit + pipeline network) the primary contributors to exergy destruction are: the heat exchangers at the Heat Stations (heat transfer), the fired heater (mixing and combustion), and the control valves and the pipelines (friction and expansion during transportation). The exergy losses in the heat distribution system (pipelines + heat exchangers) were estimated for the hot and cold tap water pipes and for the heat transfer in the Heat Stations as well, considering Micro-PUR insulation. The following sources of exergy losses were identified:

- Heat losses in the pipeline network;
- Water transportation (electrical energy is used in the pump to generate the water flow rate and this energy is used to overcome the flow resistance in the pipeline network, through which is transformed into heat [64]);
- Irreversibility during heat transfer in the heat exchangers at the Heat Stations.

Total exergy available in the hot tap water exiting the Upgrading Unit is 40940 kWatt. Table 5.4 presents the results of the calculations, which can be found in Appendix 21. The exergy losses in the heat exchangers at the districts represent 98% of the total exergy losses in the whole distribution system. Exergy losses in the heat exchangers cannot be avoided, since it is intrinsically linked to purpose of the equipment, i.e. transfer heat to the district tap water. These results demonstrate that exergy losses can be decreased by reducing the consumption of electrical energy during the transport of hot tap water and by avoiding head losses in the pipeline network. Since there is not much room to manipulate with respect to the pump power, the heat loss in the pipeline is the most important variable to be controlled so that exergy losses are reduced. Therefore, pipe characteristics (diameter and length), insulation material and insulation thickness are the most important factors to be considered with regards to the pipeline design.

Table. 5.4 Exergy losses in the heat distribution system

Source of exergy losses	**Exergy losses (kWatt)**	**Exergy losses of total losses (%)**
Water transportation	86.3	0.45
Heat exchangers at districts	18,917.0	98.17
Heat losses in the pipelines	266.3	1.38

Energy and exergy losses in the distribution system are directly related to natural gas consumption and, as a result, tremendously affect the economic advantages of heating systems. Therefore, heat losses in the pipeline network should be reduced as much as possible.

5.9 Thermodynamic Analysis of The Process

Given the conditions mentioned in the process description and present in the PFS, Stream Tables and Equipment Spec Sheets, the expected energy parameter, the Coefficient of Performance, for this heat pump system is 3.06.

COP, the enthalpy efficiency of the system, is a general characteristic of heat pumps. It is defined as the ratio of useful heat output (heat released at high temperature in the condenser) to energy input (work in the compressors). It is important that the distribution systems be properly balanced and designed so that the system operates at the lowest temperature as possible. High values of COP are desired and may be achieved by minimizing the compressor work input for given values of heat transfer.

The exergy efficiency, η_E, for this heap pump is 0.592. It is a parameter used to evaluate quality and quantity of the upgraded heat. Calculations of COP and exergy efficiency are shown in Appendix 22.

From the energy conservation standpoint, the thermodynamic efficiency of a process should be as high as possible and the entropy generation or lost work as low as possible. The aim of thermodynamic analysis is to determine how efficiently energy is used or produced and to show quantitatively the effect of inefficiencies in each step of the process. The cost of energy is of great concern in any process, and the first step in any attempt to reduce energy requirements is to determine where energy is wasted through process irreversibility.

The thermodynamic analysis of a specific process shows the locations of the major inefficiencies, and thus the pieces of equipment or steps of the process that could be changed or replaced in order to overcome such inefficiencies.

Actually, such an analysis shows that the present design is wasteful of energy and that there is room for enhancement. Table 5.5 shows the results of entropy generation calculations, which can be found in Appendix 20. It can be seen from the figures in this table that the expansion valve and the fired heater have high entropy generation rates (irreversibility) without producing work. The compression is also an irreversible process. Expansion through a valve is an isenthalpic process creates entropy due to fluid friction in the valve, which results in pressure drop.

The efficiency of the ammonia thermal cycle could be improved if the compression was carried out reversibly and the expansion valve was replaced by an expander (an expansion turbine) that would perform the expansion isentropically and work would be generated. In this case the thermal cycle would be represented by reversible processes and would approach the ideality, i.e. Carnot cycle concept. The work produced by the expansion could be used to generate electrical energy. Besides simplicity, the biggest advantage of an expansion valve is low cost. That is why they are mostly used in small capacity units. Turbines are complex rotating machines and are employed for larger capacities.

Under the process conditions of the ammonia thermal cycle in the Upgrading Unit, an expansion turbine would produce approximately 481 kWatt and this amount of energy would save around US$640,000 per year in the electric power input to the compression work. This amount represents savings of 3.2% total product. The purchase cost of such a turbine is approximately US$ 311,000.

In this design the decision was made taking into account two factors: reliability and safety. Rotating equipment is subject to mechanical problems and requires more maintenance than a simple expansion valve, which would lead to regular shut-down periods. With respect to safety, there is no information on using ammonia in a turbine, maybe due to high risk of leak. The existing MHP with ammonia in Norway [23] has capacity of 28 TJ/annum (28% of capacity of this MHP) and uses an expansion valve in at site that is located in a military area. The Upgrading Unit will be located in a civilian industrial area nearby a rather populated neighborhood and safety is an issue of concern.

Table 5.5 Entropy generation in the process within the upgrading unit

Equipment	**Entropy generation (Watt/K)**	**Entropy generation (%)**
Pump	1.94	0.05
Expansion valve	532.93	14.13
Compressors	918.04	24.35
Fired heater	2318.00	61.47
Total	3770.91	100

Chapter 6 Process Control

In every chemical plant, there are some basic principles that guide the operation of the processing units. These basic principles include:

- Safety in the operation of the processing unit;
- Achievement of the specified production rates;
- Achievement of product quality specifications.

Since there are always some disturbances, which change the operating conditions, the process has to be controlled. Therefore, the process control is needed in order to compensate or minimize these disturbances. In practice not all disturbances are measurable and in some cases the process characteristics are not known exactly. In a well-design system, typically 90% of the corrective action is provided by feed forward and 10% by feedback [62].

6.1 Process Control of The Upgrading Unit

In the design of the Upgrading Unit of residual industrial heat for a DHS, the process control is needed to maintain the operating conditions (flow rate, pressure, temperature and composition) of the ammonia stream (used in the mechanical heat pump cycle), the combustion of natural gas, and the tap water streams (hot and cold) which will be used to distribute the heat to the district area. The control loops are shown in the Process Flow Scheme (PFS) is shown in Appendix 1.

A brief description of the process control for the equipment in the process is presented below:

Ammonia Evaporator: this heat exchanger evaporates ammonia with waste water stream. The evaporation rate has to be controlled in order to ensure that the evaporation process goes properly. The evaporation rate is controlled by ammonia level control in the evaporator. Moreover, the evaporator pressure is controlled in a cascade control manner with level control by manipulating the expansion control valve.

First Compressor: This equipment is a reciprocating compressor and increases the ammonia pressure from 6.5 to 15 bar. The inlet and outlet stream of the compressor have to be monitored in terms of pressure. Therefore, pressure controllers are required for suction and discharge. These controllers are linked to a servo-mechanism which is part of the internal control system of the compressor.
As far as the compressor capacity control system is concerned, it depends on the operating conditions, the capacity of the compressor and also on the working fluid. It is based on unloading valves, a mechanism whereby some of the suction valves remain open during discharge. Solenoid or pneumatic unloaders can be operated from the output of a control instrument. The stepwise controlled external flow rate may

need to be supplemented with controlled external by-pass to smooth out pressure fluctuations.

Intercooler: This heat exchanger cools down the discharge of the first compressor. Ammonia temperature is controlled by manipulating the cold tap water flow rate. A cascade control system is used on the cold tap water stream going into the intercooler. The flow controller of the cold tap water stream, which gets its set point from temperature control, will ensure a proper heat exchange since the temperature in the discharge of the second compressor should not exceed 190ºC.

Second Compressor: This equipment is also a reciprocating compressor and increases the ammonia pressure from 14.5 to 38 bar. Similarly to the first compressor, pressure controllers are required for suction and discharge control system.
Like in the first compressor, the capacity control system depends on the operating conditions, the capacity of the compressor and the working fluid.

Ammonia Condenser: this heat exchanger condenses ammonia with cold tap water stream. No control loop will be directly controlling any inlet or outlet stream of this equipment.

Ammonia collector vessel: this vessel collects liquid ammonia from the condenser and stores ammonia during shut down of the system.

Natural Gas (NG) fired heater: The aim of this equipment is to heat up the cold tap water stream. The temperature is the main disturbance. The temperature of the tap water exiting Upgrading Unit is controlled by manipulating the air-to-NG ratio. For that purpose, a ratio control loop involving the air and natural gas flow rate is used. The ratio control set point is tuned in a cascade control manner by an oxygen analyzer in the off-gas stream from the fired heater, which will ensure that the combustion is efficiently carried out.

The temperature of the hot tap water stream leaving the fired heater needs to be controlled because it guarantees that the required amount of heat is sent to the district heating system. Therefore, a tight control system is implemented by using a High Selector (HS) in addition to Valve Position Control (VPC). Based on comparison of the opening position of the valves placed in the heating systems by HS, VPC tunes the set point of fired heater temperature control proportional to the highest valve opening position.

It should be noted that the fluctuations in the natural gas supply pressure will be overcome by a pressure controller in the natural gas feed stream to the fired heater, which sends signal to a control valve.

Blower: This equipment supplies the fired heater with air. The air flow control is part of the air-to-natural gas ratio control described above.

In addition to standard control loops mentioned above, the control devices pertaining to safety, operational, and environmental constraints are of vital importance. These additional control loops are not shown in the PFS diagrams but they have to be presented in the Piping and Instrumentation Diagrams (P&IDs), which is a document that may be preliminary, generated in the conceptual process design, but it is mostly used and updated accordingly in the subsequent detailed engineering phases of a project. P&ID is not included on the list of the documents of this CPD. The list of additional loops may include the following:

- Critical instrumentation for the compressors is always required and is included in the compressors package (e.g. solenoid valves for capacity control, start up and shut down control devices, lubricant oil control system, safety valve, etc). In order to insure maximum system integrity local and remote alarms are usually available. The manufacturers usually provide the critical loops and local control panel. A complete set of instrumentation diagrams is also made available by the manufacturer;
- Critical instrumentation for the fired heater (e.g. natural gas safety shutoff valves);
- Ammonia leak detector to be installed in the Upgrading Unit;

6.2 Process Control of The Heat Distribution System

The temperature of the district water stream leaving each Heat Station needs to be controlled so that the heat transfer between the hot tap water stream and the district water stream is properly conducted. Therefore, the temperature controller will send signal to a flow control valve installed in the hot tap water pipeline upstream each heat exchanger.

Chapter 7 Mass and Heat Balances

7.1 Balance for Total Streams

Table below shows mass and energy balances around the system battery limits, as well as every individual equipment.

Table 7.1 Mass and energy balance around the battery limits

HEAT & MASS BALANCE FOR STREAMS TOTAL										
IN					EQUIPM. IDENTIF.	OUT				
Plant		EQUIPMENT				EQUIPMENT			Plant	
Mass kg/s	Heat kW	Mass kg/s	Heat kW	Stream Nr.		Stream Nr.	Mass kg/s	Heat kW	Mass kg/s	Heat kW
81.05	-1294000	3.89	-14205	**<8>**	**E01**	**<3>**	3.89	-10681	81.05	-1297000
			3524							
		3.89	-10681		Total		3.89	-10681		
		3.89	-10681	**<3>**	**K01**	**<4>**	3.89	-10060		
			621							
		3.89	-10060		Total		3.89	-10060		
		3.89	-10266	**<5>**	**K02**	**<6>**	3.89	-9429		
			837							
		3.89	-9429		Total		3.89	-9429		
10.35	-55	10.35	-55	**<17>**	**K03**	**<18>**	10.35	275		
			330							
		10.35	275		Total		10.35	275		
0.71	-2588	97.39	-1532000	**<14>**	**F01**	**<15>**	97.39	-1505000	11.06	-29019
			26999.8							
		97.39	-1505000		Total		97.39	-1505000		
		97.39	-1505000	**<15>**	**P01**	**<20>**	97.39	-1505000		
			75							
		97.39	-1504925		Total		97.39	-1505000		
47.34	-743126	19.49	-300288	**<21>**	**E04**	**<22>**	19.49	-305527	47.34	-737887
								5239		
		19.49	-300288		Total		19.49	-300288		

Table 7.1 Mass and energy balance around the battery limits (*continued*)

47.34	-743126	19.49	-300810	**<25>**	**E05**	**<26>**	19.49	-305527	47.34	-738409
								4717		
		19.49	-300810		Total		19.49	-300810		
47.34	-743126	19.49	-300613	**<31>**	**E06**	**<32>**	19.49	-305589	47.34	-738150
								4976		
		19.49	-300613		Total		19.49	-300613		
47.34	-743126	19.49	-300747	**<35>**	**E07**	**<36>**	19.49	-305497	47.34	-738376
								4750		
		19.49	-300747		Total		19.49	-300747		
47.34	-743126	19.49	-300722	**<39>**	**E08**	**<40>**	19.49	-305497	47.34	-738351
								4775		
		19.49	-300722		Total		19.49	-300722		
					Heat Losses in the pipeline network					4503
328.79	**-5012273**				**Total**				**328.79**	**-5012689**
OUT - IN :									0.00	**-416**

It is important to remark that heat losses in the pipeline network were counted for as energy OUT. However, the total energy balance shows a difference of 416 kWatt. The reasons for that are the following:

- Inaccuracy in the estimation of the heat losses.
- Inaccuracy due to the iterative procedure to calculate the cold tap water flow rate and the temperature profile in the pipelines. As it was mentioned in Chapter 5, based on a preliminary guessed temperature profile, the tap water mass flow rate in the MHP was calculated and this value was used as input for the steady-state simulations. With the output of the simulation, a new value of the hot tap water temperature was obtained and the heat losses were estimated. As a result, a new temperature profile was obtained and new values of temperature of cold tap water and mass flow rate were calculated. This procedure should be repeated until the last and the former values of temperature differ within an acceptable tolerance. Since the lengths of the pipes were just estimated, the iterations were not carried out until the end.
- As mentioned in chapter 2, different thermodynamic models (e.g. PR and Steam Tables) were employed in the AspenPlus simulation. Since the enthalpy calculations of these models depend on reference state conditions (pressure and temperature), the application of different thermodynamic models could cause discrepancies in the overall energy balance in this table if one adds stream enthalpies calculated by different models.

7.2 Balance for Stream Components

The mass balance around the system battery limits for all the components is given in the following tables.

Table 7.2 Mass and energy balance for stream components

STREAM Nr. : Name :	1 IN WW IN		2 OUT WW OUT		OUT-IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	0	0	0	0		
Water 18.02	81.06	4.50	81.06	4.50		
Total	81.06	4.50	81.06	4.50		
Press. Bara	3.00		2.50			
Temp. °C	27.50		18.00			
Enthalpy kW	-1293746		-1297282		-3535	

STREAM Nr. : Name :	16 IN NGSTD		17 IN AIR IN		19 OUT EMISSIONS		OUT-IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	0	0	0	0	0	0		
Water	0	0	0	0	1.15	0.06		
Methane	0.50	0.03	0	0	0	0		
Ethane	0.03	1.06e-3	0	0	0	0		
Propane	6.46e-3	1.46e-4	0	0	0	0		
n-Butane	2.12e-3	3.66e-5	0	0	0	0		
n-Pentane	2.64e-3	3.66e-5	0	0	0	0		
Oxygen	0	0	2.41	0.08	0.36	0.01		
Nitrogen	0.15	5.23e-3	7.94	0.28	8.09	0.29		
Dioxide carbon	0.01	3. 30e-4	0	0	1.45	0.03		
Total	0.71	0.04	10.35	0.36	11.05	0.39		
Press. Bara	1.00		1.00		1.3			
Temp. °C	15.56		20.00		79.85			
Enthalpy kW	-2577		-55		-29021		-26388	

STREAM Nr. : Name :	24 IN FROM DIS.I		23 OUT TO DIS. I		OUT-IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	0	0	0	0		
Water 18.02	47.34	2.63	47.34	2.63		
Total	47.34	2.63	47.34	2.63		
Press. Bara	1.00		1.00			
Temp. °C	65.00		91.42			
Enthalpy kW	-743126		-737887		5239	

STREAM Nr. : Name :	34 IN FROM DIS.II		33 OUT TO DIS. II		OUT-IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	0	0	0	0		
Water 18.02	47.34	2.63	47.34	2.63		
Total	47.34	2.63	47.34	2.63		
Press. Bara	1.00		1.00			
Temp. °C	65.00		89.09			
Enthalpy kW	-743126		-738350		4776	

Table 7.2 Mass and energy balance for stream components (*continued*)

STREAM Nr. : Name :		42 IN FROM DIS. III		41 OUT TO DIS. III		OUT-IN	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0		
Water	18.02	47.34	2.63	47.34	2.63		
Total		47.34	2.63	47.34	2.63		
Press.	Bara	1.00		1.00			
Temp.	°C	65.00		88.80			
Enthalpy	kW	-743126		-738409		4717	

STREAM Nr. : Name :		28 IN FROM DIS.IV		27 OUT TO DIS. IV		OUT-IN	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0		
Water	18.02	47.34	2.63	47.34	2.63		
Total		47.34	2.63	47.34	2.63		
Press.	Bara	1.00		1.00			
Temp.	°C	65.00		88.80			
Enthalpy	kW	-743126		-738409		4717	

STREAM Nr. : Name :		38 IN FROM DIS.V		37 OUT TO DIS. V		OUT-IN	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0		
Water	18.02	47.34	2.63	47.34	2.63		
Total		47.34	2.63	47.34	2.63		
Press.	Bara	1.00		1.00			
Temp.	°C	65.00		88.96			
Enthalpy	kW	-743126		-738376		4750	

Chapter 8 Process and Equipment Design

8.1 Integration by Process Simulation

This simulation was used to model the major unit operations within the Upgrading Unit of mechanical heat pump process. Ammonia runs as a cycle integrated with cold tap water, waste water, air, and natural gas. The output of the simulation was used as the basis for the equipment design. A list of the main types of equipment in this simulation includes: single phase heat exchangers (intercooler), fired heater, compressors, evaporator, condenser and pump.

Only one tear stream was required to run this simulation with AspenPlus. The inlet stream to ammonia evaporator (*EVAPIN*) was chosen as tear to serve as the "starting point" for the sequential modular approach, which is used by the simulator. Since the process runs cyclically, the only way to change the ammonia flow rate in the system is via modifications that can be done in the input specifications of this stream. The Wegstein convergence method (default) was successfully employed with the maximum number of iterations (MAXIT) of 60. The flowsheet diagram from AspenPlus simulation is presented in Appendix 23 and the electronic file is available in a CD. The description of each block of the simulation flowsheet is given below[1] [2].

It should be noted that the electronic version of all simulation files are available in a CD-ROM.

1- Evaporator
HEATX (*EVAP*): Ammonia evaporation with the waste water from DSM.
Input data: Inlet pressure = 6.6 bar
Pressure drop = 0.1 bar
Output : Heat duty = 3535 kW

2- Compressor
COMPR (*COMPR1*): First stage ammonia compressor.
Input data: Suction pressure = 6.5 bar,
Discharge pressure = 15.0 bar
Output : Brake horse power = 745 kW

3- Intercooler
HEATX (*INCOOLER*): Cooling first stage ammonia compressor discharge with cold tap water.
Input data: Inlet pressure = 15.0 bar
Pressure drop = 0.3 bar
Output : Heat duty = 243 kW

[1] The block tag names are shown in Italic.
[2] Inputs and outputs are referred to the nominal operation conditions.

4- Compressor
COMPR (*COMPR2*): Second stage ammonia compressor.
Input data: Suction pressure = 14.7 bar,
Discharge pressure = 38.0 bar,
Output : Brake horse power = 945 kW

5- Condenser
HEATX (*COND*): Ammonia condensation to heat up the cold tap water.
Input data: Inlet pressure = 38.0 bar
Pressure drop = 0.1 bar
Output : Heat duty = 4804 kW

6- Expansion valve
VALVE (*CONTROLV*): Ammonia expansion prior to evaporation.
Input data: Inlet pressure = 37.9 bar,
Outlet pressure = 6.5 bar.

7- Splitter
SPLITTER (*SPLIT*): Splits the cold tap water coming back from the Heat Stations into two streams going to the intercooler and the condenser.
Input data: Mass flow rate = 2 kg/sec to the intercooler

8- Mixer
MIXER (*MIXER*): Mixes the tap water outlet stream from the condenser with the tap water outlet stream from the intercooler.
Input data: None was given. No pressure change was considered in the mixer.

9- Heat Exchanger
HEATX (*FLGASHX*): Flue gases from the fired heater heat up the tap water.
Input data: Inlet pressure = 8.0 bar
Pressure drop = 0.5 bar
Output : Heat duty = 5791 kW

10- Reactor
RSTOIC (*NGFURN*): Combustion of natural gas is modeled by RSTOIC with 100 % fractional conversion.
Input data: Inlet pressure = 1.3 bar
Reaction temperature = 527ºC
Output : Heat Stream **HEAT1 =**20915 kW
This heat stream is used as an input for the block (***FIREDHEX)*** that simulates the exchange of combustion heat with the tap water stream.

11- Heater
HEATER (*FIREDHEX*): Heat integration with block NGFURN.
Input data: Inlet pressure = 6.2 bar
Pressure drop = 0.5 bar

Heat duty = 20915 kW
Output : Tap water temperature = 137 ºC

12- Compressor
COMPR (*BLOWER*): Blows air into the fired heater.
Input data: Suction pressure = 1.0 bar,
Discharge pressure = 1.3 bar
Output : Brake horse power = 367 kW

13- Pump
PUMP (*CWPUMP*): Pumps the hot tap water to the Heat Stations.
Input data: Suction pressure = 6.2 bar,
Discharge pressure = 10.0 bar
Output : Brake horse power = 61 kW

8.2 Equipment Selection and Design

The equipment selection and design heuristics are adopted from "Product & Process Design Principles" [1]. The equipment costs, presented in Appendix 3, were evaluated based on the following details:

Evaporator (E01): According to *Heuristic 25*, this equipment is a shell-and-tube heat exchanger of kettle vaporizer type using countercurrent flow. It includes 1-in. O.D., 16 BWG (Birmingham Wire Gage) carbon steel tubes, 12 ft long, on square pitch in a carbon steel shell. The shortcut design calculations are done with respect to minimum temperature approach of 3°C [62] and heat coefficient of 550 Watt/sqm.K [66]. Moreover, from *Heuristic 31*, a pressure drop of 0.1 bar is specified for ammonia side, as well as a pressure drop of 0.5 bar for waste water side.

Compressors (K01) and (K02): Since the discharge pressure is greater than 2 bar, a staged compressor system is needed according to *Heuristic 34.* However, from *Heuristic 35,* the exit temperature of compressors cannot exceed 190°C. Therefore, a 2-stage compressor with a maximum compression ratio of 4 for each stage is needed. An intercooler (E02) is placed in between compressors. Furthermore, the isentropic efficiency and mechanical efficiency of the compressors are specified to be 75% and 90%, respectively.

Based on pressure range and maximum allowable capacity, two carbon steel reciprocating compressors coupled with electric drivers were chosen [1],[23]. Reciprocating compressor is a positive displacement device. For each revolution of the motor, a given volume of the vapor is compressed. The mass flow through a compressor operating at a constant speed will change as the specific volume of the inlet gas changes. The maximum flow occurs when the inlet gas is saturated vapor. A measure that is commonly used to identify how far from this maximum flow operating point is called degrees of superheat. The more degrees of superheat existing in the gas entering the compressor, the larger the specific volume of gas and the lower the mass flow of the compressor. Manufacturers must specify the nominal conditions to establish their base capacity rating. Typically, they provide a specific superheat (which governs the mass flow and specific enthalpy of the working fluid at the evaporator outlet) and a specific sub-cooling (which governs the specific enthalpy of the working fluid to the evaporator inlet). Compressor manufacturers usually state that their ratings are based on 5.5 ºC of superheat or 5.5 ºC of subcooiling[78].

When compressing a gas, the entering stream must not contain any liquid and the existing gas must be above its dew point so that the compressor does not get damaged. To remove any entrained liquid droplets from the entering gas, a vertical knock-out drum equipped with a demister pad is placed just upstream of the compressor. Therefore, as a safety matter, a knock-out drum has to be installed in the suction line of both compressors. Many manufacturers supply such drums as part of the compressor package and include them in the total price of the compressor and auxiliary components. Since there is no basis to get the proper dimensions of the knock-out drums, these pieces of equipment are not shown in the process flowsheet and their costs are considered to be included in the total price of the compressors. In a further detailed design phase of this project these drums shall be present in the Piping and Instrumentation Diagrams (P&ID).

Intercooler (E02): According to *Heuristic 25*, this equipment is a shell-and-tube heat exchanger of floating head type using countercurrent flow. It includes 1-in. O.D., 16 BWG (Birmingham Wire Gage) carbon steel tubes, 12 ft long, on square pitch in a carbon steel shell. The shortcut design calculations are done with respect to minimum temperature approach of 3°C [62] and heat coefficient of 600 Watt/sqm.K [66]. Moreover, from *Heuristic 31*, a pressure drop of 0.1 bar is specified for ammonia side, as well as a pressure drop of 0.5 bar for tap water side.

Condenser (E03): According to *Heuristic 25*, this equipment is a shell-and-tube heat exchanger using countercurrent flow. It includes 1-in. O.D., 16 BWG (Birmingham Wire Gage) carbon steel tubes, 12 ft long, on square pitch in a carbon steel shell. The shortcut design calculations are done with respect to minimum temperature approach of 3°C [62] and heat coefficient of 175 Watt/sqm.K [66]. Moreover, from *Heuristic 31*, a pressure drop of 0.3 bar is specified for ammonia side, as well as a pressure drop of 0.5 bar for tap water side.

Ammonia collector vessel (V01): The volume of this carbon steel vessel is estimated based on the ammonia mass in the cycle, in addition to an over design factor of 15 %. The ratio of length to diameter is assumed to be 2.5 [62].

Air Blower (K03): Since the discharge pressure is lower than 2 bar, a blower is needed according to *Heuristic 34.* Based on pressure range and maximum allowable capacity, cast iron centrifugal blower coupled with electrical driver was selected. Furthermore, the isentropic efficiency and mechanical efficiency of the air blower are specified to be 75% and 90%, respectively

Fired Heater (F01): In order to overcome the fluctuations in heat demand, as well as to accomplish the amount of heat being delivered by the waste water stream in the worst case scenario, this equipment is over designed by a factor of 5. The fired heater temperature is specified to be 527°C [62]. Moreover, since vapor fraction of the flue gas stream should always be 1.0 (to avoid achieving the dew point), the convective section of the fired heater is designed to achieve flue gas temperature of no less than 80°C. From *Heuristic 31*, a pressure drop of 1.3 bar is specified for tap water side. Moreover, the material of construction of the fired heater is assumed to be carbon steel fiber insulation lining.

Pump (CWPUMP): According to *Heuristic 37*, based on flow rate and pump head figures, a cast iron radial centrifugal pump coupled with an electric motor was chosen to deliver hot tap water to the heating districts. The case-split orientation of the pump is HSC and the shaft speed is 3600 rpm. To protect the internal working parts of the motor against moisture, dust, and corrosive fumes, a totally enclosed, fan cooled (TEFC) motor enclosure was selected. Furthermore, the pump efficiency and driver efficiency are specified to be 70% and 90%, respectively. It is important to emphasize that the pump head did not take into consideration any vertical branches of pipeline since the lengths were just preliminary estimated. In a detailed engineering phase of this design, the pump design should be properly updated based on isometric drawings and more accurate values of pipe length.

Heat Exchangers E04, E05, E06, E07, E08: According to *Heuristic 25*, each of this equipment is a shell-and-tube heat exchanger of floating head type using countercurrent flow. It includes 1-in. O.D., 16 BWG (Birmingham Wire Gage) carbon steel tubes, 12 ft long, on square pitch in a carbon steel shell. The shortcut design calculations are done with respect to minimum temperature approach of 3°C [62] and heat coefficient of 1000 Watt/sqm.K [66]. Moreover, from *Heuristic 31*, a pressure drop of 0.5 bar is specified for both sides. It should be noted that the pressure of the district cold and hot water streams is unknown. Since this refers to liquid streams, the pressure does not influence the energy balance in the heat exchangers because the enthalpy of compressed liquids does not depend too much on pressure. In the simulation the pressure of those streams was assumed to be 1.0 bar.

Chapter 9 Wastes

In this project, the solid waste does not apply. The direct wastes are the following:

9.1 Liquid Wastes

The main liquid waste generated in this process is the waste water stream exiting the Upgrading Unit, 2.5 million cubic meters per annum (~291.8 ton/h) at 18ºC. Although the specification of the waste water stream from DSM was not available due to confidentiality reasons, it is reported in the previous feasibility study [22] that its quality is too low. Therefore, this stream cannot be reused. The owner(s) of the Upgrading Unit (the party that extracts the heat) will be responsible for disposing the waste water. It will be sent to the North Sea via the existing pipeline, which belongs to the Water Board.

9.2 Gaseous Wastes

Off-gases from the fired heater are released to the atmosphere at 80ºC. This is the only gaseous waste stream generated in the Upgrading Unit. Table 9.1 shows the composition of this stream. The yearly amount of the gaseous emissions generate is presented in Table 9.2.

Table 9.1 Gaseous waste stream composition

Composition	Units	Specification		Notes	Additional Information
		Available	**Design**	(1),(2),(3)	(1) Estimated with AspenPlus Simulation (RGIBBS)
Carbon Dioxide	%wt	13.73	12.83		(2) Available specification refers to the emissions in the case of fired heater nominal operation.
Carbon Monoxide	%wt	traces	traces	(4)	(3) Design specification refers to the emissions in the case of fired heater peak operation
Nitrogen Dioxide	ppm	12	19		(4) (10^{-13})
Nitrogen Monoxide	ppm	74	94		
Oxygen	%wt	2.27	3.64		
Nitrogen	%wt	73.15	73.38		
Water	%wt	10.85	10.14		
Total		100	100		

Table 9.2 Yearly amount of gaseous emissions

Component	Units	Amount	Notes	Additional Information
Carbon Dioxide	Ton/year	47874.0	(1),(2)	(1) Estimated with AspenPlus Simulation (RGIBBS)
Nitrogen Dioxide	"	0.1		(2) Figures refer to the emissions in the case of fired heater nominal operation
Nitrogen Monoxide	"	0.9		
Oxygen	"	7923.0		
Nitrogen	"	255115.0		
Water	"	37846.0		
Total	"	348759.0		

Chapter 10 Process Safety

10.1 Hazard and Operability Study (Hazop)

In order to reduce safety risks and enhance the operability of the process, a Hazard and Operability study was carried out in the Upgrading Unit, in addition to the pipeline network.

Aiming to cover the most critical pieces of equipment, the following streams were assessed:

- Stream 6: transfer ammonia from the discharge of 2nd compressor to the condenser
- Stream 3: transfer ammonia from the evaporator to compressor1
- Stream 4: transfer ammonia from the 1st compressor to the intercooler
- Stream 5: feed ammonia into compressor 2
- Stream 7: transfer ammonia liquid to the expansion valve
- Stream 8: transfer ammonia from the expansion valve to the evaporator
- Stream 17: transfer air from the blower to the fired heater
- Stream 16: transfer natural gas to the fired heater
- Stream 21: transfer hot tap water from the Central Grid 1 to the Heat Station (HEX1)
- Stream 22: transfer hot tap water from Heat Station 1 (HEX1) to the Central Grid 1(CG1)
- Stream 10: transfer cold tap water to the intercooler
- Stream 14: transfer tap water from the mixer to the furnace
- Stream 29: transfer hot tap water from the Upgrading Unit to the Central Grid 2

The Hazop study revealed that an ammonia collector vessel and a by-pass stream for waste water should be installed in the Upgrading Unit. Furthermore, a thermo-expansion relief valve should be applied for each heat exchanger in the Heating Stations. Safety valves in the ammonia compressors are included in the package provided by the manufacturers and all related safety procedures and auxiliary equipment have to be accordingly addressed during the equipment purchasing and detailed engineering phase of the project. The results of the Hazop Study can be found in Appendix 17.

10.2 Fire and Explosion Index

Fire and Explosion Index (F&EI) is a numerical value or index developed by the Dow Chemical Company, which is calculated based on the nature of the process and the properties of the process material [66]. Based on the value of this index, a degree of hazard of the process can be determined. Material factor is the basis of the F& EI. The general and special process hazards have also been taken into account to this index. The degree of hazard assessment can be judged based on Table 10.1 [82].

Table 10.1 Assessment of hazard

Fire and Explosion Index Range	Degree of Hazard
1 - 60	light
61 - 96	moderate
97 - 127	intermediate
128 - 158	heavy
> 159	severe

The F&EI for the process that takes place at the Upgrading Unit is 28. It means that the degree of hazard of this process is light. This shows that the process is safe from the fire and explosion point of view. The detailed F&EI calculations are shown in Appendix 18.

Chapter 11 Economy

11.1 Total Investment

The total investment was calculated by the Lang method [2]. The final total investment excludes connections from the Heat Stations to the houses (this was far beyond the scope of the project). The transport pipelines from the Upgrading Unit to the Heat Stations are included, and contribute for a big fraction to the total investment. The fact that the length of the transport pipelines was roughly estimated means that the total investment will change when modifications to the transport pipelines are done. The total investment can also be reduced by a fraction of the one-time-revenues explained later. The final total investment is MU$29.3. The calculations are present in Appendix 25.

11.2 Total Product Cost

The total product costs consist of the manufacturing costs and the sum of Sales, Administration Research and Engineering (SARE). The total product costs are MU$20.3 per annum. At a design production capacity, 780 TJ/annum, the delivered heat at the district represents US$26.0 per GJ. The yearly cost for the disposal of the waste water was included as a part of the raw material costs and the amount is €200,000 per year [22]. The calculations are present in Appendix 28. Utility costs, which are part of the manufacturing costs, are included in Appendix 35.

11.3 Revenues

The price of the heat delivered by a DHS at the district is higher than the energy price of gas. It means that one GJ of heat consumed costs more than one GJ of gas consumed. The reason for that is the fact that customers pay for heat that is efficiently used. There are no losses inside the houses by heat generation and the losses occur mostly in the Upgrading Unit, in the fired heater and in the distribution to the districts. Thus, all heat losses take place in processes related the DHS. The price of 1 GJ at the district is €22.04 for usage below 119 GJ including tax, updated October 2005, Purmerend Holland [79].

There is a difference in price between usages below and above 119 GJ per year. So, there is also a difference in revenues for these two consumption patterns. It was assumed 20% in the case of usage higher than 119 GJ and 80% for usage less than 119 GJ. Excluding BTW of 19% and energy tax of €6.32 per GJ, the real revenues are respectively €10.14 and €12.20 per GJ [79].

In the Netherlands, once a customer is connected to a DHS it is no longer necessary to have a boiler and replace it each 15 years in each house. Nevertheless, each house has to pay €3,700 once [80]. This amount of money is mainly used to cover part of the investment of the upgrading unit and the pipelines from this unit to the houses. It was assumed that it would also be the case for this specific design. The connections from the Heat Stations to the houses were not taken into account because the districts to be connected are unknown. It sounds reasonable to consider that the total investment so far can be lowered by one third of this one-time revenue.

Every year each house has to pay a contribution for maintenance and this contribution varies depending on the type of customer. Companies pay more and house blocks pay less. Taking an average, the price paid by regular households is around €295 per year. Hence, it results in one-time revenue of MU$13.7 and yearly revenues of MU$14.2. The calculations are in Appendix 29.

11.4 Cash Flow

The difference between the income and the operating costs led to a negative cash flow of MU$6.1. It means that this project is not profitable. Several reasons can be listed and the most remarkable will be discussed below.

A trend that has being seen in DHS nowadays is that there are mainly two sorts of designs:

- The so-called Combined Heat and Power (CombHP) system. This system is mostly chosen due to the fact that just delivering heat is not profitable whereas delivering heat and power yields profits. In the CombHP the efficiency of the entire plant increases (the over designed capacity can be used to produce electricity), because the power curve has an opposite trend compared to the heat demand curve. The CombHP plants are mainly equipped with gas turbines.
- Plants that are built for heating only are often supplied by "free of charge" sources of energy such as sea water (e.g. plants in Norway, Iceland), waste water and geothermal sources. Since the energy source is almost infinitely available, it is profitable to use the heating only concept. In this case, peak demand is covered by hot water storage buffer tanks, so that the plant can run continuously at full capacity.

The problem in the present design is that although the "free of charge" heating source exists, the amount (100 TJ/annum) is not enough to fulfill the required demand (780 TJ/annum) and it cannot be increased. In order to increase the amount of "free of charge" heating source it would be necessary to increase the waste water flow rate or the temperature of that stream. Unfortunately these two variables are fixed as design basis and it is impossible to obtain a waste water stream with higher energy content. Decreasing the demand by connecting fewer districts would be a way to overcome this problem.

Due to the fact that no profit is expected, a break-even point curve has been constructed and it is shown in Figure 11.1. As it can be seen in this figure, the break-even point is around the demand of 2300 TJ/Year.

In order to make this plant profitable some suggestions are discussed. Increasing capacity seems to be a good solution at a first glance, but the plant is already used at full capacity for some periods in a year. Besides, increasing capacity also means increasing the plant size leading to a much bigger total investment. A higher total investment leads also to slightly higher fixed costs, which will drive the break-even point a bit up to a somewhat higher value. Additionally, one may think of increasing capacity, but in Upgrading Unit plants like that the maximum demand (in this case, 780 TJ/annum) is a key design factor in and it would not be realistic to assume that the number of connected districts might be doubled or tripled.

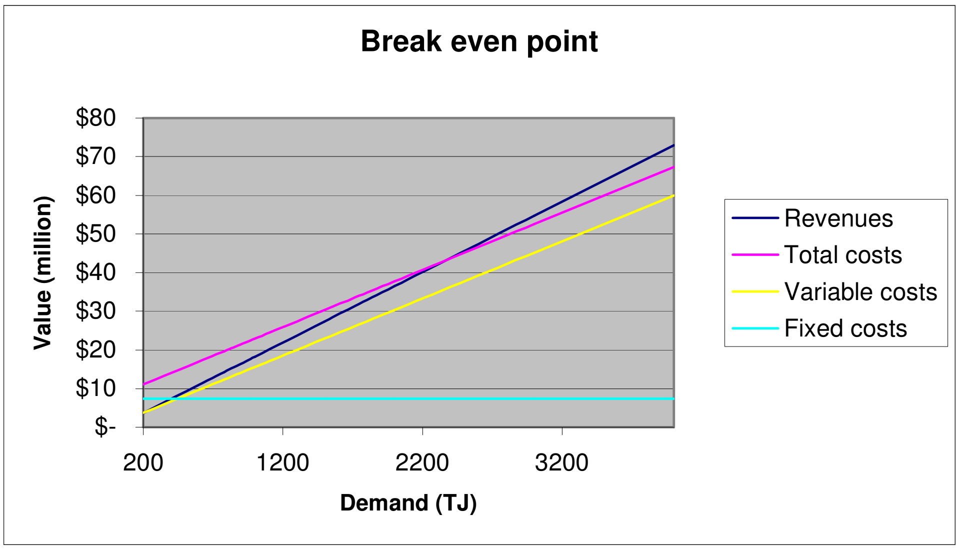

Figure 11.1 Break Even Point curve

Another alternative would be increasing the price of the delivered heat. Unfortunately one is not free to set this price, because it is officially regulated by govern. In The Netherlands there is a law called: "Niet meer dan anders" meaning that customers who are connected to a city heating may never pay more than if they were using their own boiler.

Also a better insulation, as discussed in Chapter 5, can be used in the pipeline network to provide less heat losses and less energy costs while the revenues remain constant. As a result, a higher profit (in this case less debts) would be achieved.

The Upgrading Unit can probably be made profitable by replacing the fired heater by a gas turbine. In this case the plant would be working as a combined heat and power plant. In typical combined plants almost 90% of the energy in the original fuel (in this case, natural gas) has a productive use. Energy savings can run up to 35% and this is not the only advantage. The other advantage is that electricity can be sold in periods when the heat demand is low since the price for energy is around three times as much as for gas in energy equivalent [81].

Energy is nowadays a very sensitive product and is influenced by a lot of different factors, not only demand and supply, but also by factors related to the world's energy market situation, higher prices during roaring times, lower prices in quiet times, etc. As it is well known, the price of energy tends to increase. The demand is certainly growing on a daily basis driven by dynamic economies, for instance China.

The contribution of the energy that is "for free" in this DHS will be relatively higher when energy prices are moving up. The input for this process is energy and its output is energy as well. Therefore, the profitability is not really dependent on the energy price. Actually it depends only on the profitability of the MHP (ammonia loop). The higher the price the more profitable it is.

This Upgrading Unit is also sensitive to the decisions made by DSM. If this company decides to stop delivering waste water to this MHP, e.g. by shutting down for maintenance or even by discontinuing its production in the future, an alternative should be used in order to compensate the loss of the heating generated by the ammonia cycle. The alternative is easy to implement by increasing the fired heater capacity, but the advantage of emissions reduction will be gone.

11.5 Economy Summary

The total investment and production costs were carried out by applying a factored estimation method, the Lang method, which is based on knowledge of major pieces of equipment and aims to achieve ± 25% accuracy.

The summary of economic evaluation is given in Table 11.1.

Table 11.1 Economy summary

			MUS$
Total investment			29.3
	Fixed capital		24.5
	Working capital		1.5
	Costs for transport pipelines		17.0
		Pre-insulated transport pipelines	3.5
		Laying of the transport pipelines	13.4
	One time revenues (minus)		-13.7
Start-up costs			1.5

Table 11.1 Economy summary (*continued*)

Total product costs		20.3
	Manufacturing costs	20.0
	SARE	0.4
Total yearly revenues		14.2
Cash flow		-6.1

Chapter 12 Creativity and Group Process Tools

This chapter describes the creativity and all of its tools which has been used by the team during the Conceptual Process Design project. The role of the creativity coach will also be mentioned in this chapter.

12.l Creativity

Creativity is often defined as the ability to produce a new concept or idea which has never been done ore produced before. During this CPD project, creativity is one of the main thing which has already been used for solving some problems, brainstorming, and creating some new ideas. Most of the ideas used in this CPD come from the creativity of the team members. The brainstorming sessions become more and more interesting and a lot of new ideas have been discussed in order to get the best solution.

12.2 Brainstorming

The brainstorming session is held in a rather relax situation. This is done in order to make every member feel relaxed and free to discuss some important issue. It has been proved during this CPD project that this way made everybody can think clearly and as the result, many new and creative ideas have been generated during the brainstorming session. This kind of meeting is held regularly every week. Coincidentally, if there are any key issues which has to be decided immediately, the session is also held for discussing the solution together.

12.3 Meetings for Planning

At the beginning of every week, a short meeting is set-up in order to make a weekly planning together. This is very important to control the steps that are done by the group. Having a planning will help the group members to do the things on the right track.

12.4 Wrap-up/Evaluation Meeting

Besides the meeting for planning, another short meeting is also set-up at the end of the week. The aim of the meeting is to evaluate what have been done during the week. This will avoid a kind of re-work for each group member.

12.5 Creativity Coach

A creativity coach has been pointed by the supervisor in order to help the group to be able to work on the creative and right track. In fact, several meetings with the creativity coach were held. Those meetings, many issues were discussed, especially regarding the creativity used during the project execution and the progress achieved by the group. During the meeting, alternately, each group member told the coach about what has been done, the planning and also, if any, a problem that may block the group creativity. The coach gave some inputs or feedback to the team in order to boost the group spirit and creativity. The meetings were held in an informal situation. In fact, this helps all of the members to be more relaxed after a certain time period working with the project which may cause tiredness.

12.6 Check-list of The Planning

The planning checklist is the way that the group used to evaluate all the things that have been done. This also helps the group to figure out whether some parts are done properly or missing.

12.7 Working Method

During the execution of the CPD project, some of the tasks were divided among each group member and some tasks were done together. This helped the group to manage the time well because it avoided the waste of the time and in this way, an efficient way of working was achieved. Even though some tasks were divided, this does not mean that a member was only responsible for his job. In order to overcome this matter, some informal discussions were done to address everything done by a group member. These discussions were conducted during the brainstorming sessions. In this case, some new ideas were generated by the other members and the discussions were held in such a constructive way of thinking.

The hardcopies of correspondences involving the team and the supervisor, principal, creativity coach and some people/company were also kept as a certain file. This helps the group to trace the information whenever it is needed.

Some additional meetings on Saturdays were also held if there was any pending issue which was necessary to be discussed and solved immediately.

12.8 Meeting Notes

The meeting notes were made during group meeting. The purpose of the meeting note is to list all the thing which has been done and need to be done. This avoided the team to discuss and do the same things or other unnecessary things. Thus, the work would be more effective and efficient.

Chapter 13 Conclusions and Recommendations

13.1 Conclusions

This study was focused in the validation of the feasibility of using waste heat from DSM for a DHS. A conceptual design of the DHS was carried out with emphasis on the unit for upgrading the residual waste-heat. Furthermore, in order to deliver the upgraded heat to the districts, a water pipeline distribution system was also designed.

The most widely heat pump technologies were reviewed. It was found that chemical heat pump cannot be used as an upgrading technique for this waste heat because its temperature (27.5ºC) is lower than the temperature range of the reversible reactions that could be employed for continuous process. The hydrogenation-dehydrogenation of isopropanol-acetone has the closest range temperature (80-200ºC) to the residual waste water from the DSM.

On the other hand, mechanical heat pump was found to be the best option for recovering the heat from the residual waste water. However, mechanical heat pump contributes only to approximately 15% of the total energy provided by the Upgrading Unit. Therefore, this heat pump (ammonia loop) is combined with a natural gas fired heater to provide the required heat. The fired heater can also be used as a backup system.

Energy and exergy losses in the distribution system are factors that highly affect the economic advantages of heating systems. Thus, it is essential that heat losses in the pipeline network be kept as low as possible. In order to compensate the heat losses in the pipeline network, the Upgrading Unit is designed for a capacity of 923 TJ/annum. The coefficient of performance of this heat pump is 3.1, which is within the range for the typical heat pumps with electric compression (2.5-5.0).

The total investment is MUS$29.3 and the total production cost is MUS$20.3. As a result of the economic evaluation of this design, a negative cash flow of MU$6.1 was found.

The Upgrading Unit can probably be profitable by replacing the fired heater by a gas turbine. In this case the plant would be working as a combined heat and power plant. In typical combined plants like that almost 90% of the energy in the original fuel (in this case, natural gas) has a productive use. Therefore, energy savings can run up to 35% [81] and this is not the only advantage due to the fact that the generated electricity can be sold in periods when the heat demand is low, since the price for energy is around three times as much as for natural gas in energy equivalent.

Moreover, the economic evaluation revealed that it is slightly cheaper to run the Upgrading Unit without the MHP due to the higher price of electricity compared with natural gas. However, from an environmental perspective, it is desirable to use the ammonia loop, because it yields a reduction of approximately 4% in emissions.

It should be noted that, this design had to focus mostly on designing a city heating with 780 TJ/annum capacity other than using the waste heat stream available for heating a specific number of districts with a total demand that does not exceed 140 TJ/annum.

13.2 Recommendations

The following recommendations apply for either further studies or a detailed design phase of this design:

a. Definition of the number of districts to be connected to the DHS.
   Although the required demand was given as 780 TJ/annum, the economic feasibility of a final design is highly dependent on this figure, which determines the district sizing. Moreover, pipeline network should be defined and accurately investigated based on the specification of the actual heat demand of each district because it has a big impact in the total investment.

a. Integration of the pipeline network with the district tap water system
   This issue was out of the scope of this design and should be further investigated and detailed.

b. Waste water disposal
   The existing contract between DSM and Water Board to use the pipeline to send the waste water to the North Sea should be transferred from DSM to the owner(s) of the Upgrading Unit.

c. For further study, it is recommended that the combination of heat and power plant be investigated.

List of Abbreviations

BOD	=	Basis Of Design
BWG	=	Birmingham Wire Gage
CFC	=	ChloroFluoroCarbon
CHP	=	Chemical Heat Pump
COP	=	Coefficient Of Performance
CPD	=	Conceptual Process Design
DCFROR	=	Discounted Cash Flow Rate Of Return
DHS	=	District Heating System
DSM	=	Dutch State Mines
F&EI	=	Fire and Explosion Index
HS	=	High Selector
HSC	=	Horizontal Split Case
LC	=	Lethal Concentration
LD	=	Lethal Dose
LEL	=	Lower Explosion Limit
MAC	=	Maximum Acceptable Concentration
MHP	=	Mechanical Heat Pump
OD	=	Outside Diameter
P&IDs	=	Piping and Instrumentation Diagrams
PFS	=	Process Flow Scheme
TEFC	=	Totally End-closed, Fan-Cooled
UEL	=	Upper Explosion Limit
UU	=	Upgrading Unit
VPC	=	Valve Position Control
WW	=	Waste Water

List of Symbols

ΔH = Enthalpy Change or Heat of Reaction

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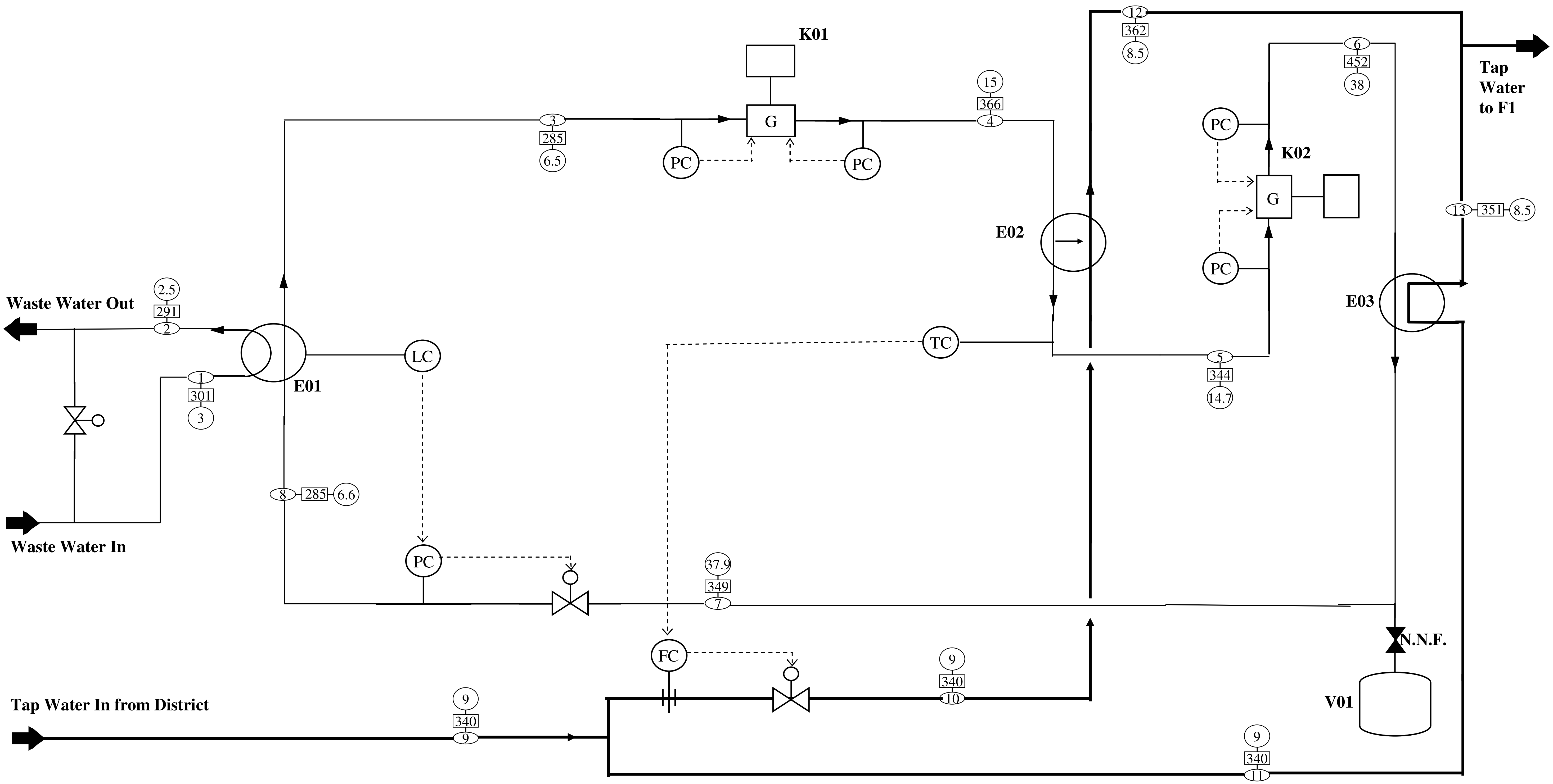

Process Equipment Summary

E01	:	Ammonia Evaporator
E02	:	Intercooler
E03	:	Ammonia Condenser
K01	:	First Ammonia Compressor
K02	:	Second Ammonia Compressor
V01	:	Ammonia Collector Vessel

Designers

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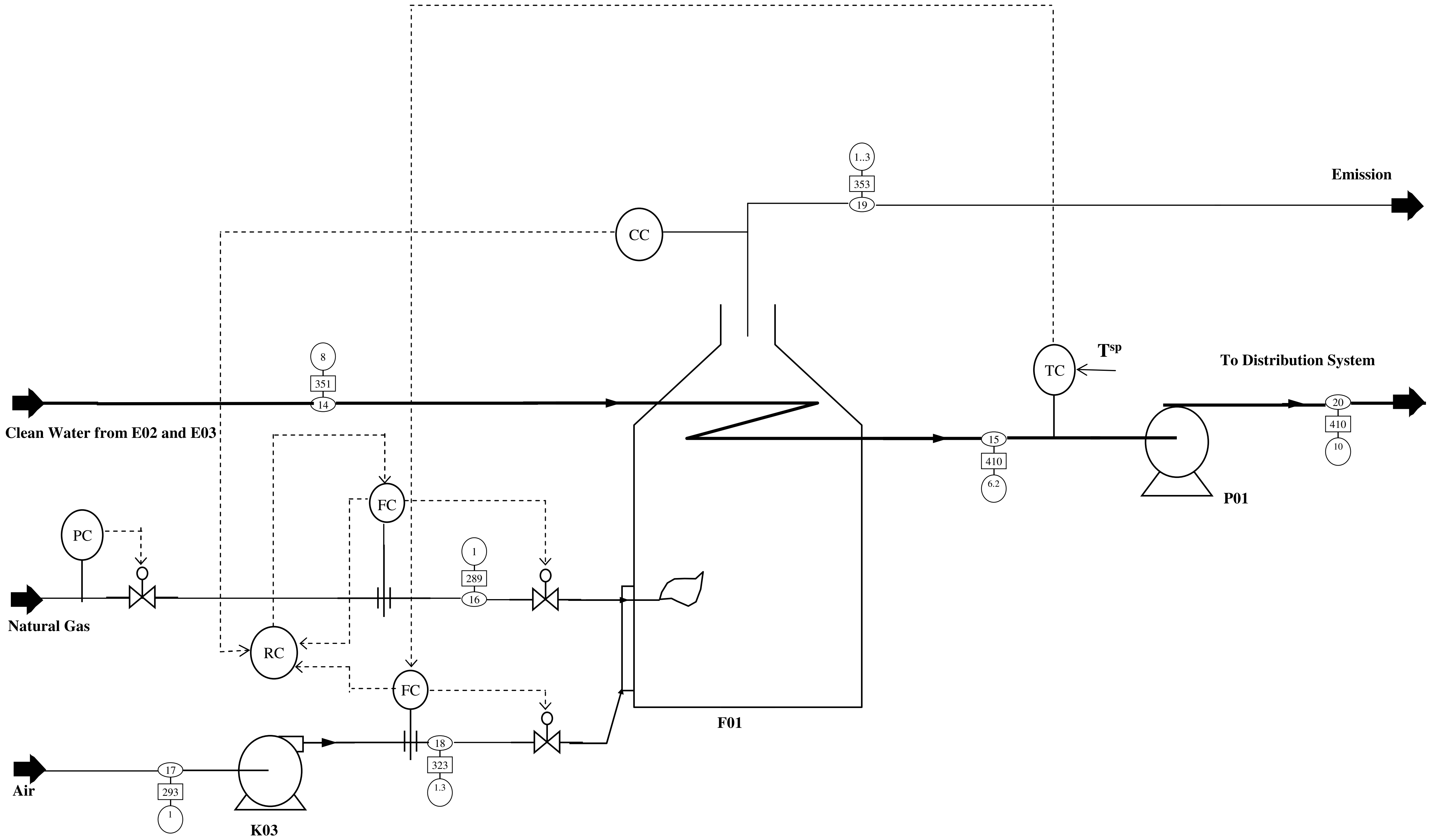
M.V. Suherman

S. Wennekes

Process Flow Scheme

Project	:	Design of A District Heating System Including The Upgrading of Residual Industrial Waste-Heat
Proj. ID Number	:	**CPD3328**
Completion Date	:	December 13th, 2005

Stream number   Temp. (K)   Pressure (Bara)



Process Equipment Summary		
F01	:	Furnace
K01	:	Air Blower
P01	:	Clean Water Pump

Designers
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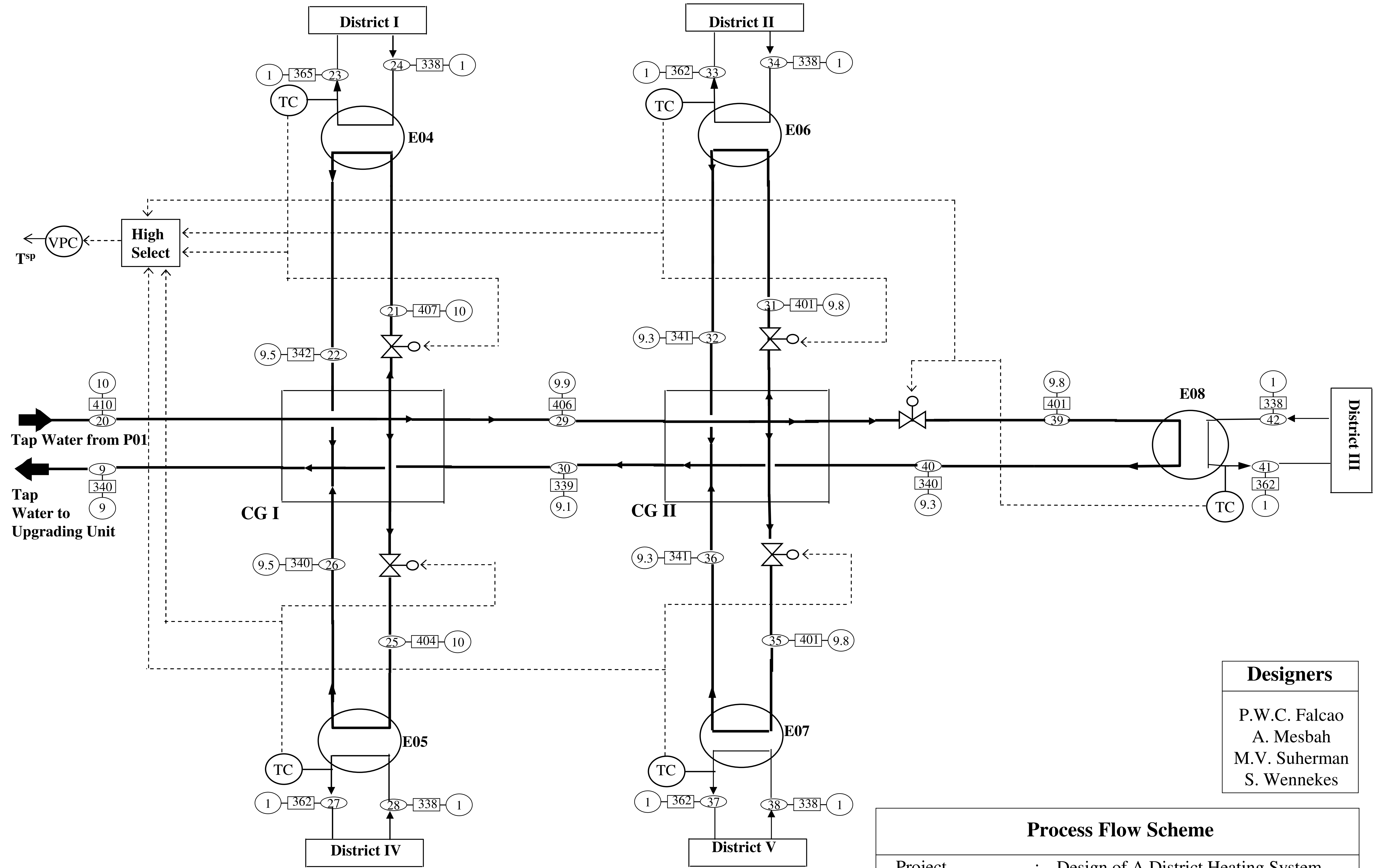
M.V. Suherman
S. Wennekes

Process Flow Scheme		
Project	:	Design of A District Heating System Including The Upgrading of Residual Industrial Waste-Heat
Proj. ID Number	:	**CPD3328**
Completion Date	:	December 13th, 2005

Stream number Temp. (K) Pressure (Bara)



Designers

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S. Wennekes

Process Equipment Summary

E04	:	Heat Exchanger I	**E07**	:	Heat Exchanger IV
E05	:	Heat Exchanger II	**E08**	:	Heat Exchanger V
E06	:	Heat Exchanger III	**CG**	:	Central Grid

Process Flow Scheme

Project	:	Design of A District Heating System Including The Upgrading of Residual Industrial Waste-Heat
Proj. ID Number	:	**CPD3328**
Completion Date	:	December 13th, 2005

Stream number   Temp. (K)   Pressure (Bara)