Appendices
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Appendix 1 Ammonia Mass in Ammonia Heat Pump

An estimative value for the inventory of ammonia in the cycle (equipment + pipes) is calculated here. The exact ammonia inventory should be provided in the detailed design.

Evaporator

Area of the evaporator:

$AEVP := 7.927 \times 10^3 \ ft^2$

The cost of this piece of equipment was calculated based on 3/4-in. or 1-in. O.D. 12ft long tubes on square or triangular pitch, thus:

$Dtube := 1 \ in$

$Ltube := 12 \ ft$

$Aouttube := \pi \cdot Dtube \cdot Ltube \cdot 0.083$

Heat transfer area for one tube is:

$Aouttube = 3.129 \ ft^2$

The number of tubes is:

$$Ntube := \frac{AEVP}{Aouttube}$$

$Ntube = 2.533 \times 10^3$

According to Table 13.6 of "Product and Process Design Principles" [1], the shell inner diameter is:

$Dshell := 58 \ in$

$Lshell := 12 \ ft$

In order to find the amount of ammonia in shell side:

$$Vtube := Ntube \cdot \frac{\pi \cdot Dtube^2 \cdot Ltube \cdot 0.083^2}{4}$$

$$Vshell := \frac{\pi \cdot Dshell^2 \cdot Lshell \cdot 0.083^2}{4}$$

$Vshell = 218.415 \ ft^3$

$Vtube = 164.485 \ ft^3$

$VammoniaEVP := Vshell - Vtube$

$VammoniaEVP = 53.93 \ ft^3$

$densityEVP := 0.062 \cdot 18.03 \ \frac{lb}{ft^3}$

$MammoniaEVP := VammoniaEVP \cdot densityEVP$

$MammoniaEVP = 60.286$ lb

It is assumed that the volume of the vapor part in the kettle type evaporator is equivalent to the 1/2 of total shell volume in the evaporator, thus:

$VvaporammoniaEVP := 0.5 \cdot Vshell$

$VvaporammoniaEVP = 109.208 \ ft^3$

$densityEVP := 0.062 \cdot (4.989) \ \frac{lb}{ft^3}$

$MvaporammoniaEVP := VvaporammoniaEVP \cdot densityEVP$

$MvaporammoniaEVP = 33.78$ lb

Condenser

Area of the condenser:

$ACOND := 2.496 \times 10^3 \ ft^2$

The cost of this piece of equipment was calculated based on 3/4-in. or 1-in. O.D. 12ft long tubes on square or triangular pitch, thus:

$Dtube := 1$ in

$Ltube := 12$ ft

$Aouttube := \pi \cdot Dtube \cdot Ltube \cdot 0.083$

Heat transfer area for one tube is:

$Aouttube = 3.129 \ ft^2$

The number of tubes is:

$$Ntube := \frac{ACOND}{Aouttube}$$

$Ntube = 797.692$

According to Table 13.6 of "Product and Process Design Principles" [1], the shell inner diameter is:

$Dshell := 37$ in

$Lshell := 12$ ft

In order to find the amount of ammonia in shell side:

$$Vtube := Ntube \cdot \frac{\pi \cdot Dtube^2 \cdot Ltube \cdot 0.083^2}{4}$$

$$Vshell := \frac{\pi \cdot Dshell^2 \cdot Lshell \cdot 0.083^2}{4}$$

$$VammoniaCOND := Vshell - Vtube$$

$$VammoniaCOND = 37.093 \quad ft^3$$

$$densityCOND := 0.062 \cdot 513.39 \quad \frac{lb}{ft^3}$$

$$MammoniaCOND := VammoniaCOND \cdot densityCOND$$

$$MammoniaCOND = 1.181 \times 10^3 \quad lb$$

It is assumed that the volume of the vapor part of the kettle type condenser is equivalent to the 1/2 of total shell volume in the condenser, thus:

$$VvaporammoniaCOND := 0.5 \cdot Vshell$$

$$VvaporammoniaCOND = 44.443 \quad ft^3$$

$$densityCOND := 0.062 \cdot (18.955) \quad \frac{lb}{ft^3}$$

$$MvaporammoniaCOND := VvaporammoniaCOND \cdot densityCOND$$

$$MvaporammoniaCOND = 52.23 \quad lb$$

Intercooler

Area of the intercooler:

$$AHEX := 2.828 \times 10^3 \quad ft^2$$

The cost of this piece of equipment was calculated based on 3/4-in. or 1-in. O.D. 12ft long tubes on square or triangular pitch, thus:

$$Dtube := 1 \quad in$$

$$Ltube := 12 \quad ft$$

$$Aouttube := \pi \cdot Dtube \cdot Ltube \cdot 0.083$$

Heat transfer area for one tube is:

$\text{Aouttube} = 3.129 \quad ft^2$

The number of tubes is:

$$\text{Ntube} := \frac{\text{AHEX}}{\text{Aouttube}}$$

$\text{Ntube} = 903.796$

According to Table 13.6 of "Product and Process Design Principles" [1], the shell inner diameter is:

$\text{Dshell} := 37 \quad in$

$\text{Lshell} := 12 \quad ft$

In order to find the amount of ammonia in shell side:

$$\text{Vtube} := \text{Ntube} \cdot \frac{\pi \cdot \text{Dtube}^2 \cdot \text{Ltube} \cdot 0.083^2}{4}$$

$$\text{Vshell} := \frac{\pi \cdot \text{Dshell}^2 \cdot \text{Lshell} \cdot 0.083^2}{4}$$

$\text{VammoniaHEX} := \text{Vshell} - \text{Vtube}$

$\text{VammoniaHEX} = 30.204 \quad ft^3$

$\text{densityHEX} := 0.062 \cdot (9.432) \quad \frac{lb}{ft^3}$

$\text{MammoniaHEX} := \text{VammoniaHEX} \cdot \text{densityHEX}$

$\text{MammoniaHEX} = 17.663 \quad lb$

Low Pressure Compressor

The net volume for ammonia vapor in the compressor is assumed to be 0.7*2.0 m3:

$\text{Vcompressor1} := 0.7 \cdot 2.0 \cdot 3.28^3$

$\text{Vcompressor1} = 49.403 \quad ft^3$

$\text{densityCOMP1} := 0.062 \cdot (9.034) \quad \frac{lb}{ft^3}$

$\text{MammoniaCOMP1} := \text{Vcompressor1} \cdot \text{densityCOMP1}$

$\text{MammoniaCOMP1} = 27.671 \quad lb$

High Pressure Compressor

The net volume for ammonia vapor in the compressor is assumed to be 0.7*2.0 m3:

$Vcompressor2 := 0.7 \cdot 2.0 \cdot 3.28^3$

$Vcompressor2 = 49.403 \quad ft^3$

$densityCOMP2 := 0.062 \cdot (18.955) \quad \frac{lb}{ft^3}$

$MammoniaCOMP2 := Vcompressor2 \cdot densityCOMP2$

$MammoniaCOMP2 = 58.058 \quad lb$

Pipe1

Pipe between condenser and evaporator is assumed to have the following length and diameter:

$Lp := 20 \quad ft$

$Dp := 5 \quad in$

To cross check the above figures, the velocity is calculated and then it is compared to the typical values for velocity according to rules of thumb:

$$Apipe := \frac{\pi \cdot Dp^2 \cdot \left(6.944 \times 10^{-3}\right)}{4}$$

$Apipe = 0.136 \quad ft^2$

Volumetric flow in pipe:

$Q := 7.575 \times 10^{-3} \cdot (35.288)$

$Q = 0.267 \quad \frac{ft^3}{sec}$

$$Velpipe := \frac{Q}{Apipe}$$

$Velpipe = 1.961 \quad \frac{ft}{sec}$

This velocity complies with the range of medium viscosity liquid velocity. Therefore the ammonia inventory in this pipe is calculated as follows:

$VammoniaPIPE1 := Apipe \cdot Lp$

$VammoniaPIPE1 = 2.727 \quad ft^3$

$$densitypipe1 := 513.395\cdot 0.062 \quad \frac{lb}{ft^3}$$

$$MammoniaPIPE1 := densitypipe1 \cdot VammoniaPIPE$$

$$MammoniaPIPE1 = 86.799 \quad lb$$

Pipe2

Pipe between evaporator and Compressor1 is assumed to have the following length and diameter :

$$Lp := 20 \quad ft$$

$$Dp := 7 \quad in$$

To cross check the above figures, the velocity is calculated and then it is compared to the rules of thumb for velocity:

$$Apipe := \frac{\pi \cdot Dp^2 \cdot \left(6.944\times 10^{-3}\right)}{4}$$

$$Apipe = 0.267 \quad ft^2$$

Volumetric flow in pipe:

$$Q := 0.779\cdot(35.288)$$

$$Q = 27.489 \quad \frac{ft^3}{sec}$$

$$Velpipe := \frac{Q}{Apipe}$$

$$Velpipe = 102.865 \quad \frac{ft}{sec}$$

This velocity is within the range for a medium pressure vapor. Therefore the ammonia inventory in this pipe is calculated as follows:

$$VammoniaPIPE2 := Apipe \cdot Lp$$

$$VammoniaPIPE2 = 5.345 \quad ft^3$$

$$densityPIPE2 := 0.062\cdot(4.989) \quad \frac{lb}{ft^3}$$

$$MammoniaPIPE2 := VammoniaPIPE2\cdot densityPIPE2$$

$$MammoniaPIPE2 = 1.653 \quad lb$$

Pipe3

Pipe between Compressor1 and intercooler is assumed to have the following length and diameter:

$Lp := 6.56$ ft

$Dp := 7$ in

To cross check the above figures, the velocity is calculated and then it is compared to the rules of thumb for velocity:

$$Apipe := \frac{\pi \cdot Dp^2 \cdot \left(6.944 \times 10^{-3}\right)}{4}$$

$Apipe = 0.267$ ft^2

Volumetric flow in pipe:

$Q := 0.43 \cdot (35.288)$

$Q = 15.174$ $\frac{ft^3}{sec}$

$$Velpipe := \frac{Q}{Apipe}$$

$Velpipe = 56.781$ $\frac{ft}{sec}$

This velocity is within the range for a medium pressure vapor. Therefore the ammonia inventory in this pipe is calculated as follows:

$VammoniaPIPE3 := Apipe \cdot Lp$

$VammoniaPIPE3 = 1.753$ ft^3

$densityPIPE3 := 0.062 \cdot (9.034)$ $\frac{lb}{ft^3}$

$MammoniaPIPE3 := VammoniaPIPE3 \cdot densityPIPE3$

$MammoniaPIPE3 = 0.982$ lb

Pipe4

Pipe between intercooler and Compressor2 is assumed to have the following length and diameter:

$Lp := 6.56$ ft

$Dp := 7$

To cross check the above figures, the velocity is calculated and then it is compared to the rule of thumb for velocity:

$$Apipe := \frac{\pi \cdot Dp^{2} \cdot \left(6.944 \times 10^{-3}\right)}{4}$$

$Apipe = 0.267 \quad ft^{2}$

Volumetric flow in pipe:

$Q := 0.412 \cdot (35.288)$

$Q = 14.539 \quad \frac{ft^{3}}{sec}$

$Velpipe := \frac{Q}{Apipe}$

$Velpipe = 54.404 \quad \frac{ft}{sec}$

This velocity is within the range for a medium pressure vapor. Therefore the ammonia inventory in this pipe is calculated as follows:

$VammoniaPIPE4 := Apipe \cdot Lp$

$VammoniaPIPE4 = 1.753 \quad ft^{3}$

$densityPIPE4 := 0.062 \cdot (9.432) \quad \frac{lb}{ft^{3}}$

$MammoniaPIPE4 := VammoniaPIPE4 \cdot densityPIPE4$

$MammoniaPIPE4 = 1.025 \quad lb$

Pipe5

Pipe between Compressor2 and condenser is assumed to have the following length and diameter:

$Lp := 20 \quad ft$

$Dp := 5 \quad in$

To cross check the above figures, the velocity is calculated and then it is compared to the rule of thumb for velocity:

$$Apipe := \frac{\pi \cdot Dp^{2} \cdot \left(6.944 \times 10^{-3}\right)}{4}$$

$Apipe = 0.136 \quad ft^{2}$

Volumetric flow in pipe:

$Q := 0.205(35.288)$

$Q = 7.234 \quad \frac{ft^3}{sec}$

$Velpipe := \frac{Q}{Apipe}$

$Velpipe = 53.057 \quad \frac{ft}{sec}$

This velocity is within the range for a medium pressure vapor. Therefore the ammonia inventory in this pipe is calculated as follows:

$VammoniaPIPE5 := Apipe \cdot Lp$

$VammoniaPIPE5 = 2.727 \quad ft^3$

$densityPIPE5 := 0.062(18.955) \quad \frac{lb}{ft^3}$

$MammoniaPIPE5 := VammoniaPIPE5 \cdot densityPIPE5$

$MammoniaPIPE5 = 3.205 \quad lb$

Total amount of ammonia in the cycle:

$SUM1 := MammoniaPIPE1 + MammoniaPIPE2 + MammoniaPIPE3 + MammoniaPIPE4$

$SUM2 := MammoniaPIPE5 + MammoniaEVP + MammoniaCOND + MvaporammoniaEVP$

$SUM3 := MvaporammoniaCOND + MammoniaCOMP1 + MammoniaCOMP2 + MammoniaHEX$

$Mtotal := SUM1 + SUM2 + SUM3$

$Mtotal = 1.524 \times 10^3 \quad lb$

$Mtotal := 0.454 Mtotal$

The rough amount of ammonia inventory in the cycle is:

$Mtotal = 691.921 \quad kg$

References

W. D. Seider, J.D. Seader, D.R. Lewin, "Product & Process Design Principles", John Wiley and Sons, Inc., 2004.

Appendix 2 Preliminary Estimation for Ammonia Cycle

Estimating the vapor pressure of ammonia

(Ref: RPP, The properties of Gases and Liquids, 4th Edition)

Parameters for Wagner equation:

$VPA := 45.327$ $VPB := 4104.67$ $VPC := -5.146$ $VPD := 615$

$t := 15$ C

$pvap := 8$ Initial guess to solve the nonlinear equation below (t in C and Ps in bar)

$$Ps(T) := \text{root}\left[\ln(pvap) - \left[(VPA) - \frac{VPB}{t+273} + VPC\ln(t+273) + \frac{VPD\cdot pvap}{(t+273)^2}\right], pvap\right]$$

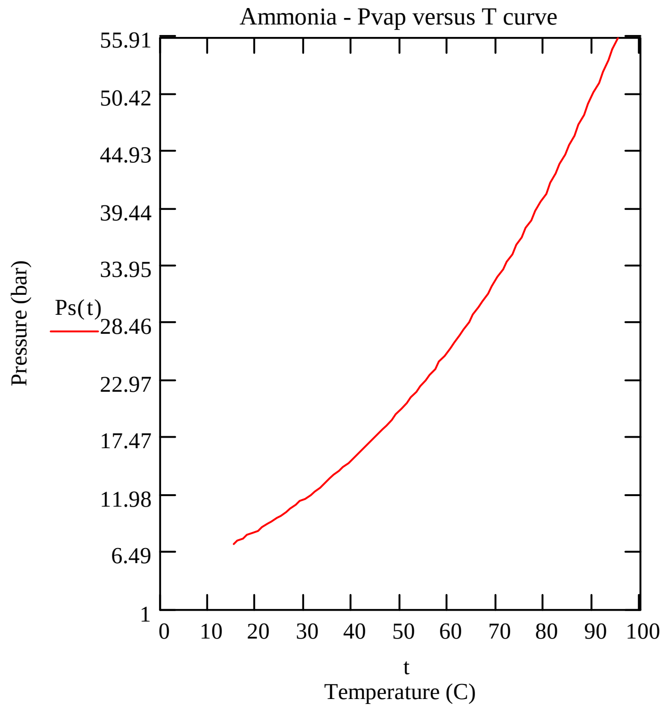
$Ps(T) = 7.297$ bar Saturation pressure at 15C

$t := 15, 16..95$ Temperature range (C) to construct the curve Pvap (bar) versus t (C)

$pvap := 8$ Initial guess to solve the nonlinear equation for all points

$$Ps(t) := \text{root}\left[\ln(pvap) - \left[(VPA) - \frac{VPB}{t+273} + VPC\ln(t+273) + \frac{VPD\cdot pvap}{(t+273)^2}\right], pvap\right]$$

Ps(t) =	t =
7.297	15
7.539	16
7.786	17
8.045	18
8.302	19
8.577	20
8.851	21
9.131	22
9.416	23
9.722	24
10.025	25
10.335	26
10.653	27
10.977	28
11.309	29
11.647	30



Calculating the mass flow rate of ammonia: energy balance in the ammonia evaporator:

mwastewater := 291800 kg/h Twastewaterin := 25 + 273 Twastewaterout := 18 + 273

Heuristics for heat exchangers: Considering minimum approach temperature of 3 C between water out and ammonia in, the vaporization of ammonia should occur at 15 C. The vapor pressure of ammonia at 15 C is 7.3 bar.

$hvap := 627.3\cdot\frac{0.252}{0.454}$ $hvap = 348.193$ kcal/kg

Ref : Smith & van Ness, 4th edition, Page 285, Table 9.2, Thermodynamic Properties of saturated ammonia

$hliq := 109.2\cdot\frac{0.252}{0.454}$ $hliq = 60.613$ kcal/kg

Values taken at 15.5C and 7.4 bar

Using a correlation CP(T) for water and ammonia

Constants for water liquid

Ref : Smith & van Ness, 4th edition,
Page 114,Table 4.3, Heat capacity of liquids
(Validity: T from 273.15 to 373.15 K)

$CPAwliq := 8.712 \qquad CPBwliq := 1.25\cdot 10^{-3} \qquad CPCwliq := -0.18\cdot 10^{-6}$

Constants for ammonia liquid

$CPAamliq := 22.626 \qquad CPBamliq := -100.75\cdot 10^{-3} \qquad CPCamliq := 192.71\cdot 10^{-6}$

$R := 8.314$ J/(mol.K)

$$CPw(T) := R\cdot\frac{1000}{18}\cdot\left(CPAwliq + CPBwliq\, T + CPCwliq\, T^2\right)$$ J/(kg.K)

$$cpwastewater := \frac{\int_{Twastewaterin}^{Twastewaterout} CPw(T)\, dT}{Twastewaterout - Twastewaterin}$$

$cpwastewater = 4.187\times 10^3$ J/(kg.K)

$$mammonia := \frac{mwastewater\cdot cpwastewater\cdot(Twastewaterin - Twastewaterout)}{(hvap - hliq)\cdot 1000\cdot 4.18}$$

$mammonia = 7.114\times 10^3$ kg/h

This is the amount of ammonia that must be circulating in the heat pump cycle. This data serve as input for a tear stream required by Aspen for the simulation of the whole system.

Ammonia will be vaporizing at ~15 C and 7.4 bar and removing heat from the waste heat stream.

The clean water (tap water) will be upgraded in the ammonia condenser.
Heuristic for the condenser: Minimum approach of ~3C. Therefore the condensation temperature will be 93-95C. Therefore the ammonia compressor must operate at a discharge pressure, which is correspondent to 93C(~200F) via isentropic compression. According to Figure 9.4 - Pressure/enthalpy for ammonia (Smith & van Ness, 4th edition, Page 284), the pressure at the compressor discharge should be ~17.22 bar.

Preliminary calculation the mass flow rate of clean water: energy balance in the ammonia condenser (this value serves only as preliminary guess).

$hg := 685 \cdot \frac{0.252}{0.454}$ $\quad hg = 380.22$ kcal/kg

Ref : Smith & van Ness, 4th edition, Page 284, Table 9.4 Pressure/enthalpy diagram for ammonia

$hL := 170 \cdot \frac{0.252}{0.454}$ $\quad hL = 94.361$ kcal/kg

Values taken at ~17 bar

$Tcleanwaterin := 60$ C $\qquad Tcleanwaterout := 90$ C

Using the same correlation CP(T) for water as above

$$cpcleanwater := \frac{\int_{Tcleanwaterin}^{Tcleanwaterout} CPw(T)\, dT}{Tcleanwaterout - Tcleanwaterin}$$

$cpcleanwater = 4.067 \times 10^{3}$ J/(kg.K)

$$mcleanwater := \frac{mammonia \cdot (hg - hL) \cdot 1000 \cdot 4.18}{cpcleanwater \cdot (Tcleanwaterout - Tcleanwaterin)}$$ kg/h

$mcleanwater = 6.968 \times 10^{4}$ kg/h

Appendix 3 Economic Evaluation of Ammonia Heat Pump

The Main Design

All the costs have been evaluated based on "Product and Process Design Principles" [1].

According to Chemical engineering Magazine (2), the Annual CE Plant Cost Index for 2000 (Base Case):

$CE1 := 394.1$

The Annual CE Plant Cost Index for 2004:

$CE2 := 444.2$

The cost of equipments are calculated, as follows:

Basis of Compressor Design:

1. Maximum discharge pressure of the compressor corresponds to a maximum temperature of 375 F (190 C) (heuristics #35, page 186 [1]).
2. Maximum compression ratio = 4 for each stage (Heuristics #36, page 186 [1]).

$Psuction := 6.5$ bar $\qquad Pdischarge := 38$ bar

These values are just initial guess for the simulation. Final values will be calculated via Aspen Simulation.

$$Ratio := \frac{Pdischarge}{Psuction}$$

$Ratio = 5.846$

Ratio > 4, 4 < Ratio < 16, from the table, page 186, the number of stages must be 2.

$nstages := 2$

For equal compression ratios, yields:

$$ratioperstage := Ratio^{\frac{1}{nstages}}$$

$ratioperstage = 2.418$

The pressure ratio in each compression stage will be 2.32

There will be two compression stages with an inter-stage receiver between them. The initial guess pressure profile will be as follows:

Psucction 1 = 6.5 bar Pdischarge 1 = 14.6 bar
P receiver inlet = 14.6 bar
Psucction 2 = 14.6 bar Pdischarge 2 = 38 bar

According to heuristics #31, the following applies:
ΔP for condenser and evaporator = 1.5 psi (0.103 bar) for boiling and condensing fluids

Inter-stage receiver oulet temperature should be higher than 100 F (38C), according to heuristics #35 [1].

Low Pressure Compressor:

Reciprocating compressor has been selected. The points to be considered with regard to this type of compressor are:
1- High efficiency (80%-90%)
2-More expensive
3-Larger in size
4-More flexible in operation
5-Large foundation and more maintenance are required
6-Less noisy
7-Must be protected by a knock-out drum

It should be mentioned that the compressor type is in compliance with reference [3]

The compressor is driven by a electric motor with a typical efficiency of 95% for 1000 hp. Thus, the driven factor for electric motor is:

$FD := 1$

Carbon Steel material factor has been applied:

$FM := 1.0$

Low pressure compressor Break HorsePower is:

$hp1 := 924.764$ hp

$CB1 := \exp(7.6084 + 0.80 \cdot \ln(hp1))$

Capital Cost for low pressure compressor:

$$CpCOMP1 := FD \cdot FM \cdot CB1 \cdot \left(\frac{CE2}{CE1}\right)$$

$CpCOMP1 = 5.359 \times 10^5$ \$

High Pressure Compressor :

Reciprocating compressor has been selected. The points to be considered with regard to this type of compressor are:
1- High efficiency (80%-90%)
2-More expensive
3-Larger in size
4-More flexible in operation
5-Large foundation and more maintenance are required
6-Less noisy
7-Must be protected by a knock-out drum

It should be mentioned that the compressor type is in compliance with reference [3].

The compressor is driven by a electric motor with a typical efficiency of 95% for 1000 hp. Thus, the driven factor for electric motor is:

$FD := 1$

Carbon Steel material factor has been applied:

$FM := 1.0$

High pressure compressor Break HorsePower is:

$hp2 := 1.268 \times 10^{3}$ hp

$CB2 := \exp(7.6084 + 0.80 \cdot \ln(hp2))$

Capital Cost for high pressure compressor :

$$CpCOMP2 := FD \cdot FM \cdot CB2 \cdot \left(\frac{CE2}{CE1}\right)$$

$CpCOMP2 = 6.898 \times 10^{5}$ $

Air Blower:

This blower is overdesigned to cope with the fluctuations in the winter time.

$hpBLOWER := 3.017 \times 10^{3}$ hp

$CBLOWER := \exp(6.6547 + 0.79 \cdot \ln(hpBLOWER))$

For cast iron, the material factor is:

$FM := 0.6$

Capital Cost for air blower:

$$CpBLOWER := FM \cdot CBLOWER \cdot \left(\frac{CE2}{CE1}\right)$$

$$CpBLOWER = 2.945\times 10^{5} \quad \$$$

Evaporator :

Area of the evaporator: $AEVP := 7.927\times 10^{3}$ ft^{2}

The material of construction of the shell and tube sides is carbon steel, thus:

$$FM := 0 + \left(\frac{AEVP}{100}\right)^{0}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PEVPShell := 81.029$ psig

$$FP := 0.9803 + 0.018\left(\frac{PEVPShell}{100}\right) + 0.0017\left(\frac{PEVPShell}{100}\right)^{2}$$

$$FP = 0.996$$

The evaporator is of Kettle Vaporizer type:

$$CBEVP := \exp\left[11.967 - 0.8709\ln(AEVP) + 0.09005(\ln(AEVP))^{2}\right]$$

The purchase cost of the evaporator is:

$$CpEVP := FP \cdot FM \cdot FL \cdot CBEVP\left(\frac{CE2}{CE1}\right)$$

$$CpEVP = 1.13\times 10^{5} \quad \$$$

Condenser:

Area of the condenser: $ACOND := 2.36\times 10^{3}$ ft^{2}

The material of construction of the shell and tube sides is carbon steel, thus:

$$FM := 0 + \left(\frac{ACOND}{100}\right)^{0}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PCONDShell := 536.447$ psig

$$FP := 0.9803 + 0.018\left(\frac{PCONDShell}{100}\right) + 0.0017\left(\frac{PCONDShell}{100}\right)^{2}$$

$$FP = 1.126$$

The condenser is of Kettle Vaporizer type:

$$CBCOND := \exp\left[11.967 - 0.8709\cdot\ln(ACOND) + 0.09005\cdot(\ln(ACOND))^{2}\right]$$

The purchase cost of the condenser is:

$$CpCOND := FP\cdot FM\cdot FL\cdot CBCOND\left(\frac{CE2}{CE1}\right)$$

$CpCOND = 5.906\times 10^{4}$ \$

Fired Heater:

The cost of the fired heater depends on absorbed heat. Since the operational pressure is not too high and the material of construction is carbon steel, there is no need to apply any correction factor.

This fired heater is designed in such a way to cope with the fluctuations in the winter time. Furthermore, the flue gas heat exchanger and the furnace are combined:

$QFurnace := 3.554 \times 10^{8} \quad \frac{Btu}{hr}$

$QFGHX := 1.449 \times 10^{8} \quad \frac{Btu}{hr}$

$Qduty := QFGHX + QFurnace$

$Qduty = 5.003 \times 10^{8} \quad \frac{Btu}{hr}$

Pressure factor: $FP := 1$

Material Factor: $FM := 1$

$CBFIRHEAT := 0.512 \cdot (Qduty)^{0.81}$

$CpFIRHEAT := FP \cdot FM \cdot CBFIRHEAT \cdot \left(\frac{CE2}{CE1}\right)$

$CpFIRHEAT = 6.421 \times 10^{6} \quad \$$

Pump:

Centrifugal pump is selected because all the requirments in terms of volumetric flow rate, developed head and NPSH are met.

$Q := 7.91 \times 10^{3} \quad gpm$

$NPSH := 116.285 \quad ft$

$H := 259.046 \quad ft$

$S := Q \cdot (H)^{0.5}$

$S = 1.273 \times 10^{5} \quad gpm \cdot ft^{0.5}$

The material of construction is cast iron, thus:

$FM := 1.0$

The type of pump is 1 stage radial centrifugal pump (HSC) with 3600 shaft rpm, thus:

$FT := 1.70$

$CBPUMP := \exp\left(9.2951 - 0.6019 \ln(S) + 0.0519 \ln(S)^{2}\right)$

$CpPump := FT \cdot FM \cdot CBPUMP \cdot \left(\frac{CE2}{CE1}\right)$

The cost of electric motor for pump is calculated as follows:

Pump brake horsepower: $PB := 650.08$ hp

$$\eta m := 0.80 + 0.0319\ln(PB) - 0.00182(\ln(PB))^2$$

$$PC := \frac{PB}{\eta m}$$

The motor type of "Totally enclosed, fan-cooled enclosure" with 3600 rpm is selected, thus the type factor for the electric motor is:

$$FTmotor := 1.4$$

$$CBMOTOR := \exp\left[5.4866 + 0.13141\ln(PC) + 0.053255(\ln(PC))^2 + 0.028628(\ln(PC))^3 - 0.0035549(\ln(PC))^4\right]$$

$$CpMotor := FTmotor \cdot CBMOTOR \cdot \left(\frac{CE2}{CE1}\right)$$

$$CpPumpTotal := CpPump + CpMotor$$

$$CpPumpTotal = 6.267 \times 10^4 \quad \$$$

A spare pump needs to be added.

Heat Exchanger Inter-Cooler:

Area of the inter-cooler: $AHXINCOOLER := 2.828 \times 10^3$ ft^2

The material of construction of the shell and tube sides is carbon steel, thus:

$$FM := 0 + \left(\frac{AHXINCOOLER}{100}\right)^0$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXINCOOLERShell := 202.861$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXINCOOLERShell}{100}\right) + 0.0017\left(\frac{PHXINCOOLERShell}{100}\right)^2$$

$$FP = 1.024$$

The heat exchanger is of floating head type:

$$CBHXINCOOLER := \exp\left[11.667 - 0.8709\ln(AHXINCOOLER) + 0.09005(\ln(AHXINCOOLER))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXINCOOLER := FP \cdot FM \cdot FL \cdot CBHXINCOOLER\left(\frac{CE2}{CE1}\right)$$

$$CpHXINCOOLER = 4.39 \times 10^4 \quad \$$$

Ammonia Collector Vessel:

The amount of ammonia in the cycle is calculated to be:

$$mammonia := 70(\quad kg$$

Therefore, considering an over design factor of 0.15, the volume of the collector vessel is calculated as follows:

$$densityammonia := 513.4 \quad \frac{kg}{m^3}$$

$$vol := \frac{mammonia \cdot 1.15}{densityammonia}$$

$$vol = 1.568 \quad m^3$$

The ratio of length to diameter is assumed to be 2.5:

$$D := 1 \qquad L := 2.5$$

Given

$$L = 2.5 \cdot D$$

$$vol = \frac{\pi \cdot D^2 \cdot L}{4}$$

$$y := Find(D, L)$$

$$y = \begin{pmatrix} 0.928 \\ 2.319 \end{pmatrix}$$

Therefore vessel's dimensions are as follows:

$D := 0.928\cdot 3.28$ ft

$L := 2.319\cdot 3.28$ ft

Density of carbon steel:

$density := 490\ \frac{lb}{ft^3}$

Since the operating pressure of vessel is 37.5 bar, the design pressure is calculated as:

$P := 537$ psig

$$Pd := \exp\left[0.60608 + 0.91615\ln(P) + 0.0015655(\ln(P))^2\right]$$

$Pd = 618.227$ psig

The operating temperature of vessel is 160 F, thus the design temperature is:

$Td := 230$ F

For SA-387B which is a commonly used low-alloy (1%Cr and 0.5% Mo) steel for non-corrosive environment including the presense of hydrogen, operating at this design temperature the maximum allowable stress is:

$S := 15000$ psi

For carbon steel the value of welding efficiency is:

$E := 0.85$

Based on the above values, the cylindrical shell wall thickness is computed from ASME pressure-vessel code as follows:

$$tp := \frac{Pd\cdot D\cdot 12}{2\cdot S\cdot E - 1.2\cdot Pd}$$

$tp = 0.912$ inch

The weight of the shell and the two heads is approximately:

$$W := \pi\cdot\left(D + \frac{tp}{12}\right)\cdot(L + 0.8\cdot D)\cdot\frac{tp\cdot density}{12}$$

$W = 3.665\times 10^3$ lb

The material of construction is carbon steel:

$FM := 1.0$

The cost of horizontal vessels are estimated by:

$$Cv := \exp\left[(8.717 - 0.2330\ln(W)) + 0.0433(\ln(W))^2\right]$$

The added cost for platforms and ladders is given by:

$$Cp := 1580 \cdot D^{0.20294}$$

Therefore the cost of this vessel is;

$$CpVESSEL := FM \cdot Cv + Cp$$

$$CpVESSEL = 1.865 \times 10^4 \quad \$$$

Heat Exchanger District 1:

Area of the HXDIS1: $AHXDIS1 := 1.676 \times 10^4 \quad ft^2$

The material of construction of the shell side and the tube side is carbon steel, thus:

$$FM := 0.00 + \left(\frac{AHXDIS1}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$$PHXDIS1Shell := 170.983 \quad psig$$

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS1Shell}{100}\right) + 0.0017\left(\frac{PHXDIS1Shell}{100}\right)^2$$

$$FP = 1.016$$

The heat exchanger is of floating head type:

$$CBHXDIS1 := \exp\left[11.667 - 0.8709\ln(AHXDIS1) + 0.09005(\ln(AHXDIS1))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS1 := FP \cdot FM \cdot FL \cdot CBHXDIS1 \left(\frac{CE2}{CE1}\right)$$

$CpHXDIS1 = 1.571 \times 10^{5}$ $

Heat Exchanger District 2:

Area of the HXDIS2: $AHXDIS2 := 1.676 \times 10^{4}$ ft^2

The material of construction of the shell side and the tube side is carbon steel, thus:

$$FM := 0.00 + \left(\frac{AHXDIS2}{100}\right)^{0.00}$$

$FM = 1$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS2Shell := 168.113$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS2Shell}{100}\right) + 0.0017\left(\frac{PHXDIS2Shell}{100}\right)^2$$

$FP = 1.015$

The heat exchanger is of floating head type:

$$CBHXDIS2 := \exp\left[11.667 - 0.8709\ln(AHXDIS2) + 0.09005(\ln(AHXDIS2))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS2 := FP \cdot FM \cdot FL \cdot CBHXDIS2\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS2 = 1.57 \times 10^{5}$ $

Heat Exchanger District 3:

Area of the HXDIS3: $AHXDIS3 := 1.676\times 10^{4}$ ft^2

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS3}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS3Shell := 127.736$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS3Shell}{100}\right) + 0.0017\left(\frac{PHXDIS3Shell}{100}\right)^{2}$$

$$FP = 1.006$$

The heat exchanger is of floating head type:

$$CBHXDIS3 := \exp\left[11.667 - 0.8709\ln(AHXDIS3) + 0.09005(\ln(AHXDIS3))^{2}\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS3 := FP\cdot FM\cdot FL\cdot CBHXDIS3\left(\frac{CE2}{CE1}\right)$$

ft^2

$CpHXDIS3 = 1.555\times 10^{5}$ \$

Heat Exchanger District 4:

Area of the HXDIS4: $AHXDIS4 := 1.676\times 10^{4}$

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS4}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS4Shell := 127.10$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS4Shell}{100}\right) + 0.0017\left(\frac{PHXDIS4Shell}{100}\right)^2$$

$FP = 1.006$

The heat exchanger is of floating head type:

$$CBHXDIS4 := \exp\left[11.667 - 0.8709\ln(AHXDIS4) + 0.09005(\ln(AHXDIS4))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS4 := FP \cdot FM \cdot FL \cdot CBHXDIS4\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS4 = 1.555 \times 10^5$ \$

Heat Exchanger District 5:

Area of the HXDIS5: $AHXDIS5 := 1.676 \times 10^4$ ft^2

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS5}{100}\right)^{0.00}$$

$FM = 1$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS5Shell := 127.42$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS5Shell}{100}\right) + 0.0017\left(\frac{PHXDIS5Shell}{100}\right)^2$$

$FP = 1.006$

The heat exchanger is of floating head type:

$$CBHXDIS5 := \exp\left[11.667 - 0.8709\ln(AHXDIS5) + 0.09005(\ln(AHXDIS5))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS5 := FP \cdot FM \cdot FL \cdot CBHXDIS5\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS5 = 1.555 \times 10^5$ \$

Total Purchase Cost (The main case):

$SUM1 := CpCOMP1 + CpCOMP2 + CpEVP + CpCOND + CpHXINCOOLEI$

$SUM2 := CpFIRHEAT + CpBLOWER + 2 \cdot CpPumpTotal + CpVESSEI$

$SUM3 := CpHXDIS1 + CpHXDIS2 + CpHXDIS3 + CpHXDIS4 + CpHXDIS5$

$capitalcost := SUM1 + SUM2 + SUM3$

$capitalcost = 9.082 \times 10^6$ \$

Costs of the Alternative Design

The costs were also calculated for an alternative case in which two tanks and a smaller furnace were used to cope with the demand fluctuations.

Air Blower:

This blower is overdesigned to cope with the fluctuations in the winter time.

$hpBLOWER := 1.81\times 10^{3}$ hp

$CBLOWER := \exp(6.6547 + 0.79\cdot \ln(hpBLOWER))$

Capital Cost for air blower:

$FM := 0.6$

$CpBLOWER := FM\cdot CBLOWER\cdot\left(\frac{CE2}{CE1}\right)$

$CpBLOWER = 1.967\times 10^{5}$ \$

Fired Heater:

The cost of the fired heater depends on absorbed heat. Since the operational pressure is not too high and the material of construction is carbon steel, there is no need to apply any correction factor.

The flue gas heat exchanger and the furnace are combined:

$QFurnace := 2.132\times 10^{8}\ \frac{Btu}{hr}$

$QFGHX := 1.449\times 10^{8}\ \frac{Btu}{hr}$

$Qduty := QFGHX + QFurnace$

$Qduty = 3.581\times 10^{8}\ \frac{Btu}{hr}$

Pressure factor: $FP := 1$

Material Factor: $FM := 1$

$CBFIRHEAT := 0.512\cdot(Qduty)^{0.81}$

$CpFIRHEAT := FP\cdot FM\cdot CBFIRHEAT\cdot\left(\frac{CE2}{CE1}\right)$

$CpFIRHEAT = 4.898\times 10^{6}$ \$

Tank 1 (Before Condenser) :

The volume of the tank was claculated to be as follows (10% overdesign):

$Vtank := 4200$ m^3

According to rules of thumb the length and diameter ratio is 2.5. For a vertical tank:

initial values: $D := 1$ $L := 3$

Given

$L = 2.5 \cdot D$

$$Vtank = \frac{\pi \cdot D^2 \cdot L}{4}$$

$dim := Find(D, L)$

$$dim = \begin{pmatrix} 12.885 \\ 32.212 \end{pmatrix}$$

thus the diameter and length of the tank are:

$D := 12.885 \cdot 3.281$ ft

$L := 32.212 \cdot 3.281$ ft

In order to evaluate the cost of this tank, first the weight of the tank should be found:

Density of carbon steel: $density := 490$ $\frac{lb}{ft^3}$

Since the operating pressure of the tank is approximately 6 atm (adjusted by a valve before the tank), the design pressure is calculated as follows:

$P := 73.5$ psig

$$Pd := exp\left[0.60608 + 0.91615 \ln(P) + 0.0015655\left(\ln(P)^2\right)\right]$$

$Pd = 96.732$ psig

The operating temperature of the column is approximately 150.8 F, thus the design temperature is:

$Td := 200$ F

For SA-387B which is a commonly used low-alloy (1%Cr and 0.5% Mo) steel for non-corrosive environment including the presense of hydrogen, operating at this design temperature the maximum allowable stress is:

$S := 15000$ psi

For carbon steel the value of welding efficiency is:

$E := 0.85$

Based on the above values, the cylindrical shell wall thickness is computed from ASME pressure-vessel code as follows:

$$tp := \frac{Pd \cdot D \cdot 12}{2 \cdot S \cdot E - 1.2 \cdot Pd}$$

$tp = 1.933$ inch

The weight of the shell and the two heads is approximately:

$$W1 := \pi \cdot \left(D + \frac{tp}{12}\right) \cdot (L + 0.8 \cdot D) \cdot \frac{tp \cdot density}{12}$$

$W1 = 1.468 \times 10^{6}$ lb

The cost of the tank is:

$$CvTANK1 := \exp\left(6.775 + 0.18255 \ln(W1) + 0.02297 \ln(W1)^{2}\right)$$

$CvTANK1 = 1.201 \times 10^{6}$ \$

The cost of ladders and platforms is:

$$Cp1TANK := 285.1 \cdot D^{0.73960} \cdot L^{0.70684}$$

$Cp1TANK = 1.226 \times 10^{5}$ \$

For material of construction of carbon steel:

$Fm := 1.0$

thus the total cost of this tank is:

$$CpTANK1 := (Fm \cdot CvTANK1 + Cp1TANK) \cdot \left(\frac{CE2}{CE1}\right)$$

$CpTANK1 = 1.492 \times 10^{6}$ \$

Tank 2 (Before Furnace) :

The volume of the tank was claculated to be as follows (10% overdesign):

$Vtank := 4200$ m^{3}

According to rules of thumb the length and diameter ratio is 2.5. For a vertical tank:

initial values: $D := 1$ $L := 3$

Given

$L = 2.5 \cdot D$

$Vtank = \frac{\pi \cdot D^2 \cdot L}{4}$

$dim := Find(D, L)$

$dim = \begin{pmatrix} 12.885 \\ 32.212 \end{pmatrix}$

thus the diameter and length of the tank are:

$D := 12.885 \cdot 3.281$ ft

$L := 32.212 \cdot 3.281$ ft

In order to evaluate the cost of this tank, first the weight of the tank should be found:

Density of carbon steel: $density := 490 \frac{lb}{ft^3}$

Since the operating pressure of the tank is approximately 3.5 atm (adjusted by a valve before the tank), the design pressure is calculated as follows:

$P := 36.75$ psig

$Pd := \exp\left[0.60608 + 0.91615 \ln(P) + 0.0015655 \left(\ln(P)^2\right)\right]$

$Pd = 50.823$ psig

The operating temperature of the column is approximately 280 F, thus the design temperature is:

$Td := 320$ F

For SA-387B which is a commonly used low-alloy (1%Cr and 0.5% Mo) steel for non-corrosive environment including the presense of hydrogen, operating at this design temperature the maximum allowable stress is:

$S := 15000$ psi

For carbon steel the value of welding efficiency is:

$E := 0.85$

Based on the above values, the cylindrical shell wall thickness is computed from ASME pressure-vessel code as follows:

$tp := \frac{Pd \cdot D \cdot 12}{2 \cdot S \cdot E - 1.2 \cdot Pd}$

$tp = 1.014$ inch

The weight of the shell and the two heads is approximately:

$$W2 := \pi \cdot \left(D + \frac{tp}{12}\right) \cdot (L + 0.8 \cdot D) \cdot \frac{tp \cdot density}{12}$$

$$W2 = 7.683 \times 10^{5} \quad \text{lb}$$

The cost of the tank is:

$$CvTANK2 := \exp\left(6.775 + 0.18255 \ln(W2) + 0.02297 \ln(W2)^{2}\right)$$

$$CvTANK2 = 7.061 \times 10^{5} \quad \$$$

The cost of ladders and platforms is:

$$Cp2TANK := 285.1 \cdot D^{0.73960} \cdot L^{0.70684}$$

$$Cp2TANK = 1.226 \times 10^{5} \quad \$$$

For material of construction of carbon steel:

$$Fm := 1.0$$

thus the total cost of this tank is:

$$CpTANK2 := (Fm \cdot CvTANK2 + Cp2TANK) \cdot \left(\frac{CE2}{CE1}\right)$$

$$CpTANK2 = 9.34 \times 10^{5} \quad \$$$

Total Purchase Cost (The alternative case):

$$SUM1 := CpCOMP1 + CpCOMP2 + CpEVP + CpCOND + CpHXINCOOLER + CpVESSEL$$

$$SUM2 := CpFIRHEAT + CpBLOWER + 2 \cdot CpPumpTotal + CpTANK1 + CpTANK2$$

$$SUM3 := CpHXDIS1 + CpHXDIS2 + CpHXDIS3 + CpHXDIS4 + CpHXDIS5$$

$$capitalcost := SUM1 + SUM2 + SUM3$$

$$capitalcost = 9.886 \times 10^{6} \quad \$$$

References

1- W. D. Seider, J.D. Seader, D.R. Lewin, "Product & Process Design Principles", John Wiley and Sons, Inc., 2004.

2- Economic Indicators (July 2005): WWW.CHE.COM

3-W. Dimoplon,"What process engineers need to know about compressors", Hydrocrabon Processing , May 1978.

Appendix 4 Basis for The Compressors Simulation

Reference: Seider, W.D., Seader, J.D., Lewin, D. R., Product and Process Design Principles

1. Maximum discharge pressure of the compressor corresponds to a maximum temperature of 375 F (190 C) (heuristics #35, page 186, from the reference mentioned above).

2. Maximum compression ratio = 4 for each stage (Heuristics #36, page 186, from the reference mentioned above).

$Psuction := 6.5$ bar $Pdischarge := 35$ bar

These values are just initial guess for the simulation. Final values will be calculated via Aspen Simulation.

$Ratio := \frac{Pdischarge}{Psuction}$ $Ratio = 5.385$ Ratio > 4, 4 < Ratio < 16, from the table, page 186, the number of stages must be 2.

$nstages := 2$

For equal compression ratios, yields:

$ratioperstage := Ratio^{\frac{1}{nstages}}$ $ratioperstage = 2.32$ The pressure ratio in each compression stage will be 2.32

There will be two compression stages with an inter-stage receiver between them. The initial guess pressure profile will be as follows:

Psucction 1 = 6.5 bar Pdischarge 1 = 14.6 bar
P receiver inlet = 14.6 bar
Psucction 2 = 14.6 bar Pdischarge 2 = 38 bar

According to heuristics #31, the following applies:
ΔP for condenser and evaporator = 1.5 psi (0.103 bar) for boiling and condensing fluids

Inter-stage receiver oulet temperature should be higher than 100 F (38C), according to heuristics #35.

Appendix 5 Compressor Scheme

An illustration of a two-stage compressor unit with intermediate cooling system is shown in Figures A5.1 and Figure A5.2

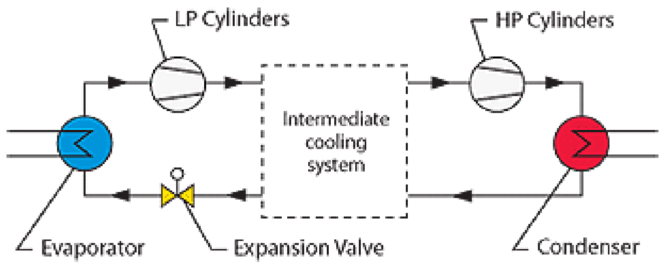


Figure A5.1 Simplified scheme of two-stage compressor with intermediate gas cooling

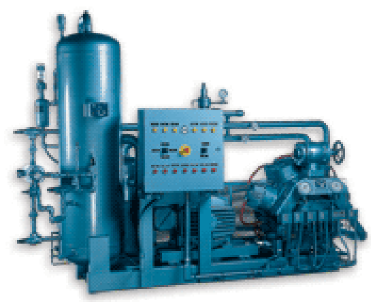


Figure A5.2 Two stage compressor unit model TSMC 108 with intermediate cooling system

Reference :

http://www.mmrefrigeration.com/mmsite/recip_cooling.html

Appendix 6 Piping Calculations

References:

1. Crane, Flow of Fluids Through Valves, Fittings and Pipe - Technical Pape No. 410, 25th Printing, Technical1988- Crane Co.
2. Coulson, J.M., Richardson, J.F., Chemical Engineering; Volume 6 (SI Units), BPCC Wheatons Ltd. Exeter: 1991
3. Coulson, J.M., Richardson, J.F., Chemical Engineering; Volume 1; Flow, Heat Transfer and Mass Transfer, Sixth edition The Bath Press, Bath:1999
4. Walas, Stanley M., *Chemical Process Equipment*, Butterworth-Heinemann, Series in Chemical Engineering, USA, 1983

Preliminary Assumptions :

1. Total pipe length from the Upgrading Unit to districts (**main header of hot tap water pipe**) : 10 km (taking into consideration the distribution of districts to be connected over the area shown in the maps of Delft available on the net [http://www.map24.nl].
2. Total pipe length from the districts to the Upgrading Unit (**main header of cold tap water pipe**) : 10 km.
3. This calculation does not take into consideration the branching for pipes inside the districts. Futher detailed calculations in this regard must be carried during the detailed design phase.
4. Maximum tap water flow rate 5.32 $*10^6$ kg/h. Calculated based on the worst case scenario (two furnaces operating in winter time)
5. Minimum tap water flow rate 2.13 $*10^5$ kg/h. Calculated based on the highest demand peak in the summer.

Piping calculations for the maximum capacity:

Estimating the piping diameter for hot tap water:

$Wmflowrate := 5.32\times 10^6$ kg/h Water mass flow rate

$\rho hotwaterin := 963.68$ kg/m^3 Density of water at 92 C = 963 kg/m^3

$$92\cdot 1.8 + 32 = 197.6 \qquad 60.107\frac{0.454}{0.3048^3} = 963.687$$

$\rho hotwaterout := 967.51$ kg/m^3 Density of water at 90 C = 967.519 kg/m^3

$$90\cdot 1.8 + 32 = 194 \qquad 60.343\frac{0.454}{0.3048^3} = 967.471$$

Data gathered from Crane:

$$\rho hotwater := \frac{\rho hotwaterin + \rho hotwaterout}{2}$$ $\rho hotwater = 965.603$ kg/m3

$$Qhotwater := \frac{Wmflowrate}{\rho hotwater}$$ $Qhotwater = 5.51\times 10^{3}$ m3/h

$$waterflowrate := Wmflowrate\cdot\frac{1}{3600}$$ $waterflowrate = 1.478\times 10^{3}$ kg/s

$$d := 260(waterflowrate)^{0.52}\cdot\rho hotwater^{-0.3}$$ $d = 1.471\times 10^{3}$ mm

The first estimation for internal diameter of the hot tap water header is 1471 mm (58 inch). This diameter exceeds the maximum commercial pipe dimension available in the PIPE data bank from Crane [1]. According to this databank, the maximum commercial diameter is 36" (894.1mm or 35.2in).

$\frac{1.471\times 10^{3}}{25.4} = 57.913$ inch

$35.2\cdot 25.4 = 894.08$ mm

$$velocity := \frac{Qhotwater}{\left[\frac{\pi\cdot\left(641.4\, 10^{-3}\right)^{2}}{4}\right]\cdot 3600}$$

$velocity = 4.737$ m/s

$4.37\cdot 0.3048 = 1.332$ ft/s

Rule of thumb: for water (pump discharge line - long line), velocity ranging from 4-7 ft/s (1.22 -.2.13 m/s). Let's calculate the diameter for 7 ft/s (2.134m/s).

$velocity := 2.134$ m/s

$$D := 1000\sqrt{\frac{4\cdot waterflowrate}{\pi\cdot\rho hotwater\cdot velocity}}$$ $D = 955.571$ mm $\frac{955}{25.4} = 37.598$ inch

This diameter is also higher than the maximum available in Crane [1] databank.
Let's use the maximum diameter,i.e., 36" (914.4mm).

Rule of thumb. Ref: Chemical Process Equipment, Schedule number=1000*PSIG/St

Psig is the internal pressure, in psig, St is the allowable working stress ~10000 psig for A120 carbon steel at 500F (932 C)

$Psig := 7 \cdot 14.7$ $\quad Psig = 102.9$ psig Considering the pressure in the pipeline ~8 bara

$St := 10000$ psig

$$Schedulenumber := \frac{1000\,Psig}{St} \qquad Schedulenumber = 10.29$$

The chosen commercial external diameter is: 914.4 mm (36 in), Schedule 10, with internal diameter of 898.6 mm (35.376 in), wall thickness 7.92 mm (0.312 in)

Estimating the piping diameter for cold tap water:

$Wmflowrate := 5.32 \times 10^{6}$ kg/h Water mass flow rate

$\rho coldwaterin := 977.844$ kg/m^3 Density of water at 70 C = 977.8 kg/m^3

$$70 \cdot 1.8 + 32 = 158 \qquad 60.99 \cdot \frac{0.454}{0.3048^3} = 977.844$$

$\rho coldwaterout := 981.2$ kg/m^3 Density of water at 68 C = 981.2 kg/m^3

$$68 \cdot 1.8 + 32 = 154.4 \qquad 61.2 \cdot \frac{0.454}{0.3048^3} = 981.211$$

Data gathered from Crane:

$$\rho coldwater := \frac{\rho coldwaterin + \rho coldwaterout}{2} \qquad \rho coldwater = 979.522 \quad kg/m^3$$

$$Qcoldwater := \frac{Wmflowrate}{\rho coldwater} \qquad Qcoldwater = 5.431 \times 10^{3} \quad m^3/h$$

$$waterflowrate := Wmflowrate \cdot \frac{1}{3600} \qquad waterflowrate = 1.478 \times 10^{3} \quad kg/s$$

$$d := 260(waterflowrate)^{0.52} \cdot \rho coldwater^{-0.3} \qquad d = 1.465 \times 10^{3} \quad mm$$

$\frac{1465}{25.4} = 57.677$ inch

The first estimation for internal diameter of the cold tap water header is 1465 mm (57.7 inch). This diameter exceeds the maximum commercial pipe dimension available in the PIPE data bank from Crane [1]. According to this databank, the maximum commercial diameter is 36" (894.1mm or 35.2in).

$35.2 \cdot 25.4 = 894.08$ mm

$$velocity := \frac{Qcoldwater}{\left[\frac{\pi \cdot \left(894.08\,10^{-3}\right)^2}{4}\right] \cdot 3600}$$

$velocity = 2.403$ m/s

$4.669\,0.3048 = 1.423$ ft/s

Rule of thumb: for water (pump discharge line - long line), velocity ranging from 4-7 ft/s (1.22 -.2.13 m/s). Let's calculate the diameter for 7 ft/s (2.134m/s).

$velocity := 2.134$ m/s

$$D := 1000\sqrt{\frac{4 \cdot waterflowrate}{\pi \cdot \rho coldwater \cdot velocity}}$$

$D = 948.758$ mm

$\frac{948.758}{25.4} = 37.353$ inch

This diameter is also higher than the maximum available in Crane [1] databank.
Let's use the maximum diameter,i.e., 36" (914.4mm).

Rule of thumb. Ref : Chemical Process Equipment, Schedule number=1000*PSIG/St

Psig is the internal pressure, in psig, St is the allowable working stress ~10000 psig for A120 carbon steel at 500F (932 C)

$Psig := 7 \cdot 14.7$ $Psig = 102.9$ psig Considering the pressure in the pipeline ~8 bara

$St := 10000$ psig

$$Schedulenumber := \frac{1000\,Psig}{St}$$

$Schedulenumber = 10.29$

The chosen commercial external diameter is: 914.4 mm (36 in), Schedule 10, with internal diameter of 898.6 mm (35.376 in), wall thickness 7.92 mm (0.312 in)

Appendix 7 Estimation of The Enthalpy and Entropy of The Hot Tap Water at 137 °C and 13 bar

Ref: Program termoprop1 from Sandler
http://www.che.udel.edu/thermo

The program was adapted with respect to the input data entry method, which here uses a matrix other than a data file as in the original program.

THERMODYNAMIC PROPERTIES CALCULATION USING THE
PENG-ROBINSON EQUATION OF STATE FOR A GIVEN T AND P

Property Data given in the matrix M.
Tc (in K), Pc (in bar), omega, Tb
Cp1, Cp2, Cp3, Cp4
Tref (in K), Pref (in bar), Tref, Pref

The matrix should be as follows
(last entry is not used, but must be there)
(In eqn Cp=Cp0+Cp1*T+Cp2*T^2+Cp3*T^4)
(reference conditions, last two entries are there because MATHCAD requires all matrix elements to be filled.)

$$M := \begin{pmatrix} 647.3 & 220.5 & 0.344 & 373.2 \\ 32.24 & 0.001924 & 1.055\,10^{-5} & -3.596\,10^{-9} \\ 298.15 & 1 & 298.15 & 1 \end{pmatrix}$$

$i := 0..3 \qquad j := 0..3$

$Tc := M_{0,0} \qquad Pc := M_{0,1} \qquad om := M_{0,2}$

$Cp_i := M_{1,i} \qquad Trs := M_{2,0} \qquad Prs := M_{2,1}$

$$kap := 0.37464 + 1.54226\,om - 0.26992\,om^2$$

Peng-Robinson Constants: $R := 0.0000831$ $\qquad b := 0.07780\dfrac{R\cdot Tc}{Pc} \qquad ac := 0.45724\dfrac{R^2\cdot Tc^2}{Pc}$

Note that these are being defined as a function of temperature since we will need to interate on temperature.

$$alf(T) := 1\cdot\left[1 + kap\cdot\left(1 - \sqrt{\frac{T}{Tc}}\right)\right]^2 \qquad a(T) := ac\cdot alf(T)$$

$$CA(T,P) := \frac{a(T)\cdot P}{(R\cdot T)^2} \qquad CB(T,P) := \frac{P\cdot b}{R\cdot T} \qquad Da(T) := \frac{d}{dT}a(T)$$

Subroutine for solving the cubic equation of state.

$$Z(T,P) := \left| \begin{array}{l} A \leftarrow CA(T,P) \\ B \leftarrow CB(T,P) \\ V \leftarrow \begin{bmatrix} -\left(A\cdot B - B^2 - B^3\right) \\ A - 3\cdot B^2 - 2\cdot B \\ -(1-B) \\ 1 \end{bmatrix} \\ ZZ \leftarrow \text{polyroots}(V) \\ \text{for } i \in 0..2 \\ \quad \left(ZZ_i \leftarrow 0\right) \text{ if } \left(\text{Im}\left(ZZ_i\right) \neq 0\right) \\ ZZ \leftarrow \text{sort}(ZZ) \\ ZZ_0 \leftarrow ZZ_2 \text{ if } \left(\left|ZZ_0\right| < 10^{-5}\right) \\ ZZ_2 \leftarrow ZZ_0 \text{ if } \left(\left|ZZ_2\right| < 10^{-5}\right) \\ ZZ \end{array} \right.$$

Vector of coefficients in the PR equation
in the form
0=-(A*B-B^2-B^3)+(A-3*B^2-2*B)*Z-(1-B)*Z^2+Z^3

Solution to the cubic

Set any imaginary roots to zero
Sort the roots

Set the value of any imaginary roots
to value of the real root

Enter temperature T, and pressure P for thermodynamic properties calculation.

$T := 137$ **C**

$T := 273.15 + T$ **K** $P := 13$ **bar**

Fugacity expressions [actually ln(f/P)] for the liquid fl and vapor fv

$$fl(T,P) := \left(Z(T,P)_0 - 1\right) - \ln\left(Z(T,P)_0 - CB(T,P)\right) - \frac{CA(T,P)}{2\cdot\sqrt{2}\cdot CB(T,P)}\cdot\ln\left[\frac{Z(T,P)_0 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_0 + \left(1-\sqrt{2}\right)\cdot CB(T,P)}\right]$$

$$fv(T,P) := \left(Z(T,P)_2 - 1\right) - \ln\left(Z(T,P)_2 - CB(T,P)\right) - \frac{CA(T,P)}{2\cdot\sqrt{2}\cdot CB(T,P)}\cdot\ln\left[\frac{Z(T,P)_2 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_2 + \left(1-\sqrt{2}\right)\cdot CB(T,P)}\right]$$

Fugacity | Fugacity coefficient

$fugl := P\cdot\exp(fl(T,P))$ $fugl = 3.18716$ $fl(T,P) = -1.40582$ $phil := \frac{fugl}{P}$ $phil = 0.24517$

$fugv := P\cdot\exp(fv(T,P))$ $fugv = 11.88924$ $fv(T,P) = -0.08932$ $phiv := \frac{fugv}{P}$ $phiv = 0.91456$

Residual entropy for liquid (DELSL) and vapor (DELSV) phases

$$DELSL := \left[R \cdot \ln\left(Z(T,P)_0 - CB(T,P)\right) + \frac{Da(T)}{2\cdot\sqrt{2}\cdot b} \cdot \ln\left[\frac{Z(T,P)_0 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_0 + \left(1-\sqrt{2}\right)\cdot CB(T,P)} \right] \right] \cdot 10^5$$

$$DELSV := \left[R \cdot \ln\left(Z(T,P)_2 - CB(T,P)\right) + \frac{Da(T)}{2\cdot\sqrt{2}\cdot b} \cdot \ln\left[\frac{Z(T,P)_2 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_2 + \left(1-\sqrt{2}\right)\cdot CB(T,P)} \right] \right] \cdot 10^5$$

Residual enthalpy for liquid (DELHL) and vapor (DELHV) phases

$$DELHL := \left[R \cdot T \cdot \left(Z(T,P)_0 - 1\right) + \frac{T\cdot Da(T) - a(T)}{2\cdot\sqrt{2}\cdot b} \cdot \ln\left[\frac{Z(T,P)_0 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_0 + \left(1-\sqrt{2}\right)\cdot CB(T,P)} \right] \right] \cdot 10^5$$

$$DELHV := \left[R \cdot T \cdot \left(Z(T,P)_2 - 1\right) + \frac{T\cdot Da(T) - a(T)}{2\cdot\sqrt{2}\cdot b} \cdot \ln\left[\frac{Z(T,P)_2 + \left(1+\sqrt{2}\right)\cdot CB(T,P)}{Z(T,P)_2 + \left(1-\sqrt{2}\right)\cdot CB(T,P)} \right] \right] \cdot 10^5$$

Ideal gas property changes relative to the reference state

$$DELHIG := Cp_0\cdot(T - Trs) + \frac{Cp_1\cdot\left(T^2 - Trs^2\right)}{2} + \frac{Cp_2\cdot\left(T^3 - Trs^3\right)}{3} + \frac{Cp_3\cdot\left(T^4 - Trs^4\right)}{4}$$

$$DELSIG := Cp_0\cdot\ln\left(\frac{T}{Trs}\right) + Cp_1\cdot(T - Trs) + \frac{Cp_2\cdot\left(T^2 - Trs^2\right)}{2} + \frac{Cp_3\cdot\left(T^3 - Trs^3\right)}{3} - R\cdot 10^5\cdot\ln\left(\frac{P}{Prs}\right)$$

Total entropy and enthalpy relative to ideal gas reference state

$SL := DELSIG + DELSI$ $\quad SV := DELSIG + DELS\backslash$ $\quad HL := DELHIG + DELHI$

$HV := DELHIG + DELHV$

SUMMARY OF RESULTS

$T = 410.15$ K Pressure, bar $P = 13$

	LIQUID	VAPOR
Compressibility	$Z(T,P)_0 = 8.90988\times 10^{-3}$	$Z(T,P)_2 = 0.90689$
Enthalpy, J/mol	$HL = -3.64797\times 10^{4}$	$HV = 2.94801\times 10^{3}$
Entropy, J/mol K	$SL = -97.02364$	$SV = -11.83908$
Fugacity coefficient	$phil = 0.24517$	$phiv = 0.91456$
Fugacity, bar	$fugl = 3.18716$	$fugv = 11.88924$

Note: If the chosen temperature and pressure are near the saturation conditions, results for both the vapor and liquid phases will be reported. The phase with the lower fugacity, and therefore lower Gibbs free energy, is the equilibrium phase. If the temperature and pressure are far away from the saturation conditions, the properties only the equilibrium state are reported for both both phases.

Appendix 8 Tera Joules Calculation

1. Calculating the TJ for the waste water (291800 kg/h) stream at 27.5 C.

Using a correlation CP(T) for water

Constants for water liquid

Ref : Smith & van Ness, 4th edition,
Page 114,Table 4.3, Heat capacity of liquids
(Validity: T from 273.15 to 373.15 K)
T in K

$CPAwliq := 8.712 \qquad CPBwliq := 1.25 \cdot 10^{-3} \qquad CPCwliq := -0.18 \cdot 10^{-6}$

$R := 8.314$ J/(mol.K)

$$CPw(T) := R \cdot \frac{1000}{18} \cdot \left(CPAwliq + CPBwliq \cdot T + CPCwliq \cdot T^2\right)$$ J/(kg.K)

Waste water stream at 27.5 C (range: 25-30C, 27.5C was taken as the value for calculations)

$Wastewater := 291800$ kg/h **Value provided by the Principal**

The reference temperature was taken as 0 C (273 K)

$Twastewaterin := 27.5 + 273 \qquad Twastewaterin = 300.5$ K

$Treference := 273 \qquad Treference = 273$ K

$Twastewaterout := 18 + 273 \qquad Twastewaterout = 291$ K

Evaluating Cp water within the range: 25 - 0 C

$$cpwastewater := \frac{\int_{Treference}^{Twastewaterin} CPw(T)\,dT}{Twastewaterin - Treference}$$

$cpwastewater = 4.183 \times 10^{3}$ J/(kg.K)

$$TJwastewater := Wastewater \cdot cpwastewater \cdot (Twastewaterin - Twastewaterout) \cdot 365 \cdot 24 \cdot 10^{-12}$$

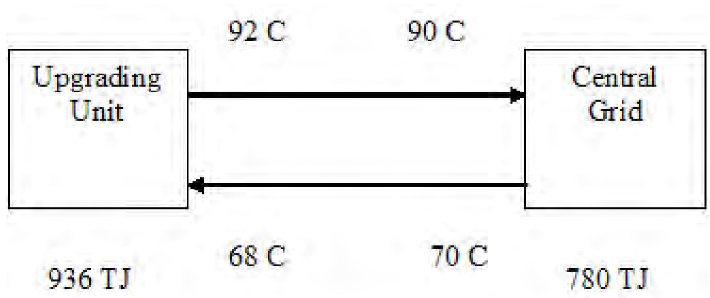
$TJwastewater = 101.571$ TJ/year

This is the heat content of the waste water stream at 27.5 C, considering 0 C as the reference temperature

2. Estimation of the temperature of the returning tap water from the Districts:

Assumptions :
1. 10% energy losses in the pipeline from the Upgrading Unit to the district (typical value of energy losses in similar systems).
2. 20^{o}C temperature approach bewteen IN and OUT tap water stream in the district

10% energy losses represents 2^{o}C difference, which added to the desired temperature, 90^{o}C, yields to 92^{o}C for the tap water outlet stream from the Upgrading Unit. Following the same reasoning, the tap water returning temperature will be 68^{o}C.



$TJCG := 780$ TJ

$\Delta TUU := 92 - 68$ $\Delta TUU = 24$ ^{o}C

$\Delta TCG := 90 - 70$ $\Delta TCG = 20$ ^{o}C

Considering the proportionality between the temperature differences and the amount of energy at the Upgrading Unit and at the Central Grid, the TJ/annum to be delivered at the Upgrading Unit can be estimated as follows:

$$\frac{TJ_{CentralGrid}}{TJ_{UpgradingUnit}} = \frac{\Delta T_{CentralGrid}}{\Delta T_{UpgradingUnit}}$$

$$TJUU := TJCG \frac{\Delta TUU}{\Delta TCG}$$

$TJUU = 936$ TJ

3. Calculating the clean water (hot tap water) flow rate in order to achieve 936 TJ/annum considering that the clean water will reach the battery limit (Central Grid) at 90 C in order to compensate any heat losses in the pipeline. The temperature of the cold tap water that reaches the Upgrading Unit is 68 C.

$TJproduct := 936$ TJ/year

$Tproductout := 92 + 273$ $\quad Tproductout = 365$ K

$Treference := 273$ $\quad Treference = 273$ K

$Tproductin := 68 + 273$ $\quad Tproductin = 341$ K

$$cpproduct := \frac{\int_{Tproductin}^{Tproductout} CPw(T)\,dT}{Tproductout - Tproductin}$$

$cpproduct = 4.217 \times 10^{3}$ J/(kg.K)

$$Product := \frac{TJproduct}{cpproduct \cdot (Tproductout - Tproductin) \cdot 365 \cdot 24 \cdot 10^{-12}}$$

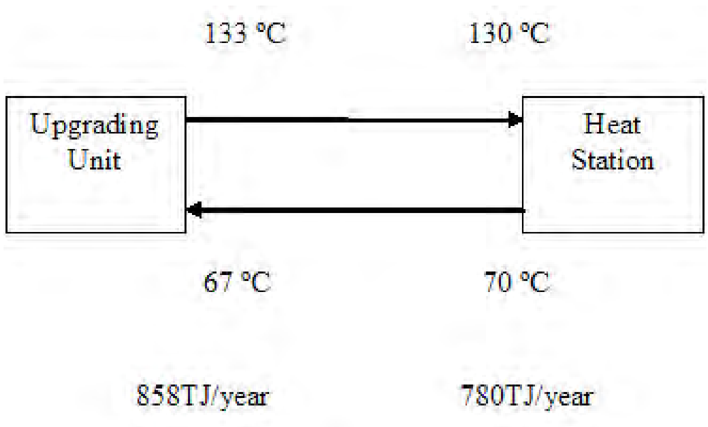
$Product = 1.056 \times 10^{6}$ kg/h

This is the flow rate of hot tap water (936TJ/annum, at 92 C), which is going to be sent through pipelines to the District Buildings.

4. Estimation of the temperature of the returning tap water from the Districts:

Assumptions :
1. 10% energy losses in the pipeline from the Upgrading Unit to the district (typical value of energy losses in similar systems).
2. 20^{o}C temperature approach bewteen IN and OUT tap water stream in the district

10% energy losses represents 2^{o}C difference, which added to the desired temperature, 90^{o}C, yields to 92^{o}C for the tap water outlet stream from the Upgrading Unit. Following the same reasoning, the tap water returning temperature will be 68^{o}C.



$TJCG := 780$ TJ

$\Delta TUU := 133 - 67$ $\qquad \Delta TUU = 66$ ^{o}C

$\Delta TCG := 130 - 70$ $\qquad \Delta TCG = 60$ ^{o}C

Considering the proportionality between the temperature differences and the amount of energy at the Upgrading Unit and at the Heat Stations, the TJ/annum to be delivered at the Upgrading Unit can be estimated as follows:

$$\frac{TJ_{CentralGrid}}{TJ_{UpgradingUnit}} = \frac{\Delta T_{CentralGrid}}{\Delta T_{UpgradingUnit}}$$

$$TJUU := TJCG \cdot \frac{\Delta TUU}{\Delta TCG}$$

$TJUU = 858$ TJ

5. Estimating the clean water (hot tap water) flow rate in order to achieve 858 TJ/annum considering that the clean water will reach the battery limit (Heat Stations) at 130 C in order to compensate any heat losses in the pipeline. The temperature of the cold tap water that reaches the Upgrading Unit is 67 C.

$TJproduct := 858$ TJ/year

$Tproductout := 133 + 273$ $\quad Tproductout = 406$ K

$Tproductin := 67 + 273$ $\quad Tproductin = 340$ K

Since the equation Cp(T) is not valid for temperature above 100C, the specific heat of water at 133 was taken from the book Transport Phenomena Data Companion, L.P.B.M. Janssen, M.M.C.G. Warmoeskerken.

$cpproductout := 4236$ J/(kg.K) Cp water at 133C

$cpproductin := CPw(67 + 273)$

$cpproductin = 4.211 \times 10^{3}$ J/(kg.K) Cp water at 67C calculated with the correlation

$$cpproduct := \frac{cpproductout + cpproductin}{2}$$

$cpproduct = 4.223 \times 10^{3}$ J/(kg.K)

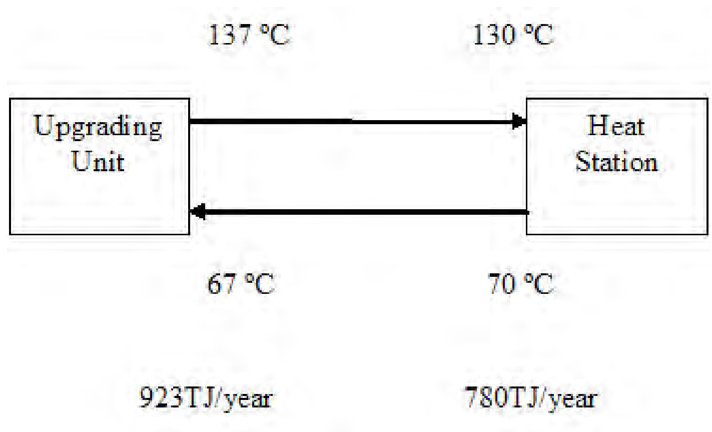
$$Product := \frac{TJproduct}{cpproduct \cdot (Tproductout - Tproductin) \cdot 365 \cdot 24 \cdot 10^{-12}}$$

$Product = 3.514 \times 10^{5}$ kg/h

This is the flow rate of hot tap water (858TJ/annum, at 133 C), which is going to be sent through pipelines to the District Buildings.

6. Estimation of the temperature of the returning tap water from the Districts taking into consideration the heat losses

After calculating the heat losses in the pipelines, an approximate temperature profile was obtained and the hot tap water outlet temperature in the Upgrading Unit was iteratively adjusted and a final value of 137 C was achieved. The same procedure was applied to the cold tap water inlet temperature at the Upgrading Unit. Therefore, the tap water temperatures to be used to calculate the required TJ in the Upgrading Unit, are:
Tout=137 C, Tin=67 C. The TJ at UU=923 TJ



$TJCG := 780$ TJ

$\Delta TUU := 137 - 66$ $\qquad \Delta TUU = 71$ °C

$\Delta TCG := 130 - 70$ $\qquad \Delta TCG = 60$ °C

Considering the proportionality between the temperature differences and the amount of energy at the Upgrading Unit and at the Heat Stations, the TJ/annum to be delivered at the Upgrading Unit can be estimated as follows:

$$\frac{TJ_{CentralGrid}}{TJ_{UpgradingUnit}} = \frac{\Delta T_{CentralGrid}}{\Delta T_{UpgradingUnit}}$$

$$TJUU := TJCG \frac{\Delta TUU}{\Delta TCG}$$

$TJUU = 923$ TJ

$$\frac{923}{365} \cdot \frac{1}{24} \cdot \frac{1}{3600} \cdot 10^{12} = 2.927 \times 10^{7}$$ Watt

This is the energy content of the hot tap water that exits the Upgrading Unit

7. Estimating the hot tap water flow rate in order to achieve 923 TJ/annum considering that the clean water will reach the battery limit (Heat Station) at 130 C in order to compensate any heat losses in the pipeline. The temperature of the cold tap water that reaches the Upgrading Unit is 67 C.

$TJproduct := 923$ TJ/year

$Tproductout := 137 + 273$ $Tproductout = 410$ K

$Tproductin := 66 + 273$ $Tproductin = 339$ K

Since the equation Cp(T) is not valid for temperature above 100C, the specific heat of water at 133 was taken from the book Transport Phenomena Data Companion, L.P.B.M. Janssen, M.M.C.G. Warmoeskerken.

$cpproductout := 4236$ J/(kg.K) Cp water at 133C

$cpproductin := CPw(67 + 273)$

$cpproductin = 4.211 \times 10^{3}$ J/(kg.K) Cp water at 67C calculated with the correlation

$$cpproduct := \frac{cpproductout + cpproductin}{2}$$

$cpproduct = 4.223 \times 10^{3}$ J/(kg.K)

$$Product := \frac{TJproduct}{cpproduct \cdot (Tproductout - Tproductin) \cdot 365 \cdot 24 \cdot 10^{-12}}$$

$Product = 3.514 \times 10^{5}$ kg/h

This is the flow rate of hot tap water (923TJ/annum, at 137 C), which is going to be sent through pipelines to the District Buildings.

Appendix 9 Overall Heat Transfer Coefficient

Reference : R.K. Sinnott, *Coulson & Richardson's Chemical Engineering*, Volume 6.
Butterworth Heinemann, Great Britain, 2003

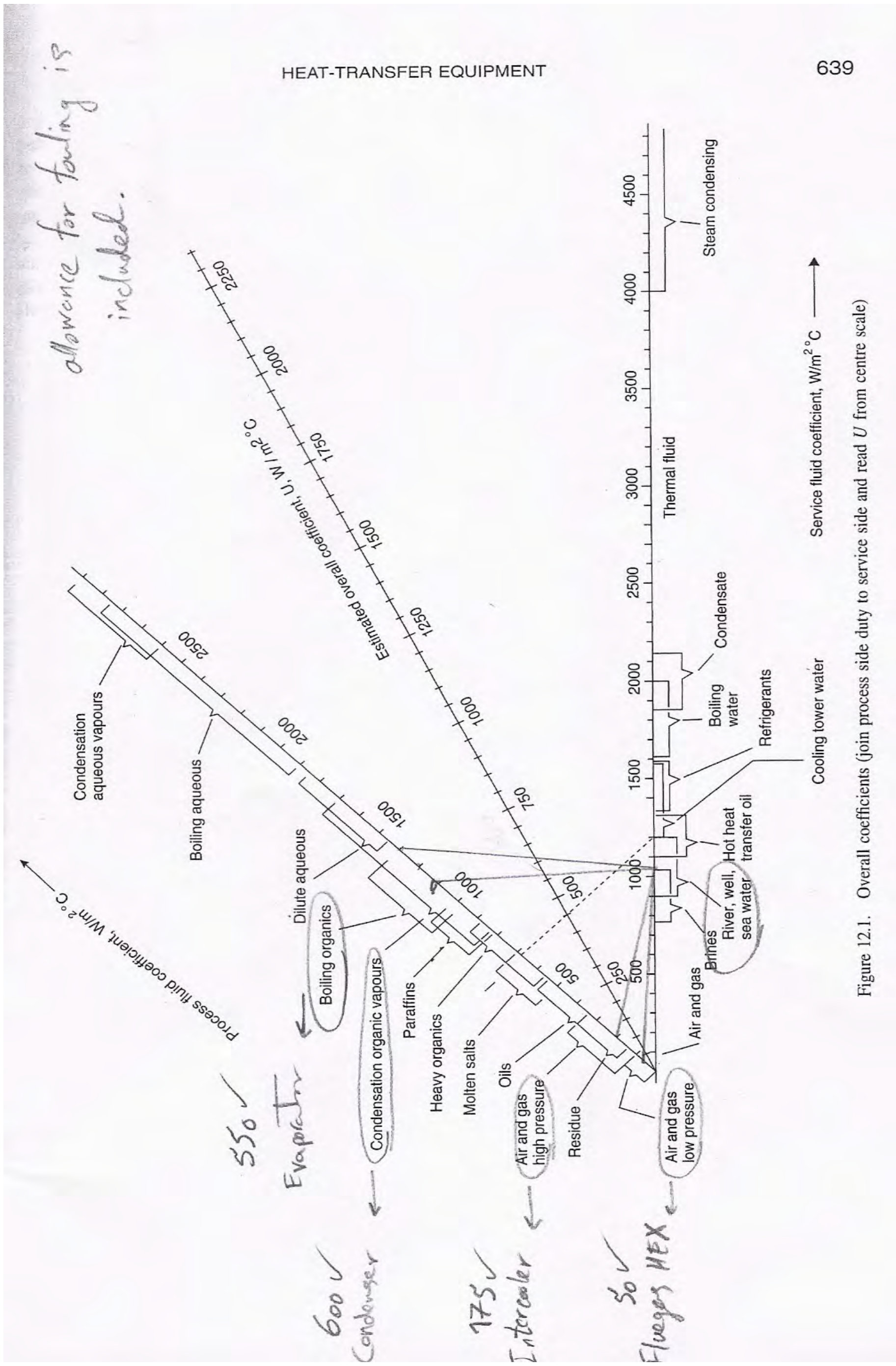


Figure 12.1. Overall coefficients (join process side duty to service side and read U from centre scale)

Appendix 10 Estimation of The Natural Gas and Air Consumption in The Fired Heater

The fired heater is the heat load necessary to increase the temperature of the tap water from 60 C to 97 C.

$Qduty := 9.355\times 10^{7}\cdot 1.15$ BTU/h

$Pfurnace := 1.3$ bar Considering Furnace within the range 1.0-1.6 bar (~1.3 bar)

$Heatvalue := 1000$ BTU/ft^3 at 60F (15.5C), 14.7 psia (1.013 bar), this serves as initial input further calculations. The natural gas flowrate will be actually calculated via Aspen Simulation and will change if any heat integration is considered.

$vNG := \frac{Qduty}{Heatvalue}$ $vNG = 1.076\times 10^{5}$ ft^3/h Volumetric flow rate of NG at 60F, 14.7 psia

North Sea (Netherlands- Groningen)
non-associated natural gas composition:

Components	Concentration (% vol.)
Methane	81.3
Ethane	2.9
Propane	0.4
N-butane	0.1
$C5^+$	0.1
Nitrogen	14.3
Carbon Dioxide	0.9

Ref: Chemical Process Technology, Moulijn et al. Wiley & Sons, 2001
which refers to the following literature sources:

1. Our Industry: Petroleum (1977), 5th ed., London,The British Petroleum Company
2. WoodCock KE, Gottlieb M (1994) 'Natural Gas' in Kroschwitz JI and Howegrant M (eds.) Kirk Othmer Encyclopedia of Chemical Technology vol.12, 4th ed., Wiley, New York, pp 318-340.

Via Aspen simulation based on the composition above the average molecular weight, density, the total mole flow rate and the mole flow rates for each component were calculated. The Aspen reference file is: CP.apw
The results are presented below:

Density at 60F(15.5C) and 14.7 psia (1.013bar) = 0.00078820 g/cm^3

MW = 18.623 kg/kmol

Total NG mass flow rate =2343.48 kg/h (125.8379 kmol/h). It was calculated with Aspen (RSTOIC) in order to satsify the conditions imposed by the fired heater and air demand. Considering the over design of 15% in order to use this fired heater as the backup system, the final values to be used for the design are: NG=2695.00 kg/h (144.829 Kmol/h). The amounts of each combustible components in NG are the following:

$nCH4 := 117.64$ kmol/h $nC2H6 := 4.197$ kmol/h

$nC3H8 := 0.5784$ kmol/h $nC4H10 := 0.149$ kmol/h

$nC5H12 := 0.149$ kmol/h Considering C5$^+$ as N-pentane

$nN2 := 20.69$ kmol/h $nCO2 := 1.3023$ kmol/h

Estimating the air flow rate:

The air flow rate is calculated considering 20% excess air for the combustion being added to the stoichiometric air flow:

Air stoichiometric:

$O2methane := 2 \cdot nCH4$ $O2methane = 235.29$ kmol/h

$O2ethane := 3.5 \cdot nC2H6$ $O2ethane = 14.691$ kmol/h

$O2propane := 5 \cdot nC3H8$ $O2propane = 2.892$ kmol/h

$O2butane := 6.5 \cdot nC4H10$ $O2butane = 0.972$ kmol/h

$O2pentane := 8 \cdot nC5H12$ $O2pentane = 1.196$ kmol/h

$$TotalO2stoich := O2methane + O2ethane + O2propane + O2butane + O2pentane$$

$TotalO2stoich = 255.041$ kmol/h

The actual flow rate of oxygen to be fed into the furnace is:

$O2actual := 1.2 \cdot TotalO2stoich$ $O2actual = 306.049$ kmol/h

Taking into account that O2 molar concentration in air is 21%, yields:

$$\text{Airmoleflowrate} := \frac{\text{O2actual}}{0.21} \qquad \text{Airmoleflowrate} = 1.457 \times 10^{3}$$ kmol/h

With the estimated air flow rate riquered for the combustion, the final blower capacity was calculated by Aspen.

The blower BHP is calculated:

$$\text{BlowerBHP} := 602.037 \frac{0.0018182}{1.356 \times 10^{-3}} \qquad \text{BlowerBHP} = 807.245$$ HP

Also via Aspen CP.apw , Toulet blower = 50 C

This capacity takes into consideration the fact that this blower can be also used to supply the heating demand as back up system.

Appendix 11 Process Flow Scheme

The Process Flow Scheme for this project design are shown in the A3 format as can be seen in the following A3 paper.

Appendix 12 Process Streams Summary

STREAM Nr. : Name :	1 WW IN		2 WW OUT		3 COMP1 IN		4 COMP1 OUT		5 COMP2 IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	0	0	0	0	3.89	0.23	3.89	0.23	3.89	0.23
Water 18.02	81.06	4.50	81.06	4.50	0	0	0	0	0	0
Total	81.06	4.50	81.06	4.50	3.89	0.23	3.89	0.23	3.89	0.23
Enthalpy kW	-1293746.38		-1297281.97		-10711.64		-10063.36		-10266.24	
Phase	L		L		V		V		V	
Press. Bara	3.00		2.50		6.50		15.00		14.70	
Temp. °C	27.50		18.00		12.00		93.25		70.85	

STREAM Nr. : Name :	6 COMP2 OUT		7 ACOND OUT		8 EVAP IN		9 UU IN		10 WCOOLER IN	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	3.89	0.23	3.89	0.23	3.89	0.23	0	0	0	0
Water 18.02	0	0	0	0	0	0	97.39	5.41	2.00	0.11
Total	3.89	0.23	3.89	0.23	3.89	0.23	97.39	5.41	2.00	0.11
Enthalpy kW	-9414.74		-14220.52		-14220.52		-1537130.33		-31566.49	
Phase	V		L		L+V		L		L	
Press. Bara	38.00		37.90		6.60		9.00		9.00	
Temp. °C	179.32		75.60		12.00		66.50		66.50	

STREAM Nr. : Name :	11 WCOND IN		12 WCOOLER OUT		13 WCOND OUT		14 FURNACE IN		15 FURNACE OUT	
COMP MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia 17.03	0	0	0	0	0	0	0	0	0	0
Water 18.02	95.39	5.30	2.00	0.11	95.39	5.30	97.39	5.41	97.39	5.41
Total	95.39	5.30	2.00	0.11	95.39	5.30	97.39	5.41	97.39	5.41
Enthalpy kW	-1505563.84		-31363.67		-1500759.33		-1532123.02		-1505416.93	
Phase	L		L		L		L		L	
Press. Bara	9.00		8.50		8.50		8.00		6.20	
Temp. °C	66.50		88.78		77.59		77.83		137.15	

STREAM Nr. : Name :	16 NGSTD		17 AIR IN		18 AIR OUT		19 EMISSIONS	
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	0	0
Water	0	0	0	0	0	0	1.15	0.06
Methane	0.50	0.03	0	0	0	0	0	0
Ethane	0.03	1.06e-3	0	0	0	0	0	0
Propane	6.46e-3	1.46e-4	0	0	0	0	0	0
n-Butane	2.12e-3	3.66e-5	0	0	0	0	0	0
n-Pentane	2.64e-3	3.66e-5	0	0	0	0	0	0
Oxygen	0	0	2.41	0.08	2.41	0.08	0.36	0.01
Nitrogen	0.15	5.23e-3	7.94	0.28	7.94	0.28	8.09	0.29
Dioxide carbon	0.01	3. 30e-4	0	0	0	0	1.45	0.03
Total	0.71	0.04	10.35	0.36	10.35	0.36	11.05	0.39
Enthalpy kW	-2577.23		-55.31		275.37		-29021.06	
Phase	V		V		V		V	
Press. Bara	1.00		1.00		1.3		1.3	
Temp. °C	15.56		20.00		51.59		79.85	

STREAM Nr. :		20		21		22		23		24	
Name :		UU OUT		TO ST. I		FROM ST. I		TO DIS. I		FROM DIS.I	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0	0	0	0	0	0	0
Water	18.02	97.39	5.41	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63
Total		97.39	5.41	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63
Enthalpy	kW	-1505356.65		-300288		-305651		-737887		-743126	
Phase		L		L		L		L		L	
Press.	Bara	10.00		10.00		9.50		1.00		1.00	
Temp.	°C	137.22		133.79		68.48		91.42		65.00	

STREAM Nr. :		25		26		27		28		29	
Name :		TO ST. IV		FROM ST. IV		TO DIS. IV		FROM DIS.IV		TO CG II	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0	0	0	0	0	0	0
Water	18.02	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63	58.47	3.25
Total		19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63	58.47	3.25
Enthalpy	kW	-300553		-305774		-738409		-743126		-901012	
Phase		L		L		L		L		L	
Press.	Bara	10.00		9.50		1.00		1.00		9.85	
Temp.	°C	130.60		66.97		88.80		65.00		133.20	

STREAM Nr. :		30		31		32		33		34	
Name :		FROM CG II		TO ST. II		FROM ST. II		TO DIS. II		FROM DIS.II	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0	0	0	0	0	0	0
Water	18.02	58.47	3.25	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63
Total		58.47	3.25	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63
Enthalpy	kW	-917639		-300812		-305781		-738350		-743126	
Phase		L		L		L		L		L	
Press.	Bara	9.10		9.80		9.30		1.00		1.00	
Temp.	°C	65.68		128.20		67.64		89.09		65.00	

STREAM Nr. :		35		36		37		38		39	
Name :		TO ST. V		FROM ST. V		TO DIS. V		FROM DIS.V		TO ST. III	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0	0	0	0	0	0	0
Water	18.02	19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63	19.49	1.08
Total		19.49	1.08	19.49	1.08	47.34	2.63	47.34	2.63	19.49	1.08
Enthalpy	kW	-300747		-305700		-738376		-743126		-300780	
Phase		L		L		L		L		L	
Press.	Bara	9.80		9.30		1.00		1.00		9.80	
Temp.	°C	127.90		67.50		88.96		65.00		127.5	

STREAM Nr. :		40		41		42	
Name :		FROM ST. III		TO DIS. III		FROM DIS. III	
COMP	MW	kg/s	kmol/s	kg/s	kmol/s	kg/s	kmol/s
Ammonia	17.03	0	0	0	0	0	0
Water	18.02	19.49	1.08	47.34	2.63	47.34	2.63
Total		19.49	1.08	47.34	2.63	47.34	2.63
Enthalpy	kW	-305716		-738409		-743126	
Phase		L		L		L	
Press.	Bara	9.30		1.00		1.00	
Temp.	°C	67.31		88.80		65.00	

Overall Component Mass Balance & Stream Heat Balance

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	

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	

Appendix 13 Equipment Summary & Spesification Sheets

In this appendix, the summary of all of the equipment and its specification sheets are made available.

REACTORS, COLUMNS & VESSELS – SUMMARY

EQUIPMENT NR. : **NAME :**	**V-01** **Ammonia Collector Vessel**				
	Vertical				
Pressure [bara] :	37.5				
Temp. [^{o}C] :	75.60				
Volume [m^3] : **Diameter [m] :** **L or H [m] :**	1.60 0.93 2.32				
Internals **- Tray Type :** **- Tray Number :** **- Fixed Packing** **Type :** **Shape :** **- Catalyst** **Type :** **Shape :**	 n.a. n.a. n.a. n.a. n.a. n.a.				
<u>Number</u> **- Series :** **- Parallel :**	 1 -				
Materials of Construction *(1)* **:**	CS				
Other :					

Remarks:
(1) CS : Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

HEAT EXCHANGERS & FURNACES – SUMMARY

EQUIPMENT NR. : NAME :	E-01 Ammonia Evaporator	E-02 Ammonia Condenser	E-03 Intercooler	E-04 Heat Exchanger District 1	E-05 Heat Exchanger District 2
Type :	Kettle Vaporizer	Kettle Vaporizer	Floating Head	Floating Head	Floating Head
Substance - **Tubes** : - **Shell** :	Water Ammonia	Water Ammonia	Water Ammonia	Water Water	Water Water
Duty [kW] :	3,535	4,804	203	23,277	23,277
Heat Exchange area [m^2] :	737	232	292	1,557	1,557
Number - **Series** : - **Parallel** :	1 -	1 -	1 -	1 -	1 -
Pressure [bara] - **Tubes** : - **Shell** :	3.00 6.60	9.00 38.00	9.00 15.00	4.00 12.80	4.00 12.60
Temperature In / Out [^{o}C] - **Tubes** : - **Shell** :	27.50 / 17.87 12.35 / 12.35	66.50 / 77.59 179.32 / 75.60	66.50 / 88.78 93.25 / 70.85	65.00 / 91.42 133.79 / 70.00	65.00 / 90.09 133.60 / 70.00
Special Materials of Construction *(1)* :	Tubes : CS Shell : CS	Tubes : CS Shell : CS	Tubes : CS Shell : CS	Tubes : CS Shell : CS	Tubes : CS Shell : CS
Other :	-	-	-	-	-

Remarks:
(1) CS = Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

HEAT EXCHANGERS & FURNACES – SUMMARY

EQUIPMENT NR. : NAME :	E-06 Heat Exchanger District 3	E-07 Heat Exchanger District 4	E-08 Heat Exchanger District 5	F-01 Natural Gas Furnace	
Type :	Floating Head	Floating Head	Floating Head	Fired Heater	
Substance - **Tubes** :	Water	Water	Water	Water	
- **Shell** :	Water	Water	Water	na	
Duty **[kW]** :	23,285	23,277	23,279	136,899 *(1)*	
Heat Exchange area **[m^2]** :	1,557	1,557	1,557	- *(2)*	
Number - **Series** :	1	1	1	1	
- **Parallel** :	-	-	-	-	
Pressure **[bara]** - **Tubes** :	4.00	4.00	4.00	8.00	
- **Shell** :	9.82	9.77	9.80	1.30	
Temperature In / Out **[^{o}C]** - **Tubes** :	65.00 / 89.09	65.00 / 88.79	65.00 / 88.98	77.83 / 137.15	
- **Shell** :	128.20 / 70.00	127.50 / 70.00	127.90 / 70.00	526.85 / 79.85	
Special Materials of Construction *(3)* :	Tubes : CS Shell : CS	Tubes : CS Shell : CS	Tubes : CS Shell : CS	Tubes : CS Shell : na	
Other :	-	-	-	-	

Remarks:
(1) This value is obtained based on the peak demand.
(2) To be specified in detailed design.
(3) CS = Carbon Steel.

Designers :	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

PUMPS, BLOWERS & COMPRESSORS – SUMMARY

EQUIPMENT NR. : NAME :	K-01 Low Pressure Compressor	K-02 High Pressure Compressor	K-03 Air Blower	P-01 A/B Hot Water Pump	
Type : **Number :**	Reciprocating 1	Reciprocating 1	Centrifugal 1	Centrifugal 2	
Medium Transferred :	Ammonia	Ammonia	Air	Water	
Capacity **[kg/s] :** **[m^3/s] :**	3.89 0.78 *(1)*	3.89 0.41 *(1)*	57.04 48.16 *(1)*	438.25 0.50	
Density [kg/m^3] :	4.99 *(1)*	9.43 *(1)*	1.18 *(1)*	878	
Pressure [bara] **Suct. / Disch. :**	6.50 / 15.00	14.50 / 38.00	Atm. / 1.30	4.00 / 13.00	
Temperature **In / Out[°C] :**	11.91 / 93.25	70.85 / 179.32	20.00 / 50.33	136.76 / 136.90	
Power [kW] **- Theor. :** **- Actual :**	621 690	852 946	1822 2025 *(3)*	485 539	
Number **- Theor. :** **- Actual :**	1	1	1	2 *(1)*	
Special Materials of Construction :	Carbon Steel	Carbon Steel	Cast Iron	Cast Iron	
Other :	-	-	-	Shaft: HT Steel	

Remarks:
(1) Calculated at inlet condition.
(2) One installed spare included.
(3) This value is obtained based on the peak demand.
(4) HT = High Tensile.

Designers :	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER :	**E-01**	**In Series** :	1
NAME :	**Ammonia Evaporator**	**In Parallel** :	none

General Data

Service	:	- ~~Heat Exchanger~~ - ~~Cooler~~ - ~~Condenser (Air cooled)~~	- **Vaporizer** - ~~Reboiler~~
Type	:	- ~~Fixed Tube Sheets~~ - ~~Floating Head~~ - ~~Hair Pin~~ - ~~Double Tube~~	- ~~Plate Heat Exchanger~~ - ~~Finned Tubes~~ - ~~Thermosyphon~~ - **Kettle Vaporizer**
Position	:	- **Horizontal** - ~~Vertical~~	

Capacity	**[kW]**	:	3,535	*(1)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	737	*(1),(2)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	:	550		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	9.7		
Passes Tube Side		:	-	*(3)*	
Passes Shell Side		:	-	*(3)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Corrected LMTD	**[^oC]**	:	8.7	*(1)*	**(Calc.)**

Process Conditions

			Shell Side	**Tube Side**
Medium		:	Ammonia	Water
Mass Stream	**[kg/s]**	:	3.89	81.06
Mass Stream to				
- **Evaporize**	**[kg/s]**	:	2.84	-
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	-	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	380	-
Temperature IN	**[^oC]**	:	12.35	27.50
Temperature OUT	**[^oC]**	:	12.35	17.87
Pressure	**[bara]**	:	6.60	3.00
Material *(4)*		:	CS	CS

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(3) To be specified in the detailed design.
(4) CS=Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER	: **E-02**	**In Series**	: 1
NAME	: **Ammonia Condenser**	**In Parallel**	: none

General Data

Service	:	- ~~Heat Exchanger~~ - ~~Cooler~~ - Condenser	- ~~Vaporizer~~ - ~~Reboiler~~
Type	:	- ~~Fixed Tube Sheets~~ - ~~Floating Head~~ - ~~Hair Pin~~ - ~~Double Tube~~	- ~~Plate Heat Exchanger~~ - ~~Finned Tubes~~ - ~~Thermosyphon~~ - Kettle Vaporizer
Position	:	- Horizontal - ~~Vertical~~	

Capacity	[kW]	:	4,804	*(1)*	(Calc.)
Heat Exchange Area	[m^2]	:	232	*(1),(2)*	(Calc.)
Overall Heat Transfer Coefficient	[$W/m^2 \cdot {}^oC$]	:	600		(Approx.)
Log. Mean Temperature Diff. (LMTD)	[^oC]	:	38.37		
Passes Tube Side		:	-	*(3)*	
Passes Shell Side		:	-	*(3)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Corrected LMTD	[^oC]	:	34.53	*(1)*	

Process Conditions

			Shell Side	Tube Side
Medium		:	Ammonia	Water
Mass Stream	[kg/s]	:	3.89	95.39
Mass Stream to				
- ~~Evaporize~~	~~[kg/s]~~	:		
- Condense	[kg/s]	:	3.89	-
Average Specific Heat	[$kJ/kg \cdot {}^oC$]	:	-	4.18
Heat of Evap. / Condensation	[kJ/kg]	:	550	-
Temperature IN	[^oC]	:	179.32	66.50
Temperature OUT	[^oC]	:	75.60	77.59
Pressure	[bara]	:	38.00	9.00
Material *(4)*		:	CS	CS

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(3) To be specified in the detailed design.
(4) CS = Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER :	**E-03**		**In Series** :	1	
NAME :	**Intercooler**		**In Parallel** :	none	
General Data					
Service :	- **Heat Exchanger**	- ~~**Vaporizer**~~			
	- ~~**Cooler**~~	- ~~**Reboiler**~~			
	- ~~**Condenser**~~				
Type :	- ~~**Fixed Tube Sheets**~~	- ~~**Plate Heat Exchanger**~~			
	- **Floating Head**	- ~~**Finned Tubes**~~			
	- ~~**Hair Pin**~~	- ~~**Thermosyphon**~~			
	- ~~**Double Tube**~~	- ~~**Kettle Vaporizer**~~			
Position :	- **Horizontal**				
	- ~~**Vertical**~~				
Capacity	**[kW]**	: 203	*(1)*	**(Calc.)**	
Heat Exchange Area	**[m^2]**	: 292	*(1),(2)*	**(Calc.)**	
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	: 175		**(Approx.)**	
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	: 4.41			
Passes Tube Side		: -	*(3)*		
Passes Shell Side		: -	*(3)*		
Correction Factor LMTD (min. 0.75)		: 0.9			
Cor**rected LMTD**	**[^oC]**	: 3.97	*(1)*		

Process Conditions

			Shell Side	Tube Side
Medium		:	Ammonia	Water
Mass Stream	**[kg/s]**	:	3.89	2.00
Mass Stream to				
- ~~**Evaporize**~~	~~**[kg/s]**~~	:		
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	2.40 *(4)*	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-
Temperature IN	**[^oC]**	:	93.25	66.50
Temperature OUT	**[^oC]**	:	70.85	88.78
Pressure	**[bara]**	:	15.00	9.00
Material *(5)*		:	CS	CS

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(3) To be specified in the detailed design.
(4) The value estimated by Aspen Plus.
(5) CS = Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13[st] 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER	:	**E-04**		**In Series**	: 1
NAME	:	**Heat Exchanger District 1**		**In Parallel**	: none

General Data

Service	:	- **Heat Exchanger** - ~~**Cooler**~~ - ~~**Condenser**~~	- ~~**Vaporizer**~~ - ~~**Reboiler**~~
Type	:	- ~~**Fixed Tube Sheets**~~ - **Floating Head** - ~~**Hair Pin**~~ - ~~**Double Tube**~~	- ~~**Plate Heat Exchanger**~~ - ~~**Finned Tubes**~~ - ~~**Thermosyphon**~~ - ~~**Kettle Vaporizer**~~
Position	:	- **Horizontal** - ~~**Vertical**~~	

Capacity	**[kW]**	:	23,277	*(1),(2)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	1,557	*(1),(2),(3)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2.^oC$]**	:	1000		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	14.95		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	-	*(4)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Corrected LMTD	**[^oC]**	:	13.45	*(1)*	

Process Conditions

			Shell Side	Tube Side
Medium		:	Water	Water
Mass Stream	**[kg/s]**	:	19.48	47.33
Mass Stream to				
- ~~**Evaporize**~~	~~**[kg/s]**~~	:		
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg.^oC$]**	:	4.18	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-
Temperature IN	**[^oC]**	:	133.79	65.00
Temperature OUT	**[^oC]**	:	70.00	91.42
Pressure	**[bara]**	:	12.80	4.00 *(5)*
Material *(6)*		:	CS	CS

Remarks:

(1) Calculation is done by Aspen Plus Simulation.
(2) This value is obtained for the peak demand.
(3) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(4) To be specified in the detailed design.
(5) It is assumed that the tap water is delivered at this pressure.
(6) CS = Carbon Steel.

Designers	:	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER	:	**E-05**	**In Series** :	1
NAME	:	**Heat Exchanger District 2**	**In Parallel** :	none

General Data

Service	:	- **Heat Exchanger** - ~~**Cooler**~~ - ~~**Condenser**~~	- ~~**Vaporizer**~~ - ~~**Reboiler**~~
Type	:	- ~~**Fixed Tube Sheets**~~ - **Floating Head** - ~~**Hair Pin**~~ - ~~**Double Tube**~~	- ~~**Plate Heat Exchanger**~~ - ~~**Finned Tubes**~~ - ~~**Thermosyphon**~~ - ~~**Kettle Vaporizer**~~
Position	:	- **Horizontal** - ~~**Vertical**~~	

Capacity	**[kW]**	:	23,277	*(1),(2)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	1,557	*(1),(2),(3)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2\cdot{}^oC$]**	:	1000		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	16.97		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	-	*(4)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Cor**rected LMTD**	**[^oC]**	:	15.27	*(1)*	

Process Conditions

			Shell Side	Tube Side	
Medium		:	Water	Water	
Mass Stream	**[kg/s]**	:	19.48	47.33	
Mass Stream to					
- ~~**Evaporize**~~	~~**[kg/s]**~~	:			
- ~~**Condense**~~	~~**[kg/s]**~~	:			
Average Specific Heat	**[$kJ/kg\cdot{}^oC$]**	:	4.18	4.18	
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-	
Temperature IN	**[^oC]**	:	133.60	65.00	
Temperature OUT	**[^oC]**	:	70.00	90.09	
Pressure	**[bara]**	:	12.60	4.00	*(5)*
Material *(6)*		:	CS	CS	

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) This value is obtained for the peak demand.
(3) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(4) To be specified in the detailed design.
(5) It is assumed that the tap water is delivered at this pressure.
(6) CS = Carbon Steel.

Designers	:	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER	:	**E-06**	**In Series** :	1
NAME	:	**Heat Exchanger District 3**	**In Parallel** :	none

General Data

Service	:	- **Heat Exchanger** - ~~**Cooler**~~ - ~~**Condenser**~~	- ~~**Vaporizer**~~ - ~~**Reboiler**~~
Type	:	- ~~**Fixed Tube Sheets**~~ - **Floating Head** - ~~**Hair Pin**~~ - ~~**Double Tube**~~	- ~~**Plate Heat Exchanger**~~ - ~~**Finned Tubes**~~ - ~~**Thermosyphon**~~ - ~~**Kettle Vaporizer**~~
Position	:	- **Horizontal** - ~~**Vertical**~~	

Capacity	**[kW]**	:	23,285	*(1),(2)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	1,557	*(1),(2),(3)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	:	1000		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	16.58		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	-	*(4)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Cor**rected LMTD**	**[^oC]**	:	14.92	*(1)*	

Process Conditions

			Shell Side	Tube Side	
Medium		:	Water	Water	
Mass Stream	**[kg/s]**	:	19.48	47.33	
Mass Stream to					
- ~~**Evaporize**~~	~~**[kg/s]**~~	:			
- ~~**Condense**~~	~~**[kg/s]**~~	:			
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	4.18	4.18	
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-	
Temperature IN	**[^oC]**	:	128.20	65.00	
Temperature OUT	**[^oC]**	:	70.00	89.09	
Pressure	**[bara]**	:	9.82	4.00	*(5)*
Material *(6)*		:	CS	CS	

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) This value is obtained for the peak demand.
(3) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(4) To be specified in the detailed design.
(5) It is assumed that the tap water is delivered at this pressure.
(6) CS = Carbon Steel.

Designers	:	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER : **E-07**			**In Series** : 1	
NAME : **Heat Exchanger District 4**			**In Parallel** : none	

General Data

Service	:	- **Heat Exchanger** - ~~**Cooler**~~ - ~~**Condenser**~~	- ~~**Vaporizer**~~ - ~~**Reboiler**~~
Type	:	- ~~**Fixed Tube Sheets**~~ - **Floating Head** - ~~**Hair Pin**~~ - ~~**Double Tube**~~	- ~~**Plate Heat Exchanger**~~ - ~~**Finned Tubes**~~ - ~~**Thermosyphon**~~ - ~~**Kettle Vaporizer**~~
Position	:	- **Horizontal** - ~~**Vertical**~~	

Capacity	**[kW]**	:	23,277	*(1),(2)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	1,557	*(1),(2),(3)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	:	1000		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	14.95		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	-	*(4)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Cor**rected LMTD**	**[^oC]**	:	13.45	*(1)*	

Process Conditions

			Shell Side	**Tube Side**
Medium		:	Water	Water
Mass Stream	**[kg/s]**	:	19.48	47.33
Mass Stream to				
- ~~**Evaporize**~~	~~**[kg/s]**~~	:		
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	4.18	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-
Temperature IN	**[^oC]**	:	127.50	65.00
Temperature OUT	**[^oC]**	:	70.00	88.79
Pressure	**[bara]**	:	9.77	4.00 *(5)*
Material *(6)*		:	CS	CS

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) This value is obtained for the peak demand.
(3) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(4) To be specified in the detailed design.
(5) It is assumed that the tap water is delivered at this pressure.
(6) CS = Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

HEAT EXCHANGER – SPECIFICATION SHEET

EQUIPMENT NUMBER	:	**E-08**	**In Series** :	1
NAME	:	**Heat Exchanger District 5**	**In Parallel** :	none

General Data

Service	:	- **Heat Exchanger** - ~~**Cooler**~~ - ~~**Condenser**~~	- ~~**Vaporizer**~~ - ~~**Reboiler**~~
Type	:	- ~~**Fixed Tube Sheets**~~ - **Floating Head** - ~~**Hair Pin**~~ - ~~**Double Tube**~~	- ~~**Plate Heat Exchanger**~~ - ~~**Finned Tubes**~~ - ~~**Thermosyphon**~~ - ~~**Kettle Vaporizer**~~
Position	:	- **Horizontal** - ~~**Vertical**~~	

Capacity	**[kW]**	:	23,279	*(1),(2)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	1,557	*(1),(2),(3)*	**(Calc.)**
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	:	1000		**(Approx.)**
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	16.54		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	-	*(4)*	
Correction Factor LMTD (min. 0.75)		:	0.9		
Cor**rected LMTD**	**[^oC]**	:	14.89	*(1)*	

Process Conditions

			Shell Side	**Tube Side**
Medium		:	Water	Water
Mass Stream	**[kg/s]**	:	19.48	47.33
Mass Stream to				
- ~~**Evaporize**~~	~~**[kg/s]**~~	:		
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	4.18	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-
Temperature IN	**[^oC]**	:	127.90	65.00
Temperature OUT	**[^oC]**	:	70.00	88.96
Pressure	**[bara]**	:	9.80	4.00 *(5)*
Material *(6)*		:	CS	CS

Remarks:
(1) Calculation is done by Aspen Plus Simulation.
(2) This value is obtained for the peak demand.
(3) Tubes: L = 12 ft; OD = 1″ on Square Pitch; Tube Pitch=11/4″.
(4) To be specified in the detailed design.
(5) It is assumed that the tap water is delivered at this pressure.
(6) CS = Carbon Steel.

Designers :	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

FURNACE – SPECIFICATION SHEET

EQUIPMENT NUMBER	:	**F-01**		**In Series**	: 1
NAME	:	**Natural Gas Furnace**		**In Parallel**	: none

General Data

Service : Burning Natural Gas *(1)*
Type : Fired Heater

Capacity	**[kW]**	:	136899	*(2) ,(3)*	**(Calc.)**
Heat Exchange Area	**[m^2]**	:	-	*(4)*	
Overall Heat Transfer Coefficient	**[$W/m^2 \cdot {}^oC$]**	:	-	*(4)*	
Log. Mean Temperature Diff. (LMTD)	**[^oC]**	:	80.77		
Passes Tube Side		:	-	*(4)*	
Passes Shell Side		:	na		
Correction Factor LMTD (min. 0.75)		:	0.9		
Corrected LMTD	**[^oC]**	:	72.69	*(2)*	

Process Conditions

			Shell Side	**Tube Side**
Medium		:	Flue Gas	Water
Mass Stream	**[kg/s]**	:	11.06	97.39
Mass Stream to				
- ~~**Evaporize**~~	~~**[kg/s]**~~	:		
- ~~**Condense**~~	~~**[kg/s]**~~	:		
Average Specific Heat	**[$kJ/kg \cdot {}^oC$]**	:	1.23 *(2),(5)*	4.18
Heat of Evap. / Condensation	**[kJ/kg]**	:	-	-
Temperature IN	**[^oC]**	:	526.85 *(6)*	77.83
Temperature OUT	**[^oC]**	:	79.85	137.15
Pressure	**[bara]**	:	1.30 *(7)*	8.00
Material *(8) ,(9)*		:	CS	CS

Remarks:
(1) Natural gas composition (mole percent): C_1=81.3, C_2=2.9, C_3=0.4, C_4=0.1, C_5=0.1, CO_2=0.9, N_2=14.3; temperature: 60 ^oF, pressure: 14.7 psia.
(2) Calculation is done by Aspen Plus Simulation.
(3) This value is obtained based on the peak demand.
(4) To be specified in the detailed design.
(5) Calculated at outlet condition.
(6) Flue gas temperature.
(7) Flue gas pressure.
(8) CS = Carbon Steel.
(9) Insulation should be provided.

Designers	:	P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : Date :	**CPD-3328** December 13st 2005

RECIPROCATING COMPRESSOR –SPECIFICATION SHEET

EQUIPMENT NUMBER	: **K-01**	**In Series**	: 1
NAME	: **Low Pressure Compressor**	**In Parallel**	: none

Service : Compress Ammonia
Type : Reciprocating

Operating Conditions & Physical Data

Compressor Model		:	Isentropic
Compressed Gas		:	Ammonia
Cp/Cv at inlet condition		:	1.39
Cp/Cv at outlet condition		:	1.38
Density at inlet condition	**[kg/m^3]**	:	4.99
Density at outlet condition	**[kg/m^3]**	:	9.03

Power

Suction Capacity	**[m^3/s]**	:	0.78	
Discharge Capacity	**[m^3/s]**	:	0.43	
Suction Pressure	**[bara]**	:	6.50	
Discharge Pressure	**[bara]**	:	15.00	
Suction Temperature	**[^{o}C]**	:	11.91	
Discharge Temperature	**[^{o}C]**	:	93.25	
Isentropic Temperature	**[^{o}C]**	:	76.61	
Isentropic Efficiency	**[-]**	:	0.75	
Mechanical Efficiency	**[-]**	:	0.90	
Power	**[kW]**	:	690	*(1)*

Construction Details *(2),(3),(4)*

Drive	:	Electrical
Material	:	CS *(5)*

Remarks:
(1) Calculation is done by Aspen Plus simulation.
(2) Further details are provided by the manufacturer.
(3) The maximum allowable noise level is 50 db.
(4) A knockout drum should be provided with the compressor.
(5) CS: Carbon Steel.

Designers	: P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number Date	: **CPD-3328** : December 13st 2005

RECIPROCATING COMPRESSOR –SPECIFICATION SHEET

EQUIPMENT NUMBER : **K-02**		**In Series** : 1	
NAME : **High Pressure Compressor**		**In Parallel** : none	
Service : Compress Ammonia			
Type : Reciprocating			
Operating Conditions & Physical Data			
Compressor Model		:	Isentropic
Compressed Gas		:	Ammonia
Cp/Cv at inlet condition		:	1.40
Cp/Cv at outlet condition		:	1.39
Density at inlet condition	**[kg/m^3]**	:	9.43
Density at outlet condition	**[kg/m^3]**	:	18.96
Power			
Suction Capacity	**[m^3/s]**	:	0.41
Discharge Capacity	**[m^3/s]**	:	0.21
Suction Pressure	**[bara]**	:	14.50
Discharge Pressure	**[bara]**	:	38.00
Suction Temperature	**[^{o}C]**	:	70.85
Discharge Temperature	**[^{o}C]**	:	179.32
Isentropic Temperature	**[^{o}C]**	:	159.03
Isentropic Efficiency	**[-]**	:	0.75
Mechanical Efficiency	**[-]**	:	0.90
Power	**[kW]**	:	946 *(1)*
Construction Details *(2),(3),(4)*			
Drive		:	Electrical
Material		:	CS *(5)*

Remarks:
(1) Calculation is done by Aspen Plus simulation.
(2) Further details are provided by the manufacturer.
(3) The maximum allowable noise level is 50 db.
(4) A knockout drum should be provided with the compressor.
(5) CS: Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

CENTRIFUGAL BLOWER – SPECIFICATION SHEET

EQUIPMENT NUMBER	: **K-03**	**In Series**	: 1
NAME	: **Air Blower**	**In Parallel**	: none

Service : Blow Air to the Furnace
Type : Centrifugal

Operating Conditions & Physical Data

Compressor Model		:	Isentropic
Compressed Gas		:	Air
Cp/Cv at inlet condition		:	1.41
Cp/Cv at outlet condition		:	1.41
Density at inlet condition	**[kg/m^3]**	:	1.18
Density at outlet condition	**[kg/m^3]**	:	1.39

Power

Suction Capacity	**[m^3/s]**	:	48.16
Discharge Capacity	**[m^3/s]**	:	40.88
Suction Pressure	**[bara]**	:	Atm
Discharge Pressure	**[bara]**	:	1.30
Suction Temperature	**[^{o}C]**	:	20.00
Discharge Temperature	**[^{o}C]**	:	50.33
Isentropic Temperature	**[^{o}C]**	:	42.77
Isentropic Efficiency	**[-]**	:	0.75
Mechanical Efficiency	**[-]**	:	0.90
Power	**[kW]**	:	1944 *(1),(2)*

Construction Details *(3),(4)*

Drive : Electrical
Material : Cast Iron

Remarks:
(1) Calculation is done by Aspen Plus simulation.
(2) This value is obtained based on the peak demand.
(3) Further details are provided by the manufacturer.
(4) The maximum allowable noise level is 50 db.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

CENTRIFUGAL PUMP – SPECIFICATION SHEET

EQUIPMENT NUMBER :	**P-01 A/B**	**Operating** :	1
NAME :	**Hot Water Pump**	**Installed Spare** :	1

Service : Water Pump
Type : 1 Stage Radial Centrifugal Pump (HSC)
Number : 2

Operating Conditions & Physical Data

Pumped liquid			:	Water
Temperature	**(T)**	**[^oC]**	:	137.0
Density	**(ρ)**	**[kg/m^3]**	:	878
Viscosity	**(η)**	**[$N \cdot s/m^2$]**	:	0.0002
Vapour Pressure	**(p_v)**	**[bara]**	:	3.3 **at Temperature [^oC] :** 137.0

Power

Capacity	**(Φ_v)**	**[m^3/s]**	:	0.50	
Suction Pressure	**(p_s)**	**[bara]**	:	4.0	*(1)*
Discharge Pressure	**(p_d)**	**[bara]**	:	13.0	*(1)*
Brake Power		**[kW]**	:	453	*(1),(2)*
Pump Efficiency		**[-]**	:	0.7	
Power at Shaft		**[kW]**	:	503	*(1),(2)*

Construction Details *(3),(4)*

RPM		:	3600	
Drive		:	Electrical	
Type electrical motor		:	Totally Enclosed, Fan-Cooled Enclosure	
Tension	**[V]**	:	-	
Rotational direction		:	**Clock / ~~Counter Cl.~~**	
Foundation Plate		:	**Combined / ~~two parts~~**	
Flexible Coupling		:	Yes	
Pressure Gauge Suction		:	No	
Pressure Gauge Discharge		:	Yes	
N.P.S.H.	**[m]**	:	31.9	*(1), (2)*

Construction Materials

Pump House		:	Cast Iron
Pump Rotor	*(5)*	:	HT Steel
Shaft	*(5)*	:	HT Steel
Special provisions		:	none
Operating Pressure	**[bara]**	:	13.0

Remarks:
(1) This value is obtained based on the peak demand.
(2) Calculation is done by Aspen Plus Simulation
(3) Further details to be specified by Rotating Equipment specialist.
(4) The maximum allowable noise level is 50 db.
(5) HT Steel: High Tensile Steel

Designers :	P.W. Falcao A. Mesbah	Project ID-Number :	**CPD-3328**
	M.V. Suherman S. Wennekes	Date :	December 13st 2005

VESSEL – SPECIFICATION SHEET

EQUIPMENT NUMBER : **V-01**		**In Series** : 1	
NAME : **Ammonia Collector Vessel**		**In Parallel** : none	
General Data			
Service		:	Ammonia Collector
Type		:	- **Buffer** - ~~**Separation**~~ - ~~**Storage**~~ - ~~**Reaction**~~
Position		:	- ~~**Horizontal**~~ - **Vertical**
Internals		:	- ~~**Demister**~~ / ~~**Plate**~~ / ~~**Coil**~~
Heating/Cooling medium		:	- **none** / ~~**Open**~~ / ~~**Closed**~~ / ~~**External Hxgr**~~
- **Type**		:	n.a.
- **Quantity**	**[kg/s]**	:	n.a.
- **Press./Temp.'s**	**[bara/°C]**	:	n.a.
Vessel Diameter (ID)	**[m]**	:	0.93
Vessel Height	**[m]**	:	2.32
Vessel Tot. Volume	**[m^3]**	:	1.60
Vessel Material		:	CS *(1)*
Other		:	-

Process Conditions

Stream Data			**Feed**	**Bottom**
Temperature	**[°C]**	:	75.60	-
Pressure	**[bara]**	:	37.50	-
Density	**[kg/m^3]**	:	513.40	-
Mass Flow	**[kg/s]**	:	3.89	-

Remarks:
(1) CS: Carbon Steel.

Designers : P.W. Falcao A. Mesbah M.V. Suherman S. Wennekes	Project ID-Number : **CPD-3328** Date : December 13st 2005

Appendix 14 Demand Fluctuations

A maximum annual demand of 780 TJ at the districts was assumed. 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. Our 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 the over-design factor to correct for monthly and hourly peaks.

Monthly fluctuations

Figure 1 shows the monthly consumption of hot tap water and central heating and was used as basis.

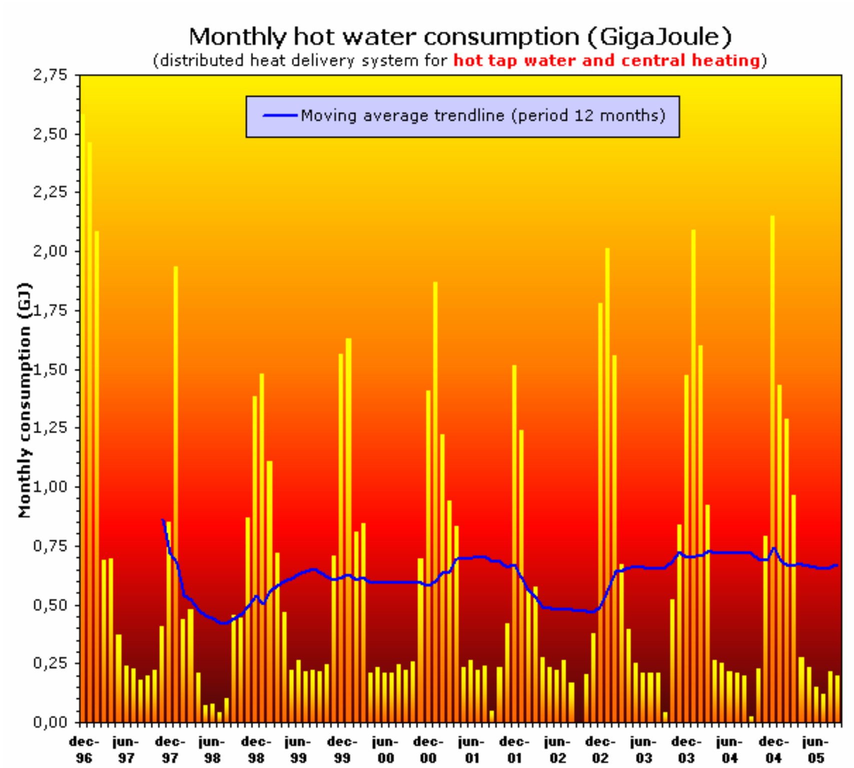


Figure 1. Monthly consumption for hot tap water and central heating, adopted from: http://www.bertie.joan.freeler.nl/totaaloverzichten_MHWC.htm#MHWCsmall

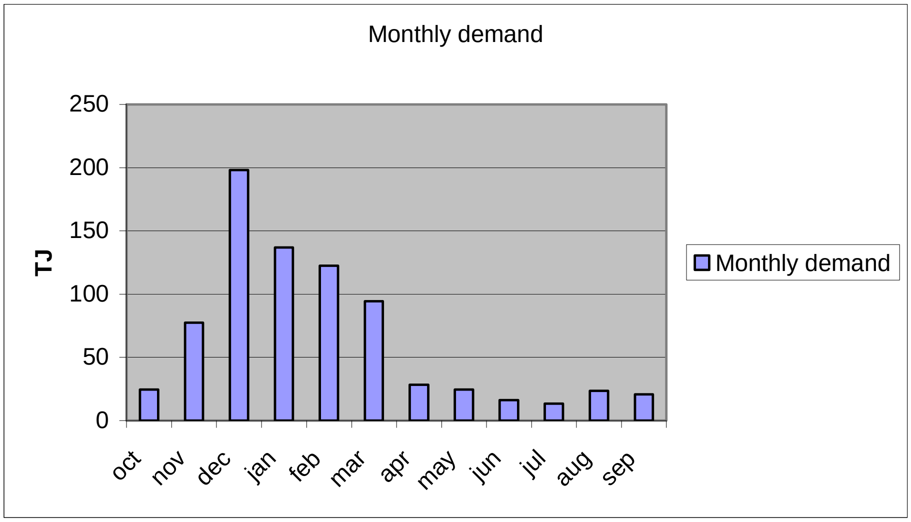


Figure 2. Monthly consumption for hot tap water and central heating

The data for the year 2004 in Figure 2 were used to calculate the ratios (month usage/year usage) and to estimate how the usage of 780 TJ would be distributed over the year based on similar ratios. The results are shown in Table 1.

Table 1. Monthly hot water consumption, hot tap water and central heating, derived from 2

2004-2005	Fraction of 780TJ/annum	Average	TJ/month	Over design
Oct	0.03	0.08	25	0.4
Nov	0.10	0.08	77	1.2
Dec	0.25	0.08	198	3.0
Jan	0.18	0.08	137	2.1
Feb	0.16	0.08	123	1.9
Mar	0.12	0.08	94	1.5
Apr	0.04	0.08	28	0.4
May	0.03	0.08	25	0.4
Jun	0.02	0.08	16	0.2
Jul	0.02	0.08	13	0.2
Aug	0.03	0.08	24	0.4
Sep	0.03	0.08	21	0.3
Total	1	1	780TJ/annum	

The figures in Table 1 show that a design for a constant energy demand will certainly not fulfill the requirements. The furnace should be over-designed for a factor of 3 in comparison with the average usage (0.08 TJ/month) to overcome the monthly fluctuations in heat usage by the district.

Hourly fluctuations

As it was mentioned before there are not only monthly fluctuations, but also hourly fluctuations during the day in the heat demand by the district. People will use less heat during night and more during daylight and evening, with maximum peaks during rush hours.

In order to fulfill this peak demands, there should be another over design, to compensate for the hourly fluctuations and peaks. To deal with these fluctuations one has to know something about the daily characteristics of the heat demand in heating districts.

Figure 3 shows the mean monthly and peak load fluctuation and was used as basis.

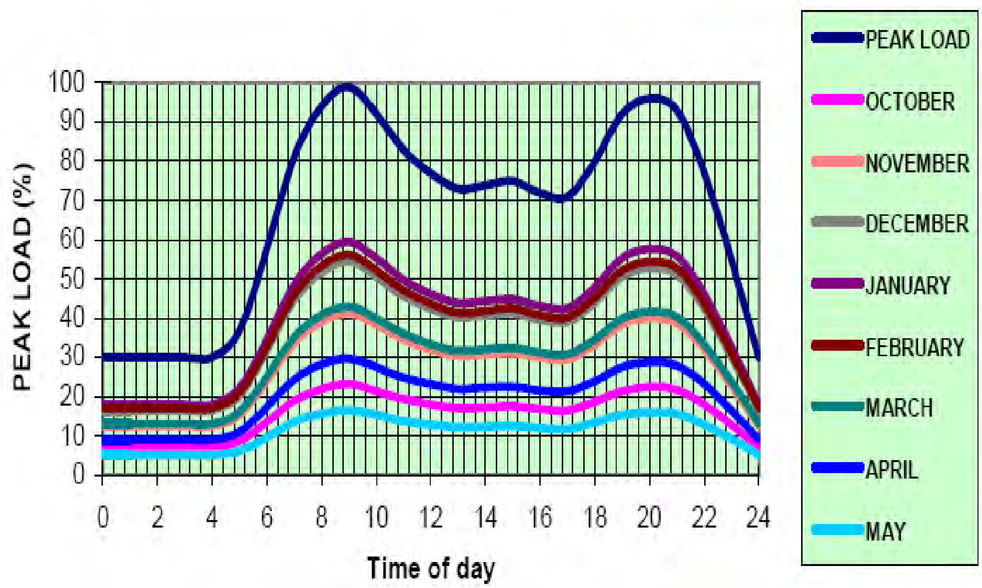


Figure 3. Mean monthly and peak load fluctuations, adopted from: http://www.opetchp.net/download/wp7/WP7_presentation_paper_CERTH_%20ISFTA.pdf

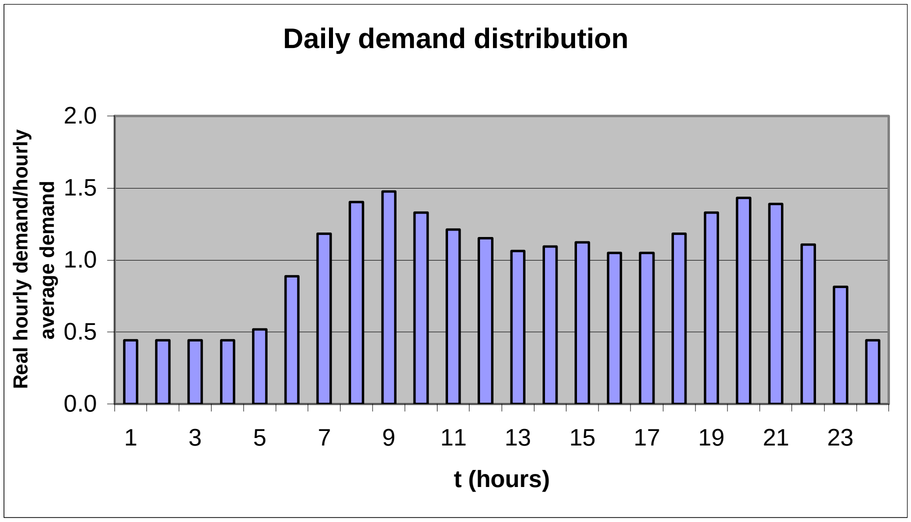


Figure 4. A Daily demand distribution

Figure 4 was used to define the ratios for the hourly energy demand in the district. The biggest peak in the graph deviates from the daily average with a factor of 1.48.

Two options were considered to overcome the hourly fluctuations:

Option 1.
The first option to overcome these fluctuations was by means of a big buffer tank. The volume of this buffer tank should be big enough to store the energy amount of the usage above the average demand, which is approximately the accumulation from 07:00 till 22:00 o'clock.

According to figures presented in Table 2, two buffer tanks with capacity of 3.57 hours 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 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 two 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 dimension calculations, as well as its cost estimation.

Table 2. Hourly over-design factors, derived from Figure 4

Hour	Factor of over design	Difference from average (relative)	Buffer utilization (hr)
1	0.44	-0.56	
2	0.44	-0.56	
3	0.44	-0.56	
4	0.44	-0.56	
5	0.52	-0.48	
6	0.89	-0.11	
7	1.18	0.18	0.18
8	1.40	0.40	0.40
9	**1.48**	0.48	0.48
10	1.33	0.33	0.33
11	1.21	0.21	0.21
12	1.15	0.15	0.15
13	1.06	0.06	0.06
14	1.09	0.09	0.09
15	1.12	0.12	0.12
16	1.05	0.05	0.05
17	1.05	0.05	0.05
18	1.18	0.18	0.18
19	1.33	0.33	0.33
20	1.43	0.43	0.43
21	1.39	0.39	0.39
22	1.11	0.11	0.11
23	0.81	-0.19	
24	0.44	-0.56	**3.57**

Option 2.
Another option and probably a better way to overcome the above average usage is to increase the capacity of the fired heater again, but now with respect to the hourly fluctuations. The hourly peaks are not that big relatively to the average usage; only a maximum factor of 1.48. Therefore, the fired heater capacity should in this case be increased in order to overcome both monthly and daily fluctuations. This is the best option due to the fact that the tanks will be big like shown before. This is a problem because in this area space is not infinitely available, and so making it an expensive solution.

One big fired heater should be used, 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. If the demand of the district is lower than the capacity of the MHP, the unloading/loading capacity of the compressors will lower the ammonia flow in the MHP and shut down the fired heater.

The disadvantage of this option is the presence of a fired heater that is not efficiently used; that is, only 20% of its capacity is utilized over the year. The idleness of the furnace 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.

Results of Fluctuations and Operability

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 of 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 of the evaluation of the demand fluctuations and studies on heat losses in the pipelines, the outputs of our design were mostly established. As a result of heat losses calculations in the pipeline network, the Upgrading Unit has to be designed for 922 TJ/annum in order to compensate the heat losses on the network.

Appendix 15 Streams To The District

Streams to the district

1	at central grid (depart)		cg	central grid
2	at sub grid (arrival)		sg	sub grid
3	at sub grid (depart)		uu	upgrading unit
4	at central grid (arrival from sub grid)			

Assumed was that each sub grid consumes as much heat, namely 156 TJ/year.

T are intermediate temperatures defined by the temperature losses (delta T) in the piping in between the districts to define the output of the upgrading unit to overcome the losses.

Insulation type: PUR

		T (C)					Q (TJ/y)			
Sub grids		1	2	3	4		1	2	3	4
	sub grid 5	135	130	70	68		176	163	7	0
	sub grid 4	136	130	70	67		179	163	7	0
	sub grid 3	135	130	70	68		175	162	6	0
	sub grid 2	136	130	70	67		180	164	8	0
	sub grid 1	133	130	70	69		168	160	4	0
		T (C)					Q (TJ/y)			
		To cg 1	From cg 1				To cg 1	From cg 1		
Central grids	grid 2	67	135				14	544		
		T (C)					Q (TJ/y)			
		To uu	To cg 2	From uu	From cg 2		To uu	To cg 2	From uu	From cg 2
	grid 1	67	139	137	66		0	574	922	0

Insulation type: Glass wool/micro PUR

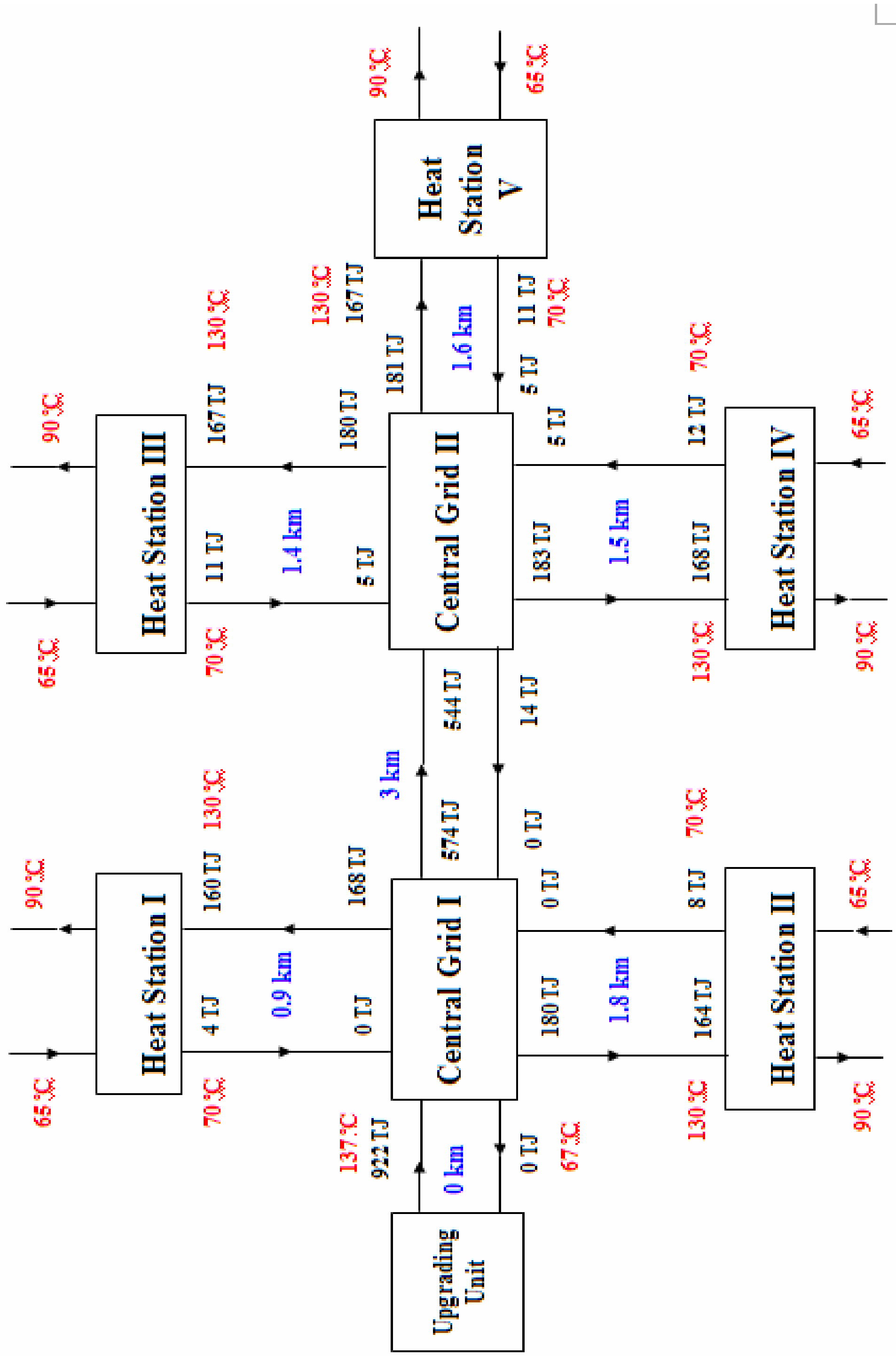
		T (C)					Q (TJ/y)			
Sub grids		1	2	3	4		1	2	3	4
	sub grid 5	131	130	70	69		161	157	1	0
	sub grid 4	131	130	70	69		161	158	2	0
	sub grid 3	131	130	70	69		160	157	1	0
	sub grid 2	131	130	70	69		162	158	2	0
	sub grid 1	131	130	70	70		159	157	1	0
		T (C)					Q (TJ/y)			
		To cg 1	From cg 1				To cg 1	From cg 1		
Central grids	grid 2	69	131				3	485		
		T (C)					Q (TJ/y)			
		To uu	To cg 2	From uu	From cg 2		To uu	To cg 2	From uu	From cg 2
	grid 1	69	132	132	69		0	492	813	0

Final energy streams in and to the district.

			Insulation type: PUR					Insulation type: Glass wool/micro PUR			
		Q (TJ/y)						Q (TJ/y)			
Sub grids		1	2	3	4			1	2	3	4
	sub grid 5	181	167	11	5			162	159	3	1
	sub grid 4	183	168	12	5			162	159	3	1
	sub grid 3	180	167	11	5			161	159	3	1
	sub grid 2	180	164	8	0			162	158	2	0
	sub grid 1	168	160	4	0			159	157	1	0
		Q (TJ/y)						Q (TJ/y)			
		To cg 1	From cg 1					To cg 1	From cg 1		
Central grids	grid 2	14	544					3	485		
		Q (TJ/y)						Q (TJ/y)			
		To uu	To cg 2	From uu	From cg 2			To uu	To cg 2	From uu	From cg 2
	grid 1	0	574	922	0			0	492	813	0

Appendix 16 District Piping System Block Scheme

Insulation Material : PUR
Thickness : 50 mm



Appendix 17 Hazard and Operability Study

Apparatus No. : - Line No. 6	Intention of apparatus: - Intention of line: transfer ammonia from the discharge of 2nd compressor to the condenser			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Failure in the compressor	1)The whole loop is down	1)Maintenance
More	FLOW	2) Not applicable	None	None
	PRESSURE	3)Plugging in the condenser 4)Block valve closed in the condenser while the compressor is running	2) Piping failure 3) The whole cycle gets affected	2) Maintenance
	TEMPERATURE	5) High pressure in the suction and discharge of the compressor 6) High suction temperature	4) Maximum allowable temperature of the 2nd compressor can b achieved	3) Maintenance
Less	FLOW	7) Compressor failure	5) Less heat delivered	4)Maintenance
	PRESSURE	8)Compressor failure	6) Less heat delivered	5)Maintenance
	TEMPERATURE	9)Low suction temperature 10)Compressor failure	7) Less heat transferred in the condenser	6)Maintenance

Apparatus No. : - Line No. 3	Intention of apparatus: - **Intention of line:** transfer ammonia from the evaporator to compressor1			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Plugging in the condenser or in the pipe 2) Pipe fracture 3) Control valve failure	1) The cycle does not work	1)Stop the compressor from running 2)Close the waste water valve 3)Provide a small tank to collect
More	FLOW	4)Not applicable	2) None	4)None
	PRESSURE	5)Compressor suction valve fails closed while the compressor is running	3)High pressure in the whole cycle	5)Maintenance
	TEMPERATURE	6) High heat transfer rate in the evaporator	4) High temperature at the compressor discharge	6) Process Control of waste water –by pass
Less	FLOW	7) Plugging in the evaporator or in the pipe 8) Control valve failure	5) No big deal	7)Compressor control loops adjustment.
	PRESSURE	9) Expansion valve failure	6) Low heat transfer in the condenser	8) Proper Maintenance in the control valve
	TEMPERATURE	10) Expansion valve failure 11) Waste water control valve fails closed	7) Low heat transfer in the condenser	9) Proper Maintenance in the control valves

Apparatus No. : - **Line No. 4**	**Intention of apparatus:** **Intention of line:** transfer ammonia from the 1st compressor to the intercooler			
Guide Word	**Deviation or disturbance**	**Possible causes**	**Consequences**	**Action required**
Not, No	FLOW	1)Failure in the compressor valves	1)The whole loop is down	1)Maintenance
More	FLOW	2) Not applicable	2)None	2)None
	PRESSURE	3)Plugging in downstream 4)Block valve closed while compressor is running	3) Piping failure 4) Low pressure in the suction of the 2nd compressor 5) The whole cycle gets affected	3) Maintenance
	TEMPERATURE	5) High pressure in the suction and discharge of the compressor 6) High suction temperature	6) Maximum allowable suction temperature of the 2nd can be achieved	4) Maintenance
Less	FLOW	7) Not applicable	7)None	5)None
	PRESSURE	8)Compressor failure	8)Low pressure in the suction of the 2nd compressor	6) Maintenance
	TEMPERATURE	9)Low suction temperature 10) Compressor failure	9) Less heat transfer in the intercooler	7) Maintenance

Apparatus No. : - Line No. 5	Intention of apparatus: - Intention of line: Feed ammonia into compressor 2			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Pipe fracture 2)1st compressor failure	1)The cycle does not work 2)Compressor 2 runs empty	1)Shutdown the compressor
More	FLOW	3)Not applicable	3)None	2) None
	PRESSURE	4)Compressor suction valve fails closed while the compressor is running	4)High pressure in the whole cycle	3) Maintenance
	TEMPERATURE	5)Intercooler does not work properly	5) High temperature at the compressor discharge	4) Process Control of tap cold water
Less	FLOW	6) Low flow down upstream	6) No big deal	5) Compressor control loops adjustment.
	PRESSURE	7) Compressor 1 failure	7) Low heat transfer in the condenser	6) Proper Maintenance in the compressor
	TEMPERATURE	7) Failure in the control system failure	8) Low heat transfer in the condenser	7) Maintenance in the control loop of the intercooler

Apparatus No. : - **Line No. 7**	**Intention of apparatus: -** **Intention of line:** transfer ammonia liquid to the expansion valve			
Guide Word	**Deviation or disturbance**	**Possible causes**	**Consequences**	**Action required**
Not, No	FLOW	1)Compressor failure 2)Block valve fails closed upstream the control valve	1) No ammonia enters the evaporator 2)The whole cycle gets affected	1)Maintenance
More	FLOW	3)Not applicable	3)None	2) None
	PRESSURE	4)Plugging in downstream 5)Block valve downstream the expansion closed while compressor is running	4) High temperature in the evaporator and in the suction of the 1st compressor	3)Check lines 4)Maintenance
	TEMPERATURE	6) High temperature at compressor	5) Two-phase flow	5) Process Control loop at the condenser
Less	FLOW	7) Flange leak	Covered by (1), (2)	Covered by (1)
	PRESSURE	8)Compressor failure	6)Two-phase flow	Covered by (1)
	TEMPERATURE	9) Insulation failure	7) No major consequences	Covered by (1)

Apparatus No. : - **Line No. 8**	**Intention of apparatus: -** **Intention of line:** transfer ammonia from the expansion valve to the evaporator			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1) Plugging in the condenser or in the pipe 2) Pipe fracture 3) Control valve failure	1)The cycle does not work	1)Stop the compressor from running 2)Close the waste water valve 3)Provide a small tank to collect ammonia
More	FLOW	4) Not applicable. The compressor delivers a certain amount that is adjusted	2)None	4)None
	PRESSURE	5)Control valve failure	3)Two-phase flow to the compressor	5) Waste water by-pass control loop linked to the control valve
	TEMPERATURE	6)Control valve failure	4)Two-phase flow to the compressor	Covered by (4)
Less	FLOW	7) Plugging in the condenser or in the pipe 8) Control valve failure	5)High temperature at compressor suction	6)Maintenance 7)Slow down the compressor
	PRESSURE	9)Compressor failure 10)Pipe leak	6)Lower suction pressure at the 1st compressor	8) Maintenance
	TEMPERATURE	11) Compressor failure 12)Pipe leak	7)Lower suction pressure at the 1st compressor	9) Maintenance

Apparatus No. : - Line No. 17	Intention of apparatus: - **Intention of line:** transfer air from the blower to the fired heater			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1) Blower failure	1) No combustion 2) Natural gas sent to the atmosphere 3) Explosion	1)Maintenance 2)Proper ratio air: NG control
More	FLOW	2) Failure in the flow control valves	4) Inefficient combustion 5) Uncontrolled emissions	3) Maintenance
	PRESSURE	3)Not likely to occur	6) None	4) None
	TEMPERATURE	4)Not applicable considering the temperature range of the fired heater	7) None	5) None
Less	FLOW	5) Piping fracture 6) Flange leak	8) Poor combustion	6) Maintenance
	PRESSURE	7)Blower failure	9) Mal function of the burner	7) Maintenance
	TEMPERATURE	8)Insulation failure	10) No hazardous for the process	8) Maintenance

Apparatus No. : - Line No. 16	Intention of apparatus: - Intention of line: transfer natural gas to the fired heater			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1) Control valve failure (fails closed)	1) No proper heat released to tap water	1)Maintenance
More	FLOW	2) Failure with flow control valve	2) No proper heat released to tap water	2) Maintenance
	PRESSURE	3) Only likely if there is higher pressure in the NG supply	3) Piping failure	3) None
	TEMPERATURE	4) Not applicable	4) None	4) None
Less	FLOW	5) Failure with flow control valve	5) No proper heat released to tap water	5) Maintenance
	PRESSURE	6)Leak in the supply header or in the pipeline	6) Mal function of the burner	6) Maintenance
	TEMPERATURE	7)Insulation failure	7) Minor hazardous for the process	7) Maintenance

Apparatus No. : - Line No. 21	Intention of apparatus: - **Intention of line:** transfer hot tap water from the Central Grid 1 to the heat station (HEX1)			
Guide Word	**Deviation or disturbance**	**Possible causes**	**Consequences**	**Action Required**
Not, No	FLOW	1) Plugging 2) Piping fracture 3) Pump failure 4) Block valve closed	1)No heat distribution, freezing of the equipment	1)Close the bock valve to the district
More	FLOW	5) Failure in the flow control valves	2) Increasing in the outcoming temperature	2)Proper maintenance
	PRESSURE	6)Plugging in downstream 7)Block valve closed while pump is running	3) Piping failure 4) Line subjected to full pump delivery or surge pressure	3)Pipe should be designed to maximum value 4)Check lines
	TEMPERATURE	8) Furnace control system failure (both temperature and flow)	5) Line fracture or flange leak 6) Two-phase flow	5)Install thermo-expansion relief valve Covered by (1), (2), (3) in the UU-IN file
Less	FLOW	9) Piping fracture 10) Flange leak 11) Pump failure	7) Low heat distribution	6)Maintenance
	PRESSURE	12)Pump failure	8)Two-phase flow 9)Pump cavitation 10)Less heat transfer (gas+iquid)	7)Proper Maintenance
	TEMPERATURE	13)Furnace failure 14)Insulation failure	11) Low heat distribution	8)Maintenance

Apparatus No. : - Line No. 22	Intention of apparatus: - **Intention of line:** transfer hot tap water from Heat Station 1 (HEX1) to the Central Grid 1(CG1)			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Plugging 2)Piping fracture 3)Pump failure 4)Block valve closed	1)No heat distribution, freezing of the equipment	1)Close the bock valve to the district
More	FLOW	5)Failure in the flow control valves	2)Increasing in the outcoming temperature	2)Proper maintenance
	PRESSURE	6)Plugging in downstream 7)Block valve closed while pump is running	3)Piping failure 4)Line subjected to full pump delivery or surge pressure	3)Pipe should be designed to maximum value 4)Check lines
	TEMPERATURE	8)Furnace control system failure (both temperature and flow) 9)Failure in the heat exchanger control system	5)Line fracture or flange leak 6)Two-phase flow	5)Install thermo-expansion relief valve Covered by (1), (2), (3) in the UU-IN file
Less	FLOW	10)Piping fracture 11)Flange leak 12)Pump failure 13)Scaling in heat exchanger	7)Low heat distribution	6)Maintenance
	PRESSURE	14)Pump failure	8)Two-phase flow 9)Pump cavitation 10)Less heat transfer (gas+liquid)	7)Maintenance
	TEMPERATURE	15)Furnace failure 16)Insulation failure	11) Low heat distribution	8)Maintenance

Apparatus No. : - Line No. 10	Intention of apparatus: - **Intention of line:** transfer cold tap water to the intercooler			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Control valve failure 2)Failure in the upstream pipeline 3)Block valve closed	1) High temperature in the suction of the 2nd compressor	1)Shut down the compressor(Trip) 2)Maintenance
More	FLOW	4)Failure in the flow control valves	2)No hazardous consequences to the process	3)Maintenance
	PRESSURE	5)Plugging in downstream 6)Block valve closed while pump is running	3)Piping failure 4)Line subjected to full pump delivery or surge pressure	4)Maintenance 5)Check lines
	TEMPERATURE	7)Furnace control system failure (both temperature and flow) 8)Failure in the heat exchanger control system	5)Line fracture or flange leak 6)Two-phase flow	6)Install thermo-expansion relief valve Covered by (1), (2), (3) in the UU-IN file
Less	FLOW	9)Piping fracture 10)Flange leak 11)Scaling in heat exchanger	7)Low heat distribution	7)Maintenance
	PRESSURE	12)Pump failure	8)Two-phase flow 9)Pump cavitation 10)Less heat transfer (gas+liquid)	Covered by (6)
	TEMPERATURE	13)Furnace failure 14)Insulation failure	11) Low heat distribution	Covered by (6)

Apparatus No. : - Line No. 14	Intention of apparatus: – Intention of line: transfer tap water from the mixer to the furnace			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Control valve failure 2)Failure in the upstream pipeline (block valve closed)	1)No heat supplied to the districts	1)Shut down the compressor 2)Maintenance
More	FLOW	3)Not applicable	2)Not applicable	3)None
	PRESSURE	4)Plugging in downstream 5)Block valve closed while pump is running	3)Piping failure 4)Line subjected to full pump delivery or surge pressure	4)Maintenance 5)Check lines
	TEMPERATURE	6)Furnace control system failure (both temperature and flow) 7)Failure in the heat exchanger control system	5)Two-phase flow	6)Install thermo-expansion relief valve Covered by (1), (2), (3) in the UU-IN file
Less	FLOW	8)Piping fracture 9)Flange leak 10)Scaling in heat exchanger (condenser and intercooler)	5)Low heat distribution to the districts	7)Maintenance
	PRESSURE	11)Pump failure 12)Flange leak	6)Two-phase flow	Covered by (6)
	TEMPERATURE	13)Inefficient heat transfer in the condenser 14)Insulation failure	7)Low heat distribution	Covered by (6)

Apparatus No. : - Line No. 29	Intention of apparatus: - Intention of line: transfer hot tap water from the Upgrading Unit to the Central Grid 2			
Guide Word	**Deviation or disturbance**	**Possible Causes**	**Consequences**	**Action Required**
Not, No	FLOW	1)Plugging 2)Piping fracture 3)Pump failure	1) No heat distribution, freezing of the equipment 2)Local hot spot in ammonia loop 3)Race in the temperature of the compressor	1)Shut-down the furnace 2)Shut-down the compressor 3)Waste water inlet should by-pass the evaporator
More	FLOW	4)Not Applicable (Closed-loop)	4)Not Applicable	4)Not Applicable
	PRESSURE	5)Plugging in the downstream 6)Block valve closed while pump is running	5)Piping failure 6)Line subjected to full pump delivery or surge pressure	5)Pipe should be designed to maximum value 6)Check lines
	TEMPERATURE	7)Furnace control system failure (both temperature and flow) 8)Failure in the district distribution system	7)Line fracture or flange leak 8)High pressure in transfer line	7)Install thermo-expansion relief valve
Less	FLOW	9)Piping fracture 10)Flange leak 11)Pump failure	9)Low heat distribution	8)Maintenance
	PRESSURE	12)Pump failure	10)Two-phase flow 11)Pump cavitation 12)Less heat transfer (gas+liquid)	9)Proper Maintenance
	TEMPERATURE	13)Furnace failure 14)Insulation failure	13) Low heat distribution	10)Maintenance

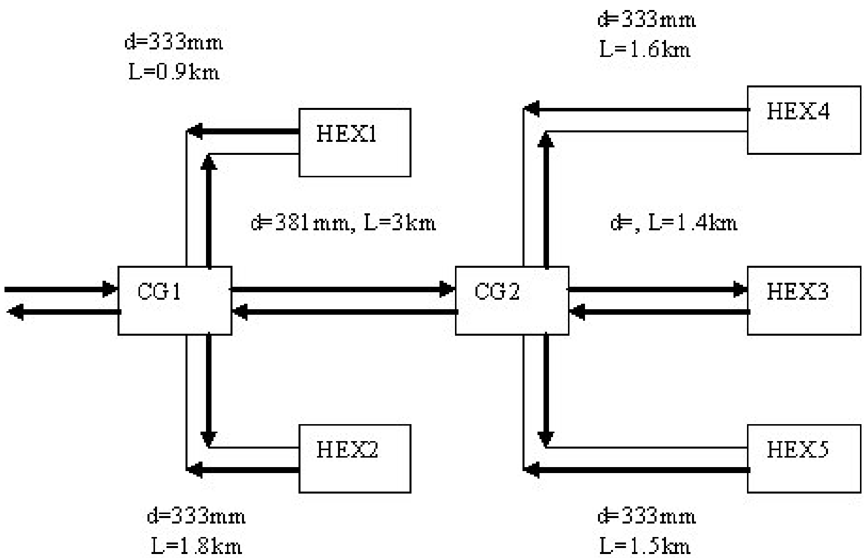
Appendix 18 Fire and Explosion Index Calculation

Area/Country: The Netherlands	**Division:** -	**Location** -	**Date** 12/1/2005
Site -	**Manufacturing Unit** Upgrading Unit of District Heating System (DHS)	**Process Unit** Heat Upgrading Unit	
Materials in Process Unit Ammonia, Natural Gas, Air and Water			
State of Operation Ammonia Thermal Cycle, Water Fired Heater and Water Pipeline Network	**Basic Materials for Material Factor** Ammonia		

Material Factor		4.00
1. General Process Hazards	**Penalty Factor Range**	**Penalty Used**
Base Factor	1.00	1.00
A. Exothermic Chemical Reactions	0.30 - 1.25	-
B. Endothermic Processes	0.20 - 0.40	-
C. Material Handling and Transfer	0.25 - 1.05	-
D. Enclosed or Indoor Process Units	0.25 - 0.90	0.30
E. Acces	0.20 - 0.35	0.35
F. Drainage and Spill Control	0.25 - 0.50	0.25
General Process Hazards Factor (F1)		1.90
2. Special Process Hazards	**Penalty Factor Range**	**Penalty Used**
Base Factor	1.00	1.00
A. Toxic Material(s)	0.20 - 0.80	0.60
B. Sub-Atmosferic Pressure (< 500 mm Hg)	0.50	-
C. Operation In or Near Flammable Range		
1. Tank Farms Storage Flammable Liquids	0.50	-
2. Process Upset or Purge Failure	0.30	0.30
3. Always in Flammable Range	0.80	0.80
D. Dust Explosion	0.25 - 2.00	-
E. Pressure — Operating Pressure: 3800 kPa		-
Relief Setting: kPa		-
F. Low Temperature	0.20 - 0.30	-
G. Quantity of Flammable Material: 2195 kg; Hc = 11315.6 kcal/kg		
1. Liquids or Gases in Process		0.16
2. Liquids or Gases in Storage		-
3. Combustible Solids in Storage, Dust in Process		-
H. Corrosion and Erosion	0.10 - 0.75	-
I. Leakage - Joints and Packing	0.10 - 1.50	0.30
J. Use of Fired Equipment		-
K. Hot Oil Heat Exchange System	0.15 - 1.15	-

L. Rotating Equipment	0.50	0.50
Special Process Hazards Factor (F2)		3.66
Process Units Hazards Factor (F1 x F2) = F3		6.95
Fire and Explosion Index (F3 x MF = F&EI)		**28**

Appendix 19 Heat Losses Calculation (Way In and Return)



WAY IN (hot tap water) at 133C

Nominal diameters: 16" and 14"
Schedule 40 carbon steel pipes

Pipe internal and external diameters for each branch, inches

$$D := \begin{pmatrix} 16 \\ 14 \\ 14 \\ 14 \\ 14 \\ 14 \end{pmatrix} \qquad din := \begin{pmatrix} 15 \\ 13.124 \\ 13.124 \\ 13.124 \\ 13.124 \\ 13.124 \end{pmatrix} \qquad dext := \begin{pmatrix} 16 \\ 14 \\ 14 \\ 14 \\ 14 \\ 14 \end{pmatrix}$$

Mass flowrate in each branch, kg/s

$$massflowrate := \begin{pmatrix} 97.39 \\ 32.47 \\ 32.47 \\ 32.47 \\ 32.47 \\ 32.47 \end{pmatrix} \quad \text{kg/s}$$

The diameters and flow rates in each branch were calculated with Aspen Pipeline model.

Pipe length in each branch, m

$$Length := \begin{pmatrix} 3000 \\ 900 \\ 1800 \\ 1400 \\ 1600 \\ 1500 \end{pmatrix} \quad m$$

Calculating the diameters in meter

$$din := din \cdot 2.54 \cdot 10^{-2} \qquad din = \begin{pmatrix} 0.381 \\ 0.333 \\ 0.333 \\ 0.333 \\ 0.333 \\ 0.333 \end{pmatrix} \; m \qquad dext := dext \cdot 2.54 \cdot 10^{-2} \qquad dext = \begin{pmatrix} 0.406 \\ 0.356 \\ 0.356 \\ 0.356 \\ 0.356 \\ 0.356 \end{pmatrix} \; m$$

Physical Properties of water at bulk temperature, T=133^{o}C:

$T := 133$ C Bulk temperature

Density

$\rho := 882.05$ kg/m^3 Source: Aspen

Dynamic viscosity

$\mu := 0.25 \cdot 10^{-2}$ g/(cm.s) Source: Aspen

Thermal conductivity

$kw := 0.6856$ Watt/(m.K) Source: Aspen

Specific heat

$cp := 4253$ J/(kg.K) Source: Transport Phenomena Data Companion L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

Dimensionless Numbers (Re, Pr, Nu) were calculated at the bulk temperature, T=133 C

$Pr := 1.325$ $Pr = 1.325$ Prandtl Number at bulk temperature Source:Transport Phenomena Data Companion

Velocity in the pipe, m/s

$$vel := \overrightarrow{\frac{massflowrate}{\rho \cdot \pi \cdot \frac{din^2}{4}}}$$

$$vel = \begin{pmatrix} 0.968 \\ 0.422 \\ 0.422 \\ 0.422 \\ 0.422 \\ 0.422 \end{pmatrix}$$ m/s

$$Re := \overrightarrow{\frac{din \cdot vel \cdot \rho}{\mu} \cdot \frac{1000}{100}}$$

$$Re = \begin{pmatrix} 1.302\times 10^6 \\ 4.961\times 10^5 \\ 4.961\times 10^5 \\ 4.961\times 10^5 \\ 4.961\times 10^5 \\ 4.961\times 10^5 \end{pmatrix}$$

Since Re>10^4, there is turbulent flow

$$Nu := 0.023 Re^{0.8} \cdot Pr^{0.33}$$

$$Nu = \begin{pmatrix} 1.967\times 10^3 \\ 908.868 \\ 908.868 \\ 908.868 \\ 908.868 \\ 908.868 \end{pmatrix}$$

Nusselt Number for forced convection

Reference :
Coulson and Rihardson ś, Chemical Engineering, Flow, Heat Transfer and Mass Transfer, 6th edition

$x := 0.05$ m Insulation thickness was taken as 5 cm

Convective heat transfer coefficient

$$h := \overrightarrow{\left(\frac{kw}{dext} \cdot Nu\right)}$$

$$h = \begin{pmatrix} 3.318\times 10^3 \\ 1.752\times 10^3 \\ 1.752\times 10^3 \\ 1.752\times 10^3 \\ 1.752\times 10^3 \\ 1.752\times 10^3 \end{pmatrix}$$ Watt/(m^2.K)

Insulation thermal conductivity

$ki := 0.031$ Watt/(m.K)

For Micro-PUR packed with cyclopenthane gas, K=0.031 Watt/(m.K)
Source: Article "Evolutie in buizen voor stadsverwarming, 2003"

For PUR,Polyurethane, K=0.36 Watt/(m.K)
Source: Transport Phenomena Data Companion

For Glass wool, K=0.041 Watt/(m.K)
Source:Coulson and Richardson's

Estimation of the insulation outlet temperature:

$Tg := 15$ C Ground surface temperature, average over the year

$$aa := \overrightarrow{\left[\left(\frac{din}{2}+x\right)\cdot\frac{h}{ki}\cdot\ln\left(\frac{\frac{din}{2}+x}{\frac{din}{2}}\right)\right]}$$

$$aa = \begin{pmatrix} 5.999\times10^{3} \\ 3.213\times10^{3} \\ 3.213\times10^{3} \\ 3.213\times10^{3} \\ 3.213\times10^{3} \\ 3.213\times10^{3} \end{pmatrix}$$

$$TL := \overrightarrow{\frac{T + aa\cdot Tg}{aa+1}}$$

$$TL = \begin{pmatrix} 15.02 \\ 15.037 \\ 15.037 \\ 15.037 \\ 15.037 \\ 15.037 \end{pmatrix} \quad C$$

Ref: Coulson and Richardson's, Chemical Engineering, Flow, Heat Transfer and Mass Transfer,6th edition

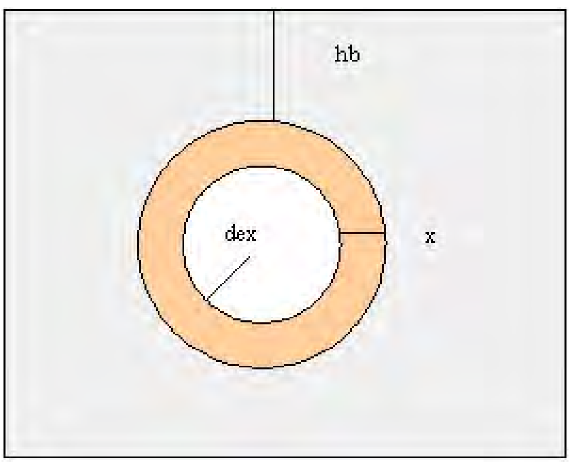
Estimation of the heat losses through the insulation only (considering pipe located in a pipe way on the ground):

$$Qlossinsulation := \overrightarrow{\frac{2\cdot\pi\cdot ki\cdot(T - TL)}{\ln\left(\dfrac{\frac{dext}{2} + x}{\frac{dext}{2}}\right)}} \qquad Qlossinsulation = \begin{pmatrix} 104.46 \\ 92.72 \\ 92.72 \\ 92.72 \\ 92.72 \\ 92.72 \end{pmatrix} \quad W/m$$

Estimation of the heat losses through the insulation and the ground (pipe insulated and undergrounded):

There are two layers: insutation (5cm) + ground (80cm deep)

$hb := 0.8$ m Height buried



hb : height buried
dex : pipe external diameter
x : insulation thickness

$$ksoil := 1.5\cdot\left(\frac{252}{3600}\cdot\frac{4.18}{0.3048}\cdot 1.8\right) \qquad ksoil = 2.592 \quad Watt/(m.K)$$

Source: Kreith, F. Principles of Heat Transfer, 2nd edition, International Textbok Company, Scranton, Pa,1965

Soil thermal conductivity is strongly dependent on the kind of soil (e.g. dry or wet)
For wet soil, k=1.5 BTU/(h.ft.F), which was the type of soil considered.

$$QT := \overrightarrow{\frac{T - Tg}{\frac{1}{2\cdot\pi}\cdot\left[\frac{1}{ki}\cdot\ln\left(\frac{\frac{dext}{2}+x}{\frac{dext}{2}}\right)+\frac{1}{ksoil}\cdot\ln\left[\left[\frac{hb+\left(\frac{dext}{2}+x\right)}{\left(\frac{dext}{2}+x\right)}+\left[\left[\frac{hb+\left(\frac{dext}{2}+x\right)}{\left(\frac{dext}{2}+x\right)}\right]^{2}-1\right]^{\frac{1}{2}}\right]\right]\right]}}$$

The resistance to heat transfer in the film as well as in the pipe wall were considered to be negligible.

$$QT = \begin{pmatrix} 95.253 \\ 85.131 \\ 85.131 \\ 85.131 \\ 85.131 \\ 85.131 \end{pmatrix} \text{Watt/m}$$

Heat losses per length unit for each branch, Watt/m

$$QLoss := \overrightarrow{(QT\cdot Length)} \qquad QLoss = \begin{pmatrix} 2.858\times 10^{5} \\ 7.662\times 10^{4} \\ 1.532\times 10^{5} \\ 1.192\times 10^{5} \\ 1.362\times 10^{5} \\ 1.277\times 10^{5} \end{pmatrix} \text{Watt}$$

Heat losses in Watt

Estimation of final temperature in each branch considering T initial as 133 C

$$tf := \overrightarrow{\left(T - \frac{QLoss}{massflowrate\cdot cp}\right)} \qquad tf = \begin{pmatrix} 132.31 \\ 132.445 \\ 131.89 \\ 132.137 \\ 132.014 \\ 132.075 \end{pmatrix} \text{C}$$

$$\Delta T := \overrightarrow{(T - tf)} \qquad \Delta T = \begin{pmatrix} 0.69 \\ 0.555 \\ 1.11 \\ 0.863 \\ 0.986 \\ 0.925 \end{pmatrix} \text{C}$$

ΔT per branch, C

Branch from CG1 to CG2:

$TfinalCG1CG2 := tf_0$ $TfinalCG1CG2 = 132.31$ C

$\Delta TCG1CG2 := \Delta T_0$ $\Delta TCG1CG2 = 0.69$ C

$QLossCG1CG2 := QLoss_0$ $QLossCG1CG2 = 2.858 \times 10^5$ Watt

Branch from CG1 to HEX1:

$TfinalCG1HEX1 := tf_1$ $TfinalCG1HEX1 = 132.445$ C

$\Delta ThotCG1HEX1 := \Delta T_1$ $\Delta ThotCG1HEX1 = 0.555$ C

$QLhotCG1HEX1 := QLoss_1$ $QLhotCG1HEX1 = 7.662 \times 10^4$ Watt

Branch from CG1 to HEX2:

$TfinalCG1HEX2 := tf_2$ $TfinalCG1HEX2 = 131.89$ C

$\Delta ThotCG1HEX2 := \Delta T_2$ $\Delta ThotCG1HEX2 = 1.11$ C

$QLhotCG1HEX2 := QLoss_2$ $QLhotCG1HEX2 = 1.532 \times 10^5$ Watt

Branch from CG2 to HEX3:

$TfinalCG2HEX3 := tf_3$ $TfinalCG2HEX3 = 132.137$ C

$\Delta ThotCG2HEX3 := \Delta T_3$ $\Delta ThotCG2HEX3 = 0.863$ C

$QLhotCG2HEX3 := QLoss_3$ $QLhotCG2HEX3 = 1.192 \times 10^5$ Watt

Branch from CG2 to HEX4:

$TfinalCG2HEX4 := tf_4$ $TfinalCG2HEX4 = 132.014$ C

$\Delta ThotCG2HEX4 := \Delta T_4$ $\Delta ThotCG2HEX4 = 0.986$ C

$QLhotCG2HEX4 := QLoss_4$ $QLhotCG2HEX4 = 1.362 \times 10^5$ Watt

Branch from CG2 to HEX5:

$TfinalCG2HEX5 := tf_5$ $\qquad$ $TfinalCG2HEX5 = 132.075$ C

$\Delta ThotCG2HEX5 := \Delta T_5$ $\qquad$ $\Delta ThotCG2HEX5 = 0.925$ C

$QLhotCG2HEX5 := QLoss_5$ $\qquad$ $QLhotCG2HEX5 = 1.277 \times 10^5$ Watt

Total heat losses in the pipeline (hot tap water stream) = $1671.3*10^3$ Watt

Total content of hot tap water leaving the Upgrading Unit = $2.927*10^7$ Watt

Percentage of the energy content lost in the pipeline is the following:

$$\frac{1671.3 \cdot 10^3}{2.927 \cdot 10^7} \cdot 100 = 5.71$$

5.7% of the input energy available in the hot tap water exiting the Upgrading Unit is lost in the pipeline

Returning (cold tap water) at 70 C

All pipe data (i.e., diameter, length, insulation thickness), mass flowrates, etc, are the same as in the Way In branch (hot tap water)

Physical Properties of water at bulk temperature, T=70^{o}C:

$T := 70$ C Bulk temperature

Density

$\rho := 977.9$ kg/m^3 $\qquad$ Source: Transport Phenomena Data Companion L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

Dynamic viscosity

$\mu := 0.404 \cdot 10^{-2}$ g/(cm.s) $\qquad$ Source: Transport Phenomena Data Companion L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

Thermal conductivity

$kw := 0.665$ Watt/(m.K) $\qquad$ Source: Transport Phenomena Data Companion L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

Specific heat

$cp := 4194$ J/(kg.K) Source: Transport Phenomena Data Companion
L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

Dimensionless Numbers (Re, Pr, Nu) were calculated at the bulk temperature, T=70 C

$Pr := 2.57$ $Pr = 2.57$ Prandtl Number at bulk temperature
Source:Transport Phenomena Data Companion

Velocity in the pipe, m/s

$$vel := \overrightarrow{\frac{massflowrate}{\rho \cdot \pi \cdot \frac{din^2}{4}}} \qquad vel = \begin{pmatrix} 0.874 \\ 0.38 \\ 0.38 \\ 0.38 \\ 0.38 \\ 0.38 \end{pmatrix}$$ m/s

$$Re := \overrightarrow{\frac{din \cdot vel \cdot \rho}{\mu}} \cdot \frac{1000}{100} \qquad Re = \begin{pmatrix} 8.056\times 10^5 \\ 3.07\times 10^5 \\ 3.07\times 10^5 \\ 3.07\times 10^5 \\ 3.07\times 10^5 \\ 3.07\times 10^5 \end{pmatrix}$$

Since Re>10^4, there is turbulent flow

$$Nu := 0.023 Re^{0.8} \cdot Pr^{0.33} \qquad Nu = \begin{pmatrix} 1.667\times 10^3 \\ 770.362 \\ 770.362 \\ 770.362 \\ 770.362 \\ 770.362 \end{pmatrix}$$

Nusselt Number for forced convection

$x := 0.05$ m Insulation thickness was taken as 5 cm

Convective heat transfer coefficient

$$h := \overrightarrow{\left(\frac{kw}{dext}\cdot Nu\right)} \qquad h = \begin{pmatrix} 2.728\times 10^{3} \\ 1.441\times 10^{3} \\ 1.441\times 10^{3} \\ 1.441\times 10^{3} \\ 1.441\times 10^{3} \\ 1.441\times 10^{3} \end{pmatrix} \qquad \text{Watt/(m}^2\text{.K)}$$

$$aa := \overrightarrow{\left[\left(\frac{din}{2}+x\right)\cdot\frac{h}{ki}\cdot\ln\left(\frac{\frac{din}{2}+x}{\frac{din}{2}}\right)\right]} \qquad aa = \begin{pmatrix} 4.932\times 10^{3} \\ 2.642\times 10^{3} \\ 2.642\times 10^{3} \\ 2.642\times 10^{3} \\ 2.642\times 10^{3} \\ 2.642\times 10^{3} \end{pmatrix}$$

$$TL := \overrightarrow{\frac{T + aa\cdot Tg}{aa+1}} \qquad TL = \begin{pmatrix} 15.011 \\ 15.021 \\ 15.021 \\ 15.021 \\ 15.021 \\ 15.021 \end{pmatrix} \qquad C$$

Estimation of the heat losses through the insulation only (considering pipe located in a pipe way on the ground):

$$Qlossinsulation := \overrightarrow{\frac{2\cdot\pi\cdot ki\cdot(T-TL)}{\ln\left(\frac{\frac{dext}{2}+x}{\frac{dext}{2}}\right)}} \qquad Qlossinsulation = \begin{pmatrix} 48.687 \\ 43.214 \\ 43.214 \\ 43.214 \\ 43.214 \\ 43.214 \end{pmatrix} \qquad W/m$$

Estimation of the heat losses through the insulation and the ground (pipe insulated and undergrounded):

There are two layers: insutation (15cm) + ground (80cm deep)

$hb := 0.8$ m Height buried

$$QT := \overrightarrow{\frac{T - Tg}{\frac{1}{2\cdot\pi}\cdot\left[\frac{1}{ki}\cdot\ln\left(\frac{\frac{dext}{2}+x}{\frac{dext}{2}}\right)+\frac{1}{ksoil}\cdot\ln\left[\left[\frac{hb}{\left(\frac{dext}{2}+x\right)}+\left[\left(\frac{hb}{\left(\frac{dext}{2}+x\right)}\right)^{2}-1\right]^{\frac{1}{2}}\right]\right]\right]}}$$

$$QT = \begin{pmatrix} 44.952 \\ 40.08 \\ 40.08 \\ 40.08 \\ 40.08 \\ 40.08 \end{pmatrix}$$ Watt/m

Heat losses per length unit for each branch, Watt/m

$$QLoss := \overrightarrow{(QT\cdot Length)}$$

$$QLoss = \begin{pmatrix} 1.349\times 10^{5} \\ 3.607\times 10^{4} \\ 7.214\times 10^{4} \\ 5.611\times 10^{4} \\ 6.413\times 10^{4} \\ 6.012\times 10^{4} \end{pmatrix}$$

Heat losses in Watt

Watt

Estimation of final temperature in each branch considering T initial as 133 C :

$$tf := \overrightarrow{\left(T - \frac{QLoss}{massflowrate\cdot cp}\right)}$$

$$tf = \begin{pmatrix} 69.67 \\ 69.735 \\ 69.47 \\ 69.588 \\ 69.529 \\ 69.559 \end{pmatrix}$$ C

$\Delta T := \overrightarrow{(T - tf)}$

$$\Delta T = \begin{pmatrix} 0.33 \\ 0.265 \\ 0.53 \\ 0.412 \\ 0.471 \\ 0.441 \end{pmatrix}$$ C

ΔT per branch, C

Branch from CG1 to CG2:

$TfinalCG1CG2 := tf_0$	$TfinalCG1CG2 = 69.67$	C
$\Delta TcoldCG1CG2 := \Delta T_0$	$\Delta TcoldCG1CG2 = 0.33$	C
$QLcoldCG1CG2 := QLoss_0$	$QLcoldCG1CG2 = 1.349 \times 10^5$	Watt

Branch from CG1 to HEX1:

$TfinalCG1HEX1 := tf_1$	$TfinalCG1HEX1 = 69.735$	C
$\Delta TcoldCG1HEX1 := \Delta T_1$	$\Delta TcoldCG1HEX1 = 0.265$	C
$QLcoldCG1HEX1 := QLoss_1$	$QLcoldCG1HEX1 = 3.607 \times 10^4$	Watt

Branch from CG1 to HEX2:

$TfinalCG1HEX2 := tf_2$	$TfinalCG1HEX2 = 69.47$	C
$\Delta TcoldCG1HEX2 := \Delta T_2$	$\Delta TcoldCG1HEX2 = 0.53$	C
$QLcoldCG1HEX2 := QLoss_2$	$QLcoldCG1HEX2 = 7.214 \times 10^4$	Watt

Branch from CG2 to HEX3:

$TfinalCG2HEX3 := tf_3$	$TfinalCG2HEX3 = 69.588$	C
$\Delta TcoldCG2HEX3 := \Delta T_3$	$\Delta TcoldCG2HEX3 = 0.412$	C
$QLcoldCG2HEX3 := QLoss_3$	$QLcoldCG2HEX3 = 5.611 \times 10^4$	Watt

Branch from CG2 to HEX4:

$TfinalCG2HEX4 := tf_4$ $\qquad$ $TfinalCG2HEX4 = 69.529$ C

$\Delta TcoldCG2HEX4 := \Delta T_4$ $\qquad$ $\Delta TcoldCG2HEX4 = 0.471$ C

$QLcoldCG2HEX4 := QLoss_4$ $\qquad$ $QLcoldCG2HEX4 = 6.413 \times 10^4$ Watt

Branch from CG2 to HEX5:

$TfinalCG2HEX5 := tf_5$ $\qquad$ $TfinalCG2HEX5 = 69.559$ C

$\Delta TcoldCG2HEX5 := \Delta T_5$ $\qquad$ $\Delta TcoldCG2HEX5 = 0.441$ C

$QLcoldCG2HEX5 := QLoss_5$ $\qquad$ $QLcoldCG2HEX5 = 6.012 \times 10^4$ Watt

Reference used :

1. Neher, J.H., 1949, The temperature rise of burried cables and pipes.Trans.A.I.E.E. 68, (1):9-21

2. Sissom, L.E., Pitts, D. R., Elements of Transport Phenomena, 1972, McGraw-Hill, Inc.

3. Kreith, F. Principles of Heat Transfer, 2nd edition, International Textbok Company, Scranton, Pa,1965

4. Coulson and Richardson's, Chemical Engineering, Flow, Heat Transfer and Mass Transfer, 6th edition

5. Transport Phenomena Data Companion,L.P.B.M. Janssen, M.M.C.G. Warmoeskerken

6. Article: Evolutie in buizen voor stadsverwarming, 2003

Appendix 20 Exergy Analysis of The Process

References:
Smith, J.M., Van Ness,H.C., Introduction to Chemical Engineering Thermodynamics 4th edition, 1987,McGraw-Hill, USA.

Since ammonia undergoes a cyclic process, the only changes that need to be considered for calculation of the ideal work are those of the gases passing through the furnace.

Ideal work is calculated by the following expression:

Source: Aspen

$ExEMISSION := -2.653\cdot 10^{6}$	J/kg	$massflowgases := 11.05$	kg/s
$ExNGSTD := -2.534\cdot 10^{6}$	J/kg	$massflowair := 10.346$	kg/s
$ExAIRIN := -4.5295\cdot 10^{4}$	J/kg	$massflowNG := 0.7130$	kg/s

$$Wideal := -[massflowgases\cdot ExEMISSION - (massflowNG\cdot ExNGSTD + massflowair\cdot ExAIRIN)]$$

$Wideal = 2.704\times 10^{7}$ Watt Eq.16.5 Van Ness

Calculating the rate of entropy generation for each of the units of the cycle:

$T0 := 298.15$ K

W lost at the furnace

$SEMISSION := 98.1459$ J/(kg.K)

$SNGSTD := -3674.03$ J/(kg.K)

$SAIRIN := 133.9976$ J/(kg.K)

$$SgenerationFurnace := massflowgases\cdot SEMISSION - (massflowNG\cdot SNGSTD + massflowair\cdot SAIRIN)$$

$SgenerationFurnace = 2.318\times 10^{3}$ Watt/K

$$WlostFurnace := T0\cdot[massflowgases\cdot SEMISSION - (massflowNG\cdot SNGSTD + massflowair\cdot SAIRIN)]$$

Eq 16.15 Van Ness

$WlostFurnace = 6.91 \times 10^{5}$ Watt

W lost at the compressors

$massflowrateammonia := 3.89$ kg/s

$SCOMP1OUT := -6.753 \cdot 10^{3}$ J/(kg.K) $SCOMP1IN := -6.865 \cdot 10^{3}$ J/(kg.K)

$SgenerationCompr1 := massflowrateammonia \cdot (SCOMP1OUT - SCOMP1IN)$

$SgenerationCompr1 = 435.68$ Watt/K

$WlostCompr1 := massflowrateammonia \cdot T0 \cdot (SCOMP1OUT - SCOMP1IN)$

$WlostCompr1 = 1.299 \times 10^{5}$ Watt

$SCOMP2OUT := -6.761 \cdot 10^{3}$ J/(kg.K) $SCOMP2IN := -6.885 \cdot 10^{3}$ J/(kg.K)

$SgenerationCompr2 := massflowrateammonia \cdot (SCOMP2OUT - SCOMP2IN)$

$SgenerationCompr2 = 482.36$ Watt/K

$WlostCompr2 := massflowrateammonia \cdot T0 \cdot (SCOMP2OUT - SCOMP2IN)$

$WlostCompr2 = 1.438 \times 10^{5}$ Watt

$SgenerationCompression := SgenerationCompr1 + SgenerationCompr2$

$SgenerationCompression = 918.04$ Watt/K

$WlostCompression := WlostCompr1 + WlostCompr2$

$WlostCompression = 2.737 \times 10^{5}$ Watt

W lost at the control valve

$Svout := -1.006 \cdot 10^{4}$ J/(kg.K) $Svin := -1.0197 \cdot 10^{4}$ J/(kg.K)

$SgenerationCvalve := massflowrateammonia \cdot (Svout - Svin)$

$SgenerationCvalve = 532.93$ Watt/K

$WlostValve := massflowrateammonia \cdot T0 \cdot (Svout - Svin)$

$WlostValve = 1.589\times 10^{5}$ Watt

W lost at the pump

$SCWOUT := -7.8587\cdot 10^{3}$ J/(kg.K) $SFURNOUT := -7.8592\cdot 10^{3}$ J/(kg.K)

$$SgenerationPump := massflowrateammonia\cdot(SCWOUT - SFURNOUT)$$

$SgenerationPump = 1.945$ Watt/K

$$WlostPump := massflowrateammonia\cdot T0\cdot(SCWOUT - SFURNOUT)$$

$WlostPump = 579.902$ Watt

$$STotal := SgenerationPump + SgenerationCvalve + SgenerationCompression + SgenerationFurnace$$

$SP\% := \frac{SgenerationPump}{STotal}\cdot 100$ $SP\% = 0.052$

$SCV\% := \frac{SgenerationCvalve}{STotal}\cdot 100$ $SCV\% = 14.134$

$SCompr\% := \frac{SgenerationCompression}{STotal}\cdot 100$ $SCompr\% = 24.347$

$SFurn\% := \frac{SgenerationFurnace}{STotal}\cdot 100$ $SFurn\% = 61.468$

$WlostPump\% := \frac{WlostPump}{Wideal}\cdot 100$ $WlostPump\% = 2.145\times 10^{-3}$

$$WlostFurn\% := \frac{WlostFurnace}{Wideal} \cdot 100 \qquad WlostFurn\% = 2.556$$

$$WlostComp\% := \frac{WlostFurnace}{Wideal} \cdot 100 \qquad WlostFurn\% = 2.556$$

$$WlostCvalve\% := \frac{WlostValve}{Wideal} \cdot 100 \qquad WlostCvalve\% = 0.588$$

Appendix 21 Exergy Losses Calculation

References :

1. Article: Comakli, K. Yuksel,B., Camakli, O., Dept. f Mechanical Engineering, Ataturk University, Turkey, *Evaluation of energy and exergy losses in district heating network*, 2003, Applied Thermal Engineering 24(2004) 1009-1017.

2. K. Iyer, C. Van Wijmen, E. Babbe. *Feasibility Study of District Heating of Delft Using Residual Waste Heat of DSM*. 3 June 2004. TU Delft, The Netherlands

3. Kotas, T.J., The Exergy Method of Thermal Plant Analysis, 2nd edition, Kriger Publishing , Company, USA, 1995

The average pressure in the main network is approximately 13 bar.

The exergy balance is written as follows:

Exergy of hot water = Exergy lost in the pipeline due to heat losses +
Exergy lost due to hot water transportation +
Exergy lost during the heat transfer in the heat exchangers

Heat losses were calculated in Appendix 19, considering the Micro-PUR as insulation and insulation thickness of 50mm

Exergy Losses in the main pipeline
PPG2 branch - From central grid 1 (CCG1) to central grid 2 (CCG2)
(16" Sch 40, 3000m), both ways

Exergy losses due to heat losses in the pipe:

$LosshotCG1CG2 := 2.858\times 10^{5}$ Watt

$LosscoldCG1CG2 := 1.349\times 10^{5}$ Watt

$Qlossesmain := LosshotCG1CG2 + LosscoldCG1CG2$

$Qlossesmain = 4.207\times 10^{5}$ Watt

$T0 := 25 + 273$ $T0 = 298$ K Average temperature of the surroundings taken as 25C

$Thotwater := 133 + 273$ $Thotwater = 406$ K

$THotwaterin := 133 + 273$ $THotwaterin = 406$ K Hot stream (IN)

$THotwaterout := 70 + 273$ $THotwaterout = 343$ K Cold stream (Return)

$$Tw := \frac{THotwaterin + THotwaterout}{2} \qquad Tw = 374.5 \quad K \qquad 372.5 - 273 = 99.5$$

$$ExLossmain := Qlossesmain \cdot \left(1 - \frac{T0}{Tw}\right)$$

$$ExLossmain = 8.594 \times 10^{4} \quad \text{Watt}$$

Exergy Losses in the pipeline #1
PHEX1 branch - From CCG1 to heat exchanger 1 (HEX1)
(14" Sch 40, 900m), both ways

Exergy losses due to heat losses in the pipe:

$$LosshotCG1HEX1 := 7.662 \times 10^{4} \quad \text{Watt}$$

$$LosscoldCG1HEX1 := 3.607 \times 10^{4} \quad \text{Watt}$$

$$Qlosses1 := LosshotCG1HEX1 + LosscoldCG1HEX1$$

$$Qlosses1 = 1.127 \times 10^{5} \quad \text{Watt}$$

$$THotwaterin := 133 + 273 \qquad THotwaterin = 406 \quad K \quad \text{Hot stream (IN)}$$

$$THotwaterout := 66 + 273 \qquad THotwaterout = 339 \quad K \quad \text{Cold stream (Return)}$$

$$Tw := \frac{THotwaterin + THotwaterout}{2} \qquad Tw = 372.5 \quad K$$

$$ExLoss1 := Qlosses1 \cdot \left(1 - \frac{T0}{Tw}\right)$$

$$ExLoss1 = 2.254 \times 10^{4} \quad \text{Watt}$$

Exergy Losses in the pipeline #2
PHEX2 branch - From CCG1 to heat exchanger 2 (HEX2)
(14" Sch 40, 1800m), both ways

Exergy losses due to heat losses in the pipe:

$$LosshotCG1HEX2 := 1.532 \times 10^{5} \quad \text{Watt}$$

$$LosscoldCG1HEX2 := 7.214 \times 10^{4} \quad \text{Watt}$$

$Qlosses2 := LosshotCG1HEX2 + LosscoldCG1HEX2$

$Qlosses2 = 2.253\times 10^{5}$ Watt

$THotwaterin := 133 + 273$ $THotwaterin = 406$ K Hot stream (IN)

$THotwaterout := 66 + 273$ $THotwaterout = 339$ K Cold stream (Return)

$$Tw := \frac{THotwaterin + THotwaterout}{2}$$ $Tw = 372.5$ K

$$ExLoss2 := Qlosses2\cdot\left(1 - \frac{T0}{Tw}\right)$$

$ExLoss2 = 4.507\times 10^{4}$ Watt

Exergy Losses in the pipeline #3
PHEX3 branch - From CCG2 to heat exchanger 3 (HEX3)
(14" Sch 40, 1400m), both ways

Exergy losses due to heat losses in the pipe:

$LosshotCG1HEX3 := 1.192\times 10^{5}$ Watt

$LosscoldCG1HEX3 := 5.611\times 10^{4}$ Watt

$Qlosses3 := LosshotCG1HEX3 + LosscoldCG1HEX3$

$Qlosses3 = 1.753\times 10^{5}$ Watt

$THotwaterin := 133 + 273$ $THotwaterin = 406$ K Hot stream (IN)

$THotwaterout := 66 + 273$ $THotwaterout = 339$ K Cold stream (Return)

$$Tw := \frac{THotwaterin + THotwaterout}{2}$$ $Tw = 372.5$ K

$$ExLoss3 := Qlosses3\cdot\left(1 - \frac{T0}{Tw}\right)$$

$ExLoss3 = 3.506\times 10^{4}$ Watt

Exergy Losses in the pipeline #4
PHEX4 branch - From CCG2 to heat exchanger 4 (HEX4)
(14" Sch 40, 1600m), both ways

Exergy losses due to heat losses in the pipe:

$\text{LosshotCG1HEX4} := 1.362 \times 10^{5}$ Watt

$\text{LosscoldCG1HEX4} := 6.413 \times 10^{4}$ Watt

$\text{Qlosses4} := \text{LosshotCG1HEX4} + \text{LosscoldCG1HEX4}$

$\text{Qlosses4} = 2.003 \times 10^{5}$ Watt

$\text{THotwaterin} := 133 + 273$ $\quad \text{THotwaterin} = 406$ K Hot stream (IN)

$\text{THotwaterout} := 66 + 273$ $\quad \text{THotwaterout} = 339$ K Cold stream (Return)

$$\text{Tw} := \frac{\text{THotwaterin} + \text{THotwaterout}}{2} \qquad \text{Tw} = 372.5 \text{ K}$$

$$\text{ExLoss4} := \text{Qlosses4} \cdot \left(1 - \frac{\text{T0}}{\text{Tw}}\right)$$

$\text{ExLoss4} = 4.007 \times 10^{4}$ Watt

Exergy Losses in the pipeline #5
PHEX5 branch - From CCG2 to heat exchanger 5 (HEX5)
(14" Sch 40, 1500m), both ways

Exergy losses due to heat losses in the pipe:

$\text{LosshotCG1HEX5} := 1.277 \times 10^{5}$ Watt

$\text{LosscoldCG1HEX5} := 6.012 \times 10^{4}$ Watt

$\text{Qlosses5} := \text{LosshotCG1HEX5} + \text{LosscoldCG1HEX5}$

$\text{THotwaterin} := 133 + 273$ $\quad \text{THotwaterin} = 406$ K Hot stream (IN)

$\text{THotwaterout} := 66 + 273$ $\quad \text{THotwaterout} = 339$ K Cold stream (Return)

$$Tw := \frac{THotwaterin + THotwaterout}{2} \qquad Tw = 372.5 \quad K$$

$$ExLoss5 := Qlosses5 \cdot \left(1 - \frac{T0}{Tw}\right)$$

$$ExLoss5 = 3.756 \times 10^{4}$$ Watt

Exergy losses due to hot water transportation

Electric-driven pumps are used transport hot water from the Upgrading Unit to the Districts.
Electrical energy is pure exergy and such exergy is used to overcome the flow resistance in the pipeline network, which is actually transformed into heat.

$$Wp := 107824.6$$ Watt

$$ELossTransport := Wp \cdot \left(\frac{T0}{Tw}\right) \qquad ELossTransport = 8.626 \times 10^{4}$$ Watt

Exergy losses during heat transfer in the heat exchangers (5 heat exchangers in total)

Heat transfer in a heat exchanger is an irreversible process and as a consequence exergy losses occur. The total exergy losses in the process is due to irreversibility in heat transfer
in addition to friction of both flows. According to Kotas [68], exergy losses due to friction in liquid flow are small due to small specific volume of the compressible fluids.

$Qc := 2.33 \cdot 10^{7}$ Watt — Heat duty of each heat exchanger (5 in total)

$Thotin := 133 + 273$ K — This temperature changes depending on the heat losses. Let's assume Thotin as 133 C.

$Thotout := 70 + 273$ K — Taken based on approach rules of thumb for heat exchangers.

$Tcoldin := 60 + 273$ K — Taken based on approach rules of thumb for heat exchangers.

$Tcoldout := 90 + 60$ K — Required temperature at the districts

$$Thot := \frac{Thotin + Thotout}{2} \qquad Thot = 374.5 \quad K$$

$$Tcold := \frac{Tcoldin + Tcoldout}{2} \qquad Tcold = 241.5 \quad K$$

$$4219.3(133-70)\cdot 32.47 = 8.631\times 10^{6} \qquad Qc := 8.631\times 10^{6}$$

$$ELossHeatExchangers := 5T0\cdot(-Qc)\cdot\left(\frac{1}{Thot}-\frac{1}{Tcold}\right)$$

$$ELossHeatExchangers = 1.891\times 10^{7} \quad Watt$$

Total Exergy Losses

$$ELossPipe := ExLossmain + ExLoss1 + ExLoss2 + ExLoss3 + ExLoss4 + ExLoss5$$

$$ELossPipe = 2.662\times 10^{5} \quad Watt$$

$$TExergyLoss := ELossPipe + ELossTransport + ELossHeatExchangers$$

$$TExergyLoss = 1.926\times 10^{7} \quad Watt$$

Total exergy input to hot water

Estimating the exergy of the hot water stream (discharge of the pump) exiting the Upgrading Unit

Using Peng Robinson EoS

From Program Prop Mathcad at 137C and 13 bar,
Tref=298K,
PR EoS:
H=-36479.7 J/mol
S=-97.02364 J/(mol.K)

$H := -36479.7\frac{1}{18}\cdot 100($ $\quad H = -2.027\times 10^{6}$ J/Kg

$S := -97.02364\frac{1}{18}\cdot 100($ $\quad S = -5.39\times 10^{3}$ J/(kg.K)

$\text{massflowrate} := 97.3\text{s}$ $\quad \text{massflowrate} = 97.39$ kg/s

$Ex := \text{massflowrate}\cdot(H - T0\cdot S)$

$Ex = -4.094\times 10^{7}$ J/s $\quad Ex = -4.094\times 10^{7}$ Watt

$\text{TotalExergyinput} := 4.094\times 10^{7}$ Watt $\quad$ Taken as positive value

Percentage of exergy losses:

$$\text{RatioELossPipeExergyinput} := \frac{\text{ELossPipe}}{\text{TotalExergyinput}}$$

$\text{RatioELossPipeExergyinput} = 6.503\times 10^{-3}$ fraction of input exergy lost in the pipeline due to heat losses

$$\text{RatioELossTransportExergyinput} := \frac{\text{ELossTransport}}{\text{TotalExergyinput}}$$

$\text{RatioELossTransportExergyinput} = 2.107\times 10^{-3}$ fraction of input exergy lost in the pipeline due to flow, friction, etc

$$\text{RatioELossHeatExchangersExergyinput} := \frac{\text{ELossHeatExchangers}}{\text{TotalExergyinput}}$$

$\text{RatioELossHeatExchangersExergyinput} = 0.462$ fraction of input exergy lost in the heat exchangers

Result from Aspen AVAILMX for stream CWOUT: Exergy=-1.277*10^9 Watt

Method 2. Lee-Kesler EoS
From the tables Ref: RPP, 4th edition:
H=-2445.43 J/mol
S=-27.203 J/(mol.K)

$H := -2445.43\frac{1}{18}\cdot 100($ $H = -1.359\times 10^{5}$ J/Kg

$S := -27.203\frac{1}{18}\cdot 100($ $S = -1.511\times 10^{3}$ J/(kg.K)

$massflowrate := 97.3($ $massflowrate = 97.39$ kg/s

$Ex := massflowrate\cdot(H - T0\cdot S)$

$Ex = 3.063\times 10^{7}$ J/s $Ex = 3.063\times 10^{7}$ Watt

Appendix 22 COP Calculation

Calculating the coefficient of performance of the MHP (mechanical heat pump) and for the CHP (chemical heat pump).

Mechanical heat pump (MHP):

From the simulation with Aspen the following data were gathered:

$Qhigh := 4.804 \times 10^{6}$ Watt

$Qintercooler := 2.028 \times 10^{5}$ Watt

$Qlow := 3.535 \times 10^{6}$ Watt

$Wcompression := 9.4583 \times 10^{5} + 6.896 \times 10^{5}$ Watt Energy input as work by the compressors

1. Coefficient of Performance (COP)

$$COP := \frac{Qhigh + Qintercooler}{Wcompression}$$

$COP = 3.061$

COP = 3.1, ratio useful heat output to energy input (work). Qintercooler was included since it was recovered at high tempertaure (discharge of the first compressor)

2. Exergy Efficiency (η_E)

$Tlow := 12.0 + 273$	$Tlow = 285$	K	Evaporator temperature
$Thigh := 76.0 + 273$	$Thigh = 349$	K	Condenser temperature
$Tc := 0 + 273$	$Tc = 273$	K	Reference temperature, taken as 0 C (must be lower than Tlow)

$$COPideal := \frac{1 - \frac{Tc}{Thigh}}{1 - \frac{Tc}{Tlow}}$$

$COPideal = 5.172$ This is Carnot COP

$$\eta E := \frac{COP}{COPideal}$$

$\eta E = 0.592$

This parameter indicates the quality and quantity of the upgraded heat.

Chemical heat pump (CHP):

From the simulation with Aspen the following data were gathered:

$Qreboiler := 7973.7$ kW $\quad$ $Qexo := 1788.5$ kW

$Qcondenser := 8687.8$ kW $\quad$ $Qendo := 2318.8$ kW

$Qlow := Qreboiler + Qendo$

$Qlow = 1.029 \times 10^{4}$ kW $\quad$ Qlow is the amount of heat absorbed that is used for both the endothermic reaction and the products separation in the distillation column.

$Qhigh := Qexo$ $\quad$ Qhigh is the amount of upgraded heat at the highest temperature, that's to say the exothermic reaction heat duty

The COP can be calculated by the following equation:

$COP := \frac{Qhigh}{Qlow}$ $\quad$ $COP = 0.174$ $\quad$ COP = 17.4%. It is the enthalpy efficiency of the system

The ideal COP is limited by tyhe conservation law of heat and entropy:

$Thigh := 200 + 273$ $\quad$ temperature of the exothermic reactor

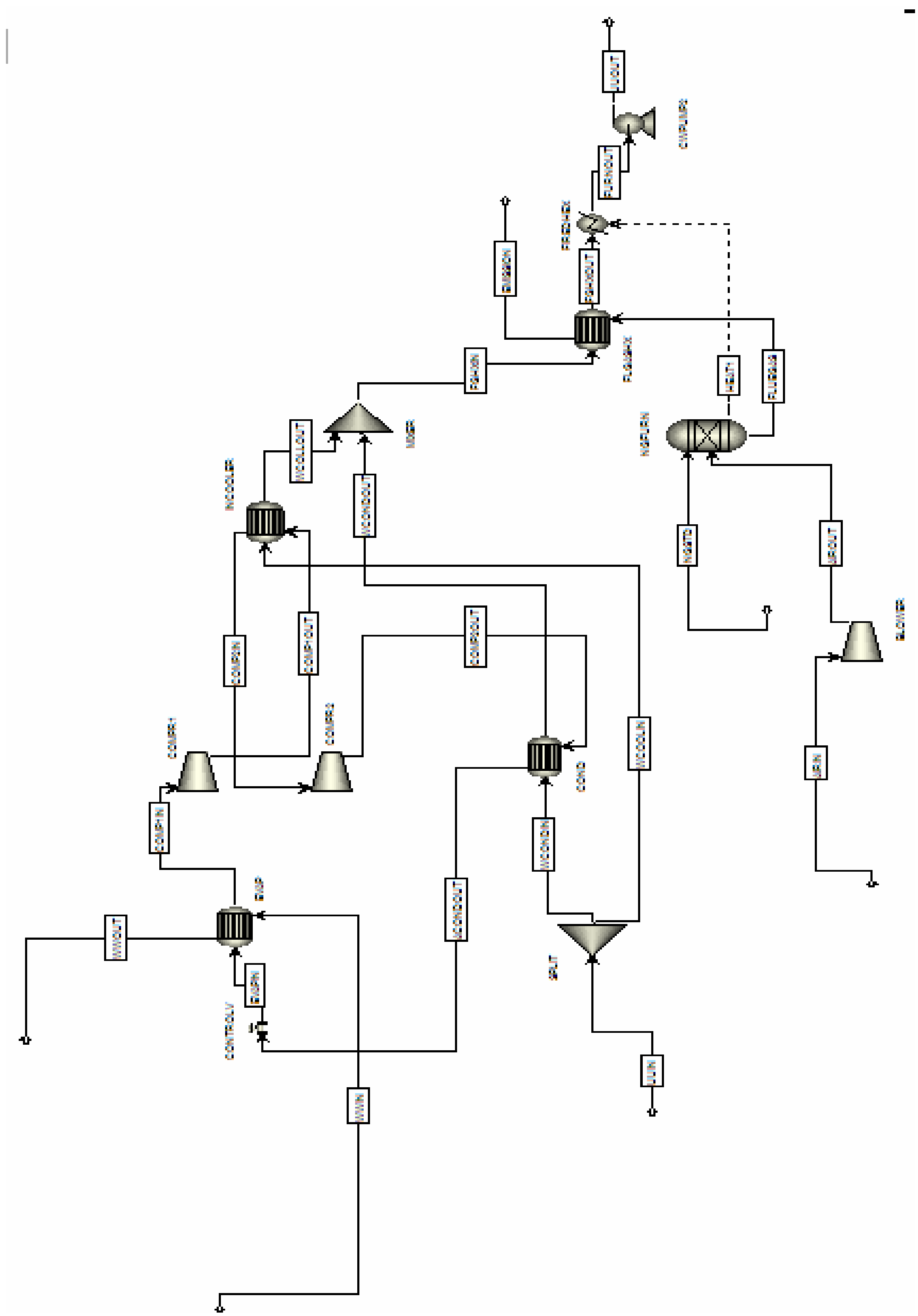
$Tc := 43.73 + 273$ $\quad$ temperature of the condenser

$Tlow := 82.5 + 273$ $\quad$ temperature of the exothermic reactor

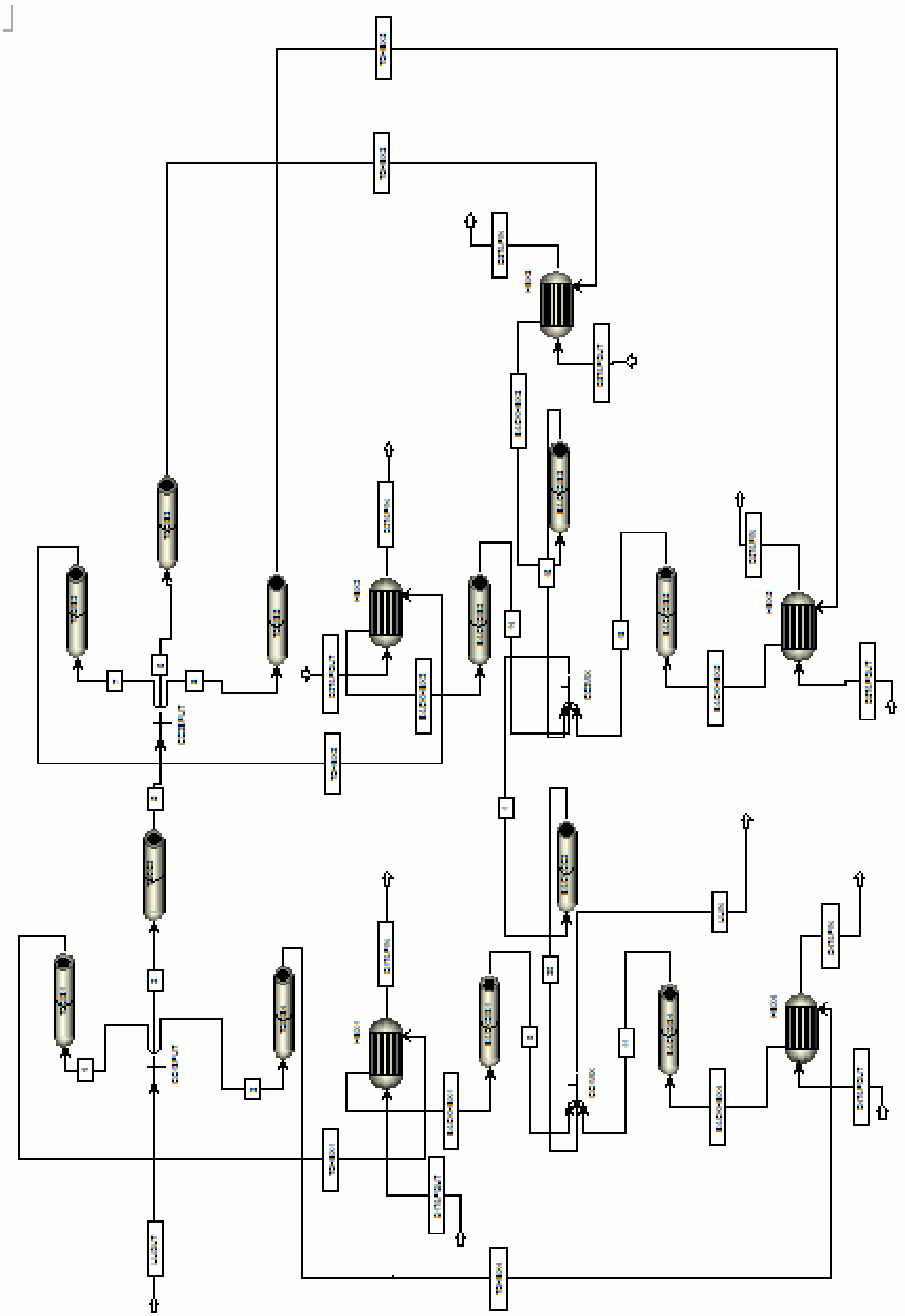
$$COPideal := \frac{1 - \left(\frac{Tc}{Tlow}\right)}{1 - \left(\frac{Tc}{Thigh}\right)} \qquad COPideal = 0.33$$

COPideal = 33.3%. It is the maximum fraction of the recovered waste heat that can be upgraded.

Appendix 23 Ammonia Heat Pump Aspen Flowsheet



Appendix 24 District Pipeline Aspen Flowsheet



Appendix 25 Total Investment of Ammonia Heat Pump

Description	Fraction	Amount
Total investment		$ 29,264,054
Fixed capital	1.00	$ 24,512,821
Direct costs	0.79	$ 19,291,590
Onsite costs		$ 16,043,641
Purchased equipment	0.37	$ 9,082,000
Purchased-equipment installation	0.17	$ 4,081,385
Instrumentation and control	0.05	$ 1,164,359
Piping UU	0.04	$ 980,513
Electrical equipment and materials	0.03	$ 735,385
Offsite costs		$ 3,247,949
Buildings	0.03	$ 735,385
Yard improvement	0.03	$ 796,667
Service facilities	0.04	$ 980,513
Land	0.03	$ 735,385
Indirect costs	0.21	$ 5,221,231
Engineering and supervision	0.08	$ 1,961,026
Construction expenses	0.05	$ 1,176,615
Contractor's fee	0.03	$ 612,821
Contingency	0.06	$ 1,470,769
Working capital		$ 1,463,203
Raw material, finished products, accounts receivable, cash on hand, account payable and taxes payable 15% of total capital investment.	0.05	$ 1,463,203
Costs for transport pipelines *		$ 16,953,225
One time revenues (minus)		-$ 13,665,194
Start-up costs		$ 1,470,769
Process modifications, start-up labor, loss in production	0.06	$ 1,470,769

Prices from (NAP prijzenboekje 20e editie februari 1999)		
The price for total steel underground transport pipelines are (1999)(16inch):	440	€ /m
The material price for under ground steel transport pipes (DN350),		
included average discount of 45%	88	€ /m
Price for laying the pipes:	352	€ /m
Index correction of 4 % per year:	445	€ /m
Prices for the pipes (by Marcel Verboven, Weijers-Waalwijk BV)		
DN350 – 355,6x5,6 / 500 (staal – PUR – PE) Standaard PUR schuim	117.68	€ /m
DN350 – 355,6x5,6 / 500 (staal – PUR – PE) MicroPUR schuim	156.98	€ /m
In the first case we used Standaard PUR, resulting at a length of 21 km in:	11,824,518	€
Including tax	$ 16,953,225	

Appendix 26 Production Cost of Ammonia Heat Pump

		Water			Ammonia				Natural Gas				Electricity	
	price	1.28	euro/m^3		307.85	euro/ton			0.18	euro/m^3			0.1	euro/kWh
	BTW	1.06							1.19				1.19	
	tax	0.146	euro/m^3		1.19				0.01	euro/m^3			0.0086	euro/kWh
									0.0057	euro/MJ				
	km											Comp1	691	kW
V pipe 16 inch	14.4	1868	m^3		700	kg		Furnace	115000	ft^3/hr				
V pipe 14 inch	6	596	m^3						3256	m^3/hr		Comp2	946	kW
												Clean w pump 2	67	kW
												Blower	367	kW
	total amount/y	2464	m^3		700	kg			35,657,930	m^3			18,138,982	kWh
	costs/y	€ 3,703			€ 256.44				€ 7,994,508				€ 2,314,534	
	total costs	€ 10,313,001												
		$ 12,311,091												
1m^3gas	31.7	MJ												
exchange course:	0.8377	euro for 1 dollar												

Appendix 27 Production Cost of Furnace

		Water				Natural Gas				Electricity	
	price	1.28	euro/m^3			0.18	euro/m^3			0.1	euro/kWh
	BTW	1.06				1.19				1.19	
	tax	0.146	euro/m^3			0.01	euro/m^3			0.0086	euro/kWh
	km				Furnace	125000	ft^3/hr		Clean w pump 1	53	kW
V pipe 16 inch	14.4	1868	m^3			3540	m^3/hr				
V pipe 14 inch	6	596	m^3						Blower	378	kW
	total amount/y	2,464	m^3			38,758,620	m^3			3,776,524	kWh
	costs/y	€ 3,703				€ 8,689,683				€ 481,884	
	total costs	€ 9,175,270									
		$ 10,952,930									
1m^3gas	31.7	MJ									
exchange course:	0.8377	$/euro									

Appendix 28 Total Product Cost

		Fraction	of		Amount
Total product costs				$	20,314,787
Manufacturing costs				$	19,958,957
Direct production cost		0.73	Total product costs	$	14,786,715
	Raw materials		Gas, waste water	$	9,743,402
	Utilities		Electricity NH3 H2O	$	2,766,279
	Maintenance	0.04	Fixed capital	$	980,513
	Operating supplies	0.15	Maintenance	$	147,077
	Labor	*	Operators	$	450,000
	Supervision	0.2	Labor	$	90,000
	Laboratory charges			$	-
	Royalties	0.03	Total product costs	$	609,444
Fixed charges		0.21	Total product costs	$	4,259,934
	Depreciation	0.1	Fixed capital	$	2,451,282
	Local taxes	0.025	Fixed capital	$	612,821
	Insurance	0.007	Fixed capital	$	171,590
	Rent	0			
	Interest	0.035	Total cap invest.	$	1,024,242
Plant OVHD		0.04	Total product costs	$	912,308
		0.72	Labor	$	324,000
		0.024	Fixed capital	$	588,308
SARE		0.025	Revenue	$	355,830

* The plant will be controlled by a 5 shifts, each shift consists of 2 persons. The salary of an operator is assumed to be € 45,000 per year.
Supervision is supposed to be 1/5 of the labor costs, equals 1 supervisor.

Appendix 29 Revenues

City heating rates for small users and small business-like users from 1 oct 2005

BTW	19%	
Energy tax	6.32	€/ GJ
exchange course:	0.8377	$/euro
Annual consumption	780000	GJ
Houses connected	11000	

Heat price for heating and hot tap water		Fraction	Inc tax		Ex tax	
Consumption less than	119 GJ per GJ	0.8	€	22.04	€	12.20
Consumption from	119 GJ per GJ	0.2	€	19.59	€	10.14
Right for usage (yearly)			€	295.07	€	247.96
One time connection contribution			€	3,715.18	€	3,122.00

Total yearly revenues		€	11,923,148	$	14,233,196
	Consumption	€	9,195,610		
	Connections	€	2,727,538		
1/3 of the one time connection revenues		€	11,447,333	$	13,665,194

Source: http://gemeente.purmerend.nl/smartsite.html?id=3923

Appendix 30 Chemical Heat Pump Simulation

In order to validate the previously done feasibility study, the Chemical Heat Pump (CHP) process was simulated in Aspen Plus. The simulation flowsheet and the distillation profiles are presented here.

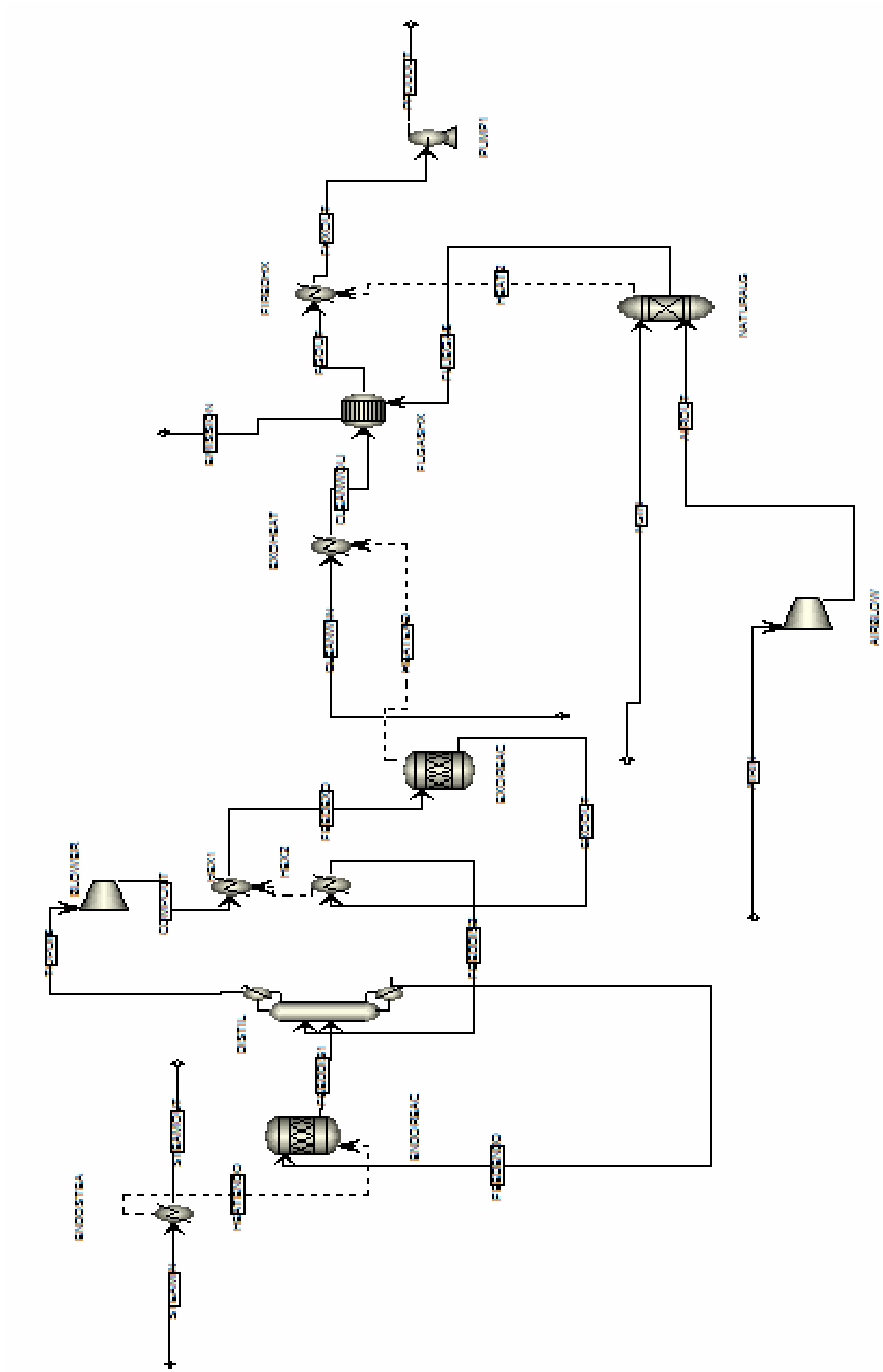
It should be noted that both endothermic and exothermic reactors were modeled by using the equilibrium reactor models (conversions were specified based on data gathered from the literature/simulation studies). The conditions of the reactors were also specified so that only the corresponding reactions were taking place. Unlike the endothermic reaction, which was simulated in the liquid phase (at boiling point), the exothermic reaction was simulated in the gas phase following the information gathered from the articles that report similar studies [10],[12],[13]. The conversions were predicted according to the chemical equilibrium conditions mentioned in the articles [10],[12],[13],[14]. The molar ratio hydrogen to acetone was set as 1.0 in accordance with the article [12].

Moreover, the distillation column was modeled with rigorous multi component distillation with partial condenser. The purity of acetone in the bottom and isopropanol in the top were specified and determined the reflux ratio and the reboiler duty.

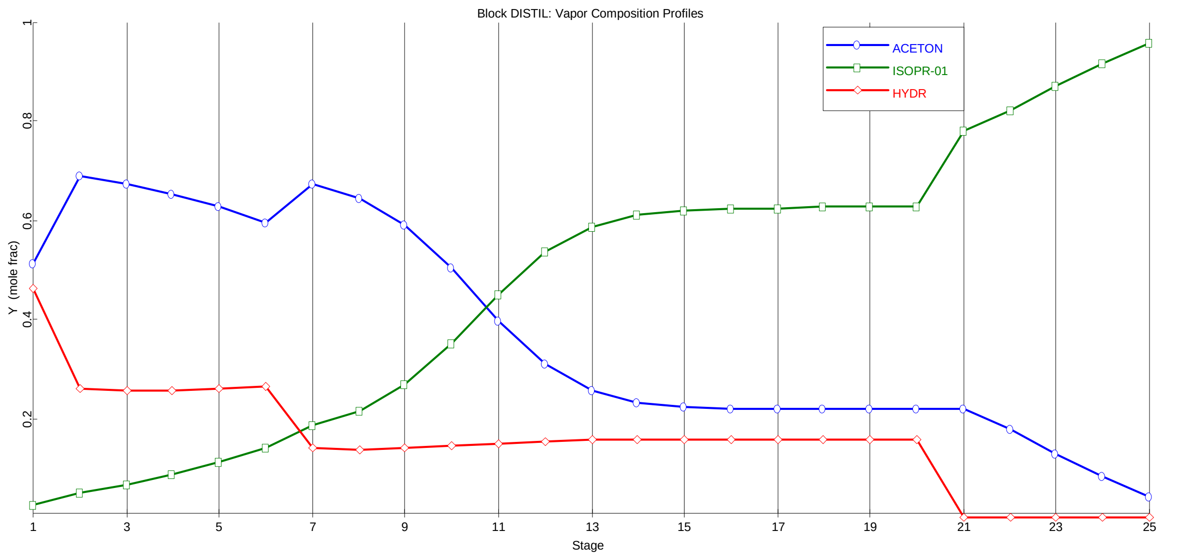
The remaining equipments (compressor and heat exchanger) were straight forwardly modeled.

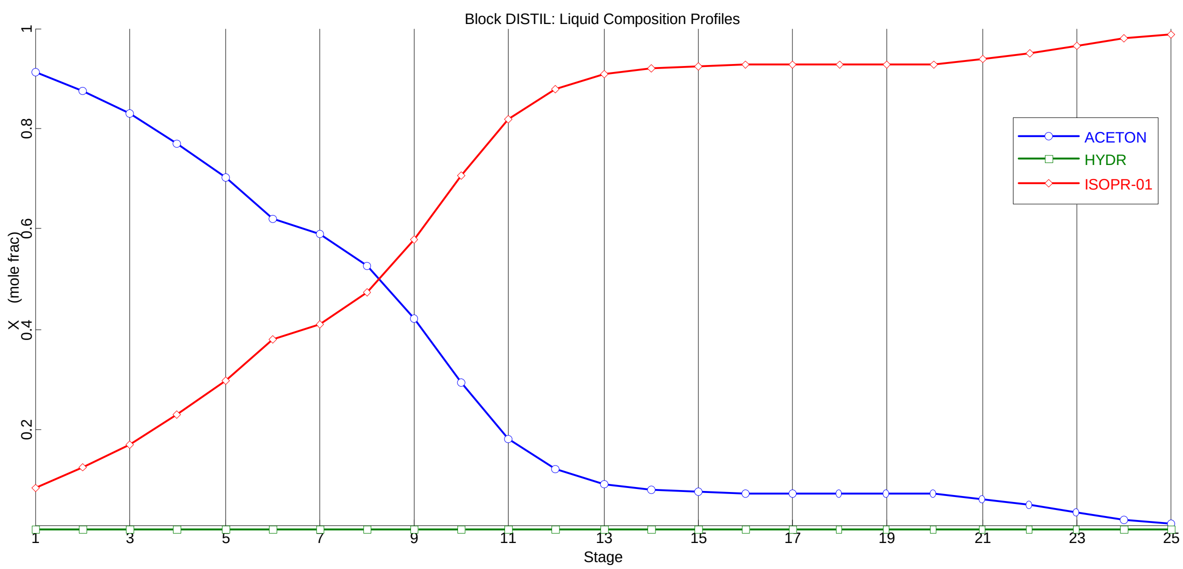
Only one tear stream (the feed to the exothermic reactor, FEEDEXO) was used to perform the simulation within the cycle. It serves as input for the total flow rate, which remains constant as the simulation runs. However, its composition, temperature, and pressure are automatically updated as a result of each run.

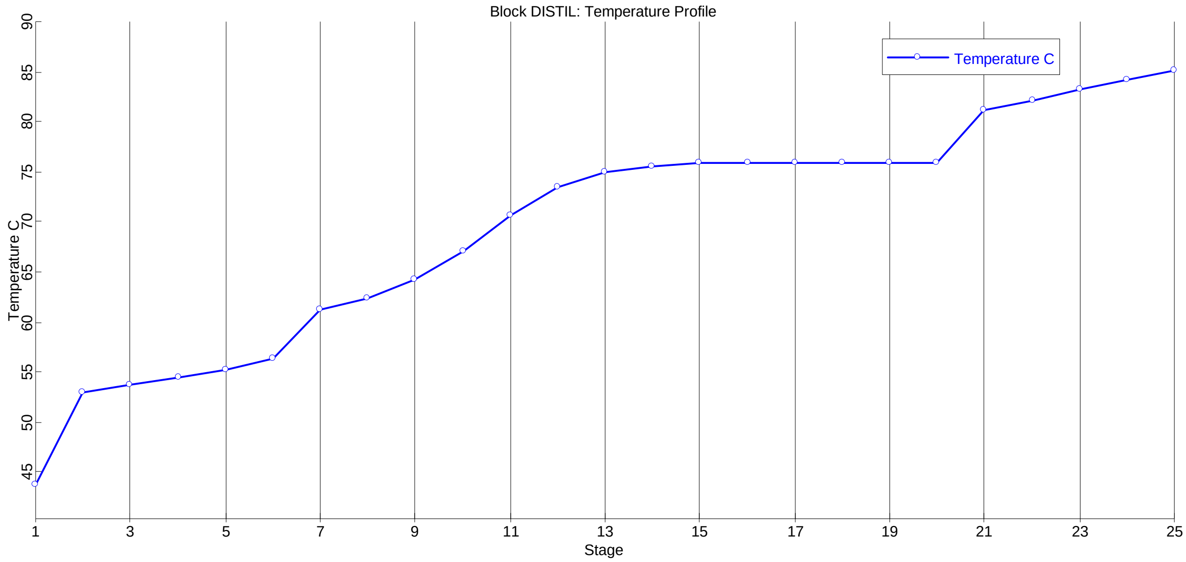
Based on heat integration studies, other simulations were run in order to better evaluate the technical feasibility of this system. However, due to low temperature level of DSM waste water, there is no way to use this stream as the heating source of isopropanol-acetone chemical heat pump.

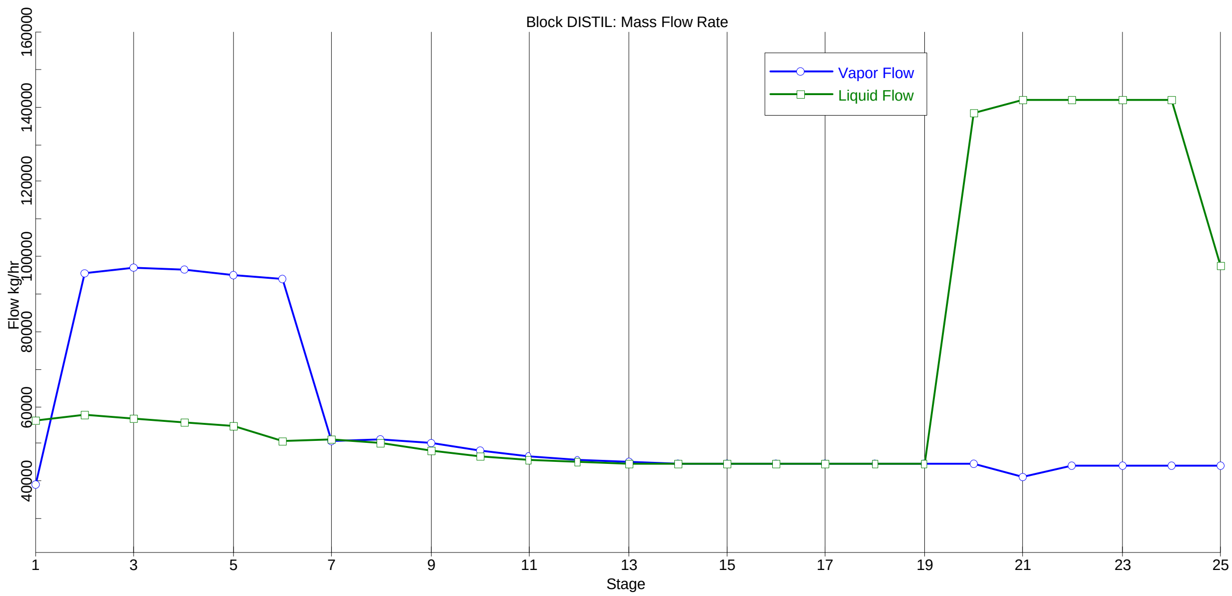


Considering the design specifications of 2.3% mole fraction of isopropanol at distillate and 1.0% mole fraction of acetone at bottoms, the distillation profiles were found to be as follows:






Block DISTIL: Temperature Profile
Temperature C
90
85
80
75
70
65
60
55
50
45
1
3
5
7
9
11
13
15
17
19
21
23
25
Stage
Temperature C


Block DISTIL: Mass Flow Rate
Vapor Flow
Liquid Flow
Flow kg/hr
160000
140000
120000
100000
80000
60000
40000
1
3
5
7
9
11
13
15
17
19
21
23
25
Stage

Appendix 31 Economic Evaluation of Chemical Heat Pump

All the costs have been evaluated based on "Product and Process Design Principles" [1].

According to Chemical engineering Magazine (2), the Annual CE Plant Cost Index for 2000 (Base Case):

$CE1 := 394.1$

The Annual CE Plant Cost Index for 2004:

$CE2 := 444.2$

The cost of equipments are calculated separately, as follows:

Blower:

Blower horsepower: $Pc := 501.96$ hp

The cost of centrifugal (turbo) blower including the cost of electric motor is:

$CB := \exp(6.6547 + 0.7900 \ln(Pc))$

For cast iron as material of construction:

$FM := 1.0$

$CpBlower := FM \cdot CB \cdot \left(\frac{CE2}{CE1}\right)$

$CpBlower = 1.19 \times 10^5$ \$

Air Blower:

This blower is overdesigned to cope with the fluctuations in the winter time.

Blower horsepower: $Pc := 2.575 \times 10^3$ hp

The cost of centrifugal (turbo) blower including the cost of electric motor is:

$CB := \exp(6.6547 + 0.7900 \ln(Pc))$

For cast iron as material of construction:

$FM := 1.0$

$CpAIRBlower := FM \cdot CB \cdot \left(\frac{CE2}{CE1}\right)$

$CpAIRBlower = 4.331 \times 10^5$ \$

Heat Exchanger:

Shell side : $Tcompout := 334.49$ K $Tfeedexo := 460.075$ K

Tube side : $Texoout := 473.15$ K $Tfeeddis2 := 348.15$ K

$$\Delta THEX := \frac{[(Tfeedexo - Tcompout) - (Texoout - Tfeeddis2)]}{\ln\left[\frac{(Tfeedexo - Tcompout)}{(Texoout - Tfeeddis2)}\right]}$$

$\Delta THEX = 125.292$ K

$QHEX := 2.694 \times 10^{6}$ Watt

According to Coulson & Richardson [3], the heat coefficient is:

$U := 200 \; \frac{Watt}{m^{2} \cdot K}$

thus, the heat exchanger area is:

$$AHEX := \frac{QHEX \cdot 10.764}{U \cdot \Delta THEX}$$

$AHEX = 1.157 \times 10^{3}$ ft^{2}

The material of construction of the shell side and the tube side is carbon steel, thus

$$FM := 0.0 + \left(\frac{AHEX}{100}\right)^{0.0}$$

$FM = 1$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure based on the shell side pressure in psig is calculated as follows:

$PHEX := 1.5$ bar

$PHEXShell := PHEX \cdot 14.5 - 14.7$

$PHEXShell = 7.05$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHEXShell}{100}\right) + 0.0017\left(\frac{PHEXShell}{100}\right)^{2}$$

$FP = 0.982$

The heat exchanger is of floating head type:

$$CBHEX := \exp\left[11.667 - 0.8709\ln(AHEX) + 0.09005(\ln(AHEX))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHEX := FP \cdot FM \cdot FL \cdot CBHEX\left(\frac{CE2}{CE1}\right)$$

$$CpHEX = 2.741\times 10^4 \quad \$$$

Distillation Condenser:

Shell side (process fluid): $Tcond := 316.88$ K $Tcwout := 302.15$ K

Tube side (cooling water) : $Tcwin := 293.15$ K

$$\Delta TCOND := Tcond - \left(\frac{Tcwin + Tcwout}{2}\right)$$

$$\Delta TCOND = 19.232 \quad K$$

$$QCOND := 8.694\times 10^6 \quad Watt$$

According to Coulson & Richardson [3], the heat coefficient is:

$$U := 600 \quad \frac{Watt}{m^2 \cdot K}$$

thus, the heat exchanger area is:

$$ACOND := \frac{QCOND\,10.764}{U \cdot \Delta TCOND}$$

$$ACOND = 8.11\times 10^3 \quad ft^2$$

The material of construction of the shell side and the tube side is carbon steel, thus

$$FM := 0.0 + \left(\frac{ACOND}{100}\right)^{0.0}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure based on the shell side pressure in psig is calculated as follows:

$PHEX := 1.15$ bar

$PHEXShell := PHEX \cdot 14.5 - 14.7$

$PHEXShell = 1.975$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHEXShell}{100}\right) + 0.0017\left(\frac{PHEXShell}{100}\right)^2$$

$FP = 0.981$

The heat exchanger is of floating head type:

$$CBCOND := \exp\left[11.667 - 0.8709\ln(ACOND) + 0.09005(\ln(ACOND))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpCOND := FP \cdot FM \cdot FL \cdot CBCOND\left(\frac{CE2}{CE1}\right)$$

$CpCOND = 8.387 \times 10^4$ \$

Distillation Reboiler:

Shell side (process fluid): $Treb := 358.195$ K

Tube side (LPsteam) : $Tsteam := 399.82$ K

$\Delta TREB := Tsteam - Treb$

$\Delta TREB = 41.625$ K

$QREB := 7.972 \times 10^6$ Watt

According to Coulson & Richardson [3], the heat coefficient is:

$U := 1000 \frac{Watt}{m^2 \cdot K}$

thus, the heat exchanger area is:

$$AREB := \frac{QREB \cdot 10.764}{U \cdot \Delta TREB}$$

$AREB = 2.062 \times 10^3$ ft^2

The material of construction of the shell side and the tube side is carbon steel, thus

$$FM := 0.0 + \left(\frac{AREB}{100}\right)^{0.0}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure based on the shell side pressure in psig is calculated as follows:

$PHEX := 1.1$ bar

$$PHEXShell := PHEX \cdot 14.5 - 14.7$$

$PHEXShell = 1.25$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHEXShell}{100}\right) + 0.0017\left(\frac{PHEXShell}{100}\right)^2$$

$$FP = 0.981$$

The heat exchanger is of floating head type:

$$CBREB := \exp\left[11.667 - 0.8709\ln(AREB) + 0.09005(\ln(AREB))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpREB := FP \cdot FM \cdot FL \cdot CBREB\left(\frac{CE2}{CE1}\right)$$

$CpREB = 3.554 \times 10^4$ $

Distillation Column:

Column diameter: $Di := 13.1$ ft

Number of trays: $Nt := 23$

Tray spacing: $Ts := 2$ ft

Length of column expect the heads:

$L := Nt \cdot Ts$

$L = 46$ ft

Density of carbon steel:

$density := 490 \dfrac{lb}{ft^3}$

Since the operating pressure of column is between 0 and 5 psig, the design pressure is:

$Pd := 10$ psig

The operating temperature of the column is 180 F, thus the design temperature is:

$Td := 230$ F

For SA-387B which is a commonly used low-alloy (1%Cr and 0.5% Mo) steel
for
non-corrosive environment including the presense of hydrogen, operating at
this
design temperature the maximum allowable stress is:

$S := 15000$ psi

For carbon steel the value of welding efficiency is:

$E := 0.85$

Based on the above values, the cylindrical shell wall thickness is computed from ASME pressure-vessel code as follows:

$$tp := \frac{Pd \cdot Di \cdot 12}{2 \cdot S \cdot E - 1.2 \cdot Pd}$$

$tp = 0.062$ inch

Since the operational pressure is low, this thickness is small to give sufficient rigidity to column, according to column diameter the following wall thickness is used:

$tp := \dfrac{5}{16}$ inch

The weight of the shell and the two heads is approximately:

$$W := \pi \cdot \left(Di + \frac{tp}{12}\right) \cdot (L + 0.8 \cdot Di) \cdot \frac{tp \cdot density}{12}$$

$W = 2.972 \times 10^4$ lb

since the material of construction is carbon steel:

$$FM := 1.0$$

$$Cv := \exp\left[7.0374 + 0.18255\ln(W) + 0.02297(\ln(W))^2\right]$$

$$Cpl := 237.1\cdot(Di)^{0.63316}\cdot(L)^{0.80161}$$

Thus, the cost of the column includingnozzles, manholes, a skirt, internals, platforms, and ladders is:

$$CP := FM\cdot Cv + Cpl$$

$$CP = 1.113\times 10^{5} \quad \$$$

The cost of trays should be included:

The number of trays is greater than 20: $Fnt := 1.0$

Valve tray is used: $Ftt := 1.18$

The material of construction of trays is carbon steel: $Ftm := 1.0$

$$Cbt := 369\cdot\exp(0.1739\,Di)$$

Thus the cost of trays is:

$$Ct := Nt\cdot Fnt\cdot Ftt\cdot Ftm\cdot Cbt$$

$$Ct = 9.772\times 10^{4} \quad \$$$

The total cost of distillation column, including trays and internals is:

$$CpDIS := (CP + Ct)\cdot\left(\frac{CE2}{CE1}\right)$$

$$CpDIS = 2.356\times 10^{5} \quad \$$$

Endo Reactor:

Since the waste water temperature is lower than the process stream temperature in the endo reactor, low pressure steam with equivalent energy content as waste water has been considered as the energy resource for the endothermic reaction. Moreover, the reactor is considered to be a shell-tube heat exchanger:

Shell side (steam) : $Tsteam := 399.82$ K $Tfeeddis1 := 336.584$ K

Tube side : $Tfeedendo := 358.195$ K

$$\Delta TENDREAC := Tsteam - \left(\frac{Tfeedendo + Tfeeddis1}{2}\right)$$

$\Delta TENDREAC = 52.43$ K

$QENDREAC := 2.326 \times 10^{6}$ Watt

According to Coulson & Richardson [3], the heat coefficient is:

$$U := 1000 \quad \frac{Watt}{m^{2} \cdot K}$$

thus, the heat exchanger area is:

$$AENDREAC := \frac{QENDREAC \cdot 10.764}{U \cdot \Delta TENDREAC}$$

$AENDREAC = 477.529 \ ft^{2}$

The material of construction of the shell side and the tube side is carbon steel, thus

$$FM := 0.0 + \left(\frac{AENDREAC}{100}\right)^{0.0}$$

$FM = 1$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure based on the shell side pressure in psig is calculated as follows:

$PHEX := 1.16$ bar

$PHEXShell := PHEX \cdot 14.5 - 14.7$

$PHEXShell = 2.12$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHEXShell}{100}\right) + 0.0017\left(\frac{PHEXShell}{100}\right)^2$$

$FP = 0.981$

The heat exchanger is of floating head type:

$$CBENDREAC := \exp\left[11.667 - 0.8709\ln(AENDREAC) + 0.09005(\ln(AENDREAC))^2\right]$$

The purchase cost of the endo reactor is:

$$CpENDREAC := FP \cdot FM \cdot FL \cdot CBENDREAC\left(\frac{CE2}{CE1}\right)$$

$CpENDREAC = 2.064 \times 10^4$ $

Exo Reactor:

The reactor is considered to be a shell-tube heat exchanger:

Shell side : $Tfeedexo := 460.075$ K $Texoout := 473.15$ K

Tube side (clean water) : $Tcleanwin := 333.15$ K $Tcleanwout := 335.368$ K

$$\Delta TEXOREAC := \frac{[(Texoout - Tfeedexo) - (Tcleanwout - Tcleanwin)]}{\ln\left[\frac{(Texoout - Tfeedexo)}{(Tcleanwout - Tcleanwin)}\right]}$$

$\Delta TEXOREAC = 6.12$ K

$QEXOREAC := 1.789 \times 10^6$ Watt

According to Coulson & Richardson [3], the heat coefficient is:

$$U := 600 \frac{Watt}{m^2 \cdot K}$$

thus, the heat exchanger area is:

$$AEXOREAC := \frac{QEXOREAC \cdot 10.764}{U \cdot \Delta TEXOREAC}$$

$AEXOREAC = 5.244 \times 10^3$ ft^2

The material of construction of the shell side and the tube side is carbon steel, thus

$$FM := 0.0 + \left(\frac{AEXOREAC}{100}\right)^{0.0}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure based on the shell side pressure in psig is calculated as follows:

$$PHEX := 1.15 \ \text{bar}$$

$$PHEXShell := PHEX \cdot 14.5 - 14.7$$

$$PHEXShell = 1.975 \ \text{psig}$$

$$FP := 0.9803 + 0.018\left(\frac{PHEXShell}{100}\right) + 0.0017\left(\frac{PHEXShell}{100}\right)^2$$

$$FP = 0.981$$

The heat exchanger is of floating head type:

$$CBEXOREAC := \exp\left[11.667 - 0.8709 \ln(AEXOREAC) + 0.09005 (\ln(AEXOREAC))^2\right]$$

The purchase cost of the endo reactor is:

$$CpEXOREAC := FP \cdot FM \cdot FL \cdot CBEXOREAC \left(\frac{CE2}{CE1}\right)$$

$$CpEXOREAC = 6.152 \times 10^4 \quad \$$$

Pump:

Centrifugal pump is selected because all the requirments in terms of volumetric flow rate, developed head and NPSH are met.

$Q := 7.91 \times 10^{3}$ gpm

$NPSH := 116.28\!\!$ ft

$H := 259.04\!\!$ ft

$S := Q \cdot (H)^{0.5}$

$S = 1.273 \times 10^{5}$ $gpm \cdot ft^{0.5}$

The material of construction is cast iron, thus:

$FM := 1.0$

The type of pump is 1 stage radial centrifugal pump (HSC) with 3600 shaft rpm, thus:

$FT := 1.70$

$$CBPUMP := \exp\left(9.2951 - 0.6019\ln(S) + 0.0519\ln(S)^{2}\right)$$

$$CpPump := FT \cdot FM \cdot CBPUMP \cdot \left(\frac{CE2}{CE1}\right)$$

The cost of electric motor for pump is calculated as follows:

Pump brake horsepower: $PB := 650.08\!\!$ hp

$$\eta m := 0.80 + 0.0319\ln(PB) - 0.00182(\ln(PB))^{2}$$

$$PC := \frac{PB}{\eta m}$$

The motor type of "Totally enclosed, fan-cooled enclosure" with 3600 rpm is selected, thus the type factor for the electric motor is:

$FTmotor := 1.4$

$$CBMOTOR := \exp\left[5.4866 + 0.13141\ln(PC) + 0.053255(\ln(PC))^{2} + 0.028628(\ln(PC))^{3} - 0.0035549(\ln(PC))^{4}\right]$$

$$CpMotor := FTmotor \cdot CBMOTOR \cdot \left(\frac{CE2}{CE1}\right)$$

$$CpPumpTotal := CpPump + CpMotor$$

$CpPumpTotal = 6.267 \times 10^{4}$ \$

A spare pump needs to be added.

Fired Heater:

The cost of the fired heater depends on absorbed heat. Since the operational pressure is not too high and the material of construction is carbon steel, there is no need to apply any correction factor.

This fired heater is designed in such a way to cope with the fluctuations in the winter time. Furthermore, the flue gas heat exchanger and the furnace are combined:

$$QFurnace := 3.554\times 10^{8} \quad \frac{Btu}{hr}$$

$$QFGHX := 1.449\times 10^{8} \quad \frac{Btu}{hr}$$

$$Qduty := QFGHX + QFurnace$$

$$Qduty = 5.003\times 10^{8} \quad \frac{Btu}{hr}$$

Pressure factor: $FP := 1$

Material Factor: $FM := 1$

$$CBFIRHEAT := 0.512\cdot(Qduty)^{0.81}$$

$$CpFIRHEAT := FP\cdot FM\cdot CBFIRHEAT\cdot\left(\frac{CE2}{CE1}\right)$$

$$CpFIRHEAT = 6.421\times 10^{6} \quad \$$$

Heat Exchanger District 1:

Area of the HXDIS1: $AHXDIS1 := 1.676 \times 10^{4}$ ft^{2}

The material of construction of the shell side and the tube side is carbon steel, thus:

$$FM := 0.00 + \left(\frac{AHXDIS1}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS1Shell := 170.983$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS1Shell}{100}\right) + 0.0017\left(\frac{PHXDIS1Shell}{100}\right)^{2}$$

$$FP = 1.016$$

The heat exchanger is of floating head type:

$$CBHXDIS1 := \exp\left[11.667 - 0.8709\ln(AHXDIS1) + 0.09005(\ln(AHXDIS1))^{2}\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS1 := FP \cdot FM \cdot FL \cdot CBHXDIS1\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS1 = 1.571 \times 10^{5}$ \$

Heat Exchanger District 2:

Area of the HXDIS2: $AHXDIS2 := 1.676\times 10^{4}$ ft^{2}

The material of construction of the shell side and the tube side is carbon steel, thus:

$$FM := 0.00 + \left(\frac{AHXDIS2}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS2Shell := 168.113$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS2Shell}{100}\right) + 0.0017\left(\frac{PHXDIS2Shell}{100}\right)^{2}$$

$$FP = 1.015$$

The heat exchanger is of floating head type:

$$CBHXDIS2 := \exp\left[11.667 - 0.8709\ln(AHXDIS2) + 0.09005(\ln(AHXDIS2))^{2}\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS2 := FP\cdot FM\cdot FL\cdot CBHXDIS2\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS2 = 1.57\times 10^{5}$ \$

Heat Exchanger District 3:

Area of the HXDIS3: $AHXDIS3 := 1.676\times 10^{4}$ ft^{2}

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS3}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS3Shell := 127.736$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS3Shell}{100}\right) + 0.0017\left(\frac{PHXDIS3Shell}{100}\right)^{2}$$

$$FP = 1.006$$

The heat exchanger is of floating head type:

$$CBHXDIS3 := \exp\left[11.667 - 0.8709\ln(AHXDIS3) + 0.09005(\ln(AHXDIS3))^{2}\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS3 := FP\cdot FM\cdot FL\cdot CBHXDIS3\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS3 = 1.555\times 10^{5}$ \$

Heat Exchanger District 4:

Area of the HXDIS4: $AHXDIS4 := 1.676\times 10^{4}$ ft^{2}

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS4}{100}\right)^{0.00}$$

$$FM = 1$$

The tube length was assumed to be 12 ft, thus:

$$FL := 1.12$$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS4Shell := 127.101$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS4Shell}{100}\right) + 0.0017\left(\frac{PHXDIS4Shell}{100}\right)^{2}$$

$$FP = 1.006$$

The heat exchanger is of floating head type:

$$CBHXDIS4 := \exp\left[11.667 - 0.8709\ln(AHXDIS4) + 0.09005(\ln(AHXDIS4))^{2}\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS4 := FP\cdot FM\cdot FL\cdot CBHXDIS4\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS4 = 1.555\times 10^{5}$ \$

Heat Exchanger District 5:

Area of the HXDIS5: $AHXDIS5 := 1.676 \times 10^4$ ft^2

The material of construction of the shell side and the tube side is carbon steel thus:

$$FM := 0.00 + \left(\frac{AHXDIS5}{100}\right)^{0.00}$$

$FM = 1$

The tube length was assumed to be 12 ft, thus:

$FL := 1.12$

The pressure factor based on the shell side pressure in psig is calculated as follows:

$PHXDIS5Shell := 127.42$ psig

$$FP := 0.9803 + 0.018\left(\frac{PHXDIS5Shell}{100}\right) + 0.0017\left(\frac{PHXDIS5Shell}{100}\right)^2$$

$FP = 1.006$

The heat exchanger is of floating head type:

$$CBHXDIS5 := \exp\left[11.667 - 0.8709\ln(AHXDIS5) + 0.09005(\ln(AHXDIS5))^2\right]$$

The purchase cost of the heat exchanger is:

$$CpHXDIS5 := FP \cdot FM \cdot FL \cdot CBHXDIS5\left(\frac{CE2}{CE1}\right)$$

$CpHXDIS5 = 1.555 \times 10^5$ \$

Total Purchase Cost:

$SUM1 := CpBlower + CpAIRBlower + CpHEX + CpREB + CpCOND$

$SUM2 := CpDIS + CpFIRHEAT + 2CpPumpTotal + CpENDREAC + CpEXOREAC$

$SUM3 := CpHXDIS1 + CpHXDIS2 + CpHXDIS3 + CpHXDIS4 + CpHXDIS5$

$CpTOTAL := SUM1 + SUM2 + SUM3$

$CpTOTAL = 8.344 \times 10^{6}$ $

References

1- W. D. Seider, J.D. Seader, D.R. Lewin, "Product & Process Design Principles", John Wiley and Sons, Inc., 2004.

2- Economic Indicators (July 2005): WWW.CHE.COM

3- R.K. Sinnott, "Coulson & Richrdson's Chemical Engineering", Vol 6, Butterworth-Heinemann Publisher, Oxford.

Appendix 32 Total Investment of Chemical Heat Pump

Description	Fraction	Amount
Total investment		$ 27,155,950
Fixed capital	1.00	$ 22,510,121
Direct costs	0.79	$ 17,715,466
Onsite costs		$ 14,732,874
Purchased equipment	0.37	$ 8,340,000
Purchased-equipment installation	0.17	$ 3,747,935
Instrumentation and control	0.05	$ 1,069,231
Piping UU	0.04	$ 900,405
Electrical equipment and materials	0.03	$ 675,304
Offsite costs		$ 2,982,591
Buildings	0.03	$ 675,304
Yard improvement	0.03	$ 731,579
Service facilities	0.04	$ 900,405
Land	0.03	$ 675,304
Indirect costs	0.21	$ 4,794,656
Engineering and supervision	0.08	$ 1,800,810
Construction expenses	0.05	$ 1,080,486
Contractor's fee	0.03	$ 562,753
Contingency	0.06	$ 1,350,607
Working capital		$ 1,357,797
Raw material, finished products, accounts receivable, cash on hand, account payable and taxes payable 15% of total capital investment.	0.05	$ 1,357,797
Costs for transport piping *		$ 16,953,225
One time revenues (minus)		-$ 13,665,194
Start-up costs		$ 1,350,607
Process modifications, start-up labor, loss in production	0.06	$ 1,350,607

Prices from (NAP prijzenboekje 20e editie februari 1999)		
The price for total steel underground transport pipelines are (1999)(16inch):	440	€ /m
The material price for under ground steel transport pipes (DN350),		
included average discount of 45%	88	€ /m
Price for laying the pipes:	352	€ /m
Index correction of 4 % per year:	445	€ /m
Prices for the pipes (by Marcel Verboven, Weijers-Waalwijk BV)		
DN350 – 355,6x5,6 / 500 (staal – PUR – PE) Standaard PUR schuim	117.68	€ /m
DN350 – 355,6x5,6 / 500 (staal – PUR – PE) MicroPUR schuim	156.98	€ /m
In the first case we used Standaard PUR, resulting at a length of 21 km in:	11,824,518	€
Including tax	$ 16,953,225	

Appendix 33 Production Cost of Chemical Heat Pump

		Water				Natural Gas				Electricity				Cooling water	
	price	1.28	euro/m^3			0.18	euro/m^3			0.1	euro/kWh			0.0041885	euro/kWh
	BTW	1.06				1.19				1.19				1.19	
	tax	0.146	euro/m^3			0.01	euro/m^3			0.0086	euro/kWh				
	km				reboiler	28,705	MJ/h		comp	374	kW		condenser	8,688	kW
V pipe 16 inch	14.4	1868	m^3			7,932,456	m^3/y		blower air	378	kW				
V pipe 14 inch	6	596	m^3		furnace	120,000	ft^3/h		pump1	53	kW				
						29,766,620	m^3/y								
	total amount/year	2,464	m^3			47,123,846	m^3			7,055,488	kWh			76105128	kWh
	costs/y	€ 3,703				€ 10,565,166				€ 900,280				€ 379,332	
	total costs/y	€ 11,469,149													
		$ 13,691,237													
1m^3gas	31.7	MJ													
exchange course:	0.8377	euro for 1 dollar													

Appendix 34 T-xy Diagram for Acetone-Isopropanol

Binary Systems with
2-Propanol C_3H_8O

45
C_3H_6O

(1) ACETONE C3H6O

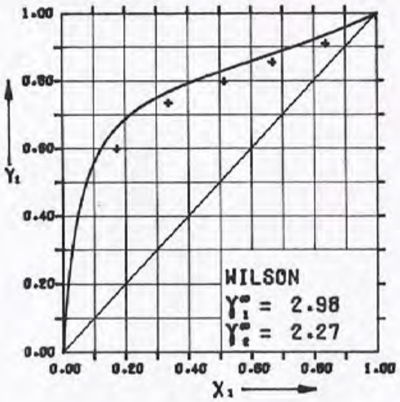
(2) 2-PROPANOL C3H8O

+++++ ANTOINE CONSTANTS REGION +++++ CONSISTENCY
(1) 7.11714 1210.595 229.664 -13- 55 C METHOD 1 -
(2) 8.87829 2010.330 252.636 -26- 83 C METHOD 2 -

TEMPERATURE= 25.00 DEGREE C

LIT: PARKS G.S.,CHAFFEE C.S.
J.PHYS.CHEM.31,439(1927).

CONSTANTS:	A12	A21	ALPHA12
MARGULES	0.9742	0.8496	
VAN LAAR	0.9927	0.8414	
WILSON	548.0277	113.5618	
NRTL	134.7562	464.9095	0.3000
UNIQUAC	-25.5373	282.0923	



EXPERIMENTAL DATA			MARGULES		VAN LAAR		WILSON		NRTL		UNIQUAC	
P MM HG	X1	Y1	DIFF P	DIFF Y1	DIFF P	DIFF Y1	DIFF P	DIFF Y1	DIFF P	DIFF Y1	DIFF P	DIFF Y1
44.30	0.0	0.0	0.00	0.0	0.00	0.0	0.00	0.0	0.00	0.0	0.00	0.0
100.00	0.1750	0.5990	-12.42	-0.0652	-12.60	-0.0653	-13.54	-0.0657	-12.66	-0.0654	-12.78	-0.0655
139.60	0.3390	0.7350	-6.57	-0.0399	-6.28	-0.0388	-5.35	-0.0342	-6.33	-0.0398	-6.19	-0.0382
167.20	0.5140	0.7980	-2.80	-0.0383	-2.40	-0.0375	-1.15	-0.0342	-2.51	-0.0376	-2.25	-0.0370
190.00	0.6690	0.8550	2.20	-0.0265	2.49	-0.0263	3.35	-0.0252	2.35	-0.0263	2.59	-0.0261
221.60	0.8390	0.9100	14.42	-0.0256	14.55	-0.0258	14.92	-0.0263	14.46	-0.0257	14.59	-0.0259
226.50	1.0000	1.0000	0.00	0.0	0.00	0.0	0.00	0.0	0.00	0.0	0.00	0.0
MEAN DEVIATION:			7.68	0.0391	7.66	0.0387	7.66	0.0371	7.66	0.0388	7.68	0.0385
MAX. DEVIATION:			14.42	0.0652	14.55	0.0653	14.92	0.0657	14.46	0.0654	14.59	0.0655

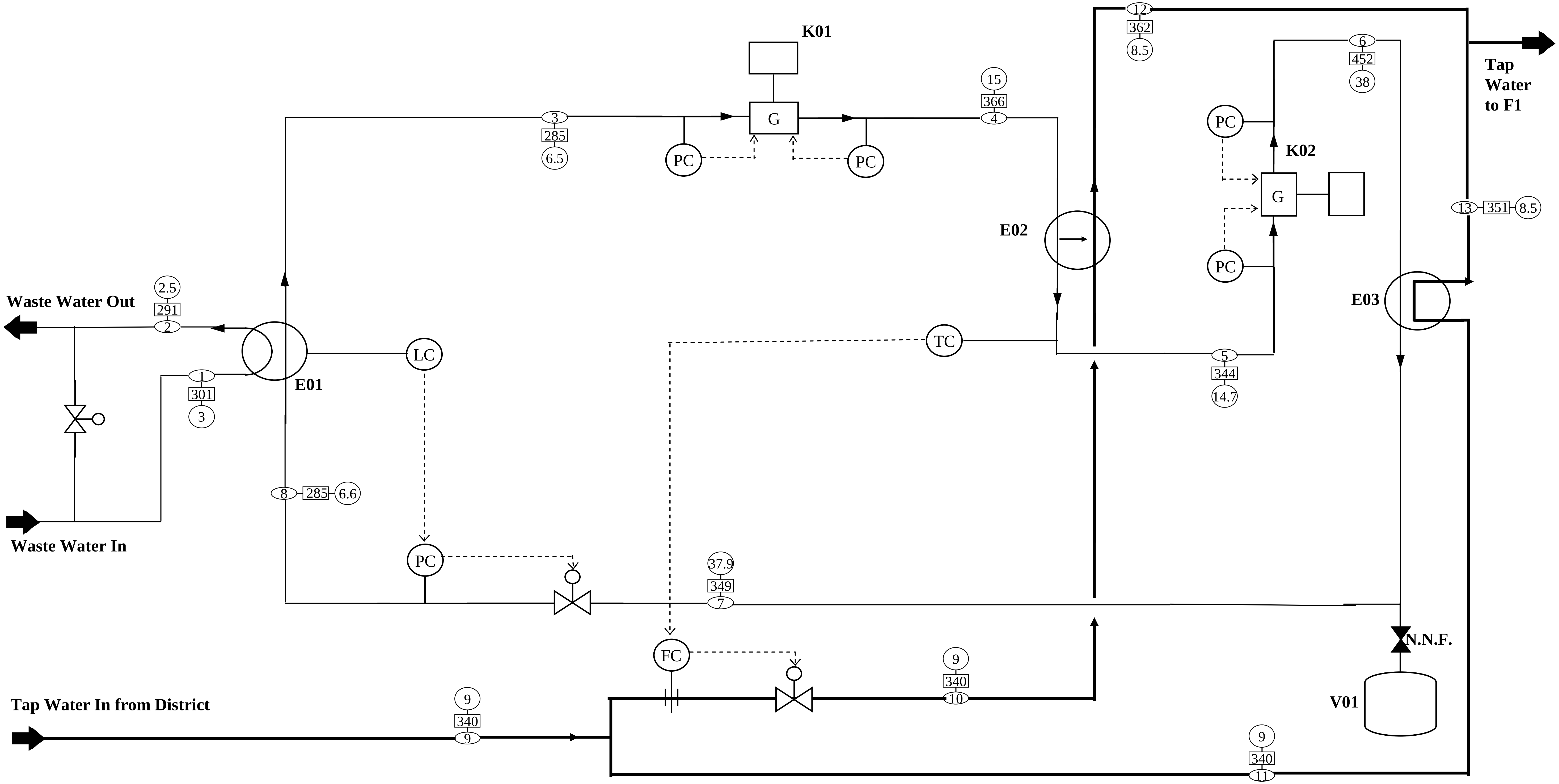
THE FOLLOWING ANTOINE CONSTANT(S) A WERE USED FOR FITTING THE DATA: COMPONENT 1 A= 7.10876
COMPONENT 2 A= 8.88729

Appendix 35 Utility Cost

SUMMARY OF UTILITIES															
EQUIPMENT		UTILITIES													
		Heating					Cooling				Power				REMARKS
Nr.	Name	Load	Consumption (t/h)				Load	Consumption (t/h)			Actual	Consumption (t/h, kWh/h)			
			Steam			Hot		Cooling	Air	Refrig.	Load	Steam (t/h)		Electr.	
		kW	LP	MP	HP	Oil	kW	Water			kW	HP	MP	kWh/h	
K 01	Compressor 1										691			691	
K 02	Compressor 2										946			946	
P 01	Clean water pump										67			67	
K 03	Blower										367			367	
TOTAL		0	0	0	0	0	0	0	0	0	2071	0	0	2071	

Project ID Number :	CPD3328
Completion Date :	3-Dec-05

Appendix 36 Team Photograph



Process Equipment Summary

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

Designers

P.W.C. Falcao

A. Mesbah

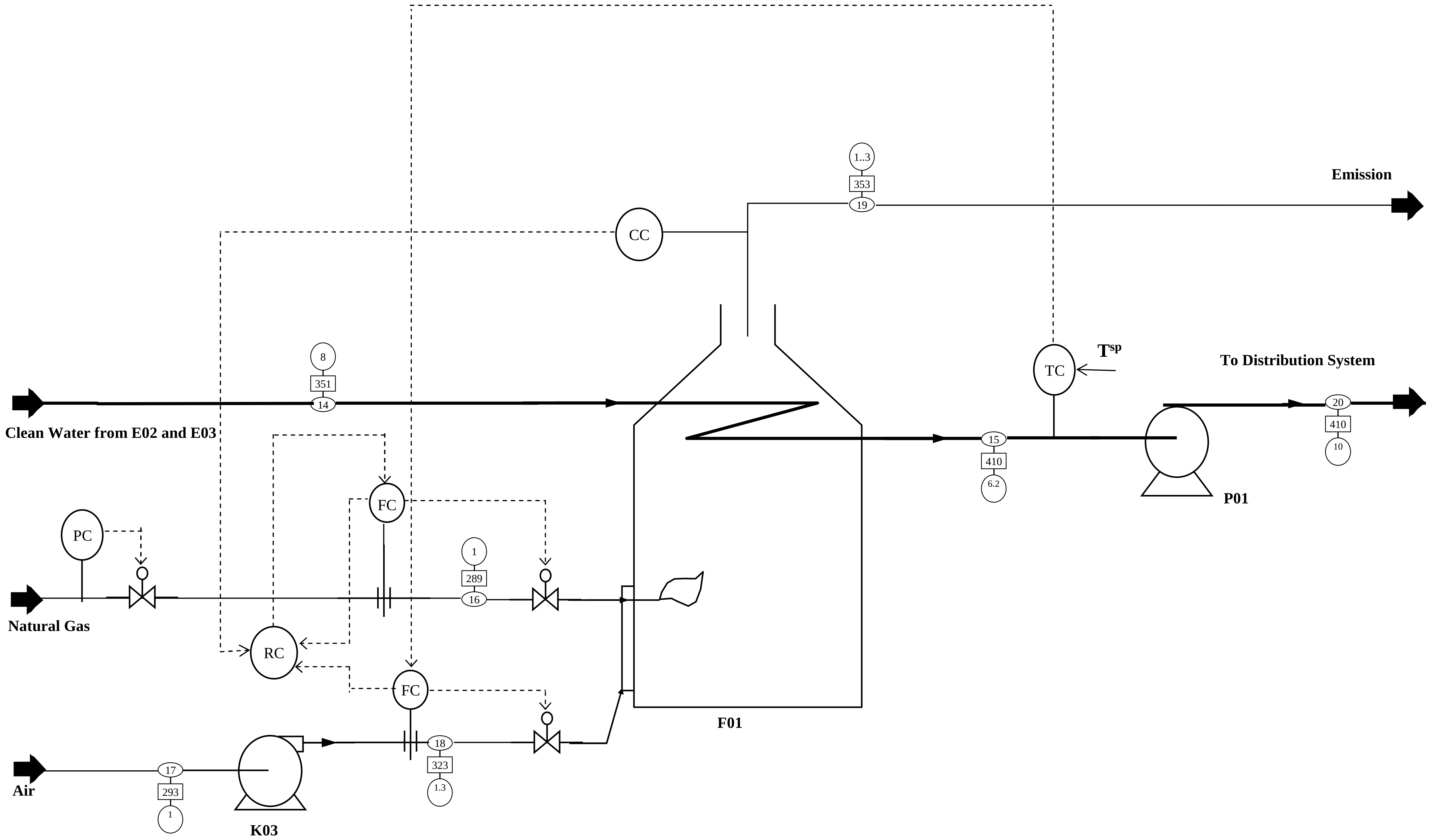
M.V. Suherman

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

P.W.C. Falcao

A. Mesbah

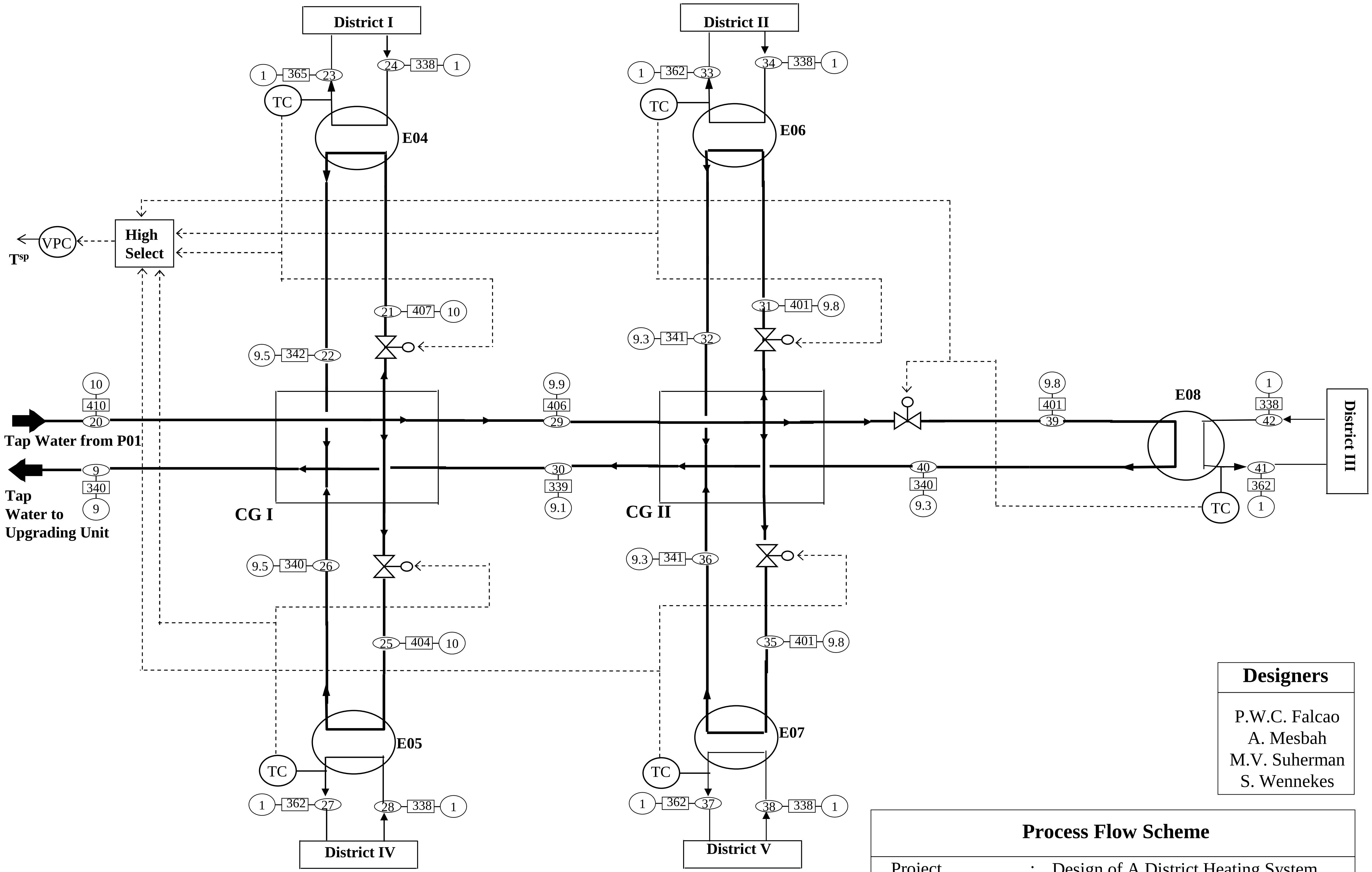
M.V. Suherman

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

P.W.C. Falcao
A. Mesbah
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

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