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Process Technology

Gas-to-gas heat exchanger for heat utilization in hot CO₂ from an electrically heated calcination process



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Summary:

Thesis was done with the objective of evaluating a gas to gas heat exchanger which will be used to recover heat from hot calciner exit gas from an electrically heated calcination process.

Shell and tube heat exchanger (STHE) was selected for design and Inconel 718 was selected as material of construction to handle high temperature. Gas flow was found to be highly dilute in terms of dust concentration, so possible problems associated with dust was assumed to be negligible for design condition.

Study of STHE for 2 different structures (1-2 STHE and 2-4 STHE) along with variation in internal tube diameter and number of STHE in parallel was done. Thermal study of STHE was done by utilizing Kern's method and cost analysis was done using capacity factor method and detailed factor method. Centrifugal radial fan and turbo blower was selected as pressure compensation equipment. Cases with inability to use both equipment was assumed to be technically infeasible. Economic feasibility was studied by calculating NPV. NPV was calculated based on total installed cost and energy savings from STHE. Study of weight, size and footprint of STHE was performed. Sensitivity analysis of NPV with equal percentage variation and more realistic variation of STHE design parameters was also done.

The project was found to be both technically and economically feasible. Heat duty was 7.6 MW for 1-2 STHE and 10 MW for 2-4 STHE. Placing 8 STHE in parallel gave almost negligible energy loss from pressure drop. NPV varied between -167 MNOK and 25.2 MNOK for different test cases. Internal tube diameter of 0.051m gave highest NPVs. Highest NPV for 1-2 STHE structure was 25.2 MNOK and was found by placing 2 STHE in parallel. Highest NPV for 2-4 STHE structure was 24.59 MNOK and was found by placing 3 STHE in parallel. Cost of electricity gave highest sensitivity for real case scenario while inlet temperature of CO₂ gave highest sensitivity for equal percentage variation.

Preface

This master's thesis titled "Gas-to-gas heat exchanger for heat utilization in hot CO₂ from an electrically heated calcination process" was done at University of South-Eastern Norway (USN), Porsgrunn. It was prepared for partial fulfilment of MS in Process Technology.

Picture in first page is taken from website of heat exchanger supplier [1].

I would like to express my sincere gratitude to my supervisor Prof. Lars Andre Tokheim for his interest, suggestions and support throughout the thesis period. I would like to thank Norcem, Brevik to provide me an opportunity with this thesis topic.

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Nomenclature

List of symbols:

SI No	Description	Unit	Symbol
1	Solid loading ratio	--	ϕ
2	Pressure loss factor	--	α
3	Density of shell side fluid in STHE	kg/m ³	ρ_s
4	Density of tube side fluid in STHE	kg/m ³	ρ_t
5	Viscosity of shell side fluid in STHE	Ns/m ²	μ_s
6	Viscosity of tube side fluid in STHE	Ns/m ²	μ_t
7	Efficiency of pressure compensation equipment	--	η_f
8	Efficiency for converting heat to electricity	--	η_t
9	Density of tube material	kg/m ³	ρ_{tm}
10	Density of shell material	kg/m ³	ρ_{sm}
11	Annuity factor	--	a_f
12	Capacity factor	--	c_f
13	Conversion factor from USD to NOK	--	c_{fc}
14	Capacity factor for CRF	--	$c_{f,rf}$
15	Capacity factor for TB	--	$c_{f,tb}$
16	Capacity factor for STHE	--	$c_{f,HE}$
17	Cost index ratio between 2019 to 2002	--	c_{CI}
18	Cost factor for CS equipment from DFT	--	$f_{EQ,CS}$
19	Cost factor for CS piping read from DFT	--	$f_{PI,CS}$
20	Total installed cost factor for required equipment	--	f_{TIC}
21	Total installed cost factor for CS material	--	$f_{TIC,CS}$
22	Equivalent shell diameter of STHE	m	d_e
23	Internal tube diameter of STHE	m	d_i

Nomenclature

24	Internal tube diameter of reference STHE	m	$d_{i,rHE}$
25	External tube diameter of STHE	m	d_o
26	External tube diameter of reference STHE	m	$d_{o,rHE}$
27	Shell side friction factor in STHE	--	f_s
28	Tube side friction factor in STHE	--	f_t
29	Shell side heat transfer coefficient in STHE	W/m ² K	h_s
30	Tube side heat transfer coefficient in STHE	W/m ² K	h_t
31	Rate of interest	--	i
32	Thermal conductivity of STHE material	W/mK	k_m
33	Thermal conductivity shell side fluid in STHE	W/mK	k_s
34	Thermal conductivity tube side fluid in STHE	W/mK	k_t
35	Mass flow rate of cold fluid in STHE	kg/hr	m_c
36	Design basis mass flow rate of cold gas (air)	ton/hr	$m_{c,init}$
37	Material factor used in DFT	--	m_f
38	Mass flow rate of hot fluid in STHE	kg/hr	m_h
39	Design basis mass flow rate of hot gas (CO ₂)	ton/hr	$m_{h,init}$
40	Mass of shell side gas in STHE at a given time	kg	m_{sg}
41	Mass of tube side gas in STHE at a given time	kg	m_{tg}
42	Mass flow rate of shell side fluid in STHE	kg/hr	m_s
43	Mass of shell material in STHE	kg	m_{sm}
44	Mass flow rate of tube side fluid in STHE	kg/hr	m_t
45	Mass of tube material in STHE	kg	m_{tm}
46	Total mass of all STHEs in parallel	ton	m_{HE}
47	Mass flow rate of solid particles in gas	kg/sec	m_{PM}
48	Number of tube passes in STHE	--	n
49	Number of operating hours per year	hr	$n_{hr,yr}$
50	Number of years of operation	yr	n_{yr}

Nomenclature

51	Heat duty in STHE	MW	q
52	Net heat transferred in STHE	MW	q_{net}
53	Equivalent electrical energy of net heat transferred in STHE	MW	$q_{net,el}$
54	Rating of each HE based on each criterion	--	r_i
55	Total ratings of each HE	--	r_t
56	Shell thickness in STHE	m	t_s
57	Shell thickness in reference STHE	m	$t_{s,rHE}$
58	Tube thickness in STHE	m	t_t
59	Tube thickness in reference STHE	m	$t_{t,rHE}$
60	Time spend by shell side fluid inside STHE	m	t_{ts}
61	Time spend by tube side fluid inside STHE	m	t_{tt}
62	Velocity of shell side fluid in STHE	m/sec	v_s
63	Velocity of tube side fluid in STHE	m/sec	v_t
64	Weight factor of each criteria to select HE	--	w_{tf}
65	Weight of each criteria to select HE	--	w_i
66	Overall heat transfer area in STHE	m ²	A
67	Surface area of ends in reference STHE	m ²	$A_{e,rHE}$
68	Overall heat transfer area of reference STHE	m ²	A_{rHE}
69	Shell side cross-sectional area	m ²	A_s
70	Surface area of shell in reference STHE	m ²	$A_{s,rHE}$
71	Surface area of shell type CS sheet	m ²	$A_{sm,CS}$
72	Surface area of shell type IN sheet	m ²	$A_{sm,IN}$
73	Surface area of tube in reference STHE	m ²	$A_{t,rHE}$
74	Surface area of tube type CS sheet	m ²	$A_{tm,CS}$
75	Surface area of tube type IN sheet	m ²	$A_{tm,IN}$
76	Baffle spacing in STHE	m	B

Nomenclature

77	USD cost index in 2002	--	C_{2002}
78	USD cost index in 2019	--	C_{2019}
79	Design basis dust concentration in cold gas (air)	mg/Nm ³	$C_{c,init}$
80	NPV of equivalent electricity during operating years	MNOK	C_{el}
81	Cost of electricity per kWh	NOK/kWh	$C_{el,kwh}$
82	Cost of equivalent electricity per year	MNOK/yr	$C_{el,yr}$
83	Design basis dust concentration in hot gas (CO ₂)	mg/Nm ³	$C_{h,init}$
84	Specific heat capacity of cold fluid	kJ/kgK	$C_{p,c}$
85	Specific heat capacity of hot fluid	kJ/kgK	$C_{p,h}$
86	Cost of reference CRF	USD	$C_{r,crf}$
87	Cost of reference STHE	USD	C_{rHE}
88	Current cost of reference STHE made of CS material	kNOK	$C_{rHE,CS}$
89	Current cost of reference STHE made of IN material	kNOK	$C_{rHE,IN}$
90	Total installed cost of all shell side CRF	MNOK	$C_{s,crf}$
91	Cost of shell side CRF after adjusting time, currency and capacity	kNOK	$C_{s,i,crf}$
92	Cost of each shell type CS sheet	USD	$C_{sm,CS}$
93	Cost of each shell type IN sheet	USD	$C_{sm,IN}$
94	Total installed cost of all tube side CRF	MNOK	$C_{t,crf}$
95	Cost of tube side CRF after adjusting time, currency and capacity	kNOK	$C_{t,i,crf}$
96	Cost of each tube type CS sheet	USD	$C_{tm,CS}$
97	Cost of each tube type IN sheet	USD	$C_{tm,IN}$
98	Fabrication cost	kNOK	C_{FC}
99	Total installed cost of all STHE in series	MNOK	C_{HE}

Nomenclature

100	Cost of STHE made of CS material after adjusting for capacity	kNOK	C_{HECS}
101	Total cost of CS material for STHE construction	kNOK	C_{MCS}
102	Total cost of IN material for STHE construction	kNOK	C_{MIN}
103	Non-discounted cash flow	MNOK	C_{ND}
104	Shell diameter of STHE	m	D_s
105	Shell diameter of reference STHE	m	$D_{s,rHE}$
106	Degree of freedom	--	DOF
107	Equivalent annual cost	MNOK/yr	EAC
108	Equivalent annual cost for capital expenditure	MNOK/yr	EAC_{CAPEX}
109	Equivalent annual cost for operation expenditure	MNOK/yr	EAC_{OPEX}
110	Total footprint of all STHE	m ²	F_{HE}
111	LMTD correction factor	--	F_T
112	Equivalent heat loss by fan on cold fluid to compensate pressure drop	MW	$H_{l,c}$
113	Equivalent heat loss by fan on hot fluid to compensate pressure drop	MW	$H_{l,h}$
114	Equivalent heat loss by fan to compensate total pressure drop in each STHE	MW	$H_{l,t}$
115	Tube length in STHE	m	L
116	Tube length of reference STHE	m	L_{rHE}
117	Molecular mass of cold side fluid	kg/mol	M_c
118	Molecular mass of hot side fluid	kg/mol	M_h
119	Required number of shell type CS sheet for shell end	--	$N_{em,CS}$
120	Required number of end type IN sheet for shell end	--	$N_{em,IN}$
121	Number of relations	--	N_r
122	Required number of shell type CS sheet for shell surface	--	$N_{sm,CS}$

Nomenclature

123	Required number of shell type IN sheet for shell surface	--	$N_{sm,IN}$
124	Number of tubes in STHE	--	N_t
125	Required number of tube type CS sheet for tube surface	--	$N_{tm,CS}$
126	Required number of tube type IN sheet for tube surface	--	$N_{tm,IN}$
127	Number of tubes in reference STHE	--	$N_{t,rHE}$
128	Number of variables	--	N_v
129	Number of STHE in parallel	--	N_{HE}
130	Number of shells in STHE	--	N_S
131	Nusselt's number	--	Nu
132	Net present value	MNOK	NPV
133	Net present value of capital expenditure	MNOK	NPV_{CAPEX}
134	Net present value of operational expenditure	MNOK	NPV_{OPEX}
135	Inlet pressure of cold fluid	bar	$P_{c,in}$
136	Inlet pressure of hot fluid	bar	$P_{h,in}$
137	Pressure drop (tube side or shell side)	bar	ΔP_{eq}
138	Pressure drop of pure gas in dusty flow	bar	ΔP_g
139	Pressure drop along shell in STHE	bar	ΔP_s
140	Pressure drop along tube in STHE	bar	ΔP_t
141	Total pressure drop of system during dusty flow	bar	ΔP_{tot}
142	Pressure in normal pressure temperature condition	Pa	P_{NTP}
143	Prandtl's number of shell side fluid in STHE	--	Pr_s
144	Prandtl's number of tube side fluid in STHE	--	Pr_t
145	Universal gas constant	J/molK	R_G
146	Reynold's number of shell side fluid in STHE	--	Re_s
147	Reynold's number of tube side fluid in STHE	--	Re_t

Nomenclature

148	Tube pitch in STHE	m	S_t
149	Tube pitch in reference STHE	m	$S_{t,rHE}$
150	Cold fluid inlet temperature in STHE	°C	$T_{c,in}$
151	Cold fluid outlet temperature in STHE	°C	$T_{c,out}$
152	Hot fluid inlet temperature in STHE	°C	$T_{h,in}$
153	Hot fluid outlet temperature in STHE	°C	$T_{h,out}$
154	Minimum temperature difference	K	ΔT_{min}
155	Log mean temperature difference	K	ΔT_{LMTD}
156	Temperature in normal pressure temperature condition	K	T_{NTP}
157	Total installed cost of all equipment		TIC
158	Overall heat transfer coefficient in STHE	W/m ² K	U
159	Volumetric flow rate of cold fluid in STHE	m ³ /sec	V_c
160	Volumetric flow rate of hot fluid in STHE	m ³ /sec	V_h
161	Volumetric flow rate of reference CRF	m ³ /sec	V_{rcrf}
162	Volume of shell material in STHE	m ³	V_{sm}
163	Volume of tube material in STHE	m ³	V_{tm}
164	Total volume occupied by all STHE	m ³	V_{HE}
165	Actual work done by fan on cold fluid to compensate pressure drop	MW	W_c
166	Isentropic work done by fan on cold fluid to compensate pressure drop	MW	$W_{is,c}$
167	Isentropic work done by fan on hot fluid to compensate pressure drop	MW	$W_{is,h}$
168	Actual work done by fan on hot fluid to compensate pressure drop	MW	W_h

List of abbreviations:

1-2 STHE	STHE with 1 shell and 2 passes
2-4 STHE	STHE with 2 shells and 4 passes
CS	Carbon Steel
CRF	Centrifugal radial fan
DFT	Detailed factor table
HE	Heat exchanger
IN	Inconel 718
LMTD	Log mean temperature difference
SS	Stainless Steel
STHE	Shell and tube heat exchanger
TB	Turbo blower
TIC	Total installed cost

1 Introduction

“Nature, in providing us with combustibles on all sides, has given us the power to produce, at all times and in all places, heat and the impelling power which is the result of it”

-Nicolas Leonard Sadi Carnot

This chapter discusses about background, goals, objectives and organization of the report.

1.1 Background

Cement industry contributes for about 7% of global CO₂ emission [2]. CO₂ emission in cement production mainly comes from burning of fossil fuel and calcination of limestone. Burning fossil fuel is required to fulfil high energy demand for calcination process. But burning fossil fuel produces other flue gas such as NO_x and SO_x along with CO₂. Hence, the flue gas coming out is a mixture of gases, making it difficult for carbon capture.

If most of the electricity in future is produced from renewable source, cement production using electrically heated calciner could be one of the solutions to this problem. With this method, flue gas generated from fossil fuel can be avoided and more concentrated CO₂ from calcination process will be present in calciner exit gas.

But the gas exits at around 900°C and the high temperature of the exit gas makes it favourable to recover the heat. A flow diagram of this process along with task description is presented in Appendix A.

1.2 Goals and Objectives

With the goal of recovering heat from calciner exit gas, this thesis evaluates a gas to gas heat exchanger to determine its technical and economic feasibility.

To achieve this goal the objectives of this thesis are:

1. Evaluate different types of gas to gas heat exchanger suitable for a gas at high temperature (~900 °C) with the possibility of dust in it
2. Suggest a heat exchanger along with design method based on gas flow rate and temperature
3. Calculate outlet temperatures, overall heat transfer coefficient and heat transfer area
4. Estimate overall size, footprint and weight of the heat exchanger
5. Estimate the cost of the heat exchanger
6. Study the variation of calculated results with respect of variation in independent design variables

These objectives can be achieved by answering the following questions:

1. What are different types of heat exchanger?
2. Which type of heat exchanger is suitable for gas to gas heat exchange?
3. Is gas to gas heat exchanger suitable with dust content in the gas?
4. Is high temperature of CO₂ a problem? What type of material should be chosen for such high temperature?
5. What are the different design parameters of heat exchanger? How will you utilize design method to find different parameters?
6. Is one heat exchanger enough or multiple heat exchanger is required?
7. How much space will heat exchanger take and how much will it weigh?

8. What is the estimated cost of the heat exchanger? Is it economically viable to recover the heat?
9. What are the different parameters that affect the calculated results and how much does it affect the results?

Work breakdown structure to achieve all the objectives is presented in Appendix B.

1.3 Organization of report

This report is divided in 4 main parts. First part consists of chapter 2 and 3 which covers the theory of heat exchangers and cost estimation. Second part consists of chapter 4 and 5 which covers selection of various parameters, methods for calculations and calculation example of heat exchanger design parameters and its estimated cost. Third part consists of chapter 6 which covers results and discussion of calculated parameters. Fourth part consists of chapter 7 which concludes the thesis.

This is designed to cover all the objectives discussed in section 1.2. 1st objective is mainly covered in sections 2.1, 2.2, 2.3, 4.2. 2nd objective is mainly covered in sections 2.2, 2.4, 2.5, 4.2 and 4.5.3. 3rd objective is mainly covered in sections 4.5 and 6.1. 4th objective is also mainly covered in sections 4.5 and 6.1. 5th objective is covered in chapters 3, 5 and section 6.2. 6th objective is covered in section 6.3.

2 Heat Exchanger theory

Heat exchangers (HE) are used in industries to recover heat from process fluids. They are most commonly used in process, power, transportation, air-conditioning, refrigeration and various other industries [3]. Common examples from our day to day life are automotive radiators, condensers and air preheats.

This section starts with general overview of various types of heat exchangers and then discuss various theories required for thermo-hydraulic design of shell and tube heat exchanger (STHE).

2.1 Heat Exchanger classification

Heat exchangers are generally classified based on contacting technique, construction, flow arrangement or surface compactness [4].

Heat exchanger classification based on these principles are shown in Figure 2.1 [4].

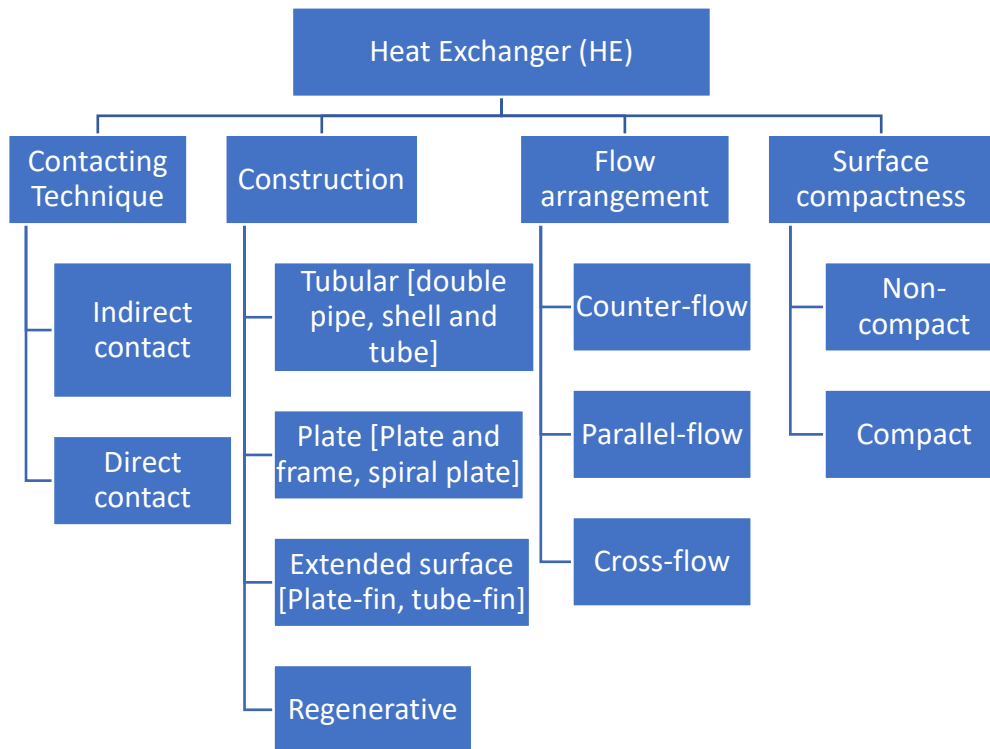


Figure 2.1: Heat exchanger classification

Indirect contact heat exchangers are heat exchangers in which fluid streams remain separate and are not in contact with each other. The heat transfer is attained with the help of separate wall between both fluids or with the help of a matrix. For example: shell and tube heat exchanger.

Direct contact heat exchangers are heat exchangers in which there is a direct contact between fluids for heat transfer. They are not separated by any medium. For example: cooling tower.

Parallel-flow arrangement is shown in Figure 2.2.a. In this arrangement, hot and cold fluid enter at the same end, flow in the same direction and leaves at the same end. This arrangement has lowest efficiency of heat transfer.

Counter-flow arrangement is shown in Figure 2.2.b. In this arrangement, hot and cold fluid enter at the opposite end, flow in the opposite direction and leaves at the opposite end. This arrangement has highest efficiency of heat transfer.

Cross-flow arrangement is shown in Figure 2.2.c. In this arrangement, hot and cold fluid enter at direction perpendicular to each other, flow in perpendicular direction and leaves at direction perpendicular to each. This arrangement has heat transfer efficiency between parallel and counter flow arrangement.

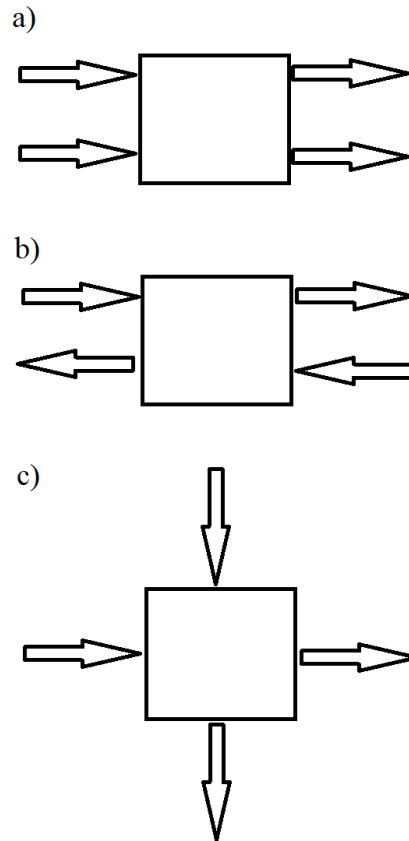


Figure 2.2: Heat exchanger classification based on flow arrangement. a) parallel flow, b) counter flow, c) cross flow

Non-compact heat exchangers have low heat transfer area per volume. In other words, it has low area density. This type of design is mainly suitable for fluid flow with high fouling tendency.

Compact heat exchangers have high heat transfer area per volume or it has high area density. This type of design is mainly suitable for fluids with low heat transfer coefficient.

Tubular heat exchangers are the most common type of heat exchanger available today. Heat exchangers with heat transfer between tubes are generally called tubular heat exchangers. Double pipe heat exchangers and shell and tube heat exchangers are the most common type of tubular heat exchangers.

Double pipe heat exchangers are made of two concentric pipes and usually have a U-bend. Flow arrangement for them can be either counter or parallel flow depending on application. Temperature cross (condition when outlet temperature of cold stream is higher than outlet temperature of hot stream) can occur for this type of heat exchanger when flow arrangement is pure counter-current. They can also be connected in series or in parallel. They are usually suitable for high pressure and high temperature applications where heat duty is low. They are

most commonly used for high-fouling services such as slurries with small heat duties. Schematic of this unit is shown in Figure 2.3.

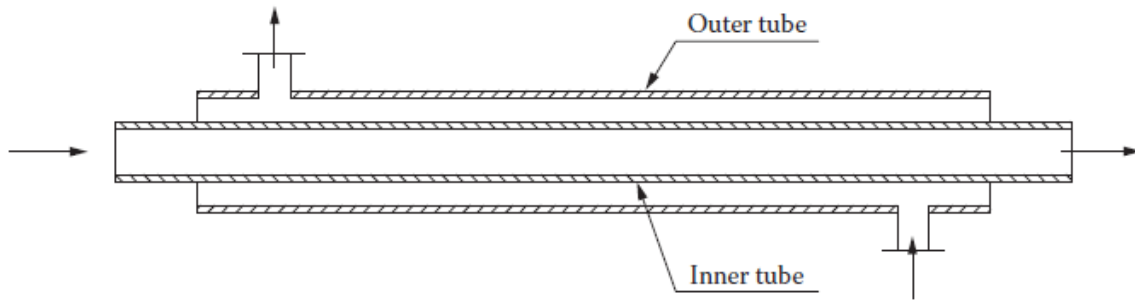


Figure 2.3: Schematic of double pipe heat exchanger [3]

Plate heat exchangers are less widely used compared to tubular heat exchangers. They usually comprise of a stack of metal plates in mutual contact. Each plate has an inlet and outlet section. Hot and cold fluids flow in adjoining plates. They have high surface area to volume ratio. When temperature and pressure of fluid is low, they are usually selected. They have a high temperature approach and heat recovery can be as high as 93% [3]. A diagram with constructional details is shown in Figure 2.4.

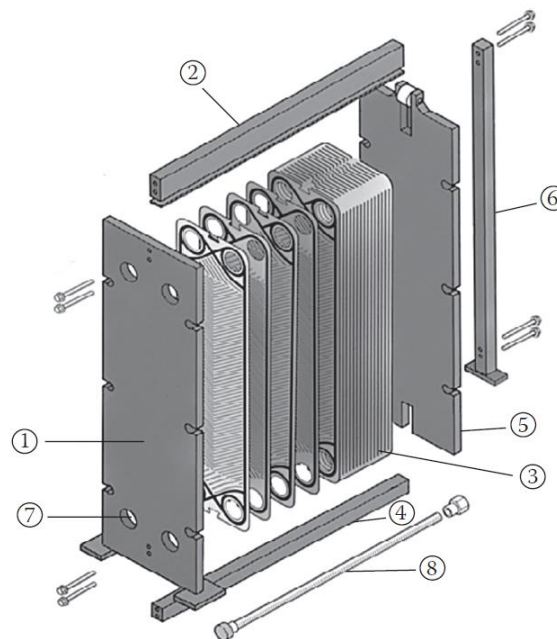


Figure 2.4: Construction of plate heat exchangers. 1: Fixed frame plate; 2: Top carrying bar; 3: Plate pack; 4: Bottom carrying bar; 5: Movable pressure plate; 6: Support column; 7: Fluid port; 8: Tightening bolt [3]

Extended surface heat exchangers are usually used when heat coefficients are quite low, and a large heat transfer area is required. Large heat transfer areas are achieved with the help of fins attached to the primary surface. They are quite useful when one of the fluids for heat transfer is gas. Tube fin heat exchanger and plate fin heat exchangers are the most common type of extended surface heat exchanger.

Tube fin heat exchanger is usually selected when extended surface is needed only in one fluid side. But their usage is limited by operating pressure and operating temperature. Most common type of tube fin heat exchanger is radiators. An example of tube fin heat exchanger is shown in Figure 2.5.

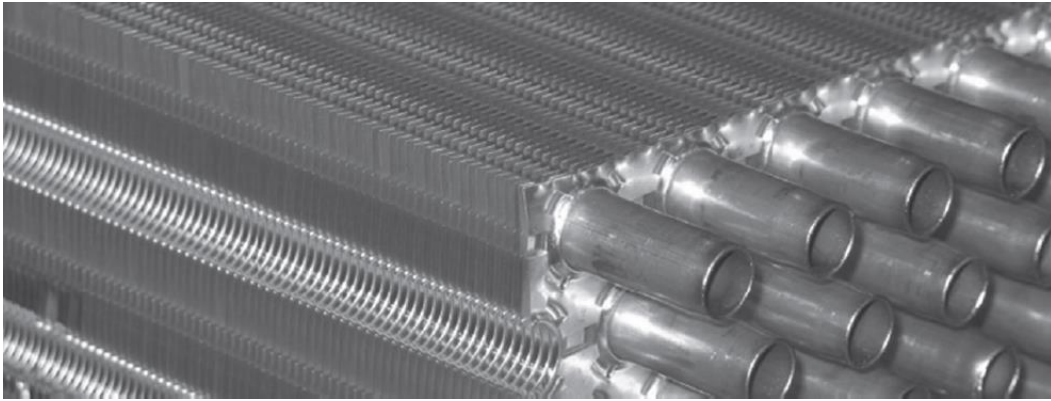


Figure 2.5: Continuous finned compact cooler [3]

Plate fin heat exchangers are commonly used in gas-gas heat exchange application. ALPEMA standard is the most important source for specifying design standards. They consist of stacks of alternate flat plates called “parting sheets” with fin corrugation. Based on requirement, they can either have cross-flow arrangement or counter-flow arrangement. But its usage is limited by operating pressure and operating temperature. Its basic elements and different types of flow arrangement are shown in Figure 2.6.

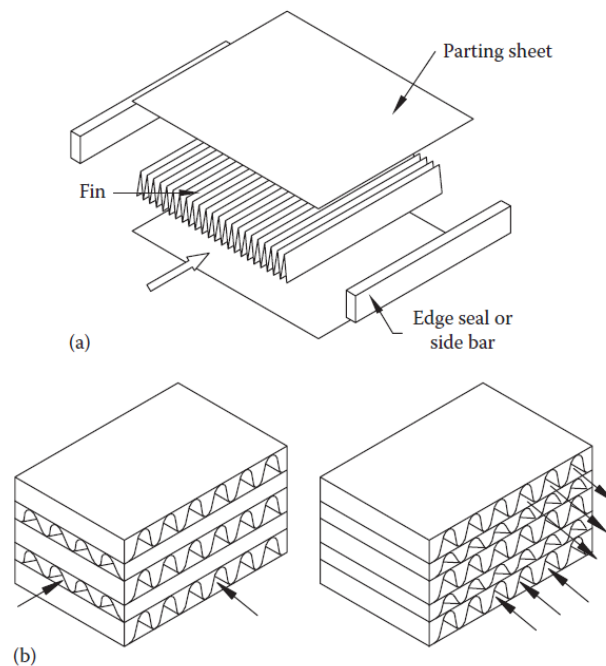


Figure 2.6: Plate fin heat exchanger. (a) basic elements, (b) two types of flow arrangements [3]

Regenerators are exclusively used for gas to gas application [5]. They are designed for low pressure and high temperature application (around atmospheric pressure). It consists of a metal matrix which acts as a surface for heat transfer. First, thermal energy from hot fluid is passed to metal matrix by making fluid flow over metal matrix. Then, thermal energy from matrix is transferred to cold fluid by switching the flow of fluid from hot to cold. The two fluids may flow either in parallel or in counter flow condition. It is designed for high temperature application and has self-cleaning characteristic. Change in flow cycle (when fluid flow is changed from hot to cold or vice versa) during operation gives a self-cleaning characteristic but this also results in mixing of fluids. The main advantage of regenerator compared to plate fin heat exchanger is that it has higher heat transfer surface per unit volume

and lower cost. But the major disadvantage is the problems associated with mixing of fluids. Rotatory and fixed matrix are 2 main types of regenerators.

Rotatory regenerators can be used for temperature range from 870 to 1100°C. But for higher temperature (upto 2000 °C) a fixed-matrix regenerator is used. Rotatory regenerators consist of a revolving solid matrix. With this, the flow of fluid in metal matrix keeps on changing as the matrix is rotated. But fixed matrix regenerators have no moving parts. Here, the flow of fluid is changed with the help of valve arrangement. A schematic diagram of rotatory regenerators is shown in Figure 2.7 and diagram of fixed matrix regenerators is shown in Figure 2.8.

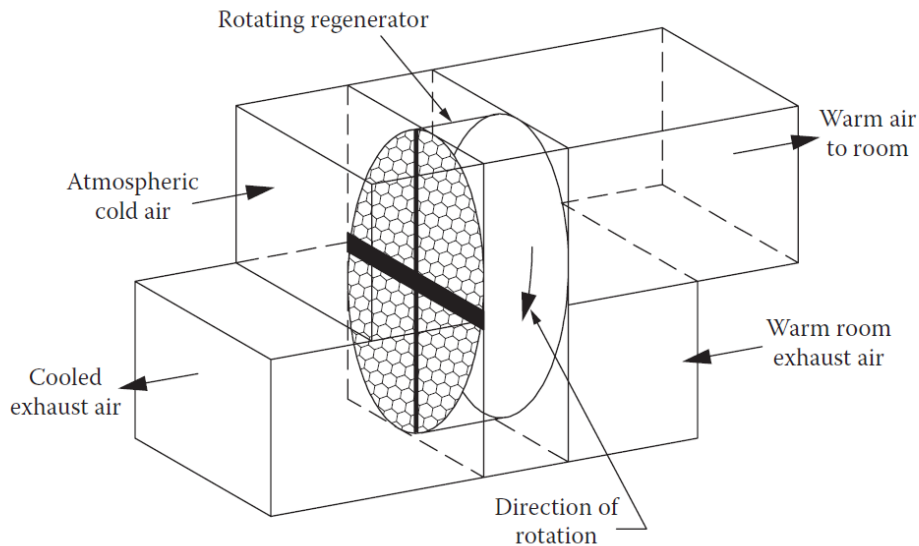


Figure 2.7: Schematic diagram of rotatory regenerator [3]

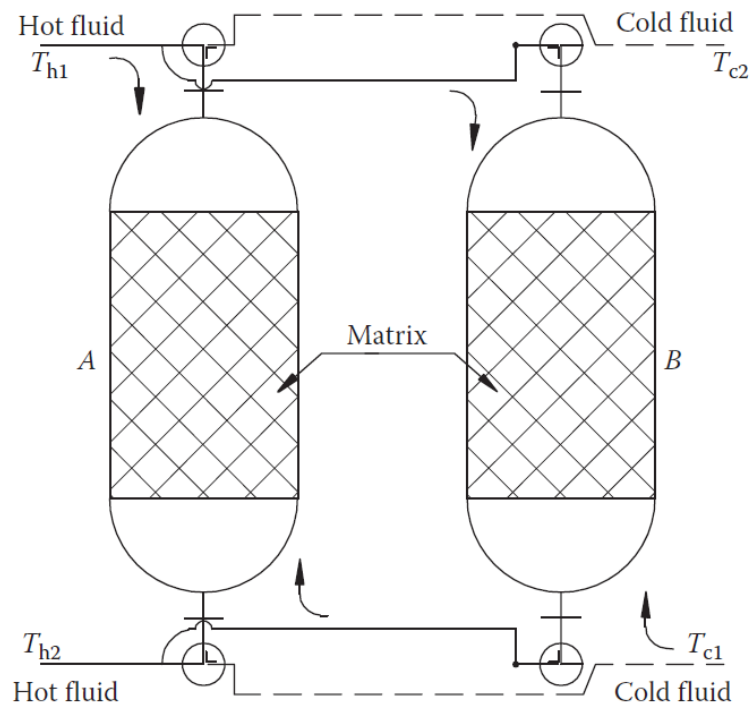


Figure 2.8: Schematic diagram of fixed matrix regenerator [3]

2.2 Selection of heat exchanger

Heat exchanger selection depends on the type of application. Choice of heat exchanger for intended application is shown in Table 2.1 [3].

Table 2.1: Choice of heat exchanger for intended application [3].

Choice of Heat Exchanger Type for Intended Applications

Application	Remarks
Low-viscosity fluids	For high temperature/pressures, use STHE or double-pipe heat exchanger. Use PHE or LHE for low temperature/pressure applications.
Low-viscosity liquid to steam	Use STHE in carbon steel.
Medium-viscosity fluids	Use PHE or with high solids content, use SPHE.
High-viscosity fluids	PHE offers the advantages of good flow distribution. For extreme viscosities, the SPHE is preferred.
Fouling liquids	Use STHE with removable tube bundle. SPHE or PHE is preferred due to good flow distribution. Use PHE if easy access is of importance. Also consider Philips RODbaffle heat exchanger, <i>TWISTED TUBE® heat exchanger</i> and Helixchanger® heat exchanger, and EMBaffle® heat exchanger to improve flow velocity on the shellside, enhance heat transfer performance, and reduce fouling tendencies on shellside.
Slurries, suspensions, and pulps	SPHE offers the best characteristics. Also consider free flow PHE or wide gap PHE, or scraped surface heat exchanger.
Heat-sensitive liquids	PHE fulfills the requirements best. Also consider SPHE.
Cooling with air	Extended surface types like tube-fin heat exchanger or PFHE.
Gas or air under pressure	Use STHE with extended surface on the gas side or brazed plate-fin exchanger made of stainless steel or nickel alloys.
Cryogenic applications	Brazed aluminum plate-fin exchanger, coiled tube heat exchangers, or PCHE.
Vapor condensation	Surface condensers of STHE in carbon steel are preferred. Also consider SPHE or brazed plate heat exchanger.
Vapor/gas partial condensation	Choose SPHE.
Refrigeration and air conditioning applications	Finned tube heat exchangers, special types of PHEs, brazed PHE up to 200°C.
Air-air or gas-gas applications	Regenerators and plate-fin heat exchangers. Also consider STHE.
Viscous products, aseptic products, jam, food and meat processing, heat sensitive products and particulate laden products	Scraped surface heat exchanger.

Note: STHE, shell and tube heat exchanger; PHE, gasketed plate heat exchanger; SPHE, spiral plate heat exchanger; LHE, lamella heat exchanger; PCHE, printed circuit heat exchangers; CTHE, coiled tube heat exchanger; PFHE, plate-fin heat exchanger.

Heat exchangers suitable for gas to gas heat exchange are regenerators, plate-fin heat exchangers and shell and tube heat exchanger (STHE) according to Table 2.1.

Since STHE has a well-established procedure for design, many years of satisfactory service and availability of codes and standards for design and fabrication, it was the first choice for design.

2.3 Impact of dust

Potential impacts of dusty gas flow in heat exchanger are as follows:

1. Reduced efficiency of heat transfer surface due to fouling
2. Blockage of flow
3. Pressure drop increase in the equipment
4. Erosion of equipment by solid particles

These problems mainly depend on the type of flow of dusty gas in the equipment. Type of flow in a horizontal pipe is generally categorized into [6]:

1. Flow of solid particles in uniformly dispersed phase (dilute phase flow)
2. Solid material occupies lower part of horizontal pipe during flow
3. Density of dusty gas approaches the bulk density of solid particles (dense phase flow)

If the flow is of type 2 or type 3, then the chances of deposit of solid particles is higher. In this case, potential impact of dusty gas flow is applicable. But if the flow is of type 1, then the chances of deposit is lower. Hence, potential impact due to dusty gas can be neglected in extreme case of type 1 flow.

The type of flow described above and their transition from one type to another mainly depends on three factors. They are:

1. Velocity of carrier gas:

When the velocity of the gas is reduced, the flow generally moves from dilute to dense phase flow. Minimum gas velocity at which flow of solid particles occurs is termed as minimum transport velocity. Since this depends on many parameters, it is difficult to predict theoretically and is usually found from experiments. It's usually between 5-10 m/sec for a dense phase flow [6].

2. Solid loading ratio (ϕ):

It is the ratio between mass flow rate of solids to mass flow rate of gas. When the solid loading ratio is increased flow moves from dilute to dense phase flow due to increased amount of solid. Actual solid loading ratio of three categories is generally difficult to predict since it is dependent on many parameters. But typically solid loading ratio of dilute phase flow is less than 5 and dense phase flow can go from about 25 to up to several hundred [6].

3. Nature of particulate material:

Nature of particulate material also plays a role in deciding the category of flow. If the size of solid material increases, then it moves from dilute to dense phase flow.

Pressure drop due to solid particles in gas:

Presence of dust particles in gas increases the pressure drop of the system compared to pure gas flow. Total pressure drop (ΔP_{tot}) arising from dusty gas flow can be shown by equation (2.1).

$$\Delta P_{tot} = \Delta P_g(1 + \alpha) \quad (2.1)$$

Here, α is pressure loss factor and ΔP_g pressure drop in pure gas flow. As a rule of thumb, pressure loss factor (α) can be assumed to be equal to solid loading ratio (ϕ).

Plot of dependence of pressure loss factor on gas velocity (or conveying velocity) and solid loading ratio is given by Figure 2.9. Figure shows that at higher velocity, pressure loss factor is equivalent to solid loading ratio.

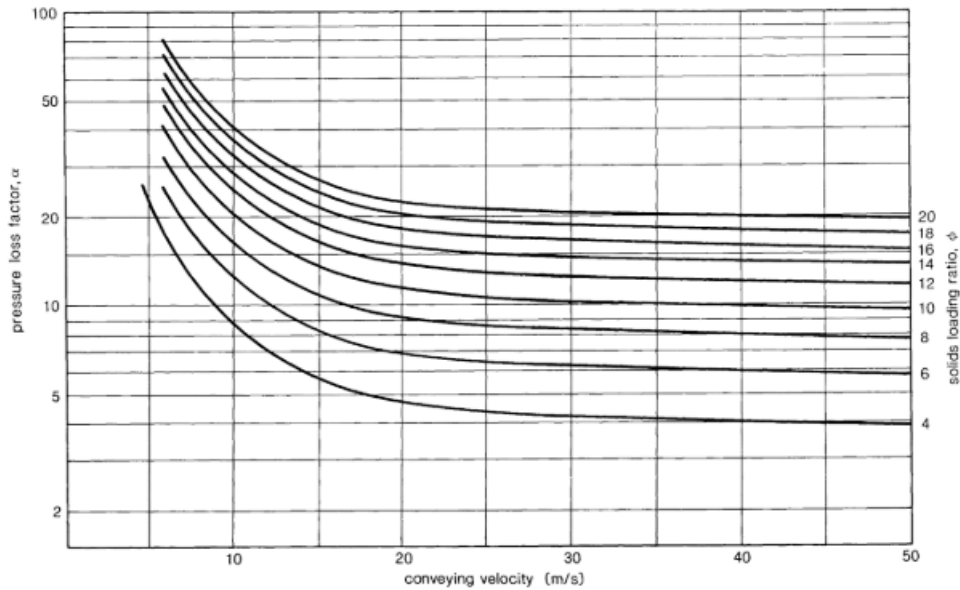


Figure 2.9: Plot to show dependence of pressure loss factor on conveying velocity and solid loading ratio [6]

Erosion of equipment by solid particles:

Erosive wear of equipment can occur if the solid particles present in the gas stream is highly abrasive. This phenomenon is higher during dense flow and high gas velocity.

Abrasiveness of a solid particles be estimated from its hardness. Abrasiveness is higher for particles with higher value of hardness.

A semi-quantitative scale of hardness was proposed by F.Mohs who selected ten materials beginning with the softest (talc) and ending with hardest (diamond). He rated each material from 1 to 10 where 1 is softest and 10 is hardest. Other material in terms of Mohs hardness is shown in Figure 2.10.

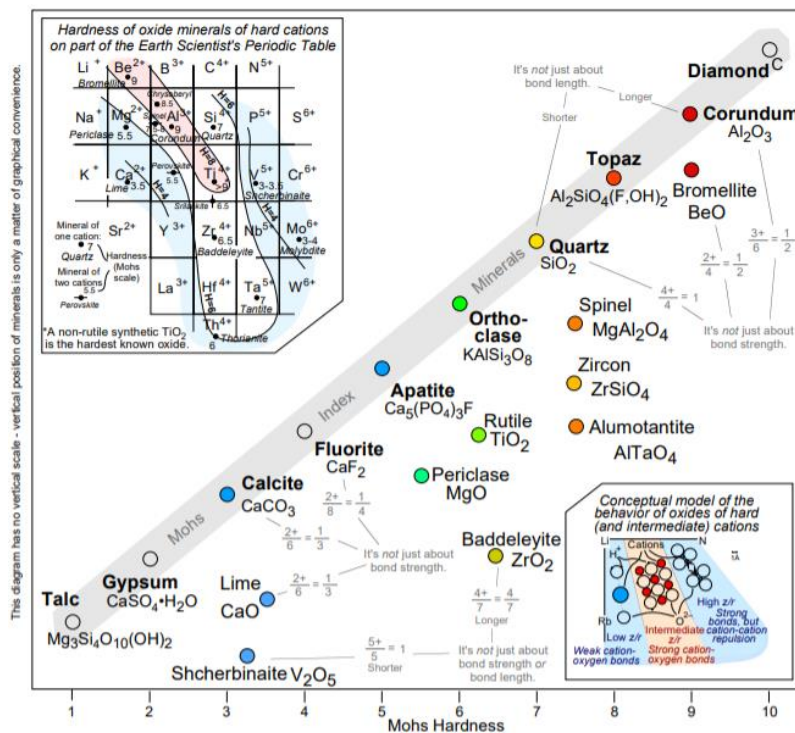


Figure 2.10: Mohs hardness of minerals: variation among oxides and oxyalts [7]

2.4 Material selection criteria

Selection of material for heat exchanger in simple terms, requires consideration of the following factors [3]:

1. Compatibility of material with process fluids
2. Compatibility of material with other component materials
3. Ease of manufacture and fabrication
4. Material strength and ability to withstand pressure and temperature conditions
5. Cost
6. Availability

Discussion on the material selection criteria can be wide. The scope of its discussion for this thesis is limited to high temperature application.

High temperature heat exchanger generally requires costly materials. This results in cost of heat exchanger increasing significantly with temperature range above 675°C. [8]

At elevated temperature, yield strength and creep resistance decreases which can result in material failure. These properties are closely related to the ductility of the material and can be represented with the help of toughness parameter.

Charpy V-Notch test is the most common test to check for material toughness. Using this test for a material, typical relationship between hardness and the temperature is given by Figure 2.11 [9].

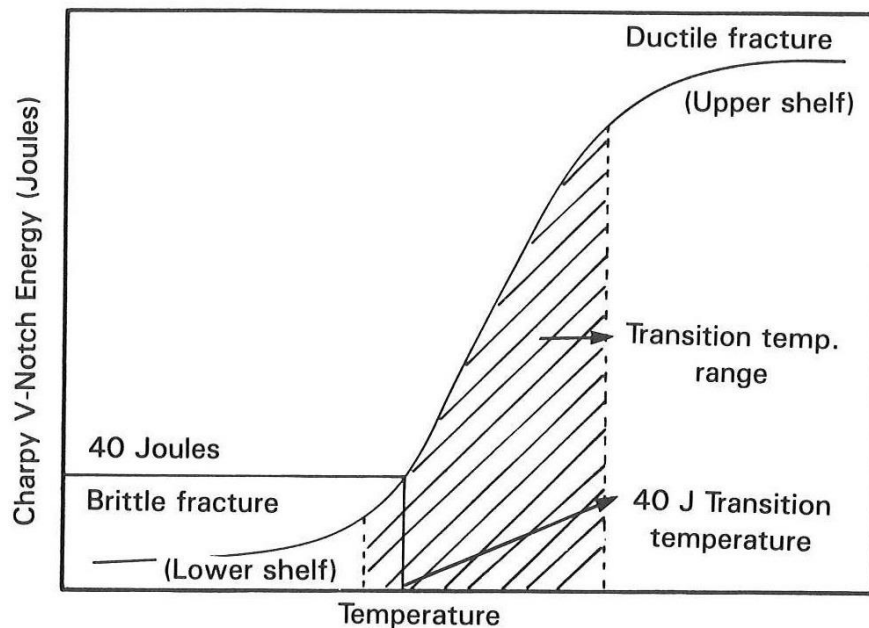


Figure 2.11: Typical relationship between toughness (measured by the Charpy V-notch test) and temperature [9]

Hence, Figure 2.11 shows that low temperature operation can result in brittle fracture and high temperature operation can result in ductile fracture. Selection of the material should be made such that Charpy V-Notch Energy lies in the transition temperature to avoid failure.

Material suitable for high temperature application:

Enhancing yield strength and creep resistance for steel at elevated temperature can be done by addition of [9]:

- Molybdenum

- Chromium
- Vanadium

Higher temperature can result in higher oxidation rate. Chromium is added to enhance oxidation resistance. It also improves strength at elevated temperature. Addition of 0.5% Chromium and 0.5% Molybdenum can work up to 540°C. Addition of 2.25% Chromium and 1% Molybdenum can work up to 650°C [9].

Nickel is added to ferric alloy steel to improve strength, toughness and ductility of the steel [10].

2.5 Shell and tube heat exchanger

Shell and tube heat exchangers (STHE) are the most common type of heat exchanger used for various industrial applications. More than 90% of heat exchangers in industries are of this type [3].

Tubular Exchangers Manufacturers Associations (TEMA) has devised standards covering details of material, methods of construction, technique of design and dimensions for exchangers[11]. These standards are widely adopted in process industries.

2.5.1 Structure

If several smaller tubes inserted in one big tube, the weight of material required for construction becomes so large that it requires a special design for construction. This special design is known as shell and tube heat exchanger. Big tube act as a shell and several smaller tubes act as tubes of a shell and tube heat exchanger. Fluids for heat transfer flows either in the shell or in the tubes. The basis of fluid allocation is discussed in Section 2.5.2.

Baffles are added in the shell side to improve heat transfer coefficient by raising the average velocity of shell side fluid. It also acts as a support for the tubes. But the main support to the tubes are provided by tube sheets. Simplest form of shell and tube heat exchanger design is shown in Figure 2.12.

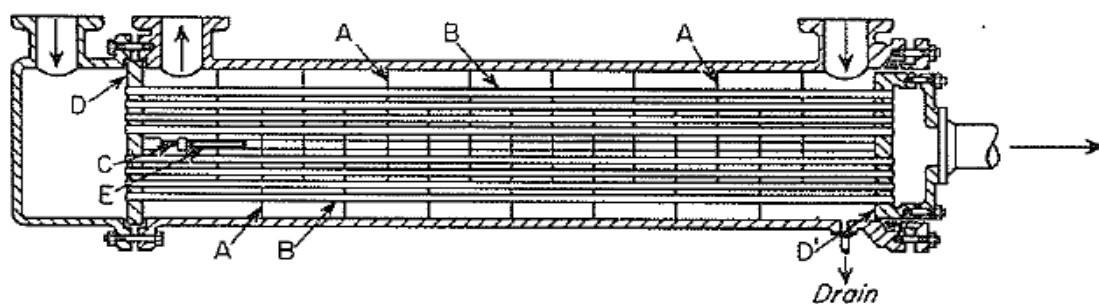


Figure 2.12: Single pass 1-1 heat exchanger. A: Baffles; B: Tubes; C: Guide rods; D, D': Tube sheets; E: Baffle spacers [11].

Single pass 1-1 heat exchanger is the simplest form of shell and tube heat exchanger and is shown in Figure 2.12. It has a single shell and tube pass. This arrangement can sometimes have low velocity and low velocity corresponds to reduction in heat transfer coefficient.

One way to improve velocity could be to reduce number of tubes. Reducing number of tubes may result in increase in tube length. But sometimes the required increased length could be so high that the design becomes impractical.

Multiple pass heat exchanger was developed to tackle this problem. Here the tubes have multiple passes. Using this design, number of tubes can be reduced, and the length can be increased as the tubes pass through the shell more than once. 1-2 shell and tube heat exchanger (1-2 STHE) is the simplest example of multiple pass heat exchanger and is shown in Figure 2.13.

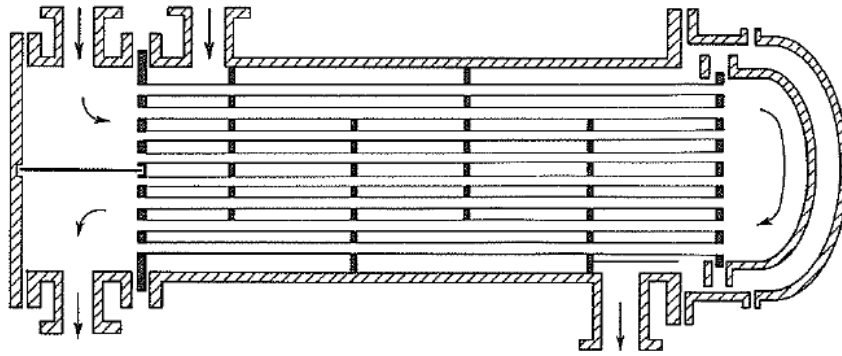


Figure 2.13: 1-2 shell and tube heat exchanger (1-2 STHE) [11]

1-2 STHE have 1 shell and 2 passes. This design makes it practical to increase the velocity of tube side fluid. But this comes at some disadvantages such as:

1. Construction becomes slightly more complicated.
2. Parallel flow in some section of STHE which can result in reduced efficiency of heat transfer area.
3. Additional pressure drop due to addition of turns.

2-4 shell and tube heat exchanger (2-4 STHE) was proposed to overcome reduction of efficiency of heat transfer area due to parallel flow. It has 2 shells and 4 passes. Addition of longitudinal baffle in 1-2 STHE along with 2 extra passes of tubes will accomplish this design.

With this design, fluids flowing in counter flow direction improves compared to 1-2 STHE. Hence, efficiency of heat transfer area is also improved. But this comes at a disadvantage of increased pressure drop and a more complicated design. Hence, according to design basis a suitable design is chosen.

A typical 2-4 STHE is shown in Figure 2.14.

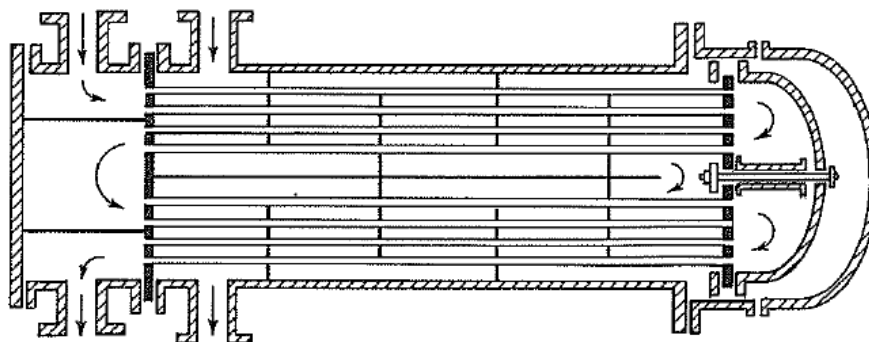


Figure 2.14: 2-4 shell and tube heat exchanger (2-4 STHE) [11]

A comparison of 1-2 STHE and 2-4 STHE is shown in Table 2.2.

Table 2.2: Comparison of 1-2 heat exchanger and 2-4 heat exchanger

1-2 shell and tube heat exchanger	2-4 shell and tube heat exchanger
Lower pressure drop	Higher pressure drop
Less complicated design	More complicated design
Lower velocity	Higher velocity
Lower efficiency of heat transfer area	Higher efficiency of heat transfer area

Series arrangement of two 1-2 STHE gives same performance as 2-4 STHE [12]. This arrangement increases the efficiency of heat transfer area which in turn improves duty of STHE. But this comes at an expense of higher pressure drop due to increased velocity and increased number of turns for the fluid flow.

Parallel arrangement of two 1-2 STHE helps with this problem by controlling the pressure drop in system but at an expense of higher capital cost. Reduction in pressure drop is mainly due to reduction in velocity at lower mass flow rates and higher capital cost is due to requirement of an extra bigger STHE.

2.5.2 Fluid flow allocation basis

Fluids can either flow in the shell or in the tube. Correct placement of fluids is important to reduce the overall cost of STHE. General placement guidelines are as follows [13]:

1. Tube side
 - a. High temperature fluid requiring alloy materials
 - b. High pressure fluids
 - c. Fluids for which pressure drop is most critical
 - d. Toxic and lethal fluids to minimize leakage
 - e. Corrosive fluids
 - f. Fouling fluids (as increased velocity in tube flow minimized fouling but can enhance erosion)
2. Shell side
 - a. Viscous fluids to improve overall heat transfer coefficient
 - b. Fluids with lower flow rate
 - c. Condensing or boiling fluids

2.5.3 Design fundamentals

Designing a shell and tube heat exchanger mainly comes under 2 categories. They are thermo-hydraulic design and mechanical design. This thesis will mainly focus on thermo-hydraulic design.

Design for better thermal performance is done through thermo-hydraulic design and design for better mechanical performance is done through mechanical design. Thermal performance is checked by estimating overall heat transfer coefficient, heat duty, heat transfer area and pressure drop. To start thermal analysis some mechanical parameters such as tube diameter, shell diameter, baffle spacing may be required. Exact mechanical parameter required depends on the design basis and the procedure used for thermo-hydraulic design.

Mechanical parameters and calculation methods are nowadays highly standardized and most of it can be found from TEMA standards.

Kern's method is one of the most widely used methods for thermo-hydraulic design of STHE. It specifies correlations to estimate shell side and tubes side heat transfer coefficients and pressure drops.

2.5.4 Sizing parameters

Shell and tube heat exchanger (STHE) is comprised of many different parts. Designing of these parts fall under mechanical design. Currently it is highly standardized and appropriate tables for selection of sizing parameters and their correlations can be found in TEMA handbook. This type of design is out of the scope of this thesis. But to estimate thermal performance of STHE, rough values of some parts is required.

Internal diameter of tube (d_i) is the most basic design parameter. As the name suggests, it gives internal diameter of tubes present inside STHE. Selection of appropriate diameter is important to maximize thermal efficiency. Small tube diameter generally gives higher transfer of heat. This is mainly because of two reason:

1. Contact surface between flowing fluid and tube surface increases
2. At lower diameter, the fluid has higher velocity. This increases heat transfer coefficient.

But this can also result in higher pressure drop. So, this parameter must be optimized to maximize heat transfer with a check in pressure drop.

Shell diameter (D_s) is the next parameter to be decided once tube diameter has been determined. Shell diameter gives the internal diameter of shell, inside which tubes are placed. Higher the shell diameter, higher will be capacity of STHE as it is directly related to size of STHE. If the ratio between shell diameter and tube diameter is too high, practical problems like inability for mechanical cleaning may arise. But if the ratio is too small less tubes are placed inside and heat transfer duty is reduced. Hence it is highly standardized. Some of the combination of tube diameter and shell diameter is shown in Figure 2.15.

$D_s \backslash D_t$	100	200	300	500	700	1000	1500
6							
10							
14							
20							
25							
38							
51							

Not Acceptable for Mechanical Cleaning

↑

Figure 2.15: Recommended combination of shell and tube diameters. Shaded regions are recommended region for combination. All dimensions in milli-meters [9]

From Figure 2.15, rough relation between shell diameter and tube diameter is shown in equation (2.2).

$$D_s = 40 \times d_i \quad (2.2)$$

Baffle spacing (B) is also an important parameter to be decided once shell diameter is decided. It is the distance between adjacent baffles inside the shell. It is an important parameter as it contributes to the turbulence of the fluid inside the shell. Reducing baffle spacing increases shell side heat transfer coefficient but it also increases pressure drop. Hence baffle spacing needs to be optimised. It was recommended to use baffle spacing between shell diameter value and $1/5^{\text{th}}$ of shell diameter value [12]. For simplicity, baffle spacing was assumed to be equal to shell diameter. Hence, it can be given by equation (2.3).

$$B = D_s \quad (2.3)$$

Tube thickness (t_t) and **shell thickness (t_s)** could be found out using Table 2.3 and Table 2.4 once tube diameter and shell diameter is determined. Thickness doesn't have significant contribution in overall heat transfer coefficient as it is usually low, and conductivity of metal is much higher than fluids in most cases.

Table 2.3: Correlation between tube diameter and tube wall thickness [12]

Tube OD, in.	BWG	Wall thickness, in.	ID, in.	Flow area per tube, in. ²	Surface per lin ft, ft ²		Weight per lin ft, lb steel
					Outside	Inside	
½	12	0.109	0.282	0.0625	0.1309	0.0748	0.493
	14	0.083	0.334	0.0876		0.0874	0.403
	16	0.065	0.370	0.1076		0.0969	0.329
	18	0.049	0.402	0.127		0.1052	0.258
¾	20	0.035	0.430	0.145	0.1963	0.1125	0.190
	10	0.134	0.482	0.182		0.1263	0.965
	11	0.120	0.510	0.204		0.1335	0.884
	12	0.109	0.532	0.223		0.1393	0.817
	13	0.095	0.560	0.247		0.1466	0.727
	14	0.083	0.584	0.268		0.1529	0.647
	15	0.072	0.606	0.289		0.1587	0.571
	16	0.065	0.620	0.302		0.1623	0.520
1	17	0.058	0.634	0.314	0.1660	0.469	
	18	0.049	0.652	0.334	0.1707	0.401	
	8	0.165	0.670	0.355	0.2618	0.1754	1.61
	9	0.148	0.704	0.389		0.1843	1.47
	10	0.134	0.732	0.421		0.1916	1.36
	11	0.120	0.760	0.455		0.1990	1.23
	12	0.109	0.782	0.479		0.2048	1.14
	13	0.095	0.810	0.515		0.2121	1.00
14	0.083	0.834	0.546	0.2183		0.890	
15	0.072	0.856	0.576	0.2241		0.781	
1¼	16	0.065	0.870	0.594	0.2277	0.710	
	17	0.058	0.884	0.613	0.2314	0.639	
	18	0.049	0.902	0.639	0.2361	0.545	
	8	0.165	0.920	0.665	0.3271	0.2409	2.09
	9	0.148	0.954	0.714		0.2498	1.91
	10	0.134	0.982	0.757		0.2572	1.75
	11	0.120	1.01	0.800		0.2644	1.58
	12	0.109	1.03	0.836		0.2701	1.45
13	0.095	1.06	0.884	0.2775		1.28	
14	0.083	1.08	0.923	0.2839		1.13	
15	0.072	1.11	0.960	0.2896		0.991	
1½	16	0.065	1.12	0.985	0.2932	0.900	
	17	0.058	1.13	1.01	0.2969	0.808	
	18	0.049	1.15	1.04	0.3015	0.688	
	8	0.165	1.17	1.075	0.3925	0.3063	2.57
	9	0.148	1.20	1.14		0.3152	2.34
	10	0.134	1.23	1.19		0.3225	2.14
	11	0.120	1.26	1.25		0.3299	1.98
	12	0.109	1.28	1.29		0.3356	1.77
13	0.095	1.31	1.35	0.3430		1.56	
14	0.083	1.33	1.40	0.3492		1.37	
15	0.072	1.36	1.44	0.3555		1.20	
1¾	16	0.065	1.37	1.47	0.3587	1.09	
	17	0.058	1.38	1.50	0.3623	0.978	
	18	0.049	1.40	1.54	0.3670	0.831	

Table 2.4: Minimum shell thickness with respect to shell diameter [12]

Nominal Shell Diameter		MINIMUM SHELL THICKNESS			
		Dimensions in Inches (mm)			
		Minimum Thickness		Alloy *	
		Carbon Steel			
		Pipe	Plate		
6	(152)	SCH. 40	-	1/8	(3.2)
8-12	(203-205)	SCH. 30	-	1/8	(3.2)
13-23	(330-584)	SCH. 20	5/16 (7.9)	1/8	(3.2)
24-29	(610-737)	-	5/16 (7.9)	3/16	(4.8)
30-39	(762-991)	-	3/8 (9.5)	1/4	(6.4)
40-60	(1016-1524)	-	7/16 (11.1)	1/4	(6.4)
61-80	(1549-2032)	-	1/2 (12.7)	5/16	(7.9)
81-100	(2057-2540)	-	1/2 (12.7)	3/8	(9.5)

*Schedule 5S is permissible for 6 inch (152 mm) and 8 inch (203 mm) shell diameters.

Tubes in STHE is held with a help of tube sheet as shown in Figure 2.12. The layout of the tubes in tube sheet may vary according to applications. Some different tube layouts it shown in Figure 2.16.

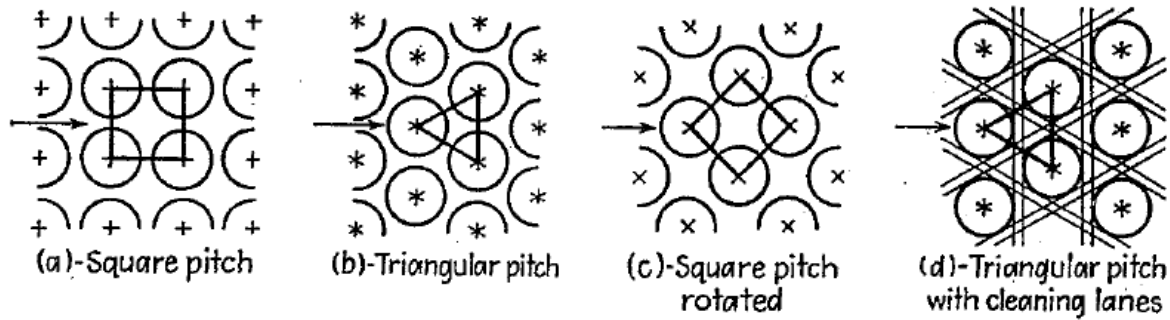


Figure 2.16: Common tube layouts in STHE [12]

As shown in Figure 2.16, tubes can be laid out in either square or triangular pattern. The advantage of square pattern is that it could easily be cleaned, and it causes lower pressure drops while the advantage of triangular pattern is that it could accommodate more tubes.

Tube pitch (S_t) is the distance between adjacent tubes in STHE. Lower the tube pitch, more the number of tubes that will fit inside the shell. But at much lower pitch, it becomes mechanically infeasible as the tube support cannot be drilled too close to each other. Hence, it is highly standardized and rough values can be estimated from equation (2.4) [9].

$$S_t = 1.25 \times d_o \quad (2.4)$$

Here, d_o is the external tube diameter in STHE.

Number of tubes (N_t) gives the count of number of tubes which is held by tube sheet. Based on the tube pitch and shell diameter, number of tubes (N_t) in STHE can be estimated from equation (2.5). A derivation to this expression is presented in Appendix C.

$$N_t = \frac{\pi D_s^2}{4S_t^2} \quad (2.5)$$

Shell side cross sectional area (A_s) is the cross-section flow area in shell side. It given by equation (2.6) [14].

$$A_s = \frac{D_s}{N_s} \times B \times \left(1 - \frac{d_o}{S_t}\right) \quad (2.6)$$

Here, N_s is number of shells.

Equivalent diameter in shell (d_e) is the diameter through which the fluids flows in the shell side. It is given by equation (2.7) for square pitch layout of tubes [14].

$$d_e = \frac{4}{\pi d_o^2} \times \left(S_t^2 - \frac{\pi d_o^2}{4}\right) \quad (2.7)$$

Tube length (L) can be estimated once heat transfer area (A) of STHE is estimated. It is given by equation (2.8).

$$L = \frac{A}{\pi d_i N_t} \quad (2.8)$$

2.5.5 Heat transfer coefficient

Heat transfer coefficients are the proportionality constant between heat flux and driving force (temperature difference). During conduction, heat flow is represented by equation (2.9) and during convection, heat flow is represented by equation (2.10).

$$q = kA \frac{dT}{dx} \quad (2.9)$$

$$q = hAdT \quad (2.10)$$

Here, q is heat transfer rate, A is heat transfer area, k is conductivity, h is convective heat transfer coefficient, dT is temperature difference and dx is distance between temperature difference sources.

Convective heat transfer coefficient can be estimated from Nusselt's correlation given in equation (2.11) [15].

$$h = \frac{Nu \times k}{d} \quad (2.11)$$

Here, Nu is Nusselt's number, k is thermal conductivity of fluid and d is effective length. Effective length depends on geometry of flow and for a tube it represents tube diameter. Nusselt's number also varies with geometry and various correlations are available for various flow geometry.

Nusselt's number for tube side flow can be given by various turbulent flow correlations. Dittus Boelter and Sieder and Tate correlations are two of the most commonly used one. They have an error of around 25% and works for Reynold's number above 10,000 [15]. Gnielinski correlation is given by equation (2.12). It has an error of around 10% and works for Reynolds number between 3,000 and 5,000,000 [15].

$$Nu = \frac{\left(\frac{f_t}{8}\right) (Re - 1000) Pr}{1 + 12.7 \sqrt{\frac{f_t}{8}} (Pr^{\frac{2}{3}} - 1)} \quad (2.12)$$

Here, Re_t is Reynold's number in tube, Pr_t is Prandlt's number in tube and f_t is tube side friction factor.

Friction factor along tube (f_t) is given by equation (2.13).

$$f_t = (1.82 \log_{10}^{Re_t} - 1.64)^{-2} \quad (2.13)$$

By combining equation (2.11) and (2.12), tube side convective heat transfer (h_t) for gases can be given by equation (2.14).

$$h_t = \frac{k_t}{d_i} \frac{\left(\frac{f_t}{8}\right) (Re - 1000) Pr}{1 + 12.7 \sqrt{\frac{f_t}{8}} (Pr^{\frac{2}{3}} - 1)} \quad (2.14)$$

Here, k_t is conductivity of tube side fluid.

For shell side flow, with $1,000,000 > Re_s > 2000$, Nusselt's can be given from Kern's formulation shown in equation (2.15) [12].

$$Nu = 0.36Re_s^{0.55}Pr_s^{0.3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (2.15)$$

Here, Re_s is shell side Reynold's number, Pr_s is shell side Prandlt's number, μ is viscosity in tube side fluid and μ_w is the viscosity near the wall.

For gases, μ is almost equal to μ_w as variation of viscosity in gases with temperature is comparatively low [Refer Table 4.6 and Table 4.8].

Equivalent diameter (d_e) discussed in section 2.5.4, is effective length for shell side in equation (2.11). Combining equations (2.11) and (2.15) along with viscosity equality condition discussed above, shell side heat transfer coefficient (h_s) for gases can be given by equation (2.16).

$$h_s = 0.36 \times \frac{k_s}{d_e} \times Re_s^{0.55} \times Pr_s^{0.3} \quad (2.16)$$

Here, k_s is conductivity of shell side fluid.

Prandlt's number is given by equation (2.17). This is valid for both shell side and tube side.

$$Pr = \frac{C_p \times \mu}{k} \quad (2.17)$$

Reynold's number in tube and shell side can be given by equations (2.18) and (2.19) respectively.

$$Re_t = \frac{\rho_t \times d_i \times v_t}{\mu_t} \quad (2.18)$$

$$Re_s = \frac{\rho_s \times d_e \times v_s}{\mu_s} \quad (2.19)$$

Here, ρ_t is tube side gas density, ρ_s is shell side gas density, d_i is internal diameter of tube, d_e is equivalent diameter of shell, μ_t is tube side gas viscosity, μ_s is shell side gas viscosity, v_t is tube side gas velocity and v_s is shell side gas velocity.

Tube side gas velocity and shell side gas velocity can be given by equations (2.20) and (2.21) respectively [14].

$$v_t = \frac{4 \times m_t \times n}{\pi d_i^2 \times \rho_t \times N_t} \quad (2.20)$$

$$v_s = \frac{m_s}{\rho_s \times A_s} \quad (2.21)$$

Here, n is number of passes, m_t is mass flow of tube side fluid, N_t is number of tubes, m_s is mass flow of shell side fluid and A_s is shell side cross-sectional area.

Once shell side and tube side heat transfer coefficients are determined, overall heat transfer (U) of STHE can be given by equation (2.22) [16].

$$U = \frac{1}{\frac{1}{h_s} + \frac{d_o}{2k_m} \ln\left(\frac{d_o}{d_i}\right) + \frac{d_o}{d_i} \times \frac{1}{h_t}} \quad (2.22)$$

Here, d_o is external diameter of tube, k_m is thermal conductivity of tube material, d_i is internal diameter of tube, h_s is shell side heat transfer coefficient and h_t is tube side heat transfer coefficient.

2.5.6 Heat balance and LMTD correction factor

Gases used in this thesis is assumed to be ideal for simplification. Hence, ideal gas law is followed for various gas correlations. Ideal gas used is shown by equation (2.23).

$$PV = nR_G T \quad (2.23)$$

Here, R_G is universal gas constant, P is pressure, V is volume, T is temperature and n is number of moles of the given gas.

Parallel arrangement of STHes divides the fluid flow. Hence, when total initial mass flow of hot ($m_{h,init}$) and cold ($m_{c,init}$) fluid is given, then mass flow of hot (m_h) and cold (m_c) fluid inside each exchanger can be calculated depending on number of heat exchangers in parallel (N_{HE}) by equations (2.24) and (2.25).

$$m_h = m_{h,init} / N_{HE} \quad (2.24)$$

$$m_c = m_{c,init} / N_{HE} \quad (2.25)$$

By energy balance, heat flow in hot side and cold side is given by equations (2.26) and (2.27).

$$q = m_h C_{p,h} (T_{h,in} - T_{h,out}) \quad (2.26)$$

$$q = m_c C_{p,c} (T_{c,out} - T_{c,in}) \quad (2.27)$$

Here, q is heat flow rate, $C_{p,h}$ is specific heat capacity of hot fluid, $C_{p,c}$ is specific heat capacity of cold fluid, $T_{h,in}$ is the temperature of hot fluid in, $T_{h,out}$ is the temperature of hot fluid out, $T_{c,in}$ is the temperature of cold fluid in and $T_{c,out}$ is the temperature of cold fluid out.

Minimum temperature difference (ΔT_{min}) is defined as the minimum temperature difference between hot and cold fluid. It could be represented by either equation (2.28) or equation (2.29).

$$\Delta T_{min} = T_{h,in} - T_{c,out} \quad (2.28)$$

$$\Delta T_{min} = T_{h,out} - T_{c,in} \quad (2.29)$$

If hot fluid has higher heat capacity, then equation (2.28) is used and if cold fluid has higher heat capacity, then equation (2.29) is used.

To know whether hot fluid or cold fluid has higher heat capacity, “ mC_p ” rule is generally employed. According to this, fluid having higher product of mass flow rate and specific heat capacity will have higher heat capacity.

Hence, if $m_h \times C_{p,h} > m_c \times C_{p,c}$ then equation (2.28) is used.

If $m_c \times C_{p,c} > m_h \times C_{p,h}$ then equation (2.29) is used.

Temperature difference between hot and cold fluid is the driving force for heat transfer. But in STHE, this temperature difference varies along the length of heat exchanger. Usual temperature profile for counterflow and parallel flow is shown in Figure 2.17.

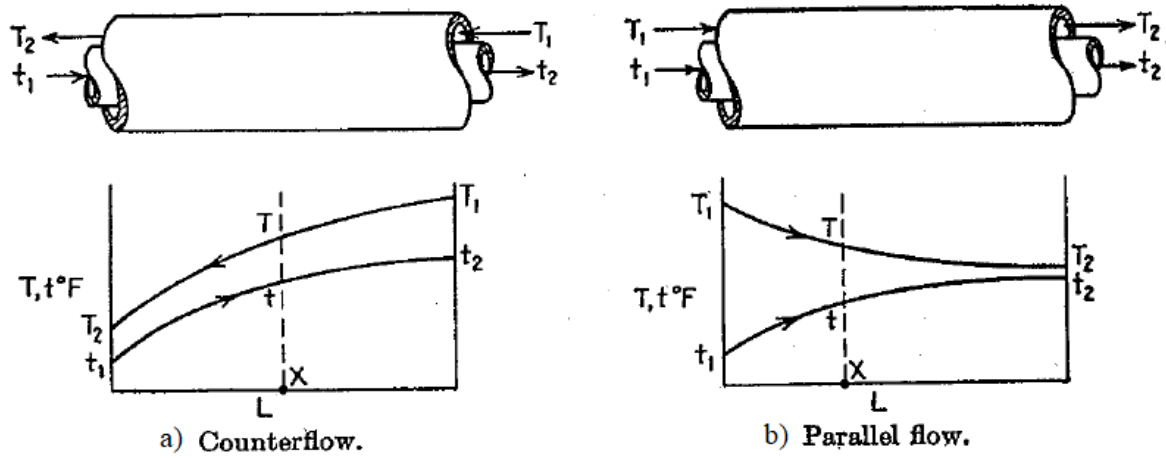


Figure 2.17: Temperature difference profile between hot and cold. a) For pure counter flow; b) For pure parallel flow [12]

Hence for design purposes, mean value of temperature difference is used. Logarithmic mean temperature difference (ΔT_{LMTD}) is the mean temperature difference derived and is shown in equation (2.30).

$$\Delta T_{LMTD} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln \left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}} \right)} \quad (2.30)$$

STHE has both counter and parallel flow happening simultaneously. Hence, equation (2.30) cannot be directly used to represent mean temperature difference and a correction factor (F_T) must be added compensate for efficiency loss from parallel flow.

LMTD correction factor (F_T) depends on the structure of STHE and can be found from literature. As a rule of thumb, F_T should be equal to or above 0.8 for an efficient system [12]. LMTD correction factor for 1-2 STHE is shown in equation (2.31) and for 2-4 STHE is shown in equation (2.32) [12].

$$F_T = \frac{\sqrt{R^2 + 1} \ln \left(\frac{1 - S}{1 - RS} \right)}{(R - 1) \ln \left(\frac{2 - S(R + 1 - \sqrt{R^2 + 1})}{2 - S(R + 1 + \sqrt{R^2 + 1})} \right)} \quad (2.31)$$

$$F_T = \frac{\frac{\sqrt{R^2 + 1}}{2(R + 1)} \ln \left(\frac{1 - S}{1 - RS} \right)}{\ln \left(\frac{\frac{2}{S} - 1 - R + \frac{2}{S} \sqrt{(1 - S)(1 - RS)} + \sqrt{R^2 + 1}}{\frac{2}{S} - 1 - R + \frac{2}{S} \sqrt{(1 - S)(1 - RS)} - \sqrt{R^2 + 1}} \right)} \quad (2.32)$$

Here, R and S are given by equation (2.33).

$$R = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{h,out}} \quad \text{and} \quad S = \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} \quad (2.33)$$

It can also be represented graphically. LMTD correction factor for 1-2 STHE is shown in Figure 2.18 and for 2-4 STHE is shown in Figure 2.19 [12].

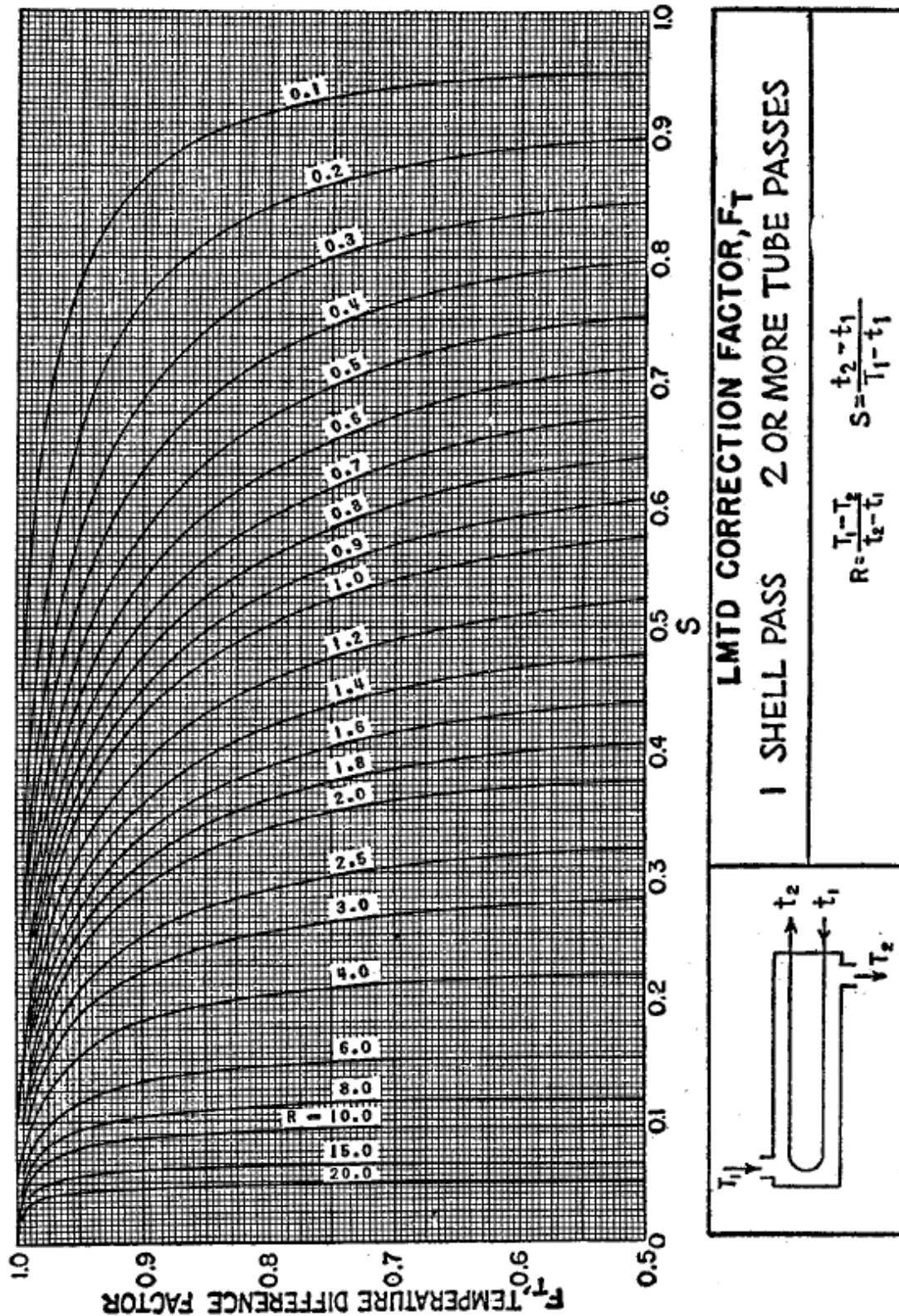


Figure 2.18: LMTD correction factor chart for 1-2 STHE [12]

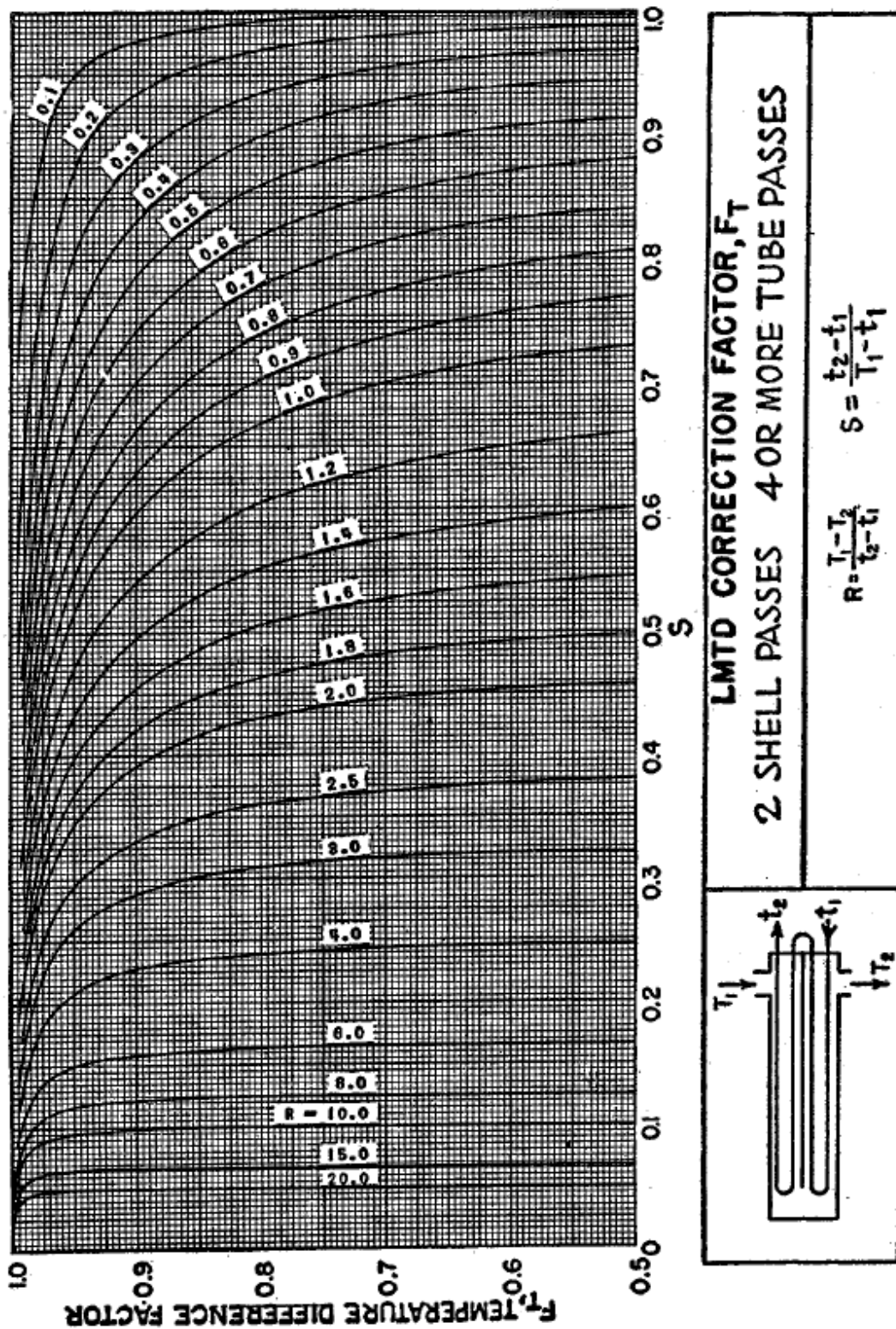


Figure 2.19: LMTD correction factor chart for 2-4 STHE [12]

The efficiency improvement when moving from 1-2 STHE structure to 2-4 STHE structure as discussed in section 2.5.1, is represented mathematically in terms of LMTD correction factor discussed above. For the same ΔT_{min} , F_T is higher in 2-4 STHE than in 1-2 STHE. Hence, the efficiency of the system is improved.

Finally, overall heat balance over STHE can be represented by equation (2.34).

$$q = UAF_T\Delta T_{LMTD} \quad (2.34)$$

Here, q is heat transfer rate in STHE, A is overall heat transfer area, F_T is LTMD correction factor and ΔT_{LMTD} is logarithmic mean temperature difference.

2.5.7 Pressure drop and net energy transferred estimation

Pressure drop determination is important to check if energy saved from using STHE is higher or lower than energy required to compensate pressure drop inside STHE.

Pressure drop for tube side:

Tube side pressure drop is the sum of pressure drop due to tube length and pressure drop induced due to change in direction inside the tubes. Taking this into account, final tube side pressure drop (ΔP_t) is given by equation (2.35) [12].

$$\Delta P_t = \frac{\rho_t v_t^2}{2} \left(\frac{L}{d_i} f_t + 4 \right) n \quad (2.35)$$

Here, ρ_t is density of tube side gas, v_t is velocity in the tube, L is tube length per pass, d_i is tube internal diameter, n is the number of passes and f_t is friction factor along tube.

Pressure drop for shell side:

Shell side pressure drop (ΔP_s) is given by equation (2.36) [14].

$$\Delta P_s = f_s \left(\frac{\rho_s v_s^2}{2} \right) \left(\frac{L}{B} \right) \left(\frac{D_s}{d_e} \right) \quad (2.36)$$

Here, ρ_s is density of cold gas, v_s is velocity in the shell, L is tube length per pass, B is baffle spacing, D_s is diameter of shell, d_e is equivalent diameter in the shell and f_s is shell side friction factor. Shell side friction factor (f_s) is given by equation (2.37) [12].

$$f_s = 2b_o Re_s^{-0.15} \quad (2.37)$$

Here, b_o is equal to 0.72 according to Kern.

Pressure compensation equipment based on pressure drop.

Based on pressure drop and volumetric flow rates, either centrifugal radial fan (CRF) or turbo blower (TB) can be used to overcome the pressure drop. It is assumed that each STHE in parallel with require 2 CRF or TB. One for tube side and one for shell side. Assumption is done for a better control over the process of heat transfer.

Centrifugal radial fan (CRF) is designed to work at lower pressure drop, but it can handle high volumetric flow rates. Turbo blower (TB) on the other hand can work at higher pressure drop, but it cannot handle high volumetric flow rates. But for general selection, centrifugal radial fan is preferred as it is cheaper. It is assumed that centrifugal radial fan can work upto pressure drop of 0.1 bar and turbo blowers can work upto pressure drop of 0.5 bar. But turbo blower has another constrain of volumetric flow rate. It is assumed that maximum volumetric flow rate of turbo blower is 15 m³/sec as it was the maximum limit used in cost estimation website [17].

Hence,

If pressure drop (ΔP_t or ΔP_s) is less than 0.1 bar, centrifugal radial fan is used.

If $0.1 \text{ bar} < \text{Pressure drop } (\Delta P_t \text{ or } \Delta P_s) < 0.5 \text{ bar}$ and Volumetric flow rate $< 15 \text{ m}^3/\text{sec}$, turbo blower is used.

Other cases are assumed to be technically infeasible.

Energy required for pressure compensation:

General isentropic work formula (W_{is}) is given in equation (2.38) [18].

$$W_{is} = C_p \times T_1 \times \left[\left(\frac{P_2}{P_1} \right)^{R_G/c_p} - 1 \right] \text{ kJ/kg} \quad (2.38)$$

If the hot air is flowing, then inlet side is at higher temperature and if cold air is flowing, outlet side has higher temperature. So, to avoid high temperature, fan should be placed in outlet during hot gas flow and in inlet during cold gas flow.

Isentropic work for hot side ($W_{is,h}$) is given by equation (2.39) [18].

$$W_{is,h} = m_h \times C_{p,h} \times T_{h,out} \times \left[\left(\frac{P_{h,in}}{P_{h,in} - \Delta P_{eq}} \right)^{R_G/c_{p,h}} - 1 \right] \quad (2.39)$$

Here, ΔP_{eq} = Pressure drop along tube or shell side, $P_{h,in}$ = inlet hot gas pressure, R_G is universal gas constant.

In actual processes there is some losses in pressure compensation equipment. This depends on fan efficiency (η_f). Efficiency of centrifugal radial fan is around 0.75 [6]. Hence for simplicity, efficiency of pressure compensation (η_f) was assumed to be constant at 0.75.

Actual work on hot side (W_h) can be calculated from equation (2.40).

$$W_h = \frac{W_{is,h}}{\eta_f} \quad (2.40)$$

Since calculation of energy saved from STHE is in form of heat, equivalent heat loss instead of work must be calculated. Conversion of work to heat depends on thermal efficiency (η_t). Gas turbine is one of the most efficient systems for conversion of heat to electricity and its efficiency is between 0.2 to 0.5 [16]. For simplicity, thermal efficiency (η_t) was assumed to be 0.4.

Hence, equivalent heat loss from hot side ($H_{l,h}$) is given by equation (2.41).

$$H_{l,h} = \frac{W_h}{\eta_t} \quad (2.41)$$

Since fan is placed at inlet of STHE for cold gas,

Isentropic work for cold side ($W_{is,c}$) is given by equation (2.42) [18].

$$W_{is,c} = m_c \times C_{p,c} \times T_{c,in} \times \left[\left(\frac{P_{c,in} + \Delta P_{eq}}{P_{c,in}} \right)^{R_G/c_{p,c}} - 1 \right] \quad (2.42)$$

Here, $P_{c,in}$ is inlet cold gas pressure.

Actual work on cold side (W_c) can be calculated from equation (2.43).

$$W_c = \frac{W_{is,c}}{\eta_{fan}} \quad (2.43)$$

Equivalent heat loss due to pressure drop ($H_{l,c}$) on cold side is given by equation (2.44).

$$H_{l,c} = \frac{W_c}{\eta_t} \quad (2.44)$$

Hence, total equivalent heat loss from each STHE ($H_{l,t}$) due to pressure drop can be estimated from equation (2.45).

$$H_{l,t} = H_{l,h} + H_{l,c} \quad (2.45)$$

Net energy transferred:

Net energy transferred (q_{net}) is the difference between heat duty of STHE and total equivalent heat loss from STHE. But when there are multiple STHE in parallel, it is assumed that each STHE has its own pressure compensation equipment for better control over the system. Hence, the difference is multiplied by number of STHE in parallel to get net energy transferred from all STHEs. Hence it can be represented by equation (2.46).

$$q_{net} = (q - H_{l,t}) \times N_{HE} \quad (2.46)$$

Here, q is STHE heat duty and N_{HE} is the number STHE in parallel.

2.5.8 Estimation of weight, size and footprint

Estimation of weight, size and footprint is important to for placement of STHE inside the system. Weight tells about the total weight of all STHEs along with fluid present inside it. Size tells about the volume occupied by all STHE. Footprint tell the floor area covered by all STHE.

Weight:

Rough weight of all STHE during operation (m_{HE}) can be given by the product of total STHE in parallel and sum of weight of tubes, shells and gases present inside STHE. It can be represented by equation (2.47).

$$m_{HE} = (m_{tm} + m_{sm} + m_{tg} + m_{sg}) \times N_{HE} \quad (2.47)$$

Here, m_{tm} is mass of tubes, m_{sm} is mass of shell, m_{tg} is mass of gas in tube side, m_{sg} is mass of gas in shell side and N_{HE} is total number of STHE in parallel.

To estimate mass of tubes, volume of tube material (V_{tm}) can be given by equation (2.48).

$$V_{tm} = \pi d_i L \times N_t \times t_t \quad (2.48)$$

Here, t_t is tube thickness, d_i is tube internal diameter, L is length of tube and N_t is number of tubes.

Hence, mass of tube (m_{tm}) can be given by equation (2.49).

$$m_{tm} = \rho_{tm} \times V_{tm} \quad (2.49)$$

Here, ρ_{tm} is density of tube material.

Assuming the shell material has flat ends for simplicity, volume of shell materials can be given from equation (2.50).

$$V_{sm} = t_s \times \left[(\pi D_s L) + \left(\frac{\pi D_s^2}{4} \right) \right] \quad (2.50)$$

Here, t_s is shell thickness and D_s is shell diameter.

Similarly, mass of shell (m_{sm}) can be given by equation (2.51).

$$m_{sm} = \rho_{sm} \times V_{sm} \quad (2.51)$$

Here, ρ_{sm} is density of shell material.

Mass of gas in tube (m_{tg}) is given by equation (2.52).

$$m_{tg} = m_t \times t_{tt} \quad (2.52)$$

Here, m_t is mass flow of gas in the tube and t_{tt} is the time spend by gas inside the tube. Time spend by gas inside the tube (t_{tt}) can be given by equation (2.53).

$$t_{tt} = \frac{L \times n}{v_t} \quad (2.53)$$

Here, n is number of passes and v_t is velocity of gas in tubes.

Similarly, mass of gas in shell (m_{sg}) is given by equation (2.54).

$$m_{sg} = m_s \times t_{ts} \quad (2.54)$$

Here, m_s is mass flow of gas in shell and t_{ts} is the time spend by gas inside the shell. Time spend by gas inside shell (t_{ts}) can be given by equation (2.55).

$$t_{ts} = \frac{L \times N_s}{v_s} \quad (2.55)$$

Here N_s is number of shell and v_s is velocity of gas in each shell.

Size:

Size can be determined by the volume occupied by all STHE. Assuming shell volume to cover entire STHE volume, total volume occupies by all STHE (V_{HE}) can be given by equation (2.56).

$$V_{HE} = \frac{\pi D_s^2 L}{4} \times N_{HE} \quad (2.56)$$

Footprint:

Footprint of all STHE can be estimated by calculating floor area occupied by all STHE. Floor area for all STHE (F_{HE}) can be given by equation (2.57).

$$F_{HE} = L \times D_s \times N_{HE} \quad (2.57)$$

3 Cost estimation theory

This section discusses different theories used for estimation of cost in this thesis.

3.1 Capacity factor method

If cost and capacity of an equipment is available and cost estimation of same equipment with different capacity is required, then this method can be used.

Assuming cost and capacity of equipment “A” is available, cost of equipment “B” with known capacity can be calculated using equation (3.1).

$$Cost_B = \left(\frac{Capacity_B}{Capacity_A} \right)^{c_f} \times Cost_A \quad (3.1)$$

Here, c_f is the capacity factor. This usually varies between 0.4 and 0.9. But if cost and capacity of both equipment is available, it can be calculated by re-arranging equation (3.1) to give equation (3.2).

$$c_f = \frac{\log \left(\frac{Cost_A}{Cost_B} \right)}{\log \left(\frac{Capacity_A}{Capacity_B} \right)} \quad (3.2)$$

3.2 Detailed factor method

This method is used to estimate total installed cost of any equipment. This method uses a table developed by Nils Henrik Eldrup at USN, Porsgrunn. The table covers various engineering disciplines, different size and type of equipment and different material types. Variation between material type in equipment is done with the help of material factor. Material factor for material A compared to reference material is the ratio between cost of equipment made of material A and cost of same equipment made of a reference material. Calculation example of material factor is shown in section 5.3.

Detailed factor table (DFT) is shown for carbon steel material. If the material type of equipment is different from carbon steel, total installed cost factor must be adjusted in the table to find total installed cost factor for required material (f_{TIC}). It can be done by adjusting factors where material is involved and can be done with equation (3.3).

$$f_{TIC} = f_{TIC,CS} - f_{EQ,CS} + (f_{EQ,CS} \times m_f) - f_{PI,CS} + (f_{PI,CS} \times m_f) \quad (3.3)$$

Here, $f_{TIC,CS}$ is the total installed cost factor of CS material read from DFT, $f_{EQ,CS}$ is cost factor for CS equipment from DFT, $f_{PI,CS}$ is cost factor for CS piping read from DFT and m_f is the material factor.

Detailed factor table used in the thesis was published during Project Management and Cost Engineering (FM 3110), 2018 examination and is shown in Table 3.1.

Table 3.1: Detailed factor table [19]

Detailed factor table

	2016-2018										Solid												
	Fluid					Solid					Fluid					Solid							
	0-20	20-100	100-500	500-1000	1000-2000	2000-5000	5000-15000	>15000	0-20	20-100	100-500	500-1000	1000-2000	2000-5000	>5000	0-20	20-100	100-500	500-1000	1000-2000	2000-5000	>5000	
Cost of equipment in carbon steel (CS) (KNOK)																							
Equipment	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Erection/installation	0.89	0.47	0.25	0.18	0.14	0.11	0.10	0.08	1.97	1.04	0.61	0.43	0.36	0.25	0.22	1.97	1.04	0.61	0.43	0.36	0.25	0.22	
Piping	3.56	1.92	1.12	0.83	0.65	0.48	0.41	0.29	0.72	0.39	0.22	0.17	0.13	0.10	0.09	0.72	0.39	0.22	0.17	0.13	0.10	0.09	
Electric	1.03	0.71	0.48	0.41	0.34	0.28	0.25	0.18	1.74	1.09	0.72	0.56	0.47	0.39	0.33	1.74	1.09	0.72	0.56	0.47	0.39	0.33	
Instrument	3.56	1.92	1.12	0.83	0.65	0.48	0.41	0.29	1.41	0.77	0.46	0.33	0.27	0.18	0.15	1.41	0.77	0.46	0.33	0.27	0.18	0.15	
Civil	0.55	0.36	0.25	0.20	0.17	0.14	0.13	0.09	1.26	0.75	0.48	0.37	0.29	0.24	0.20	1.26	0.75	0.48	0.37	0.29	0.24	0.20	
Steel & concrete	1.79	1.17	0.79	0.64	0.55	0.43	0.39	0.28	2.50	1.55	1.02	0.79	0.66	0.52	0.47	2.50	1.55	1.02	0.79	0.66	0.52	0.47	
Insulation	0.67	0.34	0.18	0.14	0.11	0.09	0.05	0.04	0.67	0.34	0.18	0.14	0.11	0.09	0.05	0.67	0.34	0.18	0.14	0.11	0.09	0.05	
Direct cost	13.04	7.88	5.19	4.21	3.60	3.02	2.74	2.24	11.27	6.94	4.68	3.78	3.29	2.78	2.51	11.27	6.94	4.68	3.78	3.29	2.78	2.51	
Engineering Process	1.23	0.43	0.24	0.18	0.15	0.13	0.11	0.09	1.23	0.43	0.24	0.18	0.15	0.13	0.11	1.23	0.43	0.24	0.18	0.15	0.13	0.11	
Engineering Mechanical	0.98	0.24	0.10	0.05	0.04	0.03	0.01	0.01	1.23	0.37	0.17	0.11	0.09	0.05	0.04	1.23	0.37	0.17	0.11	0.09	0.05	0.04	
Engineering Piping	1.08	0.58	0.34	0.25	0.18	0.14	0.13	0.09	0.22	0.11	0.05	0.04	0.03	0.03	0.03	0.22	0.11	0.05	0.04	0.03	0.03	0.03	
Engineering Electric	1.04	0.30	0.15	0.11	0.10	0.09	0.05	0.04	1.22	0.41	0.20	0.25	0.13	0.10	0.09	1.22	0.41	0.20	0.25	0.13	0.10	0.09	
Engineering Instrument	1.85	0.72	0.36	0.25	0.20	0.14	0.13	0.09	1.21	0.36	0.15	0.11	0.09	0.05	0.04	1.21	0.36	0.15	0.11	0.09	0.05	0.04	
Engineering Civil	0.39	0.11	0.04	0.03	0.03	0.01	0.01	0.01	0.50	0.17	0.09	0.05	0.04	0.03	0.03	0.50	0.17	0.09	0.05	0.04	0.03	0.03	
Engineering Steel and Concrete	0.58	0.24	0.13	0.10	0.09	0.05	0.05	0.04	0.67	0.28	0.15	0.13	0.11	0.09	0.09	0.67	0.28	0.15	0.13	0.11	0.09	0.09	
Engineering Insulation	0.27	0.09	0.03	0.01	0.01	0.01	0.01	0.01	0.27	0.09	0.03	0.01	0.01	0.01	0.01	0.27	0.09	0.03	0.01	0.01	0.01	0.01	
Engineering cost	7.43	2.73	1.38	0.99	0.80	0.60	0.51	0.38	6.54	2.21	1.08	0.89	0.65	0.48	0.43	6.54	2.21	1.08	0.89	0.65	0.48	0.43	
Procurement	1.55	0.52	0.20	0.13	0.09	0.04	0.03	0.03	1.55	0.52	0.20	0.13	0.09	0.04	0.03	1.55	0.52	0.20	0.13	0.09	0.04	0.03	
Project control	0.37	0.14	0.05	0.04	0.04	0.03	0.03	0.03	0.33	0.11	0.05	0.04	0.03	0.03	0.03	0.33	0.11	0.05	0.04	0.03	0.03	0.03	
Site Management	0.66	0.42	0.28	0.24	0.20	0.17	0.15	0.11	0.56	0.36	0.25	0.20	0.18	0.15	0.15	0.56	0.36	0.25	0.20	0.18	0.15	0.15	
Project management	0.89	0.46	0.29	0.24	0.20	0.17	0.15	0.11	0.76	0.39	0.25	0.20	0.17	0.15	0.14	0.76	0.39	0.25	0.20	0.17	0.15	0.14	
Administration cost	3.47	1.54	0.83	0.65	0.53	0.39	0.36	0.28	3.20	1.38	0.76	0.57	0.46	0.37	0.34	3.20	1.38	0.76	0.57	0.46	0.37	0.34	
Commissioning	0.72	0.33	0.17	0.10	0.10	0.05	0.05	0.04	0.62	0.29	0.15	0.11	0.09	0.05	0.04	0.62	0.29	0.15	0.11	0.09	0.05	0.04	
Total Known cost	24.66	12.48	7.57	5.95	5.04	4.06	3.65	2.94	21.64	10.83	6.68	5.36	4.48	3.68	3.32	21.64	10.83	6.68	5.36	4.48	3.68	3.32	
Contingency	4.99	2.55	1.57	1.24	1.06	0.87	0.78	0.64	4.38	2.22	1.39	1.13	0.95	0.79	0.72	4.38	2.22	1.39	1.13	0.95	0.79	0.72	
Total cost	29.65	15.03	9.13	7.20	6.10	4.93	4.44	3.59	26.02	13.05	8.07	6.48	5.43	4.47	4.04	26.02	13.05	8.07	6.48	5.43	4.47	4.04	
Material factors																							
When using other materials than CS, the factors for equipment and piping must be multiplied with the material factor																							
	Porsgrunn november 2016 Nils Henrik Eldrup																						
Material factors																							
Stainless steel (SS316) welded: 1.75																							
Stainless steel (SS316) machined: 1.30																							
GRP (plastic): 1.00																							
Exotic: 2.50																							

3.3 Net present value (NPV)

Worth of money changes with time. Future worth of money is much lower than current worth. Difference of this worth is dependent on the rate of interest (i). Higher the interest, lower is the worth of future money. If this interest is not considered, future money can be termed as non-discounted cash flow (C_{ND}).

Net present value (NPV) gives the current worth of non-discounted cash flow. If there is a cash flow from year “0” to year “ n_{yr} ”, then NPV of the cash flow is given by equation (3.4).

$$NPV = \sum_{i=0}^{n_{yr}-1} C_{ND} \times \frac{1}{(1+i)^{n_{yr}}} \quad (3.4)$$

3.4 Equivalent annual cost (EAC)

Equivalent annual cost (EAC) is the annual cost of owning, operating and maintaining the asset over its entire period of service. It is obtained by dividing NPV to annuity factor and is shown in equation (3.5).

$$EAC = \frac{NPV}{a_f} \quad (3.5)$$

Here a_f is annuity factor. Annuity factor can be calculated from equation (3.6).

$$a_f = \frac{1 - \frac{1}{(1+i)^{n_{yr}}}}{i} \quad (3.6)$$

Here, n_{yr} is the number of years and i is the rate of interest.

Capital expenditure (CAPEX) and Operational expenditure (OPEX) for each year was estimated using this method with the help of equations (3.7) and (3.8).

$$EAC_{CAPEX} = \frac{NPV_{CAPEX}}{a_f} \quad (3.7)$$

$$EAC_{OPEX} = \frac{NPV_{OPEX}}{a_f} \quad (3.8)$$

Here, NPV_{CAPEX} is net present value of capital expenditure and NPV_{OPEX} is net present value of operational expenditure. NPV_{CAPEX} is equal to total installed cost of all equipment and NPV_{OPEX} is the net present value of cost saving from electricity during operating period.

4 Heat exchanger design calculations

This section discusses the methods used for calculation of heat exchanger parameters. It also gives a step by step calculation example for better understanding of reader.

4.1 Design basis

Design basis handed at the start of thesis is attached as Appendix D and shown in Table 4.1.

Table 4.1: Design basis

Quantity	Unit	Symbol	Value	Remarks
Flow rate of hot gas (CO ₂)	t/h	$m_{h,init}$	67	Pure CO ₂ is assumed
Flow rate of cold gas (air)	t/h	$m_{c,init}$	71	Pure air is assumed
Inlet temperature of CO ₂	°C	$T_{h,in}$	900	
Inlet temperature of air	°C	$T_{c,in}$	225	
Dust concentration in CO ₂	mg/Nm ³	$C_{h,init}$	50,000	Approximate value
Dust concentration in air	mg/Nm ³	$C_{c,init}$	5	
Inlet pressure of CO ₂	bara	$P_{h,in}$	1	
Inlet pressure of air	bara	$P_{c,in}$	1	
Operation hours per year	h/y	$n_{hr,yr}$	7315	

4.2 Rating of various exchanger

Design basis given in section 4.1, sets certain criteria for selection of heat exchanger. Both inlet streams are gas. But gases have low thermal conductivity which gives low overall heat transfer coefficient in heat exchanger. Hence, large area is required for transfer of heat.

To achieve large heat transfer area, heat exchanger must be designed to accommodate maximum area per unit volume of heat exchanger (high area density) or it might be physically impossible to build. But high area density may also give high pressure drop. Hence, a design to maintain a low-pressure drop is also an important criterion.

Inlet temperature of hot gas is 900°C. Hence a heat exchanger must be designed for high temperature application. Low dust concentration for inlet hot gas is presented in design basis. Hence it is also an important criterion.

Availability of design standards also plays an important role. It directly affects the cost of building heat exchanger and reduces the risk on the designed heat exchanger as data from experience is available.

Higher heat transfer efficiency corresponds to maximum utilization of available heat. Maximum utilization of available heat increases heat duty, so it is an important criterion.

The idea of the whole project is to increase carbon capture. When there is mixing, carbon capture becomes more difficult. Hence, mixing problem is an important criterion.

Importance of various criteria for selection of heat exchanger can be represented in terms of weight factor (w_{tf}). Weight factor (w_{tf}) for each criterion can be found from equation (4.1).

$$w_{tf} = \frac{w_i}{\sum w_i} \quad (4.1)$$

Weight (w_i) in each criterion is selected from understanding of each criterion from discussion made above. Weight factor for various criteria is shown in Table 4.2.

Table 4.2: Weight factor of important criteria for selection of heat exchanger

Sl No	Criteria	Weight (w_i)	Weight Factor (w_{tf})
1	Low pressure drop	3	0.13
2	High temperature application	5	0.217
3	Low dust application	1	0.043
4	High area density	4	0.174
5	Availability of design standards	4	0.174
6	Heat transfer efficiency	3	0.13
7	Mixing problem	3	0.13

Overall ratings of a heat exchanger based on important criteria can be given as:

$$r_t = \sum w_{tf} \times r_i \quad (4.2)$$

Rating of heat exchanger is done from 0-5. If rating for any parameter is 0, then the heat exchanger is assumed to be technically infeasible.

Ratings of heat exchanger for different criteria is done from understanding of each heat exchanger discussed in section 2.1.

Overall ratings of heat exchanger according to design basis are shown in Table 4.3.

Table 4.3: Overall ratings of various heat exchangers according to design basis. Here, STHE = Shell and tube heat exchanger, PTHE = Plate type heat exchanger, RHE = Regenerator type heat exchanger, DPHE = double pipe heat exchanger

Sl No	Criteria	Weight Factor (w_{tf})	STHE rating (r_i)	PTHE rating (r_i)	RHE rating (r_i)	DPHE rating (r_i)
1	Low pressure drop	0.13	3	2	4	0
2	High temperature application	0.217	3	1	5	5
3	Low dust application	0.043	5	2	4	5
4	High area density	0.174	3	5	5	2
5	Availability of design standards	0.174	5	4	2	4
6	Heat transfer efficiency	0.13	3	5	1	2
7	Mixing problem	0.13	4	4	0	4
8	Overall Ratings (r_t)	---	3.6	3.3	0	0

Based on the ratings in given in Table 4.3, STHE and PTHE are suitable for design. But high temperature application from PTHE could be a risk. Also, STHE is much more standardised than PTHE. Hence, STHE was chosen for design.

4.3 Selection of STHE design parameters

This section discusses about selection of various parameters used in STHE design.

4.3.1 Selection of STHE structure

From various structures of shell and tube heat exchanger presented in section 2.5.1, 1-2 STHE and 2-4 STHE was chosen for design in this thesis. This was done to check the effect during series arrangement of STHE. Parallel arrangement of STHE was also done.

4.3.2 STHE material selection

Most common materials used in STHE is carbon steel (CS) and stainless steel (SS) depending on temperature conditions of the fluids. But, design basis specifies very high temperature of gas. This requires consideration of special metal alloys which can withstand high temperature conditions. Literature review on materials which can withstand high temperature conditions is presented in section 2.4. Selection of general range of materials based on high temperature application is shown in Table 4.4.

Table 4.4: Material selection for high temperature application [3]

Temperature Range	Alloys
Up to 1202°F (650°C)	Carbon steels having <1% Cr, or steels having 1%–12% Cr and Mo up to 1%.
1202°F–1472°F (650°C–800°C)	Carbon steels and low-alloy steels, Cr–Ni–Mo steels that can contain 12%–25% Cr and 5%–25% Ni, e.g., AISI type 310.
1472°F–1832°F (800°C–1000°C)	Steels with 17%–27% Cr and Mo addition, or higher Cr–Ni–Mo steels having more than 8% Ni.
1832°F–2192°F (1000°C–1200°C)	Higher alloyed nickel- or cobalt-base alloys with 18%–35% Cr; Al additions to these alloys improve resistance to high-temperature oxidation and cyclic oxidation.
>2192°F (1200°C)	Advanced ceramics and other highly refractory material.

Source: Compiled from Kane, R.D. and Cayard, M.S., *Hydrocarbon Process.*, November, 129, 1995.

Inconel 718 (IN) was the suggested material for high temperature application [8]. Chemical composition of IN is given in Table 4.5.

Table 4.5: Chemical composition of Inconel 718 [20]

Nickel (plus Cobalt)	50.00-55.00
Chromium.....	17.00-21.00
Iron	Balance*
Niobium (plus Tantalum).....	4.75-5.50
Molybdenum	2.80-3.30
Titanium.....	0.65-1.15
Aluminum	0.20-0.80
Cobalt	1.00 max.
Carbon	0.08 max.
Manganese	0.35 max.
Silicon	0.35 max.
Phosphorus.....	0.015 max.
Sulfur.....	0.015 max.
Boron	0.006 max.
Copper	0.30 max.

*Conforms to AMS specifications

*Reference to the 'balance' of a composition does not guarantee this is exclusively of the element mentioned but that it predominates and others are present only in minimal quantities.

Chemical composition of IN is also consistent with material selection criteria presented in Table 4.4. Hence, it was decided to use IN for construction of shell and tube heat exchanger.

Moreover, cost of IN was found to be around \$28-30 per kg [21].

4.3.3 Tube side and shell side fluid selection

Discussion of placing high temperature and fouling fluid in tube side is presented in section 2.5.2. Since hot CO₂ gas fulfils both criteria it was decided to put it in tube side and cold air in shell side.

4.3.4 Selection of sizing parameters

TEMA standards outlines selection of various STHE sizing parameters. Based on literature review presented in section 2.5.4, baffle spacing was assumed to be equal to shell diameter and internal diameter was varied for different test cases.

With the help of Figure 2.15, test cases for internal diameter (d_i) was decided to be 0.02m, 0.025m, 0.038m and 0.051m. With reference to Table 2.3 and Table 2.4, for simplicity thickness of tubes (t_t) and shell (t_s) was assumed to be a constant value at 0.0027 m and 0.0079 m.

Since square pitch gave lower pressure drop and easy cleaning, it was decided to use square pitch. A formula was derived in Appendix C to estimate number of tubes when a square pitch is used.

Correlations of other sizing parameters to internal diameter is discussed in section 2.5.4.

4.4 Degree of freedom analysis

Analysis of design variables and their correlations with other variables gives a good overview of design process. It helps in understanding variables which should be provided by the user and the variable which are calculated.

Degree of freedom gives the number of variables required to be specified by the user. It is the difference between number of independent variables and number of correlations between the variables.

Various independent variables used for the thermo-hydraulic analysis of STHE are:

1. N_{HE} = Number of STHE in parallel
2. N_s = Number of shells
3. D_s = Diameter of the shell
4. d_i = Internal diameter of tube
5. d_o = External diameter of tube
6. B = Baffle spacing
7. v_t = Velocity of gas inside the tube
8. N_t = Number of tubes in tube sheet
9. n = Number of passes for each tube
10. S_t = Tube pitch
11. Re_t = Reynold's number of gas inside the tube
12. h_t = Tube side heat transfer coefficient
13. t_t = Tube thickness
14. t_s = Shell thickness
15. d_e = Shell side equivalent diameter
16. A_s = Shell side cross flow area
17. v_s = Shell side velocity
18. Re_s = Shell side Reynold's number
19. h_s = Shell side heat transfer coefficient
20. U = Overall heat transfer coefficient
21. $m_{h,init}$ = Total initial mass flow rate of hot stream
22. $m_{c,init}$ = Total initial mass flow rate of cold stream
23. m_t = Mass flow rate of tube side fluid inside each HE
24. m_s = Mass flow rate of shell side fluid inside each HE
25. $T_{h,in}$ = Temperature of hot stream in
26. $T_{h,out}$ = Temperature of hot stream out
27. ΔT_{min} = Minimum temperature difference

28. $T_{c,in}$ = Temperature of cold stream in
29. $T_{c,out}$ = Temperature of cold stream out
30. q = Heat transfer rate in STHE
31. ΔT_{LMTD} = Logarithmic Mean Temperature difference
32. F_T = Correction factor for Logarithmic Mean Temperature difference
33. A = Overall heat transfer area
34. L = Length of the tube
35. f_t = Darcy friction factor or friction factor along tube
36. ΔP_t = Pressure drop across tube
37. f_s = Friction factor along shell
38. ΔP_s = Pressure drop across shell
39. $P_{h,in}$ = Inlet pressure of CO₂
40. $P_{c,in}$ = Inlet pressure of air
41. η_t = Efficiency of converting heat to electricity
42. η_f = Efficiency of fan to compensate pressure drop
43. $W_{is,h}$ = Isentropic work required on hot fluid to compensate pressure drop
44. $W_{is,c}$ = Isentropic work required on cold fluid to compensate pressure drop
45. W_h = Actual work done by fan on tube side to compensate pressure drop
46. W_c = Actual work done by fan on shell side to compensate pressure drop
47. $H_{l,c}$ = Equivalent heat loss by fan on cold fluid to compensate pressure drop
48. $H_{l,h}$ = Equivalent heat loss by fan on hot fluid to compensate pressure drop
49. $H_{l,t}$ = Equivalent heat loss by fan to compensate total pressure drop
50. q_{net} = Net energy transferred in heat exchanger
51. V_{tm} = Volume of tube material
52. V_{sm} = Volume of shell material
53. m_{tm} = Mass of tube
54. m_{sm} = Mass of shell
55. t_{tt} = Time spent by CO₂ inside tube side of HE
56. t_{ts} = Time spent by air inside shell side of HE
57. m_{tg} = Mass of CO₂ inside tube at a given time
58. m_{sg} = Mass of air inside tube at a given time
59. m_{HE} = Mass of all heat exchanger during operation
60. V_{THE} = Total volume occupied by all heat exchangers
61. F_{THE} = Total footprint of all heat exchangers

Relationship between various variables are (variables given as a function of other variables):

1. $m_t = f(m_{h,init}, N_{HE})$ [Refer equation (2.24)]
2. $m_s = f(m_{c,init}, N_{HE})$ [Refer equation (2.25)]
3. $d_o = f(d_i, t_t)$ [Thickness addition in internal diameter]
4. $n = f(N_s)$ [Decided from STHE structure]
5. $D_s = f(d_i)$ [Refer equation (2.2)]
6. $B = f(D_s)$ [Refer equation (2.3)]
7. $v_t = f(m_t, d_i, N_t, n)$ [Refer equation (2.20)]
8. $N_t = f(S_t, D_s)$ [Refer equation (2.5)]
9. $S_t = f(d_i)$ [Refer equation (2.4)]

10. $Re_t = f(v_t, d_t)$ [Refer equation (2.18)]
11. $h_t = f(Re_t, d_t)$ [Refer equation (2.14)]
12. $d_e = f(S_t, d_o)$ [Refer equation (2.7)]
13. $A_s = f(D_s, B, d_o, N_s, S_t)$ [Refer equation (2.6)]
14. $v_s = f(m_s, A_s)$ [Refer equation (2.21)]
15. $Re_s = f(d_e, v_s)$ [Refer equation (2.19)]
16. $h_s = f(Re_s, d_e)$ [Refer equation (2.16)]
17. $U = f(h_t, h_s, R_m)$ [Refer equation (2.22)]
18. $T_{c,out} = f(T_{h,in}, \Delta T_{min})$ [Refer equation (2.28)]
19. $q = f(T_{c,in}, T_{c,out}, m_s)$ [Refer equation (2.27)]
20. $T_{h,out} = f(q, T_{h,in}, m_t)$ [Refer equation (2.26)]
21. $\Delta T_{LMTD} = f(T_{h,in}, T_{h,out}, T_{c,in}, T_{c,out})$ [Refer equation (2.30)]
22. $F_T = f(T_{h,in}, T_{h,out}, T_{c,in}, T_{c,out})$ [Refer equation (2.31), (2.32)]
23. $A = f(q, \Delta T_{LMTD}, F_T, U)$ [Refer equation (2.34)]
24. $L = f(A, d_i)$ [Refer equation (2.8)]
25. $f_t = f(Re_t)$ [Refer equation (2.13)]
26. $\Delta P_t = f(v_t, L, d_i, f_t, n)$ [Refer equation (2.35)]
27. $f_s = f(Re_s)$ [Refer equation (2.13)]
28. $\Delta P_s = f(v_s, L, B, D_s, d_e)$ [Refer equation (2.36)]
29. $W_{is,h} = f(m_t, T_{h,out}, P_{h,in}, \Delta P_t)$ [Refer equation (2.39)]
30. $W_{is,c} = f(m_s, T_{c,out}, P_{c,in}, \Delta P_s)$ [Refer equation (2.42)]
31. $W_h = f(W_{is,h}, \eta_f)$ [Refer equation (2.40)]
32. $W_c = f(W_{is,c}, \eta_f)$ [Refer equation (2.43)]
33. $H_{l,h} = f(W_h, \eta_t)$ [Refer equation (2.41)]
34. $H_{l,c} = f(W_c, \eta_t)$ [Refer equation (2.44)]
35. $H_{l,t} = f(H_{l,t}, H_{l,s})$ [Refer equation (2.45)]
36. $q_{net} = f(q, H_{l,t}, N_{HE})$ [Refer equation (2.46)]
37. $V_{tm} = f(d_i, L, N_t, t_t)$ [Refer equation (2.48)]
38. $V_{sm} = f(D_s, L, t_s)$ [Refer equation (2.50)]
39. $m_{tm} = f(V_{tm})$ [Refer equation (2.49)]
40. $m_{sm} = f(V_{sm})$ [Refer equation (2.51)]
41. $t_{tt} = f(L, n, v_t)$ [Refer equation (2.53)]
42. $t_{ts} = f(L, N_s, v_s)$ [Refer equation (2.55)]
43. $m_{tg} = f(m_t, t_{tt})$ [Refer equation (2.52)]
44. $m_{sg} = f(m_s, t_{ts})$ [Refer equation (2.54)]
45. $m_{HE} = f(m_{tg}, m_{sg}, m_{tm}, m_{sm}, N_{HE})$ [Refer equation (2.47)]
46. $V_{THE} = f(D_s, L, N_{HE})$ [Refer equation (2.56)]
47. $F_{THE} = f(D_s, L, N_{HE})$ [Refer equation (2.57)]

Number of variables (N_v) = 61

Number of relations (N_r) = 47

Degree of freedom (DOF) = $N_v - N_r = 14$

Hence, the number of variables decided by the user is 14.

Design basis sets $m_{h,init}$, $m_{c,init}$, $T_{h,in}$, $T_{c,in}$, $P_{h,in}$ and $P_{c,in}$. To vary number of heat exchanger in parallel and series, number of heat exchanger (N_{HE}) and number of shells (N_S) are varied respectively. To vary size of heat exchangers, diameter of tube (d_i) is selected and varied as discussed in section 4.3.4. Thickness of tube (t_t) and shell (t_s) is selected as discussed in section 4.3.4. η_t and η_f are chosen to be 0.4 and 0.75 respectively as discussed in section 2.5.7.

LMTD correction factor (F_T) is kept as 0.8 as discussed in section 2.5.6. But, it was not possible to use F_T directly due to complicated formula given for calculations of F_T [equations (2.31) and (2.32)]. Hence, trial and error method using calculation procedure (discussed in section 4.5.3) was done to find minimum temperature difference (ΔT_{min}) when F_{LMTD} was 0.8. ΔT_{min} was found to be 315K for 1-2 STHE and 205K for 2-4 STHE.

Hence, 14 input variables are $m_{h,init}$, $m_{c,init}$, $T_{h,in}$, $T_{c,in}$, $P_{h,in}$, $P_{c,in}$, N_{HE} , N_S , d_i , t_t , t_s , ΔT_{min} , η_t and η_f . Out of these variables, to check for condition with parallel or series arrangement and size of STHE, N_{HE} , N_S and d_i were varied.

4.5 STHE design calculation procedure

This section explains calculations used in the thesis.

4.5.1 Impact of dust

Design basis specifies presence of dust in the gas. Task description specifies dust as lime (CaO). Hence, impact of dust is first evaluated. As discussed in section 2.3, potential impact of dust can be checked by calculating solid loading ratio (ϕ) and by evaluating Mohs harness of dust particles.

Solid loading ratio (ϕ)

Mass flow rate of CO₂ ($m_{h,init}$) = 67000 kg/hr = 18.61 kg/sec

Molecular mass of CO₂ (M_{CO_2}) = 44 gm/mol

Temperature of inlet CO₂ ($T_{h,in}$) = 900°C

Pressure of inlet CO₂ ($P_{h,in}$) = 1 bar

Dust concentration in CO₂ gas ($C_{h,init}$) = 50 gm/Nm³

Standard pressure (P_{NTP}) = 101325 Pa

Standard temperature (T_{NTP}) = 293.15 K

Volumetric flow rate of inlet CO₂ gas ($V_{h,in}$) [by rearranging ideal gas law]

$$V_{h,in} = \frac{m_{h,init} \times R_G \times T_{h,in}}{M_{CO_2} \times P_{h,in}} = 41.3 \text{ m}^3/\text{sec}$$

Here, R_G is universal gas constant.

Actual mass flow rate of solid particles (m_{PM}) after adjusting for gas flow rate in normal temperature pressure condition can be given by following equation:

$$m_{PM} = C_{h,init} \times V_{h,in} \times \frac{P_{h,in} \times T_{NTP}}{P_{NTP} \times T_{h,in}} = 0.509 \text{ kg/sec}$$

Solid loading ratio is the ratio between mass flow rate of dust and mass flow rate of gas. Hence, solid loading ratio (ϕ) can be determined from the formula below:

$$\phi = \frac{m_{PM}}{m_{h,init}} = 0.03$$

Since $\phi \ll 5$, the flow of CO₂ is highly dilute flow. Hence, risk of fouling and blockage is low. Moreover, by using solid loading ratio in equation (2.1) it can be proved that additional pressure drop due to solid particles is low and can be neglected.

Mohs hardness

Mohs hardness of lime (CaO) read from Figure 2.10 specifies it around 3.5. Since the flow is dilute and Mohs hardness is not too high, abrasiveness risk from solid particles can also be neglected for design condition.

4.5.2 Fitting properties in an equation

Variation of properties of the gases with respect to temperature is usually given in tabular form. But to read these properties in computer, a mathematic expression of the properties is more convenient.

Various properties of CO₂, Inconel 718 and air at different temperature is shown in Table 4.6, Table 4.7 and Table 4.8 respectively.

Table 4.6: Thermal properties of CO₂ at different temperatures [22]

Temperature (K)	Specific heat capacity (kJ/kgK)	Viscosity (Ns/m ²)	Conductivity (W/mK)
500	1.02	2.31E-05	3.25E-02
550	1.05	2.51E-05	3.66E-02
600	1.08	2.70E-05	4.07E-02
650	1.1	2.88E-05	4.45E-02
700	1.13	3.05E-05	4.81E-02
750	1.15	3.21E-05	5.17E-02
800	1.17	3.37E-05	5.51E-02

Table 4.7: Thermal properties of Inconel 718 [20]

Temperature, °F	Thermal Conductivity, ^a BTU•in/ft ² •h•°F		Electrical Resistivity, ^a ohm circ mil/ft		Mean Linear Expansion ^{b,c} , in/in/°F x 10 ⁻⁶
	Ann. 1800°F/1 hr	Ann. + Aged	Ann. 1800°F/1 hr	Ann + Aged	
-320	-	-	-	-	5.9 ^d
70	77	79	753	725	-
200	86	87	762	733	7.31
400	98	100	772	755	7.53
600	111	112	775	768	7.74
800	123	124	784	775	7.97
1000	135	136	798	788	8.09
1200	147	148	805	794	8.39
1400	160	161	802	797	8.91
1600	173	173	799	796	-
1800	185	186	801	800	-
2000	196	199	811	796	-

^aAnnealing was 1800°F/1 hr; aging was 1325°F/8 hr, F.C. 20°/hr to 1150°F, held at 1150°F for total aging time of 18 hr. Conductivity calculated from resistivity values.

^bFrom 70°F to temperature shown.

^cAnnealed 1750°F/1 hr and aged 1325°F/8 hr, F.C. to 1150°F/8 hr, A.C.

^dSamples tested were in both the annealed (1750°F/1 hr, A.C.) and annealed and aged (1750°F/1 hr + 1325°F/8 hr, F.C. to 1150°F, held at 1150°F for 10 hr, A.C.) conditions.

Table 4.8: Thermal properties of air at different temperatures [22]

Temperature (K)	Specific heat capacity (kJ/kgK)	Viscosity (Ns/m ²)	Conductivity (W/mK)
500	1.03	2.70E-05	4.07E-02
550	1.04	2.88E-05	4.39E-02
600	1.051	3.06E-05	4.69E-02
650	1.063	3.23E-05	4.97E-02
700	1.075	3.39E-05	5.24E-02
750	1.087	3.55E-05	5.49E-02
800	1.099	3.70E-05	5.73E-02
850	1.11	3.84E-05	5.96E-02
900	1.121	3.98E-05	6.20E-02
950	1.131	4.11E-05	6.43E-02
1000	1.141	4.24E-05	6.67E-02
1100	1.159	4.49E-05	7.15E-02
1200	1.175	4.73E-05	7.63E-02

Plotting these properties against temperature shows almost a straight line. Hence, all the properties are fitted in a straight line with the help of excel solver functionality. The straight line is represented by equation (4.3) and fitted parameters is shown in Table 4.9.

$$y = mx + c \quad (4.3)$$

Table 4.9: Properties fitted in a straight line

x	y	m	c
Temperature (°C)	Specific heat capacity of CO ₂ (C_{p,CO_2}) (kJ/kgK)	5E-04	9.1E-01
Temperature (°C)	Viscosity of CO ₂ (μ_h) (Ns/m ²)	3.53E-08	1.53E-05
Temperature (°C)	Conductivity of CO ₂ (k_h) (W/mK)	7.53E-05	1.58E-02
Temperature (°C)	Specific heat capacity of air ($C_{p,air}$) (kJ/kgK)	2.14E-04	9.8E-01
Temperature (°C)	Viscosity of air (μ_c) (Ns/m ²)	2.9E-08	2.13E-05
Temperature (°C)	Conductivity of air (k_c) (W/mK)	5E-05	3.05E-02
Temperature (°C)	Conductivity of Inconel 718 (k_m) (W/mK)	1.6E-02	1.09E+01

4.5.3 Calculation flow sheet

Design basis specifies inlet conditions of the gases. Based on the inlet conditions, STHE thermo-hydraulic design parameters must be calculated.

Discussion of input variables are done in section 4.4. Based on the input variables, a systematic procedure to calculate design parameters was developed. The developed procedure in form of flow sheet is shown in Figure 4.1.

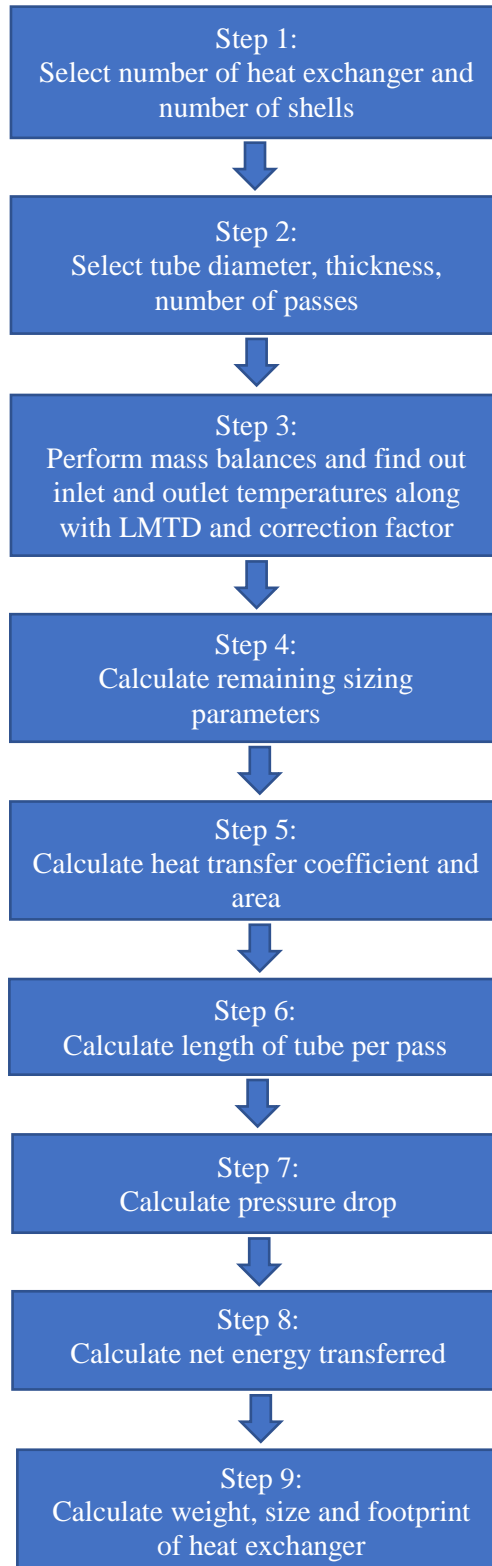


Figure 4.1: Calculation procedure for determination thermo-hydraulic parameters of shell and tube heat exchanger

4.5.4 Python code for calculation

Based on the calculation procedure, a code was developed in python 3.6 to calculate thermo-hydraulic design parameters. Properties of gas was determined from curve fitted equation discussed in section 4.5.2.

As discussed in section 4.4, N_{HE} , N_S and d_i was varied in the code and “pandas” module in python was used to print the results in an excel sheet. All results are shown in Appendix E and some results are discussed in section 6. Result includes heat exchanger design parameters and estimated cost parameters.

The developed code is attached in Appendix F.

4.5.5 Calculation example

Calculation example is presented for the case where tube internal diameter is 51mm and four 2-4 shell and tube exchangers are kept in parallel. A step by step calculation for the flow sheet shown in section 4.5.3 is presented.

Step 1:

Number of heat exchanger (N_{HE}) = 4

Number of shells (N_S) = 2

Step 2:

Tube internal diameter (d_i) = 0.051 m

Tube thickness (t_t) = 0.0027 m

Thickness of each shell (t_s) = 0.0079 m

Number of passes (n) = 4

Step 3:

Since hot fluid is placed in tube side and cold fluid is placed in shell side as discussed in section 4.3.3,

Mass flow of CO₂ (tube side) (m_t) = $\frac{m_{h,init}}{N_{HE}}$ 16750 kg/hr = 4.65 kg/sec [Refer equation (2.24)]

Mass flow of air (shell side) (m_s) = $\frac{m_{c,init}}{N_{HE}}$ = 17750 kg/hr = 4.93 kg/sec [Refer equation (2.25)]

Guessed specific heat capacity of CO₂ ($C_{p,h}$) = 1.17 kJ/kgK (at 800°C)

Guessed specific heat capacity of air ($C_{p,c}$) = 1.03 kJ/kgK (at 500°C)

CO₂ inlet temperature ($T_{h,in}$) = 900°C = 1173.15 K [Refer section 4.1]

Air inlet temperature ($T_{c,in}$) = 225°C = 498.15 K [Refer section 4.1]

ΔT_{min} = 205 K [Refer section 4.4]

When comparing, $m_t \times C_{p,h} = 5.44 > m_s \times C_{p,c} = 5.08$ [Refer section 2.5.6]

Hence,

Air outlet temperature ($T_{c,out}$) = $T_{h,in} - \Delta T_{min} = 968.15 \text{ K}$ (or $695 \text{ }^\circ\text{C}$) [Refer equation (2.28)]

Hence average temperature of cold side,

$$T_{c,avg} = \frac{T_{c,in} + T_{c,out}}{2} = 460^\circ\text{C}$$

At this temperature $C_{p,c}$ was determined from the following curve fitted equation [Refer Table 4.9]:

$$C_{p,c} = 2.14 \times 10^{-4} \times T_{c,avg} + 0.9834 = 1.082 \text{ kJ/kgK}$$

Heat rate (q) = $m_s \times C_{p,c} \times (T_{c,out} - T_{c,in}) = 2.51 \text{ MW}$ [Refer equation (2.27)]

CO₂ outlet temperature ($T_{h,out}$) [Rewriting equation (2.26)]:

$$T_{h,out} = T_{h,in} - \frac{q}{m_t \times C_{p,h}} = 439.13^\circ\text{C}$$

Hence average temperature of hot side ($T_{h,avg}$),

$$T_{h,avg} = \frac{T_{h,in} + T_{h,out}}{2} = 669.6^\circ\text{C}$$

At this temperature $C_{p,h}$ was determined from the following curve fitted equation [Refer Table 4.9]:

$$C_{p,h} = 5 \times 10^{-4} \times T_{h,avg} + 0.91 = 1.246 \text{ kJ/kgK}$$

CO₂ outlet temperature ($T_{h,out}$) was calculated again as:

$$T_{h,out} = T_{h,in} - \frac{q}{m_h \times C_{p,h}} = 467.24^\circ\text{C}$$

The procedure was repeated to find the following parameters:

$$T_{h,out} = 470.15^\circ\text{C}, C_{p,h} = 1.253 \frac{\text{kJ}}{\text{kgK}} \text{ and } T_{h,avg} = 685.1^\circ\text{C}$$

Rechecking calculation condition, $m_t \times C_{p,h} = 5.83 > m_s \times C_{p,c} = 5.33$. Hence, calculation condition remains unchanged.

Logarithmic Mean Temperature difference (ΔT_{LMTD}) [Refer equation (2.30)],

$$\Delta T_{LMTD} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln \left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}} \right)} = 224.47$$

Correction factor (F_T) [Refer equations (2.32) and (2.33)],

$$R = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{h,out}} = 1.093$$

$$S = \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} = 0.64$$

$$F_T = \frac{\frac{\sqrt{R^2 + 1}}{2(R + 1)} \ln\left(\frac{1 - S}{1 - RS}\right)}{\ln\left(\frac{\frac{2}{S} - 1 - R + \frac{2}{S}\sqrt{(1 - S)(1 - RS)} + \sqrt{R^2 + 1}}{\frac{2}{S} - 1 - R + \frac{2}{S}\sqrt{(1 - S)(1 - RS)} - \sqrt{R^2 + 1}}\right)} = 0.8$$

Step 4:

Diameter of shell (D_s) = $40 \times d_i = 2.04$ m [Refer equation (2.2)]

Tube external diameter (d_o) = $d_i + (2 \times t_t) = 0.0564$ m

Tube pitch (S_t) = $1.25 \times d_o = 0.0705$ m [Refer equation (2.4)]

Baffle length (B) = $D_s = 2.04$ m [Refer equation (2.3)]

Shell cross sectional area (A_s) [Refer equation (2.6)]

$$A_s = \frac{D_s}{N_s} \times B \times \left(1 - \frac{d_o}{S_t}\right) = 0.416 \text{ m}^2$$

Number of tubes (N_t) [Refer equation (2.5)]

$$N_t = \frac{\pi D_s^2}{4 S_t^2} = 658$$

Equivalent diameter in shell (d_e) for square pitch [Refer equation (2.7)]

$$d_e = \frac{4}{\pi d_o^2} \times \left(S_t^2 - \frac{\pi d_o^2}{4}\right) = 0.056 \text{ m}$$

Step 5:

Inlet pressure of hot CO₂ ($P_{h,in}$) and air ($P_{c,in}$) = 1 bar = 10^5 Pa

Molecular mass of CO₂ (M_h) = 0.044 kg/mol

Molecular mass of air (M_c) = 0.029 kg/mol

Density of hot CO₂ (ρ_t) [Ideal gas law]

$$\rho_t = \frac{P_{h,in} M_h}{R T_{h,avg}} = 0.55 \text{ kg/m}^3$$

Density of cold air (ρ_s) [Ideal gas law]

$$\rho_s = \frac{P_{c,in} M_c}{R T_{c,avg}} = 0.48 \text{ kg/m}^3$$

Now various other properties were determined with help of Table 4.9.

Thermal conductivity of CO₂ (k_t) was determined using curve fitted equation:

$$k_t = 7.53 \times 10^{-5} \times T_{h,avg} + 0.0158 = 0.067 \text{ W/mK}$$

Thermal conductivity of air (k_s) was determined using curve fitted equation:

$$k_s = 5 \times 10^{-5} \times T_{c,avg} + 0.0305 = 0.053 \text{ W/mK}$$

Viscosity of CO₂ (μ_t) was determined using curve fitted equation:

$$\mu_t = 3.53 \times 10^{-8} \times T_{h,avg} + 1.53 \times 10^{-5} = 3.95 \times 10^{-5} \text{ Nsec/m}^2$$

Viscosity of air (μ_s) was determined using curve fitted equation:

$$\mu_s = 2.9 \times 10^{-8} \times T_{c,avg} + 2.13 \times 10^{-5} = 3.47 \times 10^{-5} \text{ Nsec/m}^2$$

Thermal conductivity of Inconel 718 (k_m) was determined using curve fitted equation:

$$k_m = 0.016 \times T_{h,avg} + 10.87 = 21.83 \text{ W/mK}$$

Velocity in tube (v_t) [Refer equation (2.20)]

$$v_t = \frac{4 \times m_t \times n}{\pi d_i^2 \times \rho_t \times N_t} = 25.07 \text{ m/sec}$$

Reynold's number in tube (Re_t) [Refer equation (2.18)]

$$Re_t = \frac{\rho_t \times d_i \times v_t}{\mu_t} = 17875.39$$

Prandlt's Number in tube (Pr_t) [Refer equation (2.17)]

$$Pr_t = \frac{C_{p,CO2} \times \mu_t}{k_t} = 0.735$$

Tube side heat transfer coefficient (h_t) [Refer equation (2.14)]

$$h_t = \frac{k_t \left(\frac{f}{8}\right) (Re - 1000) Pr}{d_i \left(1 + 12.7 \sqrt{\frac{f}{8}} (Pr^{\frac{2}{3}} - 1)\right)} = 63.77 \text{ W/m}^2\text{K}$$

Velocity in shell (v_s) [Refer equation (2.21)]

$$v_s = \frac{m_c}{\rho_c \times A_s} = 24.9 \text{ m/sec}$$

Reynold's number in shell (Re_s) [Refer equation (2.19)]

$$Re_s = \frac{\rho_s \times d_e \times v_s}{\mu_s} = 19066.4$$

Prandlt Number in shell (Pr_s) [Refer equation (2.17)]

$$Pr_s = \frac{C_{p,air} \times \mu_s}{k_s} = 0.701$$

Shell side heat transfer coefficient (h_s) [Refer equation (2.16)]

$$h_s = 0.036 \times \frac{k_s}{d_e} \times Re_s^{0.55} \times Pr_s^{0.3} = 70.1 \text{ W/m}^2\text{K}$$

Overall heat transfer coefficient (U) [Refer equation (2.22)]

$$U = \frac{1}{\frac{1}{h_s} + \frac{d_o}{2k_m} \ln\left(\frac{d_o}{d_i}\right) + \frac{d_o}{d_i} \times \frac{1}{h_t}} = 31.51 \text{ W/m}^2\text{K}$$

Heat transfer area (A) [Refer equation (2.34)]

$$A = \frac{q}{U \times \Delta T_{LMTD} \times F_T} = 442.14 \text{ m}^2$$

Step 6:

Length of tube (L) [Refer equation (2.8)]

$$L = \frac{A}{\pi d_i N_t} = 4.19 \text{ m}$$

Step 7:

Darcy friction factor (f_t) [Refer equation (2.13)]

$$f_t = (1.82 \log_{10}^{Re_t} - 1.64)^{-2} = 0.027$$

Pressure drop across tube (ΔP_t) [Refer equation (2.35)]

$$\Delta P_t = \frac{\rho_h v_t^2}{2} \left(\frac{L}{d_i} f_t + 4 \right) n = 0.043 \text{ bar}$$

Friction factor in shell (f_s) [Refer equation (2.37)]

$$f_s = 2b_o Re_s^{-0.15} = 0.33$$

Pressure drop across shell (ΔP_s) [Refer equation (2.36)]

$$\Delta P_s = f_s \left(\frac{\rho_c v_s^2}{2} \right) \left(\frac{L}{B} \right) \left(\frac{D_s}{d_e} \right) = 0.036 \text{ bar}$$

Step 8:

Since hot fluid is in tube side and cold fluid is in shell side, pressure compensation equipment (CRF or TB) is placed at the outlet for tube side and at the inlet for shell side as discussed in section 2.5.7.

Hence, Isentropic work on tube side [Refer equation (2.39)],

$$W_{is,h} = m_t \times C_{p,CO_2} \times T_{h,out} \times \left[\left(\frac{P_{h,in}}{P_{h,in} - \Delta P_t} \right)^{R_G / C_{p,CO_2}} - 1 \right] = 0.029 \text{ MW}$$

Assume efficiency of fan (η_f) = 0.75 and thermodynamic efficiency (η_t) = 0.4,

Actual work on tube side [Refer equation (2.40)],

$$W_h = \frac{W_{is,h}}{\eta_f} = 0.039 \text{ MW}$$

Heat loss due to pressure drop on tube side [Refer equation (2.41)],

$$H_{loss,h} = \frac{W_h}{\eta_t} = 0.096 \text{ MW}$$

Isentropic work on shell side [Refer equation (2.42)],

$$W_{is,c} = m_s \times C_{p,air} \times T_{c,out} \times \left[\left(\frac{P_{c,in}}{P_{c,in} - \Delta P_s} \right)^{R_G / C_{p,air}} - 1 \right] = 0.049 \text{ MW}$$

Actual work on shell side [Refer equation (2.43)],

$$W_c = \frac{W_{is,c}}{\eta_{fan}} = 0.066 \text{ MW}$$

Heat loss due to pressure drop on shell side [Refer equation (2.44)],

$$H_{l,c} = \frac{W_c}{\eta_{thermo}} = 0.16 \text{ MW}$$

Total heat loss ($H_{l,t}$) = $H_{l,h} + H_{l,c} = 0.26 \text{ MW}$ [Refer equation (2.45)]

Net energy transferred (q_{net}) = $(q - H_{l,t}) \times N_{HE} = 8.99 \text{ MW}$ [Refer equation (2.46)]

Step 9:

Volume of tube material (V_{tm}) [Refer equation (2.48)],

$$V_{tm} = \pi d_i L \times N_t \times t_t = 1.93 \text{ m}^3$$

Mass of tube (m_{tm}) [Refer equation (2.49)],

$$m_{tm} = \rho_{tm} * V_{tm} = 9781.1 \text{ kg}$$

Volume of shell material (V_{sm}) [Refer equation (2.50)],

$$V_{sm} = t_s \times \left[(\pi D_s L) + \left(\frac{\pi D_s^2}{4} \right) \right] = 0.24 \text{ m}^3$$

Mass of shell (m_{sm}) [Refer equation (2.51)],

$$m_{sm} = \rho_{sm} * V_{sm} = 1951.3 \text{ kg}$$

Time spend by CO₂ inside the tube side of HE (t_{tt}) [Refer equation (2.53)],

$$t_{tt} = \frac{L \times n}{v_t} = 0.67 \text{ sec}$$

Mass of CO₂ inside tube at a given time (m_{tg}) [Refer equation (2.52)],

$$m_{tg} = m_t \times t_{tt} = 2.73 \text{ kg}$$

Time spend by air inside the shell side of HE (t_{ts}) [Refer equation (2.55)],

$$t_{ts} = \frac{L \times N_s}{v_s} = 0.34 \text{ sec}$$

Mass of air inside shell at a given time (m_{sg}) [Refer equation (2.54)],

$$m_{sg} = m_s \times t_{ts} = 1.66 \text{ kg}$$

Mass of each heat exchanger during operation (m_{HE}) [Refer equation (2.47)],

$$m_{HE} = (m_{tm} + m_{sm} + m_{tg} + m_{sg}) \times N_{HE} = 46.95 \text{ ton}$$

Total volume occupied by all heat exchanger (V_{HE}) [Refer equation (2.56)],

$$V_{HE} = \frac{\pi D_s^2 L}{4} \times N_{HE} = 54.83 \text{ m}^3$$

Footprint for each heat exchanger (F_{HE}) [Refer equation (2.57)],

$$F_{HE} = L \times D_s \times N_{HE} = 34.22 \text{ m}^2$$

5 Cost Estimation

This section presents method and example calculation for cost estimation.

5.1 Calculation basis

Based on design parameters calculated in chapter 4, cost of STHE and fan is estimated. This is then compared with cost of buying electrical energy to determine net present value and equivalent annual cost.

Cost of electricity in Norway is shown in Table 5.1.

Table 5.1: Electricity price in Norway [23]

Electricity prices in the end-user market, quarterly. Øre/kWh			
	4th quarter 2018	Change in per cent	
	Øre/KwH	Last 3 mos.	Last 12 mos.
Households. Total price of electricity, grid rent and taxes	123.4	0.0	25.4
Electricity price	54.8	-0.9	52.2
Grid rent	29.4	1.7	5.4
Taxes	39.2	0.0	13.6
Households. Electricity price by type of contract. Exclusive taxes			
New fixed-price contracts-1 year or less ¹	51.3	17.7	67.6
New fixed-price contracts-1 year or more ¹	43.5	4.1	26.8
All other fixed-price contracts	34.2	5.9	11.8
Contracts tied to spot price	53.1	-2.2	52.1
Variable price (not tied to spot price)	61.2	2.2	56.9
Business activity. Electricity price. Exclusive taxes			
Services	50.7	-0.6	54.6
Manufacturing excl. energy-intensive manufacturing	49.4	0.0	52.0
ManufacturiEnergy-intensive manufacturing	31.8	-1.2	10.4

¹ New fixed-price contracts are entered during the measuring period, and older fixed-price contracts are entered earlier.

Since the production of cement is highly energy intensive, from Table 5.1 cost of electricity ($C_{el,kwh}$) was assumed to be 0.318 NOK/kWh.

Moreover, capacity factor method and detailed factor method was used to estimate STHE and fan cost. Calculation was done using python code presented in Appendix E.

But before estimation of cost, capacity factor and material factor must be determined based on the selected material.

5.2 Capacity factor determination

According to design condition, equipment of different capacity is required. Hence, capacity factor must be determined to estimate cost of equipment. Data from cost estimation website for capacity factor estimation is given in Table 5.2.

Table 5.2: Equipment data from cost prediction website [17]

Equipment	Material	Size	Unit	Year	Currency	Cost
Turbo blower	---	0.69, 5	bar, m ³ /sec	2002	USD	150,274
Turbo blower	---	0.69, 14	bar, m ³ /sec	2002	USD	280,512
Centrifugal radial fan	---	5	m ³ /sec	2002	USD	5,259
Centrifugal radial fan	---	35	m ³ /sec	2002	USD	22,363
STHE	CS	200	m ²	2002	USD	18,568
STHE	CS	600	m ²	2002	USD	39,051

Capacity factor for STHE ($c_{f,HE}$) [Refer equation (3.1)]

$$c_{f,HE} = \frac{\log(\text{cost of } STHE_1 / \text{cost of } STHE_2)}{\log(\text{capacity of } STHE_1 / \text{capacity of } STHE_2)} = 0.68$$

Capacity factor for turbo blower ($c_{f,tb}$),

$$c_{f,tb} = \frac{\log(\text{cost of } TB_1 / \text{cost of } TB_2)}{\log(\text{capacity of } TB_1 / \text{capacity of } TB_2)} = 0.61$$

Capacity factor for centrifugal radial fan ($c_{f,rf}$),

$$c_{f,rf} = \frac{\log(\text{cost of } CRF_1 / \text{cost of } CRF_2)}{\log(\text{capacity of } CRF_1 / \text{capacity of } CRF_2)} = 0.74$$

5.3 Material factor determination

It was assumed that STHE are manufactured by welding metal plates together. Hence cost of plates was determined in Table 5.3 to predict material factor cost. Data for a reference STHE was also found and is shown in Table 5.4.

Table 5.3: Estimation of cost of metal plates to form shell and tubes. Thickness of plates was chosen such that it corresponds to thickness of shell and tubes

Material	Dimension [mm×mm ×mm]	Unit cost [USD/kg]	Density [kg/m ³]	Volume [m ³]	Mass [kg]	Plate cost [USD]	Reference
Carbon steel for tubes	1000×1000 ×1.65	0.9	7850	0.00165	12.95	11.66	[24]
Carbon steel for shell/ends	1000×1000 ×7.9	0.9	7850	0.0079	62.02	55.81	[24]
Inconel 718 for tubes	1000×1000 ×1.65	30	8193	0.00165	13.52	405.6	[21]
Inconel 718 for shell/ends	1000×1000 ×7.9	30	8193	0.0079	64.72	1942	[21]

Table 5.4: Data of reference heat exchanger [17]

Equipment	Material	Size	Unit	Year	Currency	Cost
Heat exchanger	CS	388.33	m ²	2002	USD	29,094

Some more data of reference STHE other than given in Table 5.4 [17],

Tube external diameter of reference STHE ($d_{o,rHE}$) = 0.019 m

Length of tube of reference STHE (L_{rHE}) = 6.1 m

Tube pitch of reference STHE ($S_{t,rHE}$) = 0.0254 m

Tube thickness of reference STHE ($t_{t,rHE}$) = 0.00165 m

Shell thickness of reference STHE ($t_{s,rHE}$) = 0.0079 m

From Table 5.4,

Cost of reference STHE (C_{rHE}) = 29094 USD

Heat transfer area of reference STHE (A_{rHE}) = 388.33 m²

Calculations for reference STHE parameters,

Tube internal diameter for reference STHE ($d_{i,rHE}$) = $d_{o,rHE} - (2 \times t_{t,rHE}) = 0.0157$ m

Number of tubes for reference STHE ($N_{t,rHE}$),

$$N_{t,rHE} = \frac{A_{rHE}}{\pi d_{i,rHE} L_{rHE}} = 1291$$

Diameter of the shell for reference STHE ($D_{s,rHE}$) from rearranging equation (2.5),

$$D_{s,rHE} = \sqrt{\frac{4S_{t,rHE}^2 N_{t,rHE}}{\pi}} = 1.03 \text{ m}$$

Tube side surface area of reference STHE ($A_{t,rHE}$) is equal to heat transfer area of reference STHE. Hence,

$$A_{t,rHE} = A_{rHE} = 388.33 \text{ m}^2$$

Shell side surface area of reference STHE ($A_{s,rHE}$),

$$A_{s,rHE} = \pi D_{s,rHE} L_{rHE} = 19.73 \text{ m}^2$$

Surface area at ends of HE ($A_{e,SHE}$),

$$A_{e,rHE} = \frac{\pi D_{s,rHE}^2}{2} = 1.67 \text{ m}^2$$

Calculations of carbon steel (CS) cost,

Surface area of tube type CS sheet ($A_{tm,CS}$) = 1 m² [From Table 5.3]

Surface area of shell/end type CS sheet ($A_{sm,CS}$) = 1 m² [From Table 5.3]

Cost of each tube type CS sheet ($C_{tm,CS}$) = 11.66 USD [From Table 5.3]

Cost of each shell/end type CS sheet ($C_{sm,CS}$) = 55.81 USD [From Table 5.3]

Number of tube type CS sheets required for tube surface ($N_{tm,CS}$),

$$N_{tm,CS} = \frac{A_{t,rHE}}{A_{tm,CS}} = 389$$

Number of shell type CS sheets required for shell surface ($N_{sm,CS}$),

$$N_{sm,CS} = \frac{A_{s,rHE}}{A_{sm,CS}} = 20$$

Number of shell type CS sheets required for shell ends ($N_{em,CS}$),

$$N_{em,CS} = \frac{A_{e,rHE}}{A_{sm,CS}} = 2$$

Conversion factor from USD to NOK (c_{fc}) was assumed to be 8.5.

Table 5.5: Inflation from 2002-2019 [25]

Year	USD Value
2002	\$100.00
2019	\$140.51

Using inflation data shown in Table 5.5,

US dollar cost index in 2002 (C_{2002}) = \$100

US dollar cost index in 2019 (C_{2019}) = \$140.51

Cost index ratio between 2019 and 2002 (c_{CI}),

$$c_{CI} = \frac{C_{2019}}{C_{2002}} = 1.405$$

Current cost of reference STHE made of CS material ($C_{rHE,CS}$),

$$C_{rHE,CS} = C_{rHE} \times c_{CI} \times c_{fc} = 346.21 \text{ kNOK}$$

Hence updated data for reference STHE made of CS material is shown in Table 5.6.

Table 5.6: Updated data for reference STHE made of CS

Equipment	Material	Size	Unit	Year	Currency	Cost
STHE	CS	388.33	m ²	2019	kNOK	346.21

Total cost of CS material for STHE construction (C_{MCS}),

$$C_{MCS} = [(C_{tm,CS} \times N_{tm,CS}) + (C_{sm,CS} \times N_{sm,CS}) + (C_{em,CS} \times N_{em,CS})] \times c_f = 48.98 \text{ kNOK}$$

Fabrication cost (C_{FC}) = $C_{rHE,CS} - C_{MCS} = 297.23$ kNOK

Calculations of Inconel 718 cost,

Surface area of tube type Inconel 718 sheet ($A_{tm,IN}$) = 1 m²

Surface area of shell/end type Inconel 718 sheet ($A_{sm,IN}$) = 1 m²

Cost of each tube type Inconel 718 sheet ($C_{tm,IN}$) = 405.6 USD

Cost of each shell/end type Inconel 718 sheet ($C_{sm,IN}$) = 1942 USD

Number of tube type Inconel 718 sheets required ($N_{tm,IN}$),

$$N_{tm,IN} = \frac{A_{t,rHE}}{A_{tm,IN}} = 389$$

Number of shell type Inconel 718 sheets required for curved surface area ($N_{sm,IN}$),

$$N_{sm,IN} = \frac{A_{s,rHE}}{A_{sm,IN}} = 20$$

Number of shell type Inconel 718 sheets required for curved surface area ($N_{em,IN}$),

$$N_{em,IN} = \frac{A_{e,rHE}}{A_{sm,IN}} = 2$$

Total cost of IN material for STHE construction (C_{MIN}),

$$C_{MIN} = [(C_{tm,IN} \times N_{tm,IN}) + (C_{sm,IN} \times N_{sm,IN}) + (C_{em,IN} \times N_{em,IN})] \times c_{fc} \\ = 1704.1 \text{ kNOK}$$

Cost of reference STHE made of IN material ($C_{rHE,IN}$),

$$C_{rHE,IN} = C_{MIN} + C_{FC} = 2001.3 \text{ kNOK}$$

Material factor of any material is ratio between equipment cost constructed with that material and equipment cost constructed with reference material (CS used in DFT). Hence, material factor of IN compared to CS (m_f),

$$m_f = \frac{C_{rHE,IN}}{C_{rHE,CS}} = 5.78$$

5.4 Cost calculation procedure

Once capacity factor and material factor is determined, capacity factor method and detailed factor method is used to determine the cost of STHE and fans. Net present value (NPV) is then found out by comparing it to electricity cost in 10 years and finally equivalent annual cost is determined.

As discussed in section 4.5.4, cost calculations are included in python code. The calculation procedure used in the code is shown in this section. STHE design parameters determined in section 4.5.5 is used to determine NPV and equivalent annual cost (EAC) for the example calculation.

From STHE design parameter calculations in section 4.5.5,

Mass flow rate of CO₂ in each STHE (m_t) = 4.65 kg/sec

Mass flow rate of air in each STHE (m_s) = 4.93 kg/sec

Density of hot CO₂ (ρ_t) = 0.55 kg/m³

Density of cold air (ρ_s) = 0.48 kg/m³

Area of each STHE (A) = 442.14 m²

Pressure drop across tube for each STHE (ΔP_t) = 0.043 bar

Pressure drop across shell for each STHE (ΔP_s) = 0.036 bar

Volumetric flow rate of hot fluid for each STHE (V_h),

$$V_h = \frac{m_t}{\rho_t} = 8.43 \text{ m}^3/\text{sec}$$

Volumetric flow rate of cold gas for each STHE (V_c),

$$V_c = \frac{m_s}{\rho_s} = 10.36 \text{ m}^3/\text{sec}$$

Calculation procedure for estimation of cost is shown in Figure 5.1.

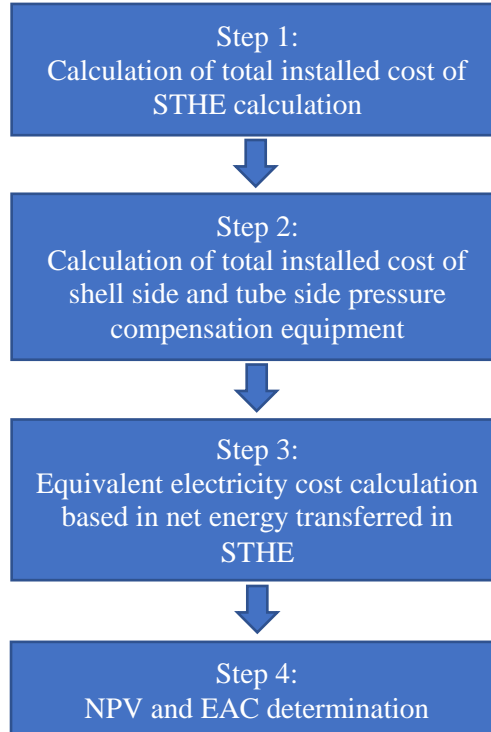


Figure 5.1: Calculation procedure for estimation of cost

5.4.1 STHE cost calculation

Material factor of Inconel 718 (IN) to carbon steel (CS) (m_f) = 5.78

From the table standard equipment data,

Heat transfer area of reference STHE (A_{rHE}) = 388.33 m² [Refer Table 5.6]

Cost of reference carbon steel STHE ($C_{rHE,CS}$) = 346.21 kNOK [Refer Table 5.6]

Capacity factor for STHE ($c_{f,HE}$) = 0.68

After adjusting for capacity, cost of STHE made of CS (C_{HECS}) [Refer equation (3.1)]

$$C_{HECS} = \left(\frac{A}{A_{rHE}} \right)^{c_{f,HE}} \times C_{rHE,CS} = 378.15 \text{ kNOK}$$

Total installed cost factor for CS material ($f_{TIC,CS}$) = 9.13 [Refer Table 3.1]

Reading value of material factor for equipment ($f_{EQ,CS}$) and piping ($f_{PI,CS}$) from Table 3.1 and adjusting it for material.

Total installed cost factor of heat exchanger for Inconel 718 (f_{TIC}) [Refer equation (3.3)]

$$f_{TIC} = f_{TIC,CS} - f_{EQ,CS} + (f_{EQ,CS} \times m_f) - f_{PI,CS} + (f_{PI,CS} \times m_f) = 19.26$$

Total installed cost of all 2-4 STHE made of IN (C_{HE}),

$$C_{HE} = C_{HECS} \times f_{TIC} \times N_{HE} = 29.14 \text{ MNOK}$$

5.4.2 Pressure compensation equipment cost calculation

Along tube side,

Pressure drop across tube for each STHE (ΔP_t) = 0.043 bar

Since ΔP_t is less than 0.1 bar centrifugal radial fan (CRF) is used [Discussed in section 2.5.7]

Table 5.7: Data from cost prediction website [17]

Equipment	Material	Size	Unit	Year	Currency	Cost
Turbo blower (TB)	---	0.69, 5	bar, m ³ /sec	2002	USD	150,274
Centrifugal radial fan (CRF)	---	5	m ³ /sec	2002	USD	5,259

Reading data from Table 5.7,

Volumetric flow rate of reference CRF ($V_{r,crf}$) = 5 m³/sec

Cost of reference CRF ($C_{r,crf}$) = 5259 USD

Capacity factor for centrifugal radial fan ($c_{f,rf}$) = 0.74 [Refer section 5.2]

Conversion factor from USD to NOK (c_{fc}) = 8.5

Cost index from 2002 to 2019 (c_{CI}) = 1.405 [Refer section 5.3]

After adjusting for time, currency and capacity, cost of tube side CRF ($C_{t,i,crf}$),

$$C_{t,i,crf} = \left(\frac{V_h}{V_{r,crf}} \right)^{c_{f,rf}} \times C_{r,crf} \times c_f \times c_{CI} = 92.4 \text{ kNOK}$$

Total installed cost factor for tube side CRF (f_{TIC}) = 15.03 [Refer Table 3.1]

Since each STHE require one CRF on tube side as discussed in section 2.5.7, total installed cost of all tube side CRF ($C_{t,crf}$),

$$C_{t,crf} = C_{t,crf} \times f_{TIC} \times N_{HE} = 5.56 \text{ MNOK}$$

Along shell side,

Pressure drop across shell for each heat exchanger (ΔP_s) = 0.036 bar

Since ΔP_s is less than 0.1 bar centrifugal radial fan is used [Discussed in section 2.5.7]

Volumetric flow rate of cold fluid for each HE (V_c) = 10.36 m³/sec

After adjusting for time, currency and capacity, cost of shell side CRF ($C_{s,i,crf}$),

$$C_{s,i,crf} = \left(\frac{V_c}{V_{r,crf}} \right)^{c_{f,rf}} \times C_{r,crf} \times c_f \times c_{CI} = 107.7 \text{ kNOK}$$

Total installed cost factor for tube side centrifugal radial fan (f_{TIC}) = 9.13 [Refer Table 3.1]

Since each STHE require one CRF on shell side as discussed in section 2.5.7, total installed cost of all shell side CRF ($C_{s,crf}$),

$$C_{s,crf} = C_{s,i,crf} \times f_{TIC} \times N_{HE} = 3.93 \text{ MNOK}$$

5.4.3 Equivalent electricity cost calculation

Cost of electricity in Norway ($C_{el,kwh}$) = 0.318 NOK/kWhr

Number of working hours per year ($n_{hr,yr}$) = 7315 [Refer Table 4.1]

For 2-4 heat exchanger,

Net energy transferred (q_{net}) = 8.99 MW [Refer section 4.5.5]

Thermal efficiency of conversion from heat to electricity (η_t) = 0.4

Energy quantification in terms of electricity is required to calculate cost of purchasing same amount of electricity. Since q_{net} is given in thermal energy, equivalent electrical energy ($q_{net,el}$) from this thermal energy is,

$$q_{net,el} = q_{net} \times \eta_{thermo} = 3.6 \text{ MW}$$

Cost of electricity per kWh ($C_{el,kwh}$) = 0.318 NOK/kWh [Refer section 5.1]

Hence cost of equivalent electricity per year ($C_{el,yr}$),

$$C_{el,yr} = C_{el,kwh} \times q_{net,el} \times n_{hr,yr} = 8.36 \text{ MNOK/year}$$

5.4.4 Net present value calculation

Assume rate of interest (i) = 8%

Number of years to buy electricity (n_{yr}) = 10

NPV of 10 years of electricity (C_{el}), [Refer equation (3.4)]

$$C_{el} = C_{el,yr} + \frac{C_{el,yr}}{(1+i)^1} + \frac{C_{el,yr}}{(1+i)^2} + \dots + \frac{C_{el,yr}}{(1+i)^{n_{yr}-1}} = 60.6 \text{ MNOK}$$

Here, $C_{el,yr}$ is the non-discounted cash flow as it is the cash flow per year.

Finally, overall Net present value (NPV) = $C_{el} - C_{HE} - C_{t,crf} - C_{s,crf} = 21.97 \text{ MNOK}$

5.4.5 Equivalent annual cost calculation

Annuity factor (a_f) [Refer equation (3.6)]

$$a_f = \frac{1 - \frac{1}{(1+i)^{n_{yr}}}}{i} = 6.71$$

NPV of capital expenditure (NPV_{CAPEX}) is the total installed cost (TIC) of all equipment. This is given by the sum of cost of all STHEs, tube side CRF and shell side CRF. Hence,

$$NPV_{CAPEX} = TIC = C_{HE} + C_{t,crf} + C_{s,crf} = 38.6 \text{ MNOK}$$

Hence, equivalent annual cost for capital investment (EAC_{CAPEX}) [Refer equation (3.7)],

$$EAC_{CAPEX} = \frac{NPV_{CAPEX}}{a_f} = 5.76 \text{ MNOK/year}$$

NPV of operational expenditure (NPV_{OPEX}) is assumed only to be electricity cost (C_{el}) which is saved. Hence, equivalent annual cost for operational cost (EAC_{OPEX}) [Refer equation (3.8)],

$$EAC_{OPEX} = \frac{-C_{el}}{a_f} = -9.03 \text{ MNOK/year}$$

6 Results and discussions

Calculation method using python is discussed in section 4.5.4. Python code results and its discussions are presented in this section. Internal diameter value of 0.051m showed the most promising results. Hence, results presented in this section contains only variation in STHE structure and number of STHE kept in parallel, while internal diameter was kept constant at 0.051m. Additional results with other internal diameters can be found in Appendix E.

Sensitivity of NPV with variation in design parameters is also presented in this section.

6.1 Design results and discussions

Variation of velocity in tubes (v_t), velocity in shell (v_s), overall heat transfer coefficient (U), heat duty in each STHE (q) heat transfer area (A), pressure drop along tube (P_t), pressure drop along shell (P_s), net energy transferred (q_{net}), total weight of all STHE (m_{HE}), total volume occupied by all STHE (V_{HE}) and total footprint of all STHE (F_{HE}) when structure and number of STHE in parallel is varied is shown in Figure 6.1, Figure 6.2, Figure 6.3, Figure 6.4, Figure 6.5, Figure 6.6, Figure 6.7, Figure 6.8, Figure 6.9, Figure 6.10 and Figure 6.11 respectively.

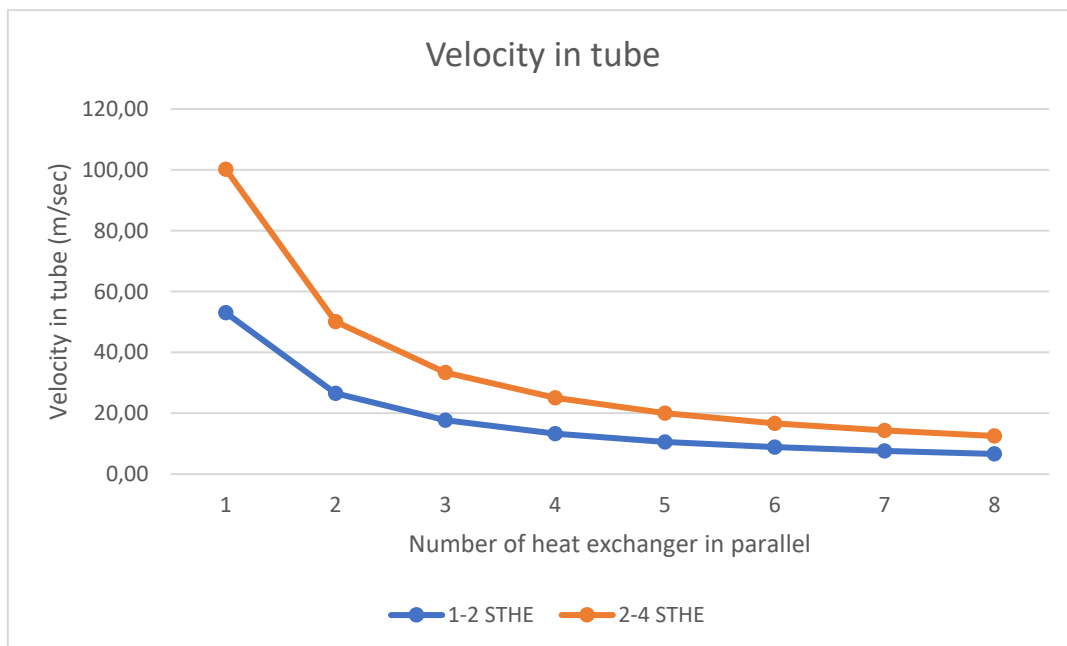


Figure 6.1: Change in fluid velocity in tube when structure of STHE and number of STHE in parallel is varied

Velocity of inside the tube (v_t) increases when structure of STHE is changed from 1-2 STHE to 2-4 STHE. This is because when number of passes increases for the same shell diameter and number of tubes in each pass reduces. This results in reduced flow area for the same amount of gas flow. Hence, velocity of the gas increases. When number of STHE in parallel is increased, amount of gas flowing in each STHE decreases. Hence, the velocity of gas decreases too.

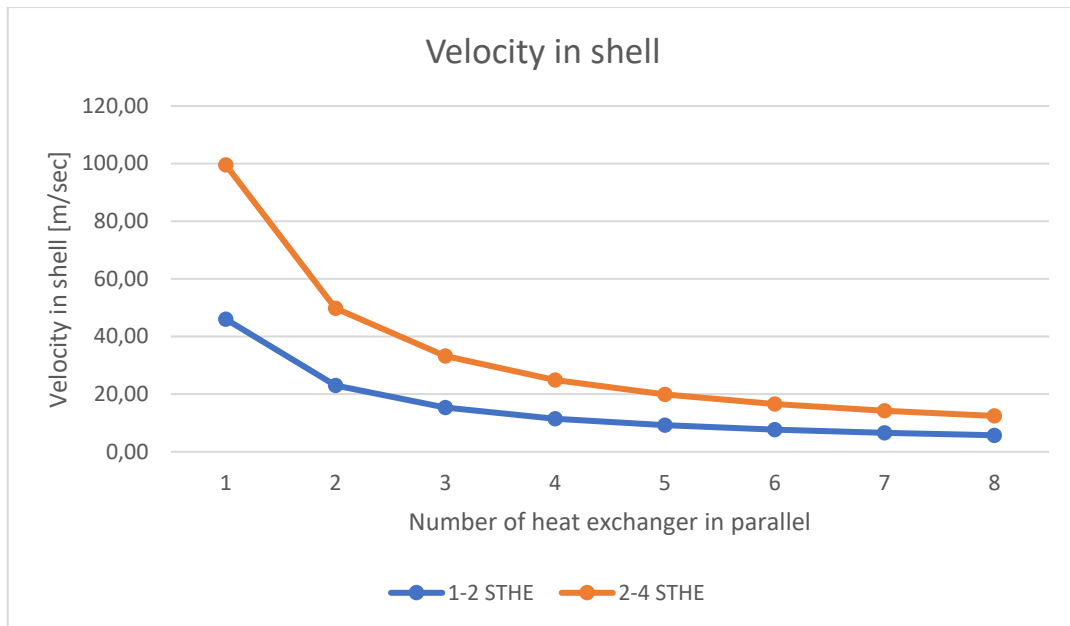


Figure 6.2: Change in fluid velocity in shell when structure of STHE and number of STHE in parallel is varied

Velocity of gas inside the shell (v_s) increases when structure of STHE is changed from 1-2 STHE to 2-4 STHE. Changing the structure is like adding a longitudinal baffle inside 1-2 STHE. This results in reduced flow area in shell side. Hence, velocity inside the shell increases. Decrease in velocity with increase in number of STHE in parallel is because of reduced mass flow of air in each STHE.

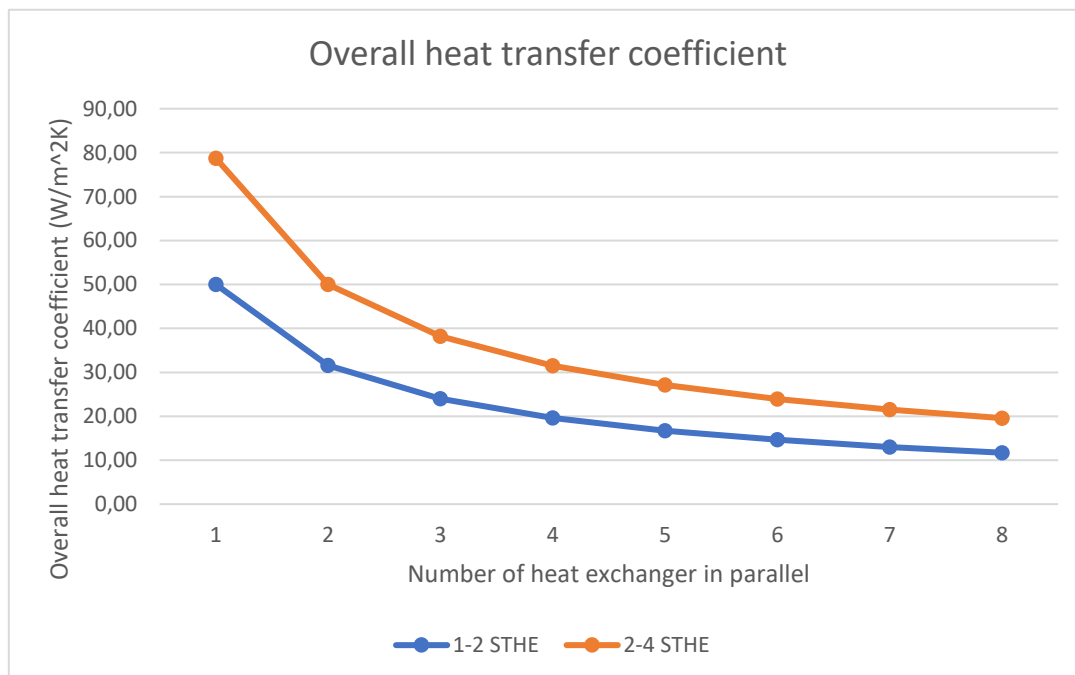


Figure 6.3: Change in overall heat transfer coefficient when structure of STHE and number of STHE in parallel is varied

Since, the velocity in both shell and tube increases when structure of STHE is changed from 1-2 STHE to 2-4 STHE, the heat transfer coefficient on both shell side and tube side increases. Hence, overall heat transfer coefficient also increases. Similarly, since the velocity in both tube and shell side decreases when number of STHE in parallel increases, overall heat transfer coefficient also reduces.

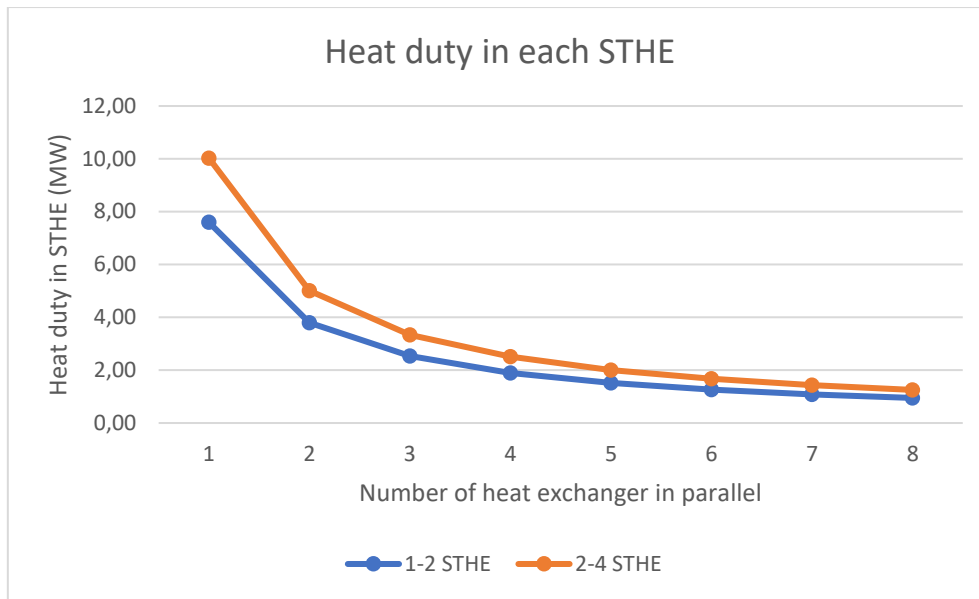


Figure 6.4: Change in duty of each STHE when structure of STHE and number of STHE in parallel is varied

When the structure of STHE is changed from 1-2 STHE to 2-4 STHE, the efficiency of heat transfer area increases too. This is represented by LMTD correction factor (F_T). But the calculations are done assuming constant F_T and instead improving temperature gradient [refer section 4.4]. Since, the temperature gradient is higher in 2-4 STHE, energy available for heat transfer is higher too. Hence, heat duty increases when structure is changed from 1-2 STHE to 2-4 STHE. But when number of STHE in parallel is increased, mass flow of gas in each STHE decreases. Hence, heat available for transfer reduces which leads to lower duty in each STHE.

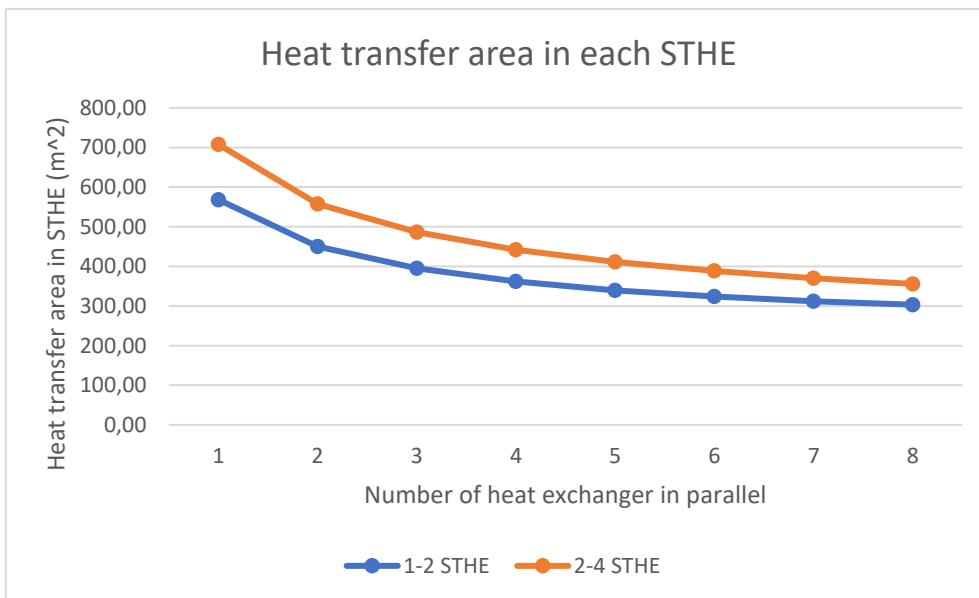


Figure 6.5: Change in heat transfer area when structure of STHE and number of STHE in parallel is varied

Changing structure of STHE from 1-2 STHE to 2-4 STHE, increases overall heat transfer coefficient and duty on each STHE. Increase in overall heat transfer coefficient leads to lower heat transfer area as conduction and convection effect improves but increase in duty leads to higher heat transfer area as larger area is required to transfer higher energy. But, the effect of increased heat duty is higher. Hence, heat transfer area increases.

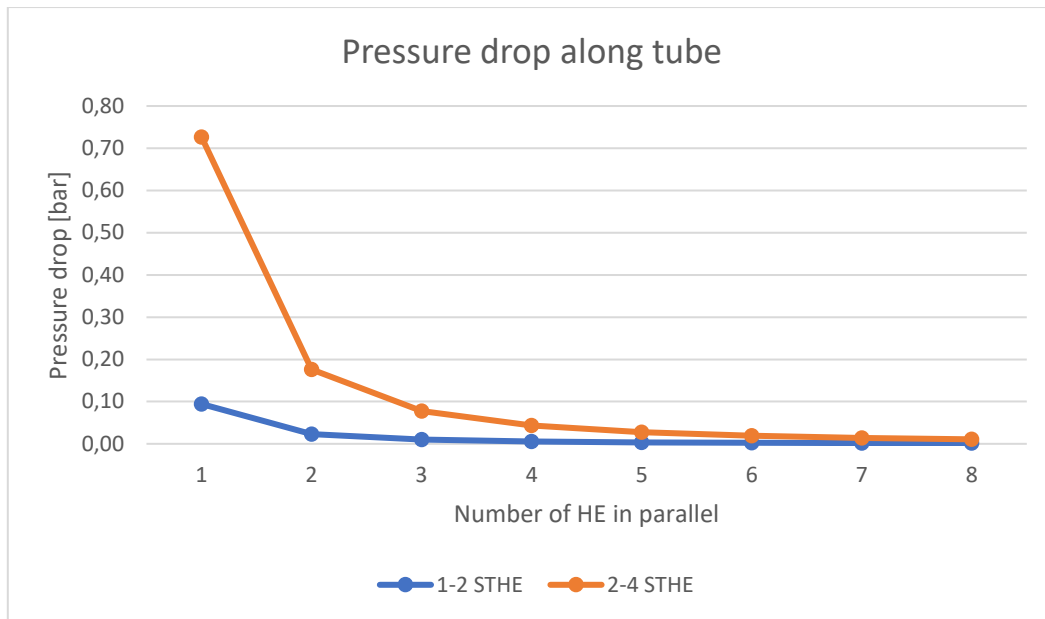


Figure 6.6: Change in pressure drop along tube when structure of STHE and number of STHE in parallel is varied

Pressure drop depends on many different variables. Pressure drop increase along tube when the structure of STHE is changed from 1-2 STHE to 2-4 STHE, is mainly due to increase in number of pass and velocity of gas. Addition of passes increases number of 180° turns and increased velocity results in decreased pressure according to Bernoulli's principle. Hence, pressure drop is much higher in 2-4 STHE. Similarly, since velocity decreases when number of STHE in parallel increases, pressure drop decreases as well.

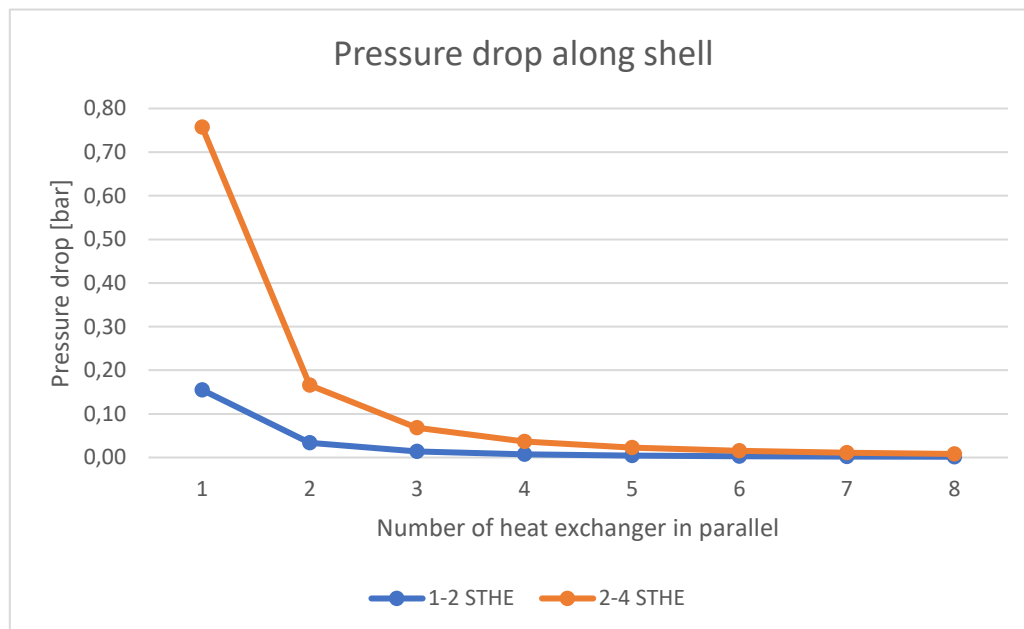


Figure 6.7: Change in pressure drop along shell when structure of STHE and number of STHE in parallel is varied

Pressure drop along shell increases when the structure is changed from 1-2 STHE to 2-4 STHE in similar way as pressure drop along tube. Increase in velocity along shell is the main reason for this. Similarly, decrease in pressure drop along shell when number is STHE in parallel is increased, is mainly due to decrease in velocity of gas in shell.

Pressure drops along shell and tube with one STHE alone of both structure and two 2-4 STHE was found to be higher than 0.1 bar. Hence, the centrifugal radial fan cannot be used. The alternative could be to use a turbo blower. But turbo blower has volumetric flow rate limitations of 15 m³/sec as discussed in section 2.5.7. Volumetric flow rate of hot gas in tube was found to be higher than this limitation value for all the 3 cases. Hence, these cases were assumed to be technically infeasible. So, further calculations for these 3 cases is not performed.

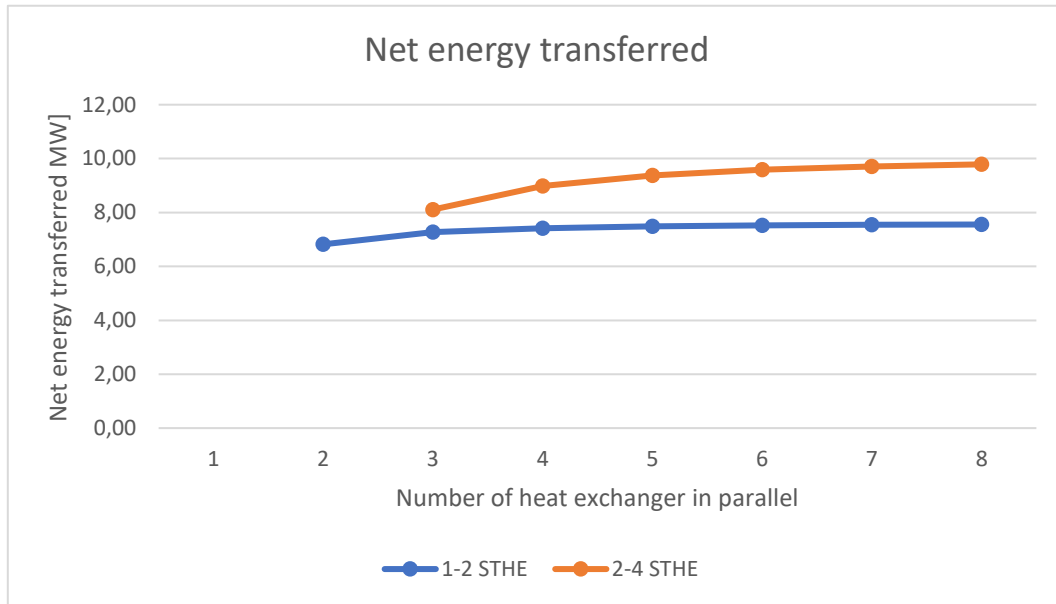


Figure 6.8: Change in net energy transferred when structure of STHE and number of STHE in parallel is varied

Net energy transferred is the balance between heat duty of STHE and energy losses to recover pressure drop in the system. When energy losses are higher than STHE duty then the project is not sustainable. Increase in net energy transferred with increase in number of STHE in parallel is mainly because of pressure drop decrease.

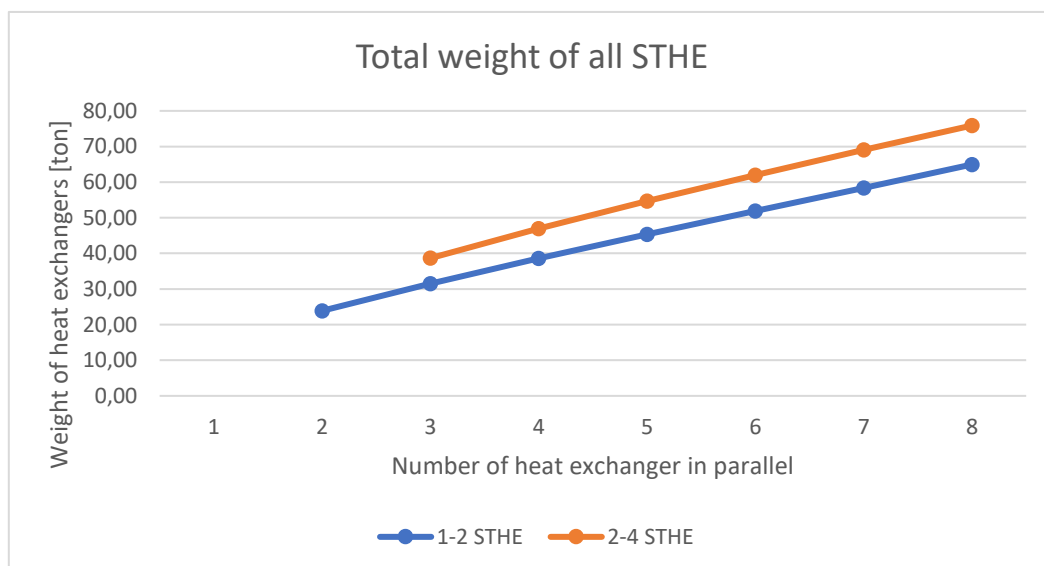


Figure 6.9: Change in weight of all heat exchangers when structure of STHE and number of STHE in parallel is varied

When the number of STHE in parallel is increased, the total weight of STHE increases too as more STHE is used. Due to bigger construction in 2-4 STHE, its weight is more than 1-2 STHE.

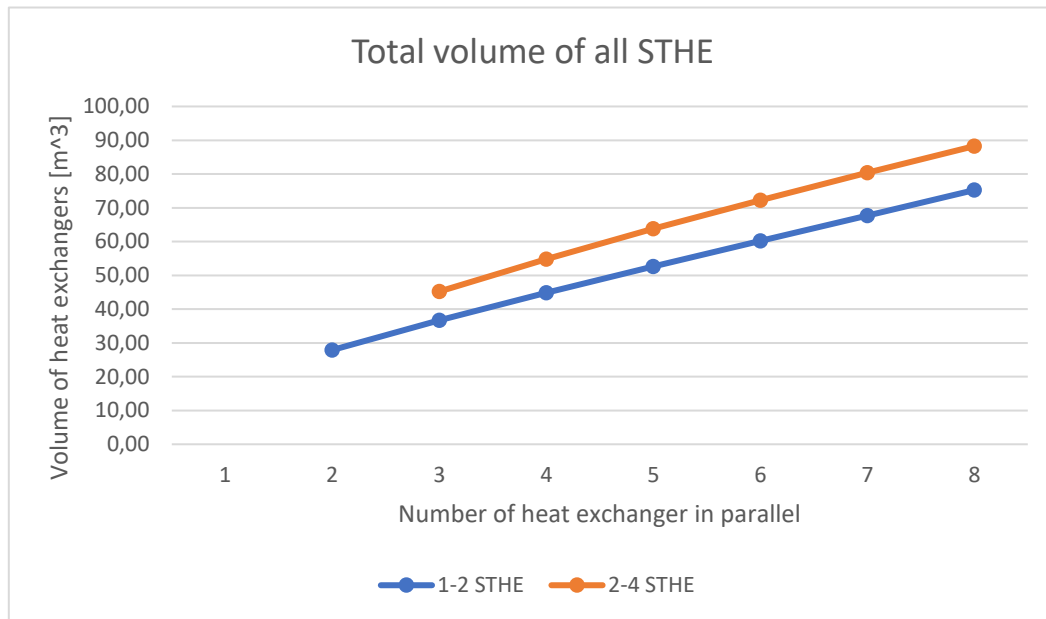


Figure 6.10: Change in total volume occupied by all STHE when structure of STHE and number of STHE in parallel is varied

Diameter of shell in both 1-2 STHE and 2-4 STHE is same but thermo-hydraulic calculation gives larger length in 2-4 STHE due to higher heat transfer area. Hence, the volume occupied by 2-4 STHE is also higher. Moreover, since number of STHE in parallel increases units of STHE, hence total volume occupied also increases.

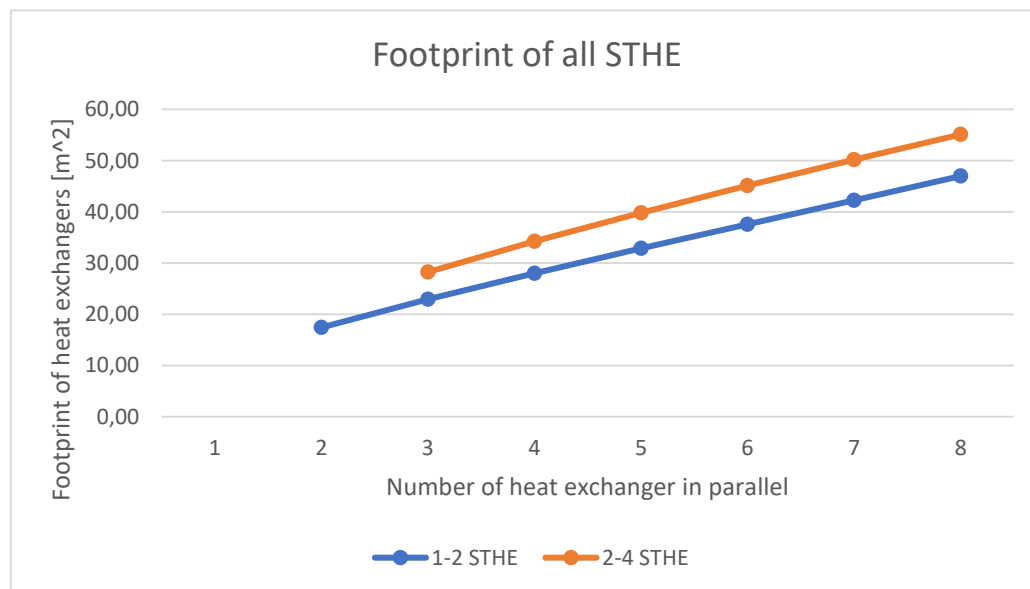


Figure 6.11: Change in total footprint occupied by all STHE when structure of STHE and number of STHE in parallel is varied

Similar to discussions in volume occupied, since the length of STHE from 2-4 STHE is higher, footprint is also higher and since number of units increases with number of STHE, footprint occupied by all STHE also increases.

6.2 Cost estimation results and discussions

Variation of total installed cost, net present value and equivalent annualized cost when structure and number of STHE in parallel is varied is shown in Figure 6.12, Figure 6.13 and Figure 6.14 respectively. As discussed in section 6.1, cost calculations were not performed for cases with one STHE of both structure in parallel and two 2-4 STHE in parallel as they were technically infeasible.

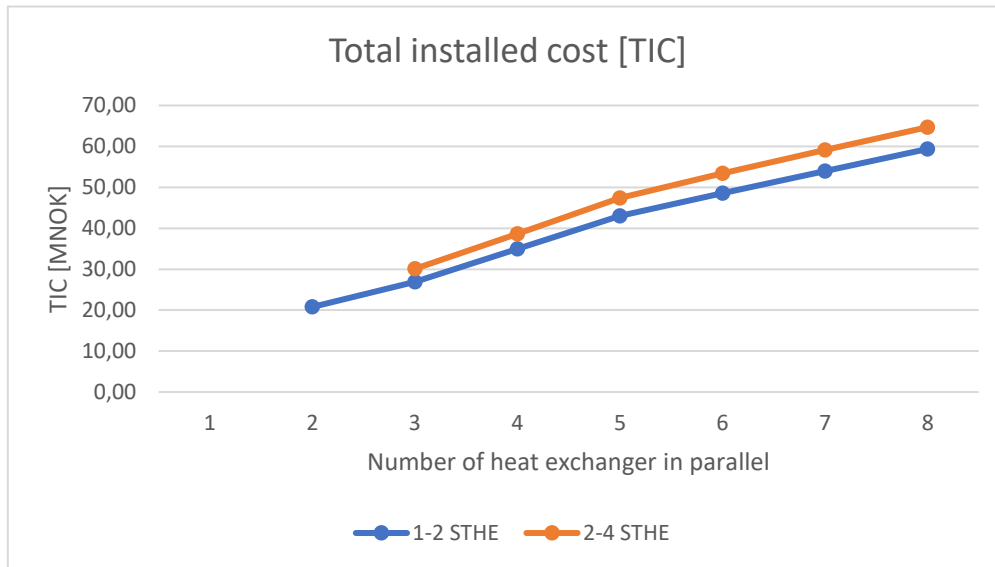


Figure 6.12: Change in total installed cost of all equipment when structure of STHE and number of STHE in parallel is varied

Total installed cost (TIC) gives the total installed cost of all STHEs and fans used. 2-4 STHEs are bigger in size and have more heat transfer area than 1-2 STHE. Hence total installed cost for 2-4 STHE is higher than 1-2 STHE. Moreover, TIC also increases with addition of extra STHE in parallel as it adds additional units of STHE.

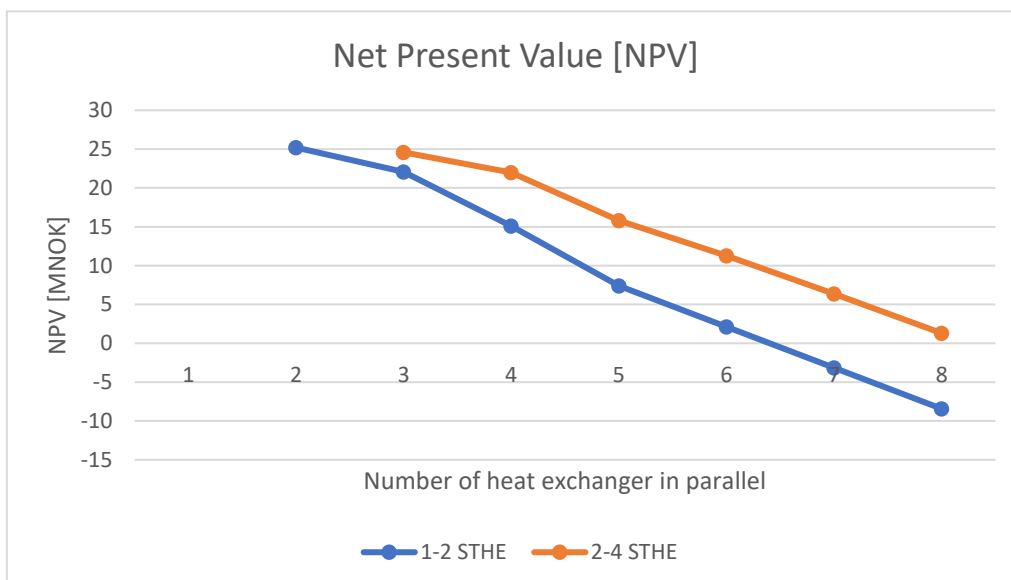


Figure 6.13: Change in NPV when structure of STHE and number of STHE in parallel is varied

Net present value considers saving from energy transferred in STHE and total installed cost of equipment. This parameter is generally optimized as when more STHE is kept in parallel,

energy savings increases, but cost of equipment goes higher. So, the most optimal point should be determined. It was found that case of 3 parallel 2-4 STHE gave the highest NPV.

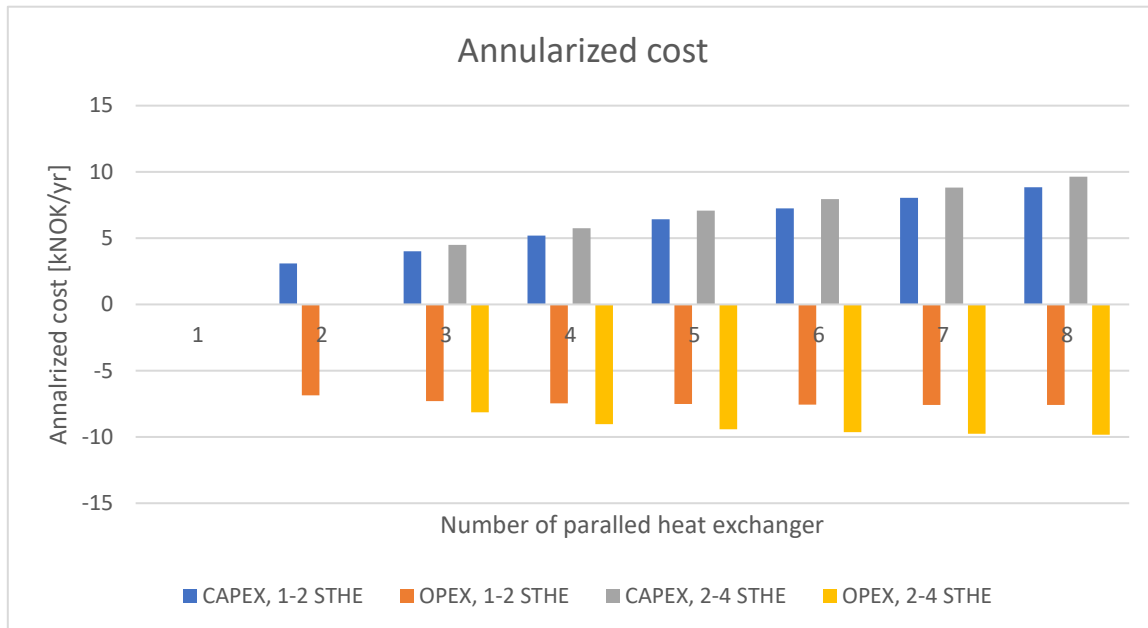


Figure 6.14: Change in equivalent annualized cost when structure of STHE and number of STHE in parallel is varied

A better understanding of difference between CAPEX and OPEX can be seen from analysing equivalent annualized cost. From the plot it's clear that CAPEX keeps on going higher but OPEX has a maximum limitation. Maximum limitation on OPEX corresponds to maximum energy saved from using STHE.

Case of 2 parallel 1-2 STHE gave highest NPV of 25.2 MNOK. For 2-4 STHE structure, highest NPV was 24.59 MNOK with 3 parallel 2-4 STHE.

To understand the effect of utilizing turbo blower as pressure compensation equipment, variation of NPV with variation in internal diameter for case of 3 parallel 2-4 STHE is shown in Table 6.1.

Table 6.1: Variation of NPV with variation in internal tube diameter for 3 parallel 2-4 STHE. "N/A" represents case which is technically infeasible

Tube internal diameter (d_i) [m]	Net Present Value (NPV) [MNOK]
0.02	N/A
0.025	N/A
0.038	-90.11
0.051	24.59

NPV has a negative value when internal diameter is 0.038m as shown in Table 6.1. This is mainly because of requirement of turbo blower (TB) for pressure compensation. Hence, high cost of TB makes total installed cost of all equipment much higher than energy saved from STHE.

6.3 Sensitivity Analysis

Even though case with 2 parallel 1-2 STHE gave best NPV, case scenario of 3 parallel 2-4 STHE with NPV of 24.59 MNOK was chosen as basis for sensitivity analysis. Higher heat transfer duty in 2-4 STHE and low difference of NPV between both cases was the reason for this. Two different tests are performed. In the first test, some parameters are increased and decreased to 10% of its value and its effect on NPV is plotted in Figure 6.15. In second test, more realistic variation in different parameters is assumed and its effect on NPV is shown in Figure 6.16.

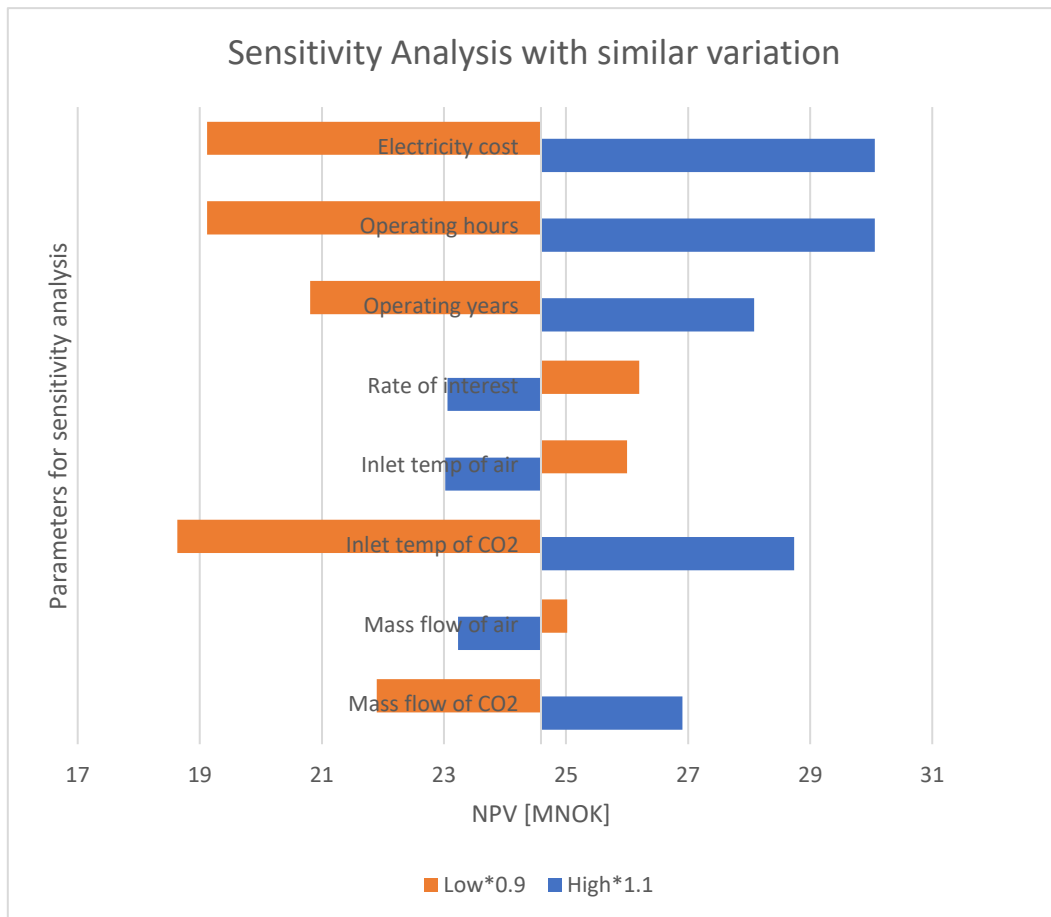


Figure 6.15: Sensitivity of NPV when different variables are varied upto $\pm 10\%$

Inlet temperature of CO₂, electricity cost, operating hours, operating years and mass flow rate of CO₂ were found to have high impact on sensitivity when all the parameters were increased by same percentage as shown in Figure 6.15. Main reason for change in NPV with each parameter can be discussed as:

1. Electricity cost

Higher the electricity cost, higher is the savings from heat transfer. Hence, NPV increases and vice versa. Since it directly affects the NPV, sensitivity is high.

2. Operating hours

Higher the operating hours, more the energy is saved. Since it directly influences the amount to energy to be bought, sensitivity is high.

3. Operating years

Higher the operating year higher the NPV as it directly affects the total cost of electricity. It is less sensitive than operating hours. Operating hours affects cost of

electricity per year. But operating year affects total cost of electricity. Since the value of money reduces with time, hence operating years is less sensitive than operating hours.

4. Rate of interest

When the interest rate is less, NPV increases as the depreciation of value of money reduces. Hence, it has opposite effect.

5. Inlet temperature of air

Higher temperature of air leads to reduction in STHE duty as the temperature gradient reduces. Hence, increased temperature gives lower NPV value

6. Inlet temperature of CO₂

When inlet temperature is increased, duty of heat exchanger increases. Since duty is directly related to cost savings, it is highly sensitive.

7. Mass flow of air

When mass flow of air increases, quantity of cold gas increases. Hence, duty of STHE decreases which leads to decrease in NPV. Increased mass flow rate of air also reverses " mC_p " condition discussed in section 2.5.6. Since heat capacity of air is much higher than heat capacity of CO₂ when mass flow rate of air is increased, NPV reduction is much higher than NPV increase when mass flow rate of air is decreased.

8. Mass flow of CO₂

When mass flow of CO₂ increases, quantity of hot gas increases. Hence, duty of STHE increases which leads to increase in NPV. Since, heat capacity of CO₂ remains higher from " mC_p " rule, its sensitivity is much higher than change in mass flow rate of air.

To have a more realistic sensitivity analysis, more realistic variations in parameters was selected and the selected variation is shown in Table 6.2.

Table 6.2: High and low realistic values of important parameters

Sl No	Parameters	Unit	Current Value	High Value	Low Value
1	Mass flow of CO ₂	kg/hr	67000	76136	54818
2	Mass flow of air	kg/hr	71000	80681	58091
3	Inlet temperature of CO ₂	°C	900	950	700
4	Inlet temperature of air	°C	225	250	150
5	Operating Hours	hr	7315	8000	6000
6	Operating years	Yr	10	25	8
7	Electricity Cost	kr/kwh	0.318	1	0.2
8	Rate of interest	----	8%	12%	3%

Selection of values in Table 6.2 was done from intuition and experience. Based on these values, sensitivity analysis was done and the results are shown in Figure 6.16.

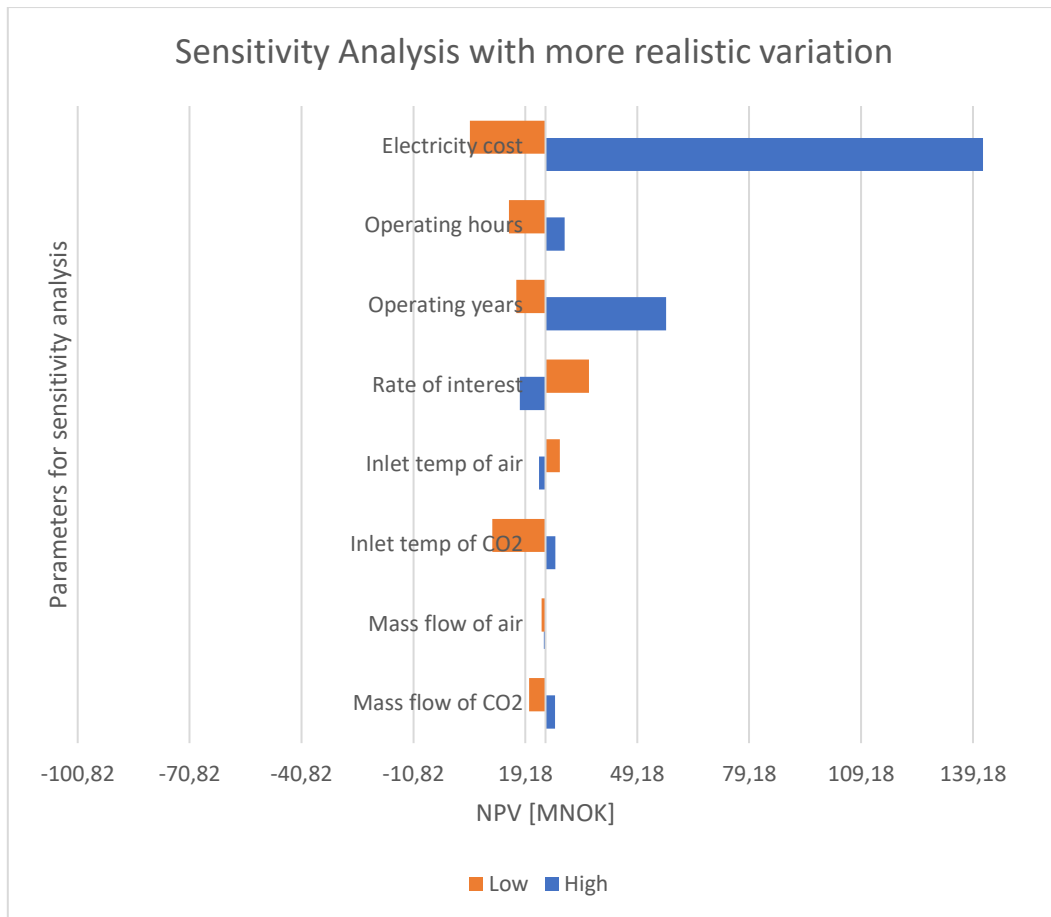


Figure 6.16: Sensitivity of NPV on more realistic variations of various parameters

Electricity cost was found to be most sensitive in real case scenario as seen from Figure 6.16. Thermal efficiency, inlet temperature of CO₂ and operating years were also found to be sensitive in a more real case scenario.

7 Conclusion

Gas to gas heat exchanger to recover heat from hot calcinated gas coming out of electrically heated calciner was studied in this thesis. Challenge was to exchange heat for gas at very high temperature with some dust presence.

Various heat exchangers were studied. Shell and tube heat exchanger and plate fin heat exchanger was found to be feasible for design condition. Shell and tube heat exchanger was chosen to be designed in this thesis as it had a well-established procedure for design and many years of satisfactory service.

Dust concentration in gas was studied and it was found that gas flow was highly dilute. Hence possible problems associated with dust could be neglected for given design basis.

Study of various materials showed that alloys with presence of Molybdenum, Chromium and Nickel could enhance yield strength at elevated temperature. Inconel 718 was found to have all these compounds, and it was also referred in literature for high temperature application. Hence, it was chosen as material of construction.

Kern's method of thermo-hydraulic design was utilized to study thermal performance of STHE. Heat exchanger duty in STHE and energy losses from pressure compensation in STHE were calculated. This was used to determine net energy transferred. Centrifugal radial fan and turbo blower was identified as equipment for pressure compensation. Cases with inability to use both equipment were considered technically infeasible. Study showed that with internal diameter of tube as 0.051m and cases with more than 1 parallel 1-2 STHE or more than 2 parallel 2-4 STHE, design of STHE was technically feasible.

Heat duty for 1-2 STHE was found to be 7.6 MW. Utilizing 2-4 STHE gave a higher duty of 10 MW. Higher efficiency of heat transfer area in 2-4 STHE than 1-2 STHE was the reason for this difference. But this increase in duty comes at an expense of higher pressure drop. Hence, net energy transferred reduces when structure is changed. Placing more STHE in parallel reduced overall pressure drop. Hence, net energy transferred increases by placing more STHE in parallel.

Tests cases up to placing 8 STHE in parallel was done in this thesis. Highest net energy transferred for 1-2 STHE was found to be 7.6 MW and for 2-4 STHE was found to be 9.8 MW. Hence, placing 8 STHE in parallel gave almost negligible losses from pressure compensation.

Study of weight, size and footprint of STHE was done and the results can be helpful while integrating STHE into the system.

Cost estimation was done using capacity factor method and detailed factor method. Net present value (NPV) calculation was done to check economic feasibility. It was done by comparing total installed cost and energy savings of STHE in terms of buying same amount of electrical energy. Positive value of NPV indicated that the project was economically feasible.

Changing STHE structure from 1-2 STHE to 2-4 STHE increases losses due to pressure compensation. Hence, total energy savings decreases. Placing more STHE in parallel reduced the losses and increases energy savings, but it also increases total installed cost. Hence for optimization, NPV was found for all cases and highest NPV gave best case scenario.

NPV depended on test cases and varied between -167 MNOK to 25.2 MNOK for various test cases. Internal tube diameter value of 0.051m gave higher NPV values compared to other

internal diameters. Highest NPV for 1-2 STHE was found to be 25.2 MNOK and this was achieved by placing 2 STHE in parallel. Highest NPV for 2-4 STHE was found to be 24.59 MNOK and this was achieved by placing 3 STHE in parallel.

Sensitivity analysis with equal percentage variation showed that inlet temperature of hot CO₂, operating hours, operating years and electricity cost was highly sensitive to NPV value. A more realistic variation showed electricity cost to be most sensitive for the success of this project.

Hence to conclude, the project was found to both technically and economically feasible.

References

- [1] D. Corporation. (2019, 03.05.2019). *Chemical Process Equipment* [Online]. Available: <http://www.doosan.com/en/business/chemical-process-equipment/heat-exchanger/>.
- [2] M. B. Ali, R. Saidur, and M. S. Hossain, "A review on emission analysis in cement industries," *Renewable and Sustainable Energy Reviews*, vol. 15, no. 5, pp. 2252-2261, 2011/06/01/ 2011.
- [3] K. Thulukkanam, *Heat Exchanger Design Handbook*. Chapman and Hall/CRC, 2013.
- [4] B. K. Dutta, *Heat Transfer Principles and Application*. PHI Learning Private Limited, 2001.
- [5] R. K. Shah and D. P. Sekulic, *Fundamentals of Heat Exchanger Design*. Wiley, 2003.
- [6] C. R. Woodcock and J. S. Mason, *Bulk Solids Handling: An Introduction to the Practice and Technology*. Springer US, 1988.
- [7] L. B. Railsback, *Some Fundamentals of Mineralogy and Geochemistry*. 2019.
- [8] B. Sunden, *Keynote Lecture: High Temperature Heat Exchangers*. 2005.
- [9] E. U. Schlünder, *Heat Exchanger Design Handbook: Vol. 1, "Heat exchanger theory" ; Vol. 2, "Fluid mechanics and heat transfer" ; Vol. 3, "Thermal and hydraulic design of heat exchangers" ; Vol. 4, "Mechanical design of heat exchangers" ; Vol. 5, "Physical properties."*. Hemisphere Publishing Corporation, 1983.
- [10] R. E. Kirk, D. F. Othmer, J. I. Kroschwitz, and M. Howe-Grant, *Encyclopedia of chemical technology*. 1991.
- [11] W. McCabe, J. Smith, and P. Harriott, *Unit Operations of Chemical Engineering*. McGraw-Hill Education, 2005.
- [12] D. Q. Kern, *Process Heat Transfer*. McGraw-Hill, 1950.
- [13] J. M. Campbell, L. L. Lilly, and R. N. Maddox, "Gas conditioning and processing : volume 2 : the equipment modules," (in English), 1984.
- [14] V. K. Patel and R. V. Rao, "Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique," *Applied Thermal Engineering*, vol. 30, no. 11, pp. 1417-1425, 2010/08/01/ 2010.
- [15] F. P. Incropera, *Fundamentals of Heat and Mass Transfer*. John Wiley & Sons Canada, Limited, 1993.
- [16] R. Smith, *Chemical Process: Design and Integration*. John Wiley & Sons, Ltd, 2005.
- [17] Mcgraw-hill. (26.04.2019). *Chemical data* [Online]. Available: <http://www.mhhe.com/engcs/chemical/peters/data/ce.html>.
- [18] J. M. Smith, H. C. Van Ness, and M. Abbott, *Introduction to Chemical Engineering Thermodynamics*. McGraw-Hill Education, 2005.
- [19] "Project Management & Cost Engineering (FM 3110)," USN, Porsgrunn, Exam Paper, 2018.

- [20] S. Metals. (2007, 26.04.2019). *INCONEL alloy 718* [Online]. Available: http://www.specialmetals.com/assets/smc/documents/inconel_alloy_718.pdf.
- [21] Alibaba. (24.04.2019). *Super alloy inconel 718 bright bars* [Online]. Available: https://www.alibaba.com/product-detail/super-alloy-inconel-718-bright-bars_60273852218.html?spm=a2700.7724857.normalList.1.55d36916QWW4QL&s=p.
- [22] T. L. Bergman, A. S. Lavine, and F. P. Incropera, *Fundamentals of Heat and Mass Transfer, 7th Edition*. John Wiley & Sons, Incorporated, 2011.
- [23] S. Norway. (26.04.2019). *Electricity prices* [Online]. Available: <https://www.ssb.no/en/elkraftpris>.
- [24] Alibaba. (26.04.2019). *ASTM Carbon Steel Hot Rolled Round Bar* [Online]. Available: https://www.alibaba.com/product-detail/ASTM-Carbon-Steel-Hot-Rolled-Round_62037605340.html?spm=a2700.7724838.2017115.1.151979af1bR2li&s=p.
- [25] C. I. Calculator. (26.04.2019). *U.S. Inflation Rate* [Online]. Available: <http://www.in2013dollars.com/2002-dollars-in-2018>.

Appendices

Appendix A: Task description

Appendix B: Work breakdown structure

Appendix C: Derivation of equation for number of tubes

Appendix D: Design basis

Appendix E: Additional STHE results

Appendix F: Developed python code

Appendix A: Task description of thesis



Faculty of Technology, Natural Sciences and Maritime Sciences, Campus Porsgrunn

FMH606 Master's Thesis

Title: Gas-to-gas heat exchanger for heat utilization in hot CO₂ from an electrically heated calcination process

USN supervisor: Lars-André Tokheim

External partner: Norcem AS Brevik (Tor Gautestad, Senior Manager Project and Process)

Task background:

USN is one of the partners in an ongoing research project, "Combined calcination and CO₂ capture in cement clinker production by use of CO₂-neutral electrical energy – Phase 1: Technical feasibility and early-phase cost-estimate (6CP)", run by Norcem AS. The goal of the project is to utilize electricity (instead of fuels) to decarbonate the raw meal in the cement kiln process. This will eliminate N₂, H₂O and (excess) O₂ in the calciner exit gas, which will be a more or less pure CO₂ stream. Besides, the fuel generated CO₂ will be eliminated, so less CO₂ is produced in the calcination process. This is expected to make it less expensive (compared with regular post-combustion) to capture a large part of the CO₂ from the plant.

Mass and energy balances made by USN indicate that utilization of the heat in the hot CO₂ stream exiting from the calciner is important for the thermal efficiency and costs of the system. One possibility of such utilization is to transfer the CO₂ heat to a relatively cold air stream, which is subsequently used for direct heating of a cold raw meal stream. The heat exchanger ("CO₂ HEX") implemented in a cement kiln system is shown in Figure 1.

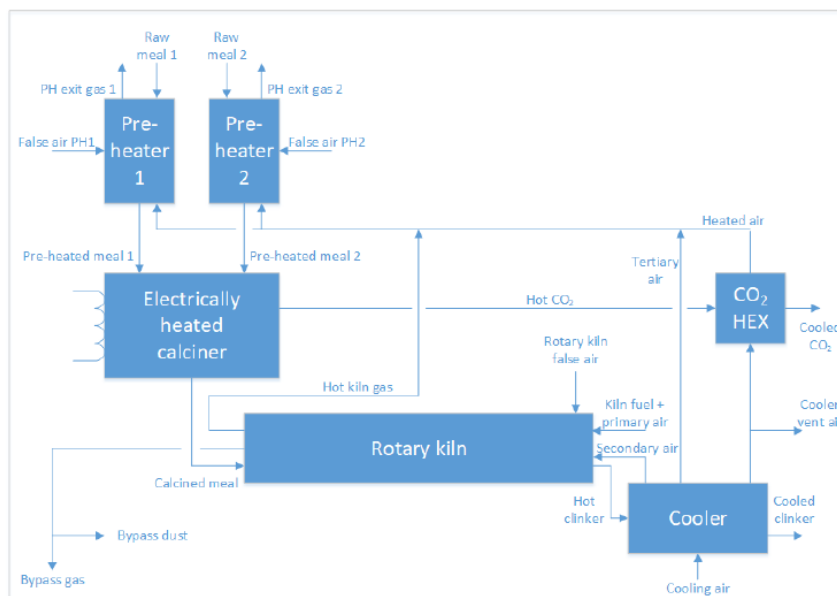


Figure 1: A cement kiln process with two preheater strings, a bypass system, an electrified calciner and a CO₂ heat exchanger.

Address: Kjølnes ring 56, NO-3918 Porsgrunn, Norway. **Phone:** 35 57 50 00. **Fax:** 35 55 75 47.

The CO₂ inlet temperature is around 900 °C, the air inlet temperature is about 200 °C, and the duty of the CO₂ HEX is in the order of 10-15 MW (depending on the outlet temperatures and the flow rates). The thermal resistance in this gas-to-gas heat exchanger will most likely be relatively high, so it is expected that a very large heat exchanger area will be required, suggesting that it will be necessary to have several units in parallel.

The CO₂ stream is expected to have a significant concentration of dust in the form of calcined meal in the size range 1-1000 µm (median ca 30 µm), which may be a challenge for the HEX operation.

Although the duty of CO₂ HEX has been calculated, the heat transfer area and the design of the heat exchanger has not yet been determined. Such information is required to be able to estimate the costs of this process unit.

Task description:

The task could include the following sub-tasks:

- Make a literature review of gas-to-gas heat exchangers in industrial applications and determine what heat exchanger design is preferred
- Assess how the very high temperature of the hot CO₂ may require special attention in the design process
- Evaluate the potential impact of high concentrations of CaO particles in the hot CO₂
- Suggest a heat exchanger design based on given gas flow rates and temperatures
- Calculate the outlet temperatures, the overall heat transfer coefficient and the required heat transfer area
- Estimate the overall size, footprint and weight of the CO₂ HEX
- Estimate the costs of the CO₂ HEX
- Execute a case study to investigate how different values of independent variables will affect the results of the calculations

Student category: EET or PT students

Practical arrangements: Visits to Norcem AS Brevik (~10 km from the USN campus) may be useful or necessary to collect info about the full-scale process.

Supervision:

As a general rule, the student is entitled to 15-20 hours of supervision. This includes necessary time for the supervisor to prepare for supervision meetings (reading material to be discussed, etc).

Signatures:

Supervisor (date and signature):

25.01.2019 

Student (write clearly in all capitalized letters): RON M JAUB

Student (date and signature):

25.01.2019 

Appendix B: Work breakdown structure

Work breakdown structure of thesis is shown in Figure B1.

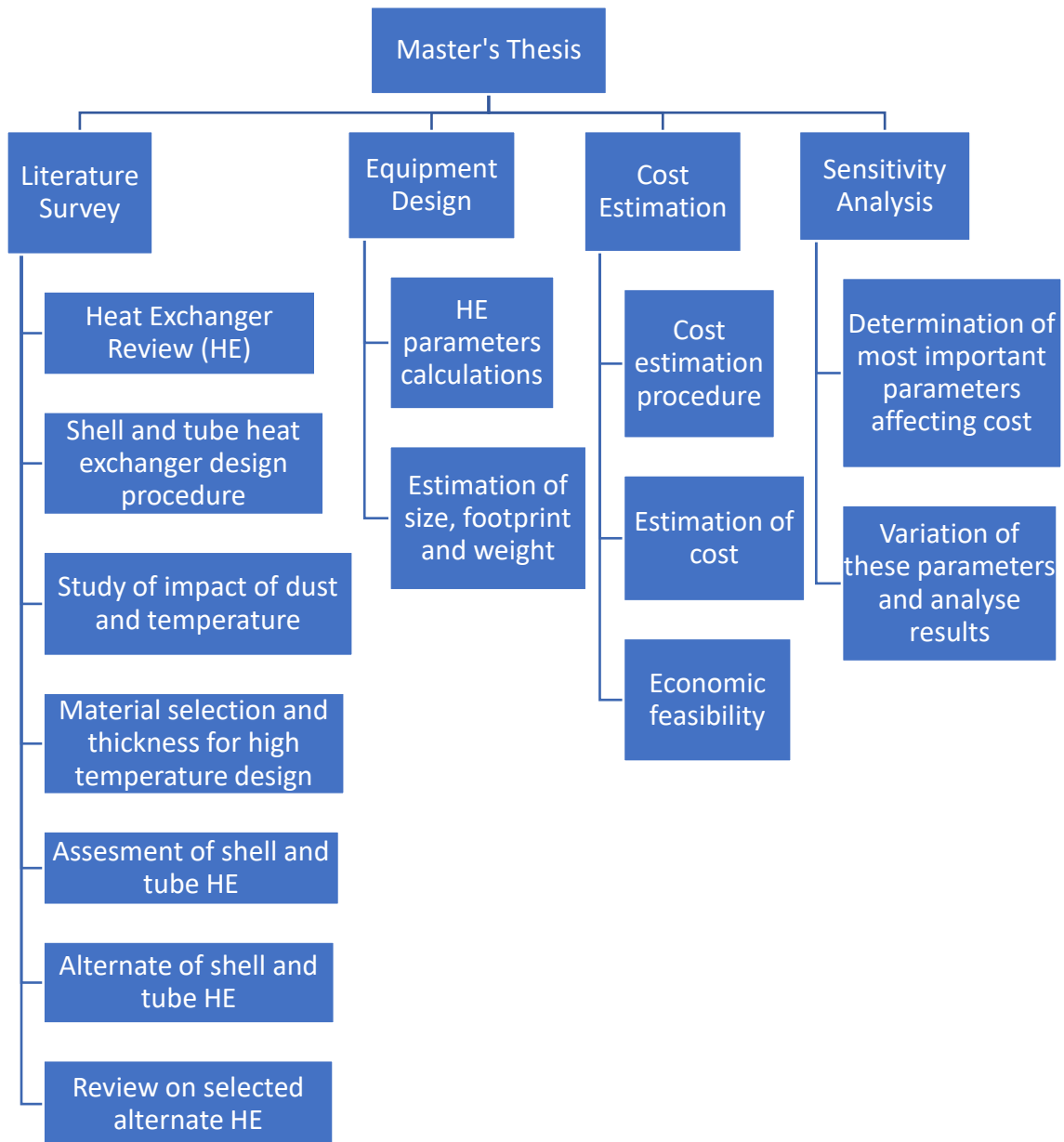


Figure B1: Thesis work breakdown structure

Appendix C: Derivation of equation for number of tubes

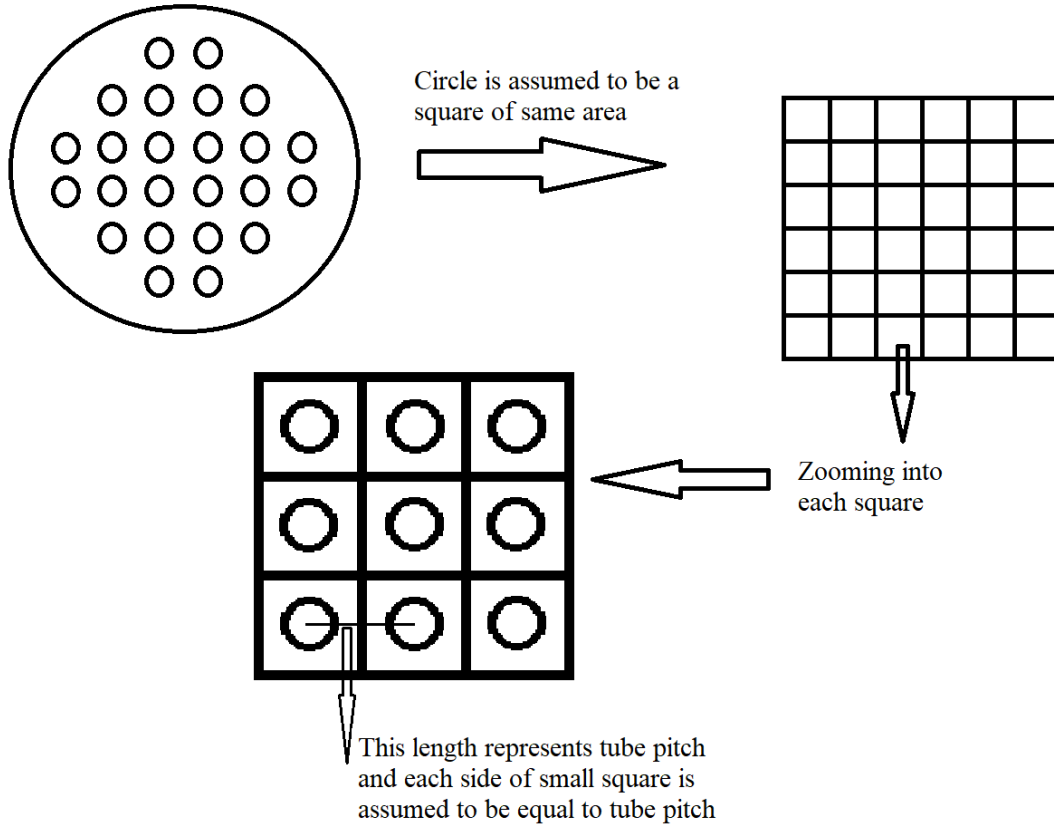


Figure C1: Tube sheet represented in square for to derive number of tubes

Number of tubes (N_t) can be derived by assuming circular region in tube sheet as a square region with the same area as shown in Figure C1. Length of the smaller squares inside the big square is assumed to be equal to tube pitch (S_t). Hence, the count of number of small squares inside the large square will be equal to number of tubes that can fit in a tube sheet.

Since, length of each small square is equal to tube pitch, area of small squares (A_{sq}) is given from equation (C.1).

$$A_{sq} = S_t^2 \tag{C.1}$$

Since, area of bigger square (A_{SQ}) is equal to cross sectional area of tube sheet, it is represented by equation (C.2).

$$A_{SQ} = \frac{\pi D_s^2}{4} \tag{C.2}$$

Here, D_s is the diameter of larger circle in Figure C1 which represents diameter of shell in STHE.

Since, number of tubes (N_t) is equal to number of small squares, it can be represented by equation (C.3).

$$N_t = \frac{A_{SQ}}{A_{sq}} = \frac{\pi D_s^2}{4 S_t^2} \tag{C.3}$$

Appendix D: Design basis

Design basis for CO₂ heat exchanger in relation to Ron Jacob's thesis work

L.A. Tokheim, 2019-01-29, updated 2019-03-08

Quantity	Unit	Value	Remark
Flow rate of hot gas (CO ₂)	t/h	67	Pure CO ₂ is assumed
Flow rate of cold gas (air)	t/h	71	Pure air is assumed
Inlet temperature of CO ₂	°C	900	
Inlet temperature of air	°C	225	
Dust concentration in CO ₂	mg/Nm ³	50 000	Approximate value
Dust concentration in air	mg/Nm ³	5	
Inlet pressure of CO ₂	bara	1	
Inlet pressure of air	bara	1	
Operational time	h/y	7315	

Design basis for CO₂ heat exchanger in relation to Ron Jacob's thesis work.xlsx

Appendix E: Additional STHE results

Additional results for STHE parameter design area presented in Table F1. “N/A” represents cases which are technically infeasible due to high pressure drop.

Additional cost estimation results are presented in Table F2. All the symbols and units used in the table can be referred from Nomenclature chapter.

Table E1: Additional results for STHE parameter design

Structure	N_{HE}	d_i	v_t	v_s	U	q	A	P_t	P_s	q_{net}	m_{HE}	V_{HE}	F_{HE}
1-2 STHE	1	0.02	455	300	230	7.6	124	8.04	9.07	N/A	N/A	N/A	N/A
1-2 STHE	1	0.03	267	192	161	7.6	177	2.64	3.39	N/A	N/A	N/A	N/A
1-2 STHE	1	0.04	102	83	81	7.6	350	0.36	0.55	N/A	N/A	N/A	N/A
1-2 STHE	1	0.05	53	46	50	7.6	568	0.09	0.15	N/A	N/A	N/A	N/A
1-2 STHE	2	0.02	227	150	148	3.8	96	1.92	1.95	N/A	N/A	N/A	N/A
1-2 STHE	2	0.03	134	96	103	3.8	138	0.63	0.73	N/A	N/A	N/A	N/A
1-2 STHE	2	0.04	51	41	52	3.8	275	0.09	0.12	N/A	N/A	N/A	N/A
1-2 STHE	2	0.05	27	23	32	3.8	450	0.02	0.03	6.8	24	28	17
1-2 STHE	3	0.02	152	100	114	2.5	83	0.83	0.80	N/A	N/A	N/A	N/A
1-2 STHE	3	0.03	89	64	79	2.5	120	0.28	0.30	-0.1	10	7	8

1-2 STHE	3	0.04	34	28	39	2.5	240	0.04	0.05	6.4	19	18	15
1-2 STHE	3	0.05	18	15	24	2.5	395	0.01	0.01	7.3	32	37	23
1-2 STHE	4	0.02	114	75	95	1.9	75	0.46	0.42	-4.9	8	5	8
1-2 STHE	4	0.03	67	48	65	1.9	109	0.15	0.16	3.5	12	8	10
1-2 STHE	4	0.04	25	21	32	1.9	219	0.02	0.03	7.0	24	22	18
1-2 STHE	4	0.05	13	12	20	1.9	362	0.01	0.01	7.4	39	45	28
1-2 STHE	5	0.02	91	60	82	1.5	69	0.29	0.26	0.2	10	6	9
1-2 STHE	5	0.03	53	38	56	1.5	101	0.10	0.10	5.0	14	9	12
1-2 STHE	5	0.04	20	17	28	1.5	204	0.01	0.02	7.2	27	25	21
1-2 STHE	5	0.05	11	9	17	1.5	340	0.00	0.00	7.5	45	53	33
1-2 STHE	6	0.02	76	50	73	1.3	65	0.20	0.17	2.6	11	6	10
1-2 STHE	6	0.03	45	32	50	1.3	95	0.07	0.07	5.9	16	10	13
1-2 STHE	6	0.04	17	14	24	1.3	193	0.01	0.01	7.3	31	29	24
1-2 STHE	6	0.05	9	8	15	1.3	324	0.00	0.00	7.5	52	60	38

1-2 STHE	7	0.02	65	43	66	1.1	62	0.15	0.12	4.1	12	7	11
1-2 STHE	7	0.03	38	27	45	1.1	90	0.05	0.05	6.3	17	12	15
1-2 STHE	7	0.04	15	12	22	1.1	185	0.01	0.01	7.4	35	32	27
1-2 STHE	7	0.05	8	7	13	1.1	312	0.00	0.00	7.5	58	68	42
1-2 STHE	8	0.02	57	37	60	0.9	59	0.11	0.09	4.9	13	8	12
1-2 STHE	8	0.03	33	24	41	0.9	86	0.04	0.04	6.6	19	13	16
1-2 STHE	8	0.04	13	10	20	0.9	179	0.01	0.01	7.4	39	35	29
1-2 STHE	8	0.05	7	6	12	0.9	303	0.00	0.00	7.6	65	75	47
2-4 STHE	1	0.02	860	648	354	10.0	157	63.47	45.2 1	N/A	N/A	N/A	N/A
2-4 STHE	1	0.03	505	415	250	10.0	223	20.65	16.7 6	N/A	N/A	N/A	N/A
2-4 STHE	1	0.04	193	179	127	10.0	437	2.78	2.69	N/A	N/A	N/A	N/A
2-4 STHE	1	0.05	100	100	79	10.0	708	0.73	0.76	N/A	N/A	N/A	N/A
2-4 STHE	2	0.02	430	324	230	5.0	121	14.98	9.65	N/A	N/A	N/A	N/A
2-4 STHE	2	0.03	252	207	161	5.0	173	4.92	3.61	N/A	N/A	N/A	N/A

2-4 STHE	2	0.04	96	90	81	5.0	343	0.67	0.58	N/A	N/A	N/A	N/A
2-4 STHE	2	0.05	50	50	50	5.0	557	0.18	0.17	N/A	N/A	N/A	N/A
2-4 STHE	3	0.02	287	216	178	3.3	104	6.47	3.93	N/A	N/A	N/A	N/A
2-4 STHE	3	0.03	168	138	124	3.3	150	2.13	1.47	N/A	N/A	N/A	N/A
2-4 STHE	3	0.04	64	60	62	3.3	298	0.29	0.24	2.9	24	22	18
2-4 STHE	3	0.05	33	33	38	3.3	486	0.08	0.07	8.1	39	45	28
2-4 STHE	4	0.02	215	162	148	2.5	94	3.58	2.08	N/A	N/A	N/A	N/A
2-4 STHE	4	0.03	126	104	103	2.5	135	1.18	0.78	N/A	N/A	N/A	N/A
2-4 STHE	4	0.04	48	45	52	2.5	270	0.16	0.13	6.2	29	27	22
2-4 STHE	4	0.05	25	25	32	2.5	442	0.04	0.04	9.0	47	55	34
2-4 STHE	5	0.02	172	130	128	2.0	87	2.26	1.27	N/A	N/A	N/A	N/A
2-4 STHE	5	0.03	101	83	89	2.0	125	0.75	0.48	N/A	N/A	N/A	N/A
2-4 STHE	5	0.04	39	36	44	2.0	251	0.10	0.08	7.7	34	31	26
2-4 STHE	5	0.05	20	20	27	2.0	411	0.03	0.02	9.4	55	64	40

2-4 STHE	6	0.02	143	108	114	1.7	81	1.56	0.85	N/A	N/A	N/A	N/A
2-4 STHE	6	0.03	84	69	79	1.7	118	0.52	0.32	N/A	N/A	N/A	N/A
2-4 STHE	6	0.04	32	30	39	1.7	236	0.07	0.05	8.4	38	35	29
2-4 STHE	6	0.05	17	17	24	1.7	388	0.02	0.02	9.6	62	72	45
2-4 STHE	7	0.02	123	93	103	1.4	77	1.13	0.61	N/A	N/A	N/A	N/A
2-4 STHE	7	0.03	72	59	71	1.4	111	0.38	0.23	1.9	21	14	18
2-4 STHE	7	0.04	28	26	35	1.4	225	0.05	0.04	8.9	42	39	32
2-4 STHE	7	0.05	14	14	21	1.4	370	0.01	0.01	9.7	69	80	50
2-4 STHE	8	0.02	107	81	95	1.3	74	0.86	0.45	N/A	N/A	N/A	N/A
2-4 STHE	8	0.03	63	52	65	1.3	107	0.29	0.17	4.1	23	16	20
2-4 STHE	8	0.04	24	22	32	1.3	215	0.04	0.03	9.2	46	42	36
2-4 STHE	8	0.05	13	12	20	1.3	356	0.01	0.01	9.8	76	88	55

Table E2: Additional results of estimated cost

Structure	N_{HE}	d_i	NPV_{CAPEX}	NPV	EAC_{CAPEX}	EAC_{OPEX}
1-2 STHE	1	0.02	N/A	N/A	N/A	N/A
1-2 STHE	1	0.03	N/A	N/A	N/A	N/A
1-2 STHE	1	0.04	N/A	N/A	N/A	N/A
1-2 STHE	1	0.05	N/A	N/A	N/A	N/A
1-2 STHE	2	0.02	N/A	N/A	N/A	N/A
1-2 STHE	2	0.03	N/A	N/A	N/A	N/A
1-2 STHE	2	0.04	N/A	N/A	N/A	N/A
1-2 STHE	2	0.05	20.8	25.2	3.1	-6.9
1-2 STHE	3	0.02	0.0	0.0	0.0	0.0
1-2 STHE	3	0.03	101.1	-101.9	15.1	0.1
1-2 STHE	3	0.04	21.2	22.1	3.2	-6.5
1-2 STHE	3	0.05	27.0	22.1	4.0	-7.3
1-2 STHE	4	0.02	111.7	-144.8	16.7	4.9
1-2 STHE	4	0.03	114.2	-90.9	17.0	-3.5
1-2 STHE	4	0.04	27.6	19.3	4.1	-7.0
1-2 STHE	4	0.05	34.9	15.1	5.2	-7.5
1-2 STHE	5	0.02	122.7	-121.5	18.3	-0.2
1-2 STHE	5	0.03	25.9	7.9	3.9	-5.0
1-2 STHE	5	0.04	34.2	14.4	5.1	-7.2
1-2 STHE	5	0.05	43.1	7.4	6.4	-7.5
1-2 STHE	6	0.02	146.5	-128.7	21.8	-2.7
1-2 STHE	6	0.03	28.6	10.9	4.3	-5.9
1-2 STHE	6	0.04	38.1	11.2	5.7	-7.4
1-2 STHE	6	0.05	48.6	2.1	7.2	-7.6
1-2 STHE	7	0.02	178.7	-151.4	26.6	-4.1
1-2 STHE	7	0.03	31.1	11.7	4.6	-6.4

1-2 STHE	7	0.04	42.0	7.9	6.3	-7.4
1-2 STHE	7	0.05	54.0	-3.1	8.0	-7.6
1-2 STHE	8	0.02	111.3	-78.1	16.6	-5.0
1-2 STHE	8	0.03	33.5	11.4	5.0	-6.7
1-2 STHE	8	0.04	45.7	4.5	6.8	-7.5
1-2 STHE	8	0.05	59.4	-8.4	8.8	-7.6
2-4 STHE	1	0.02	N/A	N/A	N/A	N/A
2-4 STHE	1	0.03	N/A	N/A	N/A	N/A
2-4 STHE	1	0.04	N/A	N/A	N/A	N/A
2-4 STHE	1	0.05	N/A	N/A	N/A	N/A
2-4 STHE	2	0.02	N/A	N/A	N/A	N/A
2-4 STHE	2	0.03	N/A	N/A	N/A	N/A
2-4 STHE	2	0.04	N/A	N/A	N/A	N/A
2-4 STHE	2	0.05	N/A	N/A	N/A	N/A
2-4 STHE	3	0.02	N/A	N/A	N/A	N/A
2-4 STHE	3	0.03	N/A	N/A	N/A	N/A
2-4 STHE	3	0.04	109.6	-90.1	16.3	-2.9
2-4 STHE	3	0.05	30.1	24.6	4.5	-8.1
2-4 STHE	4	0.02	N/A	N/A	N/A	N/A
2-4 STHE	4	0.03	N/A	N/A	N/A	N/A
2-4 STHE	4	0.04	124.7	-82.8	18.6	-6.3
2-4 STHE	4	0.05	38.6	22.0	5.8	-9.0
2-4 STHE	5	0.02	N/A	N/A	N/A	N/A
2-4 STHE	5	0.03	N/A	N/A	N/A	N/A
2-4 STHE	5	0.04	84.7	-33.0	12.6	-7.7

2-4 STHE	5	0.05	47.4	15.8	7.1	-9.4
2-4 STHE	6	0.02	N/A	N/A	N/A	N/A
2-4 STHE	6	0.03	N/A	N/A	N/A	N/A
2-4 STHE	6	0.04	41.9	14.9	6.2	-8.5
2-4 STHE	6	0.05	53.4	11.3	8.0	-9.6
2-4 STHE	7	0.02	N/A	N/A	N/A	N/A
2-4 STHE	7	0.03	179.8	-167.1	26.8	-1.9
2-4 STHE	7	0.04	46.1	13.8	6.9	-8.9
2-4 STHE	7	0.05	59.1	6.3	8.8	-9.8
2-4 STHE	8	0.02	N/A	N/A	N/A	N/A
2-4 STHE	8	0.03	190.5	-163.2	28.4	-4.1
2-4 STHE	8	0.04	50.1	11.6	7.5	-9.2
2-4 STHE	8	0.05	64.7	1.3	9.6	-9.8

Appendix F: Developed python code

```

1 ##ASSUMPTIONS##
2 #Works only for 1-2 and 2-4 heat exchanger
3 #Code starts with calculating for 1 heat exchanger and one diameter case, then 2 then so on.
4 #The result printed are in excel sheet
5 ##-----MAIN INPUT PARAMETERS-----##
6 N_HE_T = 8           #Total number of heat exchanger in parallel
7 N_S = 1             #Number of shells
8 n = N_S*2          #Number of passes    ##try 2,4
9 T_h_in = 900       #Hot temperature in [degree C]
10 T_c_in = 225      #Cold temperature in [degeree C]
11 if N_S == 1:      #Automating calcaultion to maintain f_LMTD=0.8
12     dT_h_c = 315
13 else:
14     dT_h_c = 205
15 m_h_star=67000     #mass flow rate of hot in [kg/hr]
16 m_c_star=71000     #mass flow rate of cold in [kg/hr]
17 #d_i = [0.02, 0.025, 0.038, 0.051] #internal diameter of tube in [m]
18 #t_m = [0.0027, 0.0027, 0.0027, 0.0027] #thickness of tube [m]
19 d_i = [0.051]
20 t_m = [0.0027]
21 e_f = 0.75         #Efficiency of fan
22 e_t = 0.4          #Thermodynamic efficiency for heat to work
23 t_sm=7.9e-3       #Thickness of shell material
24 rho_sm=8193.3     #Density of inconel OR material of heat exchanger (kg/m3)
25
26 ##-----Cost calculation deatils-----##
27 C_e1_kwh=0.318    #kr/year
28 n_yr=10           #No of year
29 i_yr=0.08         #Interest rate
30 hr_year=7315     #No of hours per year
31 an_f=(1-(1/(1+i_yr)**n_yr))/i_yr    #Annuity factor
32
33 ##-----Standard Equipment detials-----##
34 ##Heat exchanger
35 A_sm = 388.33     #Area of CS material HE (m2)
36 C_sm = 346.21     #Cost of CS material HE (kNOK)
37 n_cf_he=0.68     #Capacity correction
38 n_CI=1.4051      #Cost index from 2002 to 2019
39 #mc_f = 2.15     #Material cost factor #2.15
40 mc_f = 5.78

```

```

41
42 ##Tubo blower
43 P_tb=50000          #Pressure drop allowance (Pa)
44 V_sm_tb=5          #Volumetric flow rate (m3/sec)
45 C_sm_tb=150274     #Cost of turbo blower (USD)
46 n_cf_tb=0.61       #Capacity factor
47
48 ##Radial Fan
49 P_rf=10000         #Pressure drop allowance (Pa)
50 V_sm_rf=5          #Volumetric flow rate (m3/sec)
51 C_sm_rf=5259       #Cost of radial fan (USD)
52 n_cf_rf=0.74       #Capacity factor
53
54 ##-----Parameter input-----###
55 ##-----Input Tube side properties-----##
56 ocp_h=1.17 #Guess heat capacity hot kJ/kgK
57 M_h = 0.044 #Molecular mass hot
58 P_h = 100000 #Pressure hot
59 #k_t
60
61 ##-----Input Shell side properties-----##
62 ocp_c=1.03 #Guess heat capacity cold kJ/kgK
63 M_c = 0.029 #Molecular mass cold
64 P_c = 100000 #Pressure cold
65
66 def paramhot(T_h_av):
67     cp_h = (0.000499974418585001*T_h_av)+0.911585305907074
68     mu_h = (3.53e-8*T_h_av)+1.532e-5
69     k_h = (0.0000752862724867153*T_h_av)+0.0157997927149376
70     paramhot_results=[cp_h,mu_h,k_h]
71     return paramhot_results
72
73 def paramcold(T_c_av):
74     cp_c = (0.000213997503655823*T_c_av)+0.983400749478319
75     mu_c = (0.0000000289857*T_c_av)+0.0000213429554120471
76     k_c = (0.0000499720906376566*T_c_av)+0.0304952230380011
77     paramcold_results=[cp_c,mu_c,k_c]
78     return paramcold_results
79

```

```

80 ##-----MAIN CODE STARTS HERE-----
81 import numpy as np
82 import pandas as pd
83 from math import log, pi, sqrt, log10
84
85 ##----LMTD Correction factor definition----####
86 def F_LMTD(N_S,R,S):
87     if N_S == 1:
88         F_lm1=sqrt((R**2)+1)
89         F_lm2=log((1-S)/(1-R*S))
90         F_lm3=(R-1)
91         F_lm4=(2-S*(R+1-sqrt((R**2)+1)))
92         F_lm5=(2-S*(R+1+sqrt((R**2)+1)))
93         F_lm=(F_lm1*F_lm2)/(F_lm3*log(F_lm4/F_lm5))
94     else:
95         F_lm1=log((1-S)/(1-R*S))
96         F_lm2=(sqrt((R**2)+1))/(2*(R-1))
97         F_lm3=(2/S)-1-R+sqrt((R**2)+1)+((2/S)*sqrt((1-S)*(1-(R*S))))
98         F_lm4=(2/S)-1-R-sqrt((R**2)+1)+((2/S)*sqrt((1-S)*(1-(R*S))))
99         F_lm=(F_lm1*F_lm2)/(log(F_lm3/F_lm4))
100     return F_lm
101
102 ##-----Heat Balance definition-----####
103 def heat_bal(cr_t,T_h_in,T_c_in,dT_h_c,m_h,m_c):
104     if cr_t==1:
105         T_c_out = T_h_in - dT_h_c #Cold temperature out
106         T_c_av = (T_c_in+T_c_out)/2
107         paramcold_results=paramcold(T_c_av)
108         cp_c = paramcold_results[0]
109         q = (m_c/3600)*cp_c*(T_c_out-T_c_in)
110         T_h_out = T_h_in - (q/((m_h/3600)*ocp_h))
111         T_h_av = (T_h_in+T_h_out)/2
112         paramhot_results=paramhot(T_h_av)
113         cp_h = paramhot_results[0]
114         T_h_out = T_h_in - (q/((m_h/3600)*cp_h))
115         T_h_av = (T_h_in+T_h_out)/2
116         paramhot_results=paramhot(T_h_av)
117         cp_h = paramhot_results[0]
118         T_h_out = T_h_in - (q/((m_h/3600)*cp_h))
119         result = [T_h_out,T_c_out,q,cp_h,cp_c]

```



```

120 else:
121     T_h_out = T_c_in + dT_h_c #Cold temperature out
122     q = (m_h/3600)*ocp_h*(T_h_in-T_h_out)
123     T_c_out = T_c_in + (q/((m_c/3600)*ocp_c))
124     T_h_av = (T_h_in+T_h_out)/2
125     T_c_av = (T_c_in+T_c_out)/2
126     paramcold_results=paramcold(T_c_av)
127     paramhot_results=paramhot(T_h_av)
128     cp_h = paramhot_results[0]
129     cp_c = paramcold_results[0]
130     q = (m_h/3600)*ocp_h*(T_h_in-T_h_out)
131     T_c_out = T_c_in + (q/((m_c/3600)*cp_c))
132     T_h_av = (T_h_in+T_h_out)/2
133     T_c_av = (T_c_in+T_c_out)/2
134     paramcold_results=paramcold(T_c_av)
135     paramhot_results=paramhot(T_h_av)
136     cp_h = paramhot_results[0]
137     cp_c = paramcold_results[0]
138     q = (m_h/3600)*ocp_h*(T_h_in-T_h_out)
139     T_c_out = T_c_in + (q/((m_c/3600)*cp_c))
140     result = [T_h_out,T_c_out,q,cp_h,cp_c]
141 return result
142
143 ##-----Present Cost desifinition-----##
144 def presentcost(p,i,n):
145     sumcos=0
146     for j in range(0,n):
147         cost=p/((1+i)**j)
148         sumcos=sumcos+cost
149     return sumcos
150
151 ##-----Pandas Related work (Creating table)-----##
152 HE_data = pd.DataFrame(columns=['No_HE', 'd_i', 'D_s', 'Allow L', 'Actual L', 'Velocity_tube', \
153 'Velocity_shell', 'Heat transfer Area', 'P_t', 'P_s', 'q_loss', 'q_overall', 'C_pc_tube(MNOK)', \
154 'C_pc_shell(MNOK)', 'HE_cost(MNOK)', 'NPV (MNOK)', 'CAPEX(MNOK)', 'OPEX(MNOK)', 'Mass(ton)',
155 'Size(m^3)', 'Footprint(m^2)', 'Duty [MW]', 'Overall U'])
156 HE_data_final = pd.DataFrame(columns=['No_HE', 'd_i', 'D_s', 'Allow L', 'Actual L', 'Velocity_tube', \
157 'Velocity_shell', 'Heat transfer Area', 'P_t', 'P_s', 'q_loss', 'q_overall', 'C_pc_tube(MNOK)', \
158 'C_pc_shell(MNOK)', 'HE_cost(MNOK)', 'NPV (MNOK)', 'CAPEX(MNOK)', 'OPEX(MNOK)', 'Mass(ton)', 'Size(m^3)', \
159 'Footprint(m^2)', 'Duty [MW]', 'Overall U'])

```

```

160
161 ##-----Input Variables-----##
162 N_HE_TM=N_HE_T+1
163 for N_HE in range(1,N_HE_TM):
164     m_h = m_h_star/N_HE      #mass flow of hot (kg/hr)
165     m_c = m_c_star/N_HE      #mass flow of cold (kg/hr)
166     x_n = len(d_i)
167     Rs = 8.314 #Universal gas constant
168
169     ##-----Definitions-----##
170     d_o=np.zeros(x_n)
171     S_t=np.zeros(x_n)
172     D_s=np.zeros(x_n)
173     L_t=np.zeros(x_n)
174     B=np.zeros(x_n)
175     A_s=np.zeros(x_n)
176     d_e=np.zeros(x_n)
177     N_t=np.zeros(x_n)
178
179     ##-----Sizing Variable determination-----##
180     for i in range(x_n):
181         d_o[i]=d_i[i]+(2*t_m[i])
182         S_t[i]=1.25*d_o[i] #Tube pitch
183         D_s[i]=40*d_i[i] #Diameter of shell-----Other than d_i=0.02, increase it to 40*d_i
184         L_t[i]=10*D_s[i]
185         B[i]=D_s[i] #Try changing baffle to reduce turbulence
186         if N_S==1:
187             A_s[i]=D_s[i]*B[i]*(1-(d_o[i]/S_t[i]))
188         else:
189             A_s[i]=(D_s[i]/2)*B[i]*(1-(d_o[i]/S_t[i]))
190         d_e[i] = (4/(pi*d_o[i]))*((S_t[i]**2)-((pi/4)*(d_o[i]**2)))
191         N_t[i] = round((pi*D_s[i]**2)/(4*S_t[i]**2))
192
193
194     ##-----Heat Balance results-----##
195     if (m_h*ocp_h)>(m_c*ocp_c):
196         cr_t=1
197         print("mCp of hot fluid higher")
198     else:
199         cr_t=2
200         print("mCp of cold fluid is higher")

```

```

201
202 results=heat_bal(cr_t,T_h_in,T_c_in,dT_h_c,m_h,m_c)
203 T_h_out=results[0]      #degree C
204 T_c_out=results[1]      #degree C
205 q=results[2]           #kW
206 cp_h=results[3]        #kJ/kgK
207 cp_c=results[4]        #kJ/kgK
208
209 ##-----Recheck heat balance results-----##
210
211 if (m_h*cp_h)>(m_c*cp_c):
212     cr_t=1
213     print("mCp of hot fluid higher")
214 else:
215     cr_t=2
216     print("mCp of cold fluid is higher")
217
218 results=heat_bal(cr_t,T_h_in,T_c_in,dT_h_c,m_h,m_c)
219 T_h_out=results[0]      #degree C
220 T_c_out=results[1]      #degree C
221 q=results[2]           #kW
222 cp_h=results[3]        #kJ/kgK
223 cp_c=results[4]        #kJ/kgK
224
225 ##-----Tube side properties-----##
226 T_h_av = (T_h_in+T_h_out)/2
227 rho_h = (P_h*M_h)/(Rs*(T_h_av+273.15))
228 paramhot_results=paramhot(T_h_av)
229 mu_h = paramhot_results[1]
230 k_h = paramhot_results[2]
231 nu_h = mu_h/rho_h
232
233 ##-----Shell side properties-----##
234 T_c_av = (T_c_in+T_c_out)/2
235 rho_c = (P_c*M_c)/(Rs*(T_c_av+273.15))
236 paramcold_results=paramcold(T_c_av)
237 mu_c = paramcold_results[1]
238 k_c = paramcold_results[2]
239 nu_c = mu_c/rho_c
240

```

```

241  ##-----Material Properties calculation-----##
242  k_m=(0.016*T_h_av)+10.87
243
244  ##-----Definitions-----##
245  V_t=np.zeros(x_n)
246  A_t=np.zeros(x_n)
247  v_t=np.zeros(x_n)
248  Re_t=np.zeros(x_n)
249  Pr_t=np.zeros(x_n)
250  h_t=np.zeros(x_n)
251  f_t=np.zeros(x_n)
252
253  ##-----Tube side calculations-----##
254  for i in range(x_n):
255      V_t[i]=(m_h/(3600*rho_h))          #Volumetric flow rate in tube side
256      A_t[i]=(pi*(d_i[i]**2)/4)*(N_t[i]/n)
257      v_t[i]=(m_h/(3600*rho_h))/((pi*(d_i[i]**2)/4)*(N_t[i]/n))
258      Re_t[i]=(rho_h*v_t[i]*d_i[i])/mu_h
259      Pr_t[i]=(cp_h*1000*mu_h)/k_h
260      f_t[i] = ((1.82*log10(Re_t[i]))-1.64)**-2
261      h_t[i] = (k_h/d_i[i])*((Pr_t[i]*(Re_t[i]-1000)*(f_t[i]/8))/(1+(12.7\
262          *((Pr_t[i]**(2/3))-1)*(f_t[i]/8)**(1/2))))
263
264      ##-----Definitions-----##
265      V_s=np.zeros(x_n)
266      v_s=np.zeros(x_n)
267      Re_s=np.zeros(x_n)
268      Pr_s=np.zeros(x_n)
269      h_s=np.zeros(x_n)
270      f_s=np.zeros(x_n)
271
272  ##-----Shell side calculation-----##
273  for i in range(x_n):
274      V_s[i]=(m_c/3600)/(rho_c)          #Volumetric flow rate in shell side
275      v_s[i] = (m_c/3600)/(rho_c*A_s[i])
276      Re_s[i] = (d_e[i]*v_s[i]*rho_c)/mu_c
277      Pr_s[i] = (mu_c*cp_c*1000)/k_c
278      h_s[i] = 0.36*(k_c/d_e[i])*(Re_s[i]**0.55)*(Pr_s[i]**0.3)
279      f_s[i] = 2*0.72*(Re_s[i]**-0.15)

```

```

280  ##-----LMTD-----##
281  T_lm = ((T_h_in-T_c_out)-(T_h_out-T_c_in))/log((T_h_in-T_c_out)/(T_h_out-T_c_in))
282  R=(T_c_out-T_c_in)/(T_h_in-T_h_out)
283  S=(T_h_in-T_h_out)/(T_h_in-T_c_in)
284  S_max=2/(R+1+sqrt((R**2)+1))
285  F_lm=F_LMTD(N_S,R,S)
286
287  ##Preliminary printout-----##
288  print("S = %0.2f" %S)
289  print("S_max = %0.2f" %S_max)
290
291  ##-----Definitions-----##
292  U=np.zeros(x_n)
293  A=np.zeros(x_n)
294  L_t_actual=np.zeros(x_n)
295  L_total=np.zeros(x_n)
296
297  #####-----Overall heat transfer coefficient and area-----###
298  for i in range(x_n):
299      U[i] = 1/((1/h_s[i])+((1/h_t[i])*(d_o[i]/d_i[i]))+((d_o[i]/(2*k_m))*log(d_o[i]/d_i[i])))
300      A[i]=(q*1000)/(U[i]*T_lm*F_lm)
301      L_t_actual[i]=A[i]/(pi*d_i[i]*N_t[i])
302      L_total[i] = L_t_actual[i]*N_t[i]
303
304  ##-----Definitions-----##
305  P_t=np.zeros(x_n)
306  P_s=np.zeros(x_n)
307
308  ##-----Pressure Drops-----##
309  for i in range(x_n):
310      P_t[i]=(rho_h*(v_t[i]**2)*(n/2))*(4+((L_t_actual[i]*f_t[i])/d_i[i]))
311      P_s[i]=(f_s[i]*L_t_actual[i]*D_s[i]*rho_c*(v_s[i]**2))/(2*B[i]*d_e[i])
312
313  ##-----Definitions-----##
314  W_s_isen_CO2=np.zeros(x_n)
315  W_s_CO2=np.zeros(x_n)
316  q_loss_CO2=np.zeros(x_n)
317  W_s_isen_air=np.zeros(x_n)
318  W_s_air=np.zeros(x_n)
319  q_loss_air=np.zeros(x_n)

```

```

320
321 ##-----Fan Power calculations-----##
322 for i in range(x_n):
323     #Fan at exit of tube side
324     W_s_isen_CO2[i] = (m_h/3600)*cp_h*(T_h_out+273.15)*((P_h/(P_h-P_t[i]))**(Rs/(cp_h*M_h*1000))-1) #kW
325     W_s_CO2[i] = W_s_isen_CO2[i]/e_f
326     q_loss_CO2[i] = W_s_CO2[i]/e_t    #kW
327     #Fan at entrance of shell side
328     W_s_isen_air[i] = (m_c/3600)*cp_c*(T_c_out+273.15)*(((P_c+P_s[i])/P_c)**(Rs/(cp_c*M_c*1000))-1) #kW
329     W_s_air[i] = W_s_isen_air[i]/e_f
330     q_loss_air[i] = W_s_air[i]/e_t    #kW
331
332 ##-----Definitions-----##
333 q_loss_total=np.zeros(x_n)
334 q_overall=np.zeros(x_n)
335
336 ##-----Unit conversions-----##
337 q=q/1000
338 for i in range(x_n):
339     q_loss_CO2[i] = q_loss_CO2[i]/1000
340     q_loss_air[i] = q_loss_air[i]/1000
341     q_loss_total[i] = q_loss_air[i]+q_loss_CO2[i]
342     q_overall[i] = (q-q_loss_total[i])*N_HE
343
344 ###-----Estimation of weight, size and footprint of HE-----##
345 ##-----Definitions-----##
346 V1_s=np.zeros(x_n)
347 m1_s=np.zeros(x_n)
348 V1_t=np.zeros(x_n)
349 m1_t=np.zeros(x_n)
350 tm_t=np.zeros(x_n)    #time spend by tube gas in tube
351 m_mh=np.zeros(x_n)
352 tm_s=np.zeros(x_n)    #time spend by shell gass in shell
353 m_mc=np.zeros(x_n)
354 m1_tot=np.zeros(x_n)
355 V_size=np.zeros(x_n)
356 F_HE=np.zeros(x_n)
357 for i in range(x_n):
358     #Mass of shell material
359     V1_s[i]=((pi*D_s[i]*L_t_actual[i])+((pi*D_s[i]**2)/4))*t_sm
360     m1_s[i]=rho_sm*V1_s[i]

```

```

361     #Mass of tube material
362     V_l_t[i]=pi*d_i[i]*L_t_actual[i]*N_t[i]*t_m[i]
363     m_l_t[i]=rho_sm*V_l_t[i]
364     #Mass of CO2 in HE
365     tm_t[i]=(L_t_actual[i]*n)/v_t[i]
366     m_mh[i]=(m_h/3600)*tm_t[i]
367     #Mass of air in HE
368     tm_s[i]=(L_t_actual[i]*N_S)/v_s[i]
369     m_mc[i]=(m_c/3600)*tm_s[i]
370     #Total mass of all HE
371     m_l_tot[i]=(m_l_s[i]+m_l_t[i]+m_mh[i]+m_mc[i])*N_HE
372     #Total size of all HE
373     V_size[i]=((pi*L_t_actual[i]*D_s[i]**2)/4)*N_HE
374     #Footprint of all HE
375     F_HE[i]=(L_t_actual[i]*D_s[i])*N_HE
376
377     ##-----Definitions-----##
378     C_HE_CS=np.zeros(x_n)
379     C_HE=np.zeros(x_n)
380     C_pc_t=np.zeros(x_n)
381     C_pc_s=np.zeros(x_n)
382     C_pc_ft=np.zeros(x_n)
383     C_pc_fs=np.zeros(x_n)
384
385     ##-----Cost of equipment (Detailed factor method)-----##
386     for i in range(x_n):
387         if P_t[i]<=P_tb and P_t[i]>P_rf and V_t[i]<15:
388             C_pc_t[i]=8.5*n_CI*(C_sm_tb/1000)*(V_t[i]/V_sm_tb)**n_cf_tb     #kNOK
389         elif P_t[i]<=P_rf:
390             C_pc_t[i]=8.5*n_CI*(C_sm_rf/1000)*(V_t[i]/V_sm_rf)**n_cf_rf     #kNOK
391         else:
392             C_pc_t[i]=0
393         if P_s[i]<=P_tb and P_s[i]>P_rf and V_s[i]<15:
394             C_pc_s[i]=8.5*n_CI*(C_sm_tb/1000)*(V_s[i]/V_sm_tb)**n_cf_tb     #kNOK
395         elif P_s[i]<=P_rf:
396             C_pc_s[i]=8.5*n_CI*(C_sm_rf/1000)*(V_s[i]/V_sm_rf)**n_cf_rf     #kNOK
397         else:
398             C_pc_s[i]=0
399     #Cost of CS material HE calculations
400     if C_pc_s[i] != 0 and C_pc_t[i] != 0:
401         C_HE_CS[i]=(C_sm)*(A[i]/A_sm)**n_cf_he     #Cost in kNOK

```

```

402     else:
403         C_HE_CS[i]=0
404         C_pc_s[i]=0
405         C_pc_t[i]=0
406         #Actual cost of HE Calculation
407         if C_HE_CS[i]<=20:
408             cf_t=29.65
409             cf_e=1
410             cf_p=3.56
411             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
412             C_HE[i]=C_HE_CS[i]*c_f*N_HE
413         elif C_HE_CS[i]>20 and C_HE_CS[i]<=100:
414             cf_t=15.03
415             cf_e=1
416             cf_p=1.92
417             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
418             C_HE[i]=C_HE_CS[i]*c_f*N_HE
419         elif C_HE_CS[i]>100 and C_HE_CS[i]<=500:
420             cf_t=9.13
421             cf_e=1
422             cf_p=1.12
423             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
424             C_HE[i]=C_HE_CS[i]*c_f*N_HE
425         elif C_HE_CS[i]>500 and C_HE_CS[i]<=1000:
426             cf_t=7.20
427             cf_e=1
428             cf_p=0.83
429             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
430             C_HE[i]=C_HE_CS[i]*c_f*N_HE
431         elif C_HE_CS[i]>1000 and C_HE_CS[i]<=2000:
432             cf_t=6.10
433             cf_e=1
434             cf_p=0.65
435             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
436             C_HE[i]=C_HE_CS[i]*c_f*N_HE
437         elif C_HE_CS[i]>2000 and C_HE_CS[i]<=5000:
438             cf_t=4.93
439             cf_e=1
440             cf_p=0.48
441             c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)

```



```

442     C_HE[i]=C_HE_CS[i]*c_f*N_HE
443 elif C_HE_CS[i]>5000 and C_HE_CS[i]<=15000:
444     cf_t=4.44
445     cf_e=1
446     cf_p=0.41
447     c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
448     C_HE[i]=C_HE_CS[i]*c_f*N_HE
449 else:
450     cf_t=3.59
451     cf_e=1
452     cf_p=0.29
453     c_f=cf_t-cf_e+(cf_e*mc_f)-cf_p+(cf_p*mc_f)
454     C_HE[i]=C_HE_CS[i]*c_f*N_HE
455 if C_pc_t[i]<=20:
456     cf_t=29.65
457     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
458 elif C_pc_t[i]>20 and C_pc_t[i]<=100:
459     cf_t=15.03
460     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
461 elif C_pc_t[i]>100 and C_pc_t[i]<=500:
462     cf_t=9.13
463     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
464 elif C_pc_t[i]>500 and C_pc_t[i]<=1000:
465     cf_t=7.20
466     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
467 elif C_pc_t[i]>1000 and C_pc_t[i]<=2000:
468     cf_t=6.10
469     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
470 elif C_pc_t[i]>2000 and C_pc_t[i]<=5000:
471     cf_t=4.93
472     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
473 elif C_pc_t[i]>5000 and C_pc_t[i]<=15000:
474     cf_t=4.44
475     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
476 else:
477     cf_t=3.59
478     C_pc_ft[i]=(C_pc_t[i]*cf_t)*N_HE
479 if C_pc_s[i]<=20:
480     cf_t=29.65
481     C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE

```

```

482     elif C_pc_s[i]>20 and C_pc_s[i]<=100:
483         cf_t=15.03
484         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
485     elif C_pc_s[i]>100 and C_pc_s[i]<=500:
486         cf_t=9.13
487         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
488     elif C_pc_s[i]>500 and C_pc_s[i]<=1000:
489         cf_t=7.20
490         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
491     elif C_pc_s[i]>1000 and C_pc_s[i]<=2000:
492         cf_t=6.10
493         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
494     elif C_pc_s[i]>2000 and C_pc_s[i]<=5000:
495         cf_t=4.93
496         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
497     elif C_pc_s[i]>5000 and C_pc_s[i]<=15000:
498         cf_t=4.44
499         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
500     else:
501         cf_t=3.59
502         C_pc_fs[i]=(C_pc_s[i]*cf_t)*N_HE
503     C_HE[i]=C_HE[i]/1000
504     C_pc_ft[i]=C_pc_ft[i]/1000
505     C_pc_fs[i]=C_pc_fs[i]/1000
506
507     ##-----Definitions-----##
508     C_el_yr=np.zeros(x_n)
509     CP_el=np.zeros(x_n)
510     NPV=np.zeros(x_n)
511     CAPEX=np.zeros(x_n)
512     OPEX=np.zeros(x_n)
513
514     ##-----NPV determination-----##
515     for i in range(x_n):
516         if C_pc_s[i] != 0 and C_pc_t[i] != 0:
517             C_el_yr[i]=(C_el_kwh/1000)*q_overall[i]*hr_year*e_t      #(MNOK) per year
518             CP_el[i]=presentcost(C_el_yr[i],i_yr,n_yr)
519             NPV[i]=(CP_el[i]-C_HE[i]-C_pc_fs[i]-C_pc_ft[i])      #MNOK
520             CAPEX[i]=(C_HE[i]+C_pc_fs[i]+C_pc_ft[i])/(an_f)
521             OPEX[i]=-CP_el[i]/(an_f)

```

```

522     else:
523         C_el_yr[i]=0      #cost of electricity (MNOK) per year
524         CP_el[i]=0
525         NPV[i]=0      #MNOK
526         CAPEX[i]=0
527         OPEX[i]=0
528
529     ##-----Unit conversions-----##
530     for i in range(x_n):
531         P_t[i]=P_t[i]/100000
532         P_s[i]=P_s[i]/100000
533         ml_tot[i]=ml_tot[i]/1000
534
535     ##-----No of heat exchanger-----##
536     No_HE=np.zeros(x_n)
537     for i in range(x_n):
538         No_HE[i]=N_HE
539
540     ###-----RESULTS PRINTOUT IN EXCEL-----####
541     HE_data['No_HE']=No_HE
542     HE_data['d_i']=d_i
543     HE_data['D_s']=D_s
544     HE_data['Allow L']=L_t
545     HE_data['Actual L']=L_t_actual
546     HE_data['Duty [MW]']=q
547     HE_data['Overall U']=U
548     HE_data['Heat transfer Area']=A
549     HE_data['Velocity_tube']=v_t
550     HE_data['Velocity_shell']=v_s
551     HE_data['P_t']=P_t
552     HE_data['P_s']=P_s
553     HE_data['q_loss']=q_loss_total
554     HE_data['q_overall']=q_overall
555     HE_data['C_pc_tube(MNOK)']=C_pc_ft
556     HE_data['C_pc_shell(MNOK)']=C_pc_fs
557     HE_data['HE_cost(MNOK)']=C_HE
558     HE_data['NPV (MNOK)']=NPV
559     HE_data['CAPEX(MNOK)']=CAPEX
560     HE_data['OPEX(MNOK)']=OPEX
561     HE_data['Mass(ton)']=ml_tot
562
563     HE_data['Size(m^3)']=V_size
564     HE_data['Footprint(m^2)']=F_HE
565     HE_data_final=HE_data_final.append(HE_data)
566     if N_S == 1:      #Automating calculation to maintain f_LMTD=0.8
567         HE_data_final.to_excel('NPV_results_Shell_1.xlsx',index=False)
568     else:
569         HE_data_final.to_excel('NPV_results_Shell_2.xlsx',index=False)

```