CHAPTER 1

INTRODUCTION

1 LAYOUT OF THE REPORT

The report gives a detailed description of the steps involved in conducting a study for the design of a heat exchanger for this particular application. The first part gives the literature involved in carrying out this study, where different sources were found to substantiate the validity of the study. Then next part involves the methodology taken in fulfilling the objectives of the study, then follows the results and discussion of the study which will determine whether the objectives have been met. The last part concludes this whole study and gives some recommendations for designers who are looking to undertake this endeavor

1.1 Background of Study

The bio-oil upgradation system requires a heat recovery system to recover the latent heat of vaporization in the light hydrocarbons that form part of the product from the reactor (figure 1.1). The heat recovery system will condense the product mixture ($H_2 + \ Hydrocarbons$) and reheat the hydrogen gas to send it back to the reactor feed. Therefore the system will have to handle both high pressures and temperatures.

The design of a heat recovery system requires that the designer become familiar with the fluids exchanging heat in the system, depending on the state/phase of the fluids (i.e. liquid, gaseous, and solid). The designer can then start qualitatively evaluating different heat recovery systems. Therefore this report entails the qualitative evaluation of different heat recovery systems in relation to the fluids that will be entering and leaving the system. A brief background of the heat recovery system required is done to ensure a proper understanding of what is required.

Bio-oil will be processed in a reactor to react with hydrogen in the presence of a catalyst; giving an output of a mixture containing light hydrocarbons (i.e. gasoline) and unreacted Hydrogen. The mixture will then enter the heat recovery system where it is to be condensed to separate the two phases into hydrogen gas and liquid hydrocarbons. The system is required to then reheat the hydrogen gas and recycle it back into the main system where it can be fed back into the reactor. Figure 1.1 indicates a schematic layout of the main system.

1.2 Problem Statement

The product mixture requires further processing to get the desired liquid gasoline. The problem at hand is to find a heat transfer system that will:

- Cool the hydrogen to recover the hydrocarbon vapors as liquid gasoline,
- Remove H2O as liquid water,
- Recover heat from the hydrocarbon vapors and water vapor,
- Reheat the gaseous hydrogen back to the inlet condition of high temperature.

1.3 Objectives and Scope of Study

The operating conditions of this heat recovery system are of great importance to the proper designing of the system; therefore an evaluation of the requirements to be met by both the hot and cold fluids entering the system is done.

Hot fluid requirements:

The hot fluid side will contain the mixture of hydrogen and light hydrocarbons. The cooling down of the mixture will induce a condensation process that will account for the Latent Heat released by the hydrocarbons. Figure 1.2 indicates the condensation process of the mixture (orange line). The parameters required for the hot fluid side are temperature, pressure, flow rate, thermo-physical properties of both Hydrogen and the light hydrocarbon involved (i.e. gasoline) and the percentage compositions of Hydrogen and the hydrocarbon.

Cold fluid requirements:

The cold fluid side depends on the hot fluid side; it is a lot simpler to analyze as there is only one phase (cooler gaseous hydrogen). We consider the Pressure losses of the Hydrogen gas at every stage of the system, as shown in Figure1.

During the condensation process we require a point of optimum cooling load, where all the light hydrocarbons will have condensed into liquid phase. The most preferred approximation to the optimum cooling load would be in the $\Delta T \approx 5°C$ margin just after the 100% Liquid point.

The qualitative evaluation of the heat recovery systems will involve the construction of a rating table that uses the following criteria to assess the suitability of the heat recovery systems mentioned in the report: percent heat recovery, pressure drop in the heat recovery system, initial cost, ease of separation of condensed hydrocarbons.

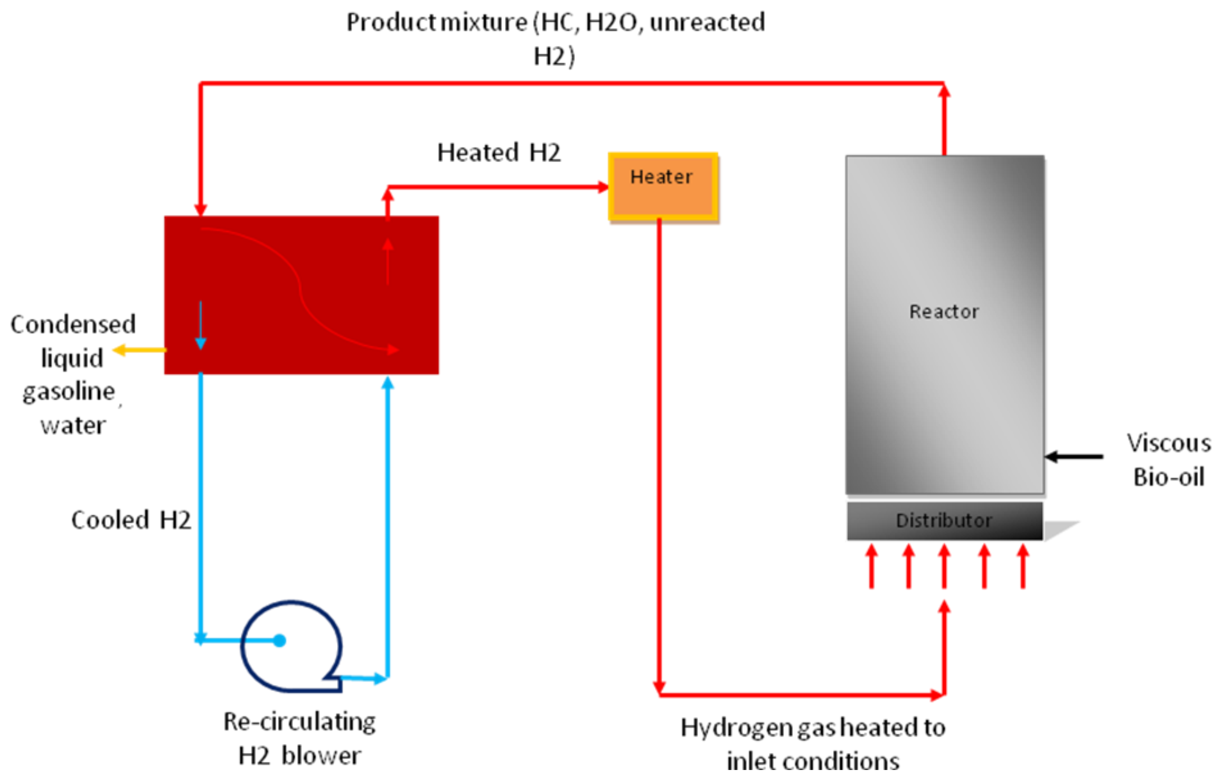

Figure 1.1: Schematic layout of the bio-oil conversion system, indicating pressure losses in the equipments.

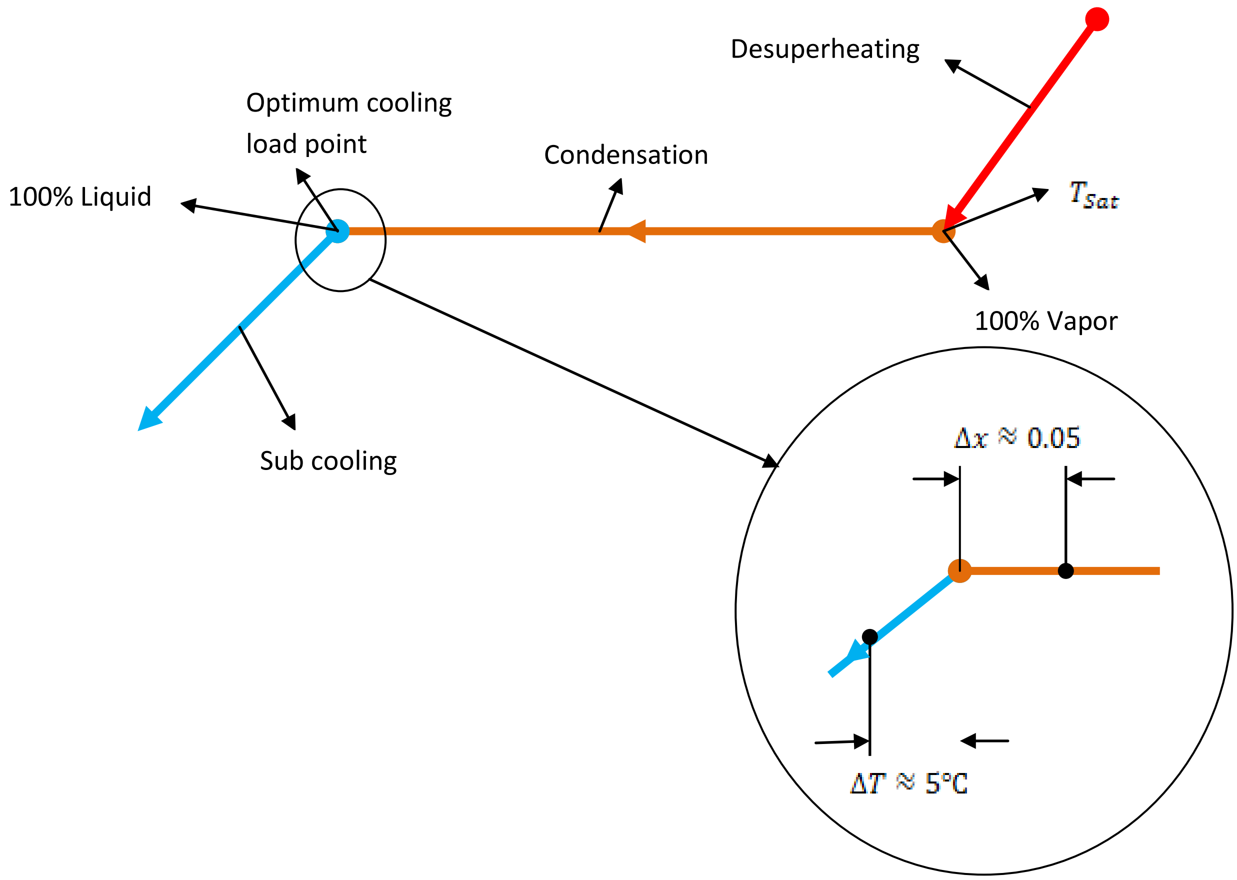

Figure 1.2: Condensation process of the mixture ($H_2 + \ Hydrocarbons$)

CHAPTER 2

THEORY

2 LITERATURE

The literature involved in gathering information about the study is split into two sections being bio-oil upgrading to acquire some typical compositions of the products from bio-oil upgrading and evaluating the designs of different heat exchangers to compare their effectiveness in a bio-oil conversion system that utilizes hydrogen. The results from different sources were taken and compiled in this section to suit the scope of this project.

2.1 Bio-oil Upgrading

The Bio-oil conversion system that is required utilizes hydrogen and therefore the literature that is discussed here is mainly focused on hydro-treating of bio-oil. Four main approaches to improve the quality of bio-oil:

a) Fluidized catalytic cracking (FCC): $C6H8O4 \rightarrow C4.5H6+H2O+1.5CO2$

b) Decarboxylation (DCO): $C6H8O4 \rightarrow C4H8+2CO2$

c) Hydrodeoxygenation (HDO): $C6H8O4+4H2 \rightarrow C6H8+4H2O$

d) Hydrotreating (HT): $C6H8O4+7H2 \rightarrow C6H14+4H2O$

The typical gas product compositions of hydrogen and lighter hydrocarbons are results from the gasification of wood powders in a circulating fluidized bed gasifier. The gas compositions of Hydrogen and the lighter Hydrocarbon are the only products required (shown in table 2.1), the other products are insignificant in the design of the heat recovery system. [4]

Temperature °C	C_nH_m %	H_2 %
630	1.3	5.5
690	3.0	5.71
685	2.1	5.34
748	1.2	5.79
760	2.1	10.78
919	1.2	13.24
974	1.0	16.32
1042	1.3	16.57

Table 2.1: Typical Compositions of Hydrogen and light hydrocarbons

2.2 Heat Exchangers

The design of the heat exchanger will be the main part of the heat recovery system, as it does the actual cooling and condensing processes in the system. Different types of heat exchangers are evaluated in terms of the range of applications and capabilities when related to the application in the heat recovery system. Literature is gathered up on the following heat exchangers that might be applied in the heat recovery system:

a) Compact Recuperative Heat Exchanger
    a. Plate-Fin Heat Exchanger
b) Shell-and-tube Heat Exchanger – Tubular Heat Exchanger
c) Run-Around Coil Regenerative Heat Exchanger
d) Two-Stage Regenerative Heat Exchanger
e) Rotary Regenerative Heat Exchanger

2.2.1 Compact Recuperative heat exchangers

These types of exchangers lack a standardized design procedure, and require that data for predicting the rate of condensation for pure vapor be found. Compact heat exchangers can be applied in condensation applications due to their capability in the following:

- They are able to vary the saturation temperature along the condensation path
- They reduce the adverse effects of the mass-transfer resistance (keeping the two-phases together)
- They enhance the process of simultaneous heat & mass transfer

Compact heat exchangers in condensation applications are developed for the simultaneous heat-mass transfer process for binary vapor mixtures, noting that mass transfer resistance reduces thermal performance. The plate-fin compact heat exchanger has a rate of heat flux at a given mean temperature difference that is two to three times that for conventional shell and tube heat exchangers [3]. The key design features of a plate-fin heat exchanger are flow path for the vapor mixture; variable fin density and distribution vent system. Figure 2.1 below indicates the configuration of the plate-fin heat exchanger. However, the maximum inlet temperature of a Stainless-steel recuperator is about 700˚C.

Plate-Fin Heat Exchanger

This type of heat exchanger has a particular significant size and weight, which is small and compact. It has a typical surface area density of $500m^2/m^3$, that can reach $1800m^2/m^3$. [3] The heat exchanger has the following characteristics:

- Maximum pressure: 90bars
- Temperature range: $-200°\text{C}$ to $150°\text{C}$ in Aluminum
- Fluids: Limited by the material
- Duties: Single & two-phase flows
- Configuration: Cross-flow, Counter-flow
- Maximum ΔT: Typically $50°\text{C}$
- ***High Effectiveness*** ε: up to 0.98.

The Plate-fin heat exchanger is constructed with aluminum as the common material, its alternatives being Nickel and Copper Alloys. Stainless steel is applied in high temperature and pressure applications. [2] Two types of flow arrangements are suggested as the more effective ones for the heat recovery system. The counter-flow arrangement is most thermally effective for heat recovery from process streams. In the cross-counter configuration, the gas stream with the large volume flow rate takes the straight path within the exchanger and the two-phase stream takes the zigzag path. The latter configuration has a high allowable pressure drop. [2]

In the plate-fin exchanger, the plates separate the two fluid streams and the fins for the individual flow passages. The fins are die/roll formed and attached to the plates by brazing, soldering, adhesive bonding, welding, mechanical fit, or extrusion. In a condensing application, fins are generally used only on the gas side. [1]

They are generally designed for moderate operating pressures (about 700kPa), but available commercially for operating pressures of up to about 8300kPa. The temperature limitation for plate-fin heat exchangers depend on the method of bonding and the materials employed (metal exchangers$Temp \approx 850°C$, ceramic exchangers$Temp \geq 1150°C$).

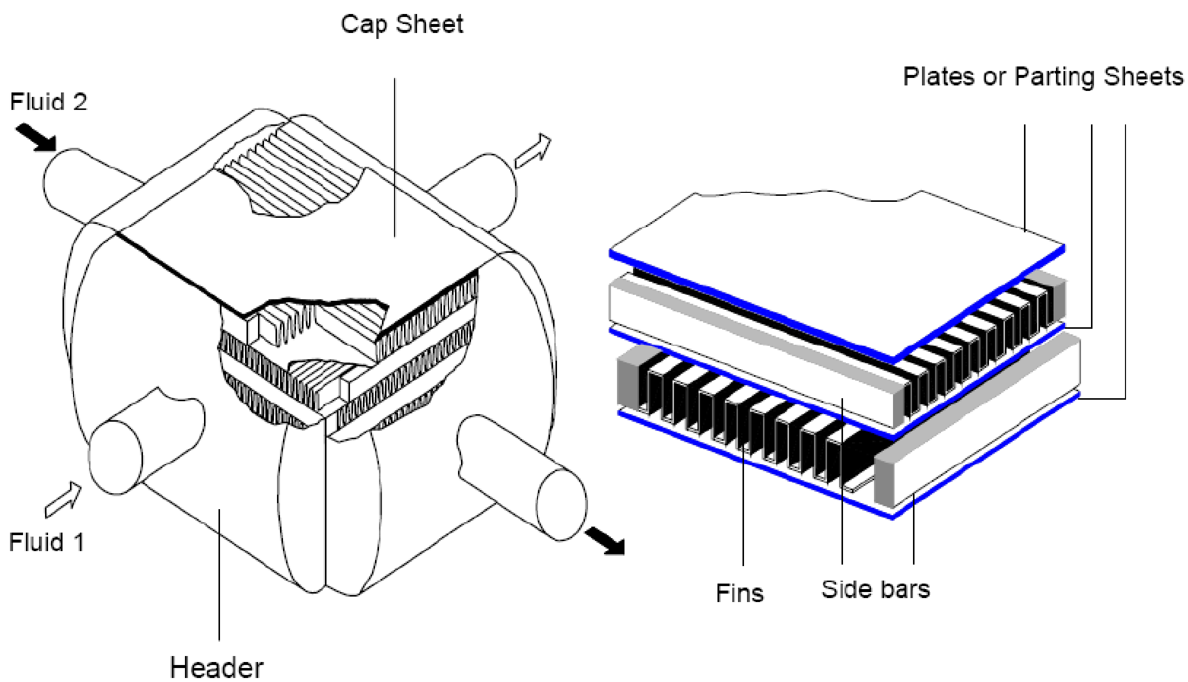

Figure 2.1: Plate-fin heat exchanger.

2.2.2 Shell-and-tube Heat Exchanger

The exchanger is generally built of a bundle of round tubes mounted in a cylindrical shell with the tubes axis parallel to that of the shell. One fluid stream flows inside the tubes, the others flows across and along the tubes. It is also applied in waste heat recovery with heat recovery from liquids and condensing fluids. They are able to withstand ultrahigh pressures (over 100MPa) and high temperatures (about $1100°C$).The shell-&-tube heat exchanger designs are standardized by the Tubular Exchanger Manufacturers Association (TEMA) [4]; with a flexible & robust design this exchanger is suited for most process applications. The usual material used is Carbon Steel, and has a typical size range of $10m^2$ to $1000m^2$. Its other advantage is the ease of maintenance and repair, which would greatly reduce operating costs. [1]

Initial Cost

Exchanger costs are usually quoted proportional to the exchanger heat transfer area. Shell-and-tube exchangers (two fluids only) report a single area which is usually the outside surface area of the tubes. Purohit [16] provides costs estimations for shell-and-tube exchangers made from carbon steel at approximately \$20/ft^2.

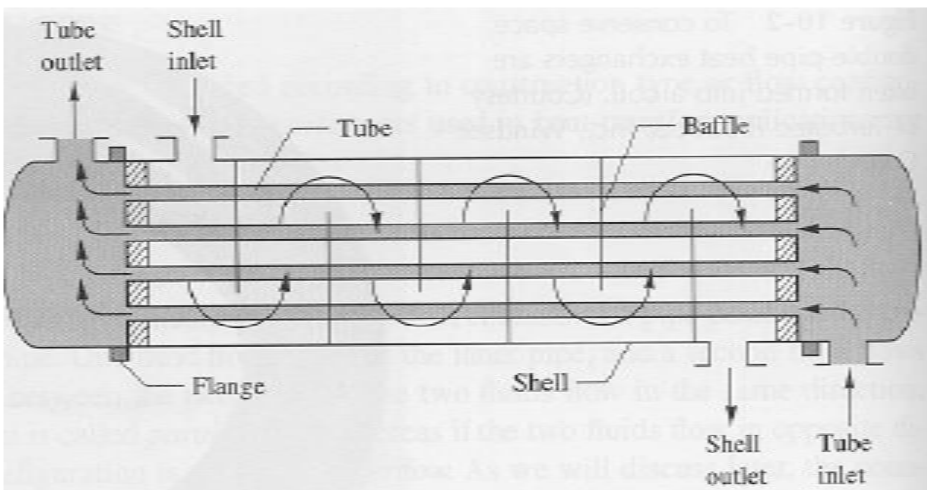

Figure 2.2: Conventional Shell-and-tube heat exchanger

For all four cases, Table 2.2 reports the mean temperature, duty, initial cost, area, and volume for both the brazed and corresponding shell-and-tube exchangers. This table suggests that brazed exchangers are more economical for small mean temperature differences and large duties.

However, for large mean temperature differences and relatively small duties, the shell-and-tube networks are the more attractive option. [17]

			Brazed			Shell-and-Tube		
Case	MTD °F	Duty MMBTU/hr	Area ft^2	Volume ft^3	Cost US$	Area ft^2	Volume ft^3	Cost US$
1	20	9.5	15200	60.6	$80,000	10250	314	$200,000
2	4	1.9	12000	41.7	$60,000	6900	105	$205,000
3	10	5.5	17100	60	$80,000	13610	297	$400,000
4	26	2.4	4420	16.6	$70,000	2520	112	$50,000

Table 2.2: Cost comparison between Compact recuperative exchanger (brazed) and shell-and-tube exchanger

2.2.3 Run-around Coil Regenerative Heat Recovery System

Introduction

A run-around coil heat recovery system uses a linkage of two recuperative heat exchangers by a third fluid which exchanges heat with each fluid in the individual recuperative heat exchangers. This type of system is used in cases where the two fluids which are required to exchange heat are too far apart to use a conventional direct recuperative heat exchanger, in this case the hot side two-phase mixture of Light hydrocarbons + unreacted hydrogen and the cold side hydrogen gas. It is also used if there is a risk of cross-contamination between the two primary fluids (e.g. when a particularly corrosive fluid is involved). The figure below indicates a schematic view of the run-around heat recovery system. [8]

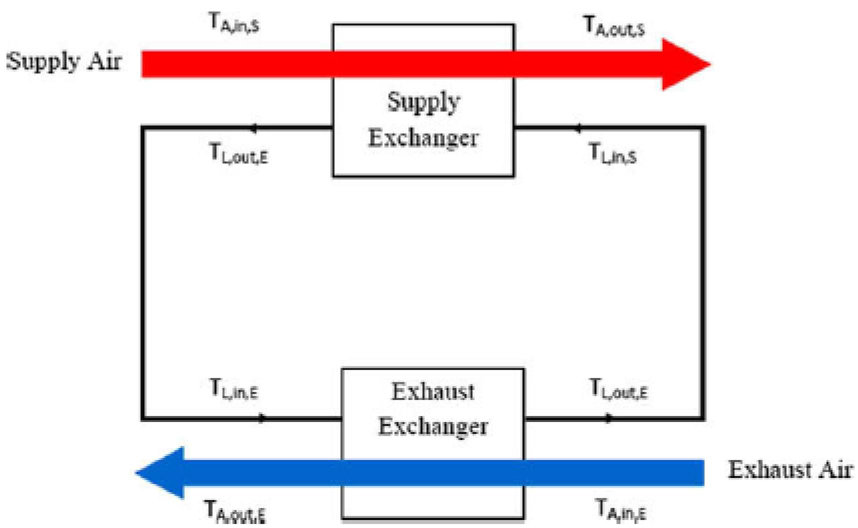

Figure 2.3: Schematic diagram showing the inlet and outlet condition of each exchanger of the run-around heat recovery system.

The run-around heat recovery system has the advantage of allowing us to choose the working fluid freely, but has a disadvantage of having low heat exchange effectiveness. In this application we require a heat recovery system that recovers heat between two fluids of different thermal capacity. The next figure illustrates a schematic view of a heat recovery system applied for a system that has two fluids of different thermal capacity, and the process it the reaction occurring in the reactor.

Alireza Vali [8] mentions that the effectiveness of the individual recuperative heat exchangers is given by:

$$\varepsilon = \frac{C_A(T_{A,in} - T_{A,out})}{C_{min}(T_{A,in} - T_{L,in})}$$

Effectiveness

The **effectiveness** of this type of heat recovery system increases as Number of transfer units (NTU) increases at constant Heat Capacity Ratio (Cr), it also increases as Cr decreases at constant NTU. The figure below clearly indicates this, with both the simulated and correlated results given for different Cr values. The run-around coil literature also gave a direct comparison of three different heat exchangers used in the heat recovery system: cross flow, counter-flow and counter/cross flow with variable NTU & fixed Cr=1.

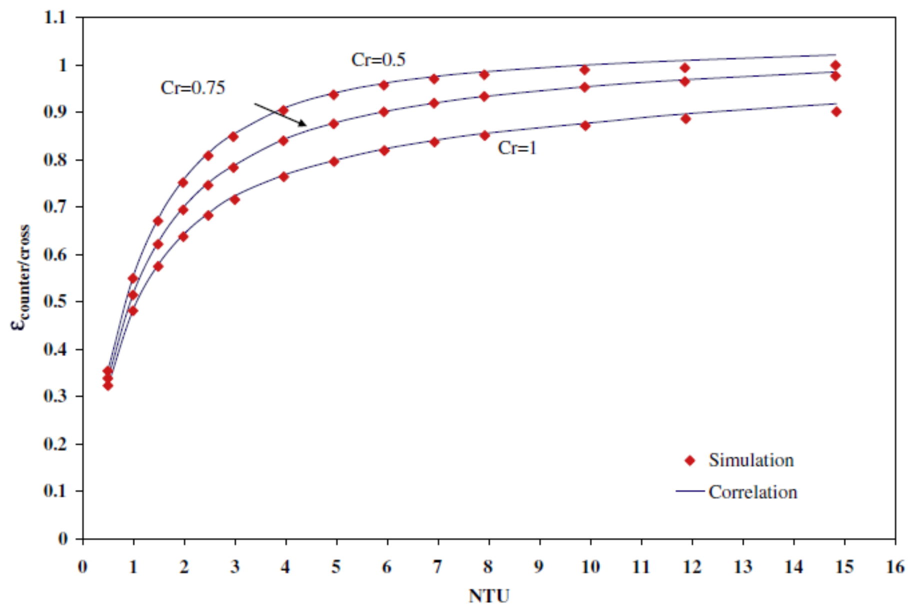

Figure 2.4: Results of the variation of the effectiveness of a counter/cross flow heat exchangers with NTU & Cr, $\frac{y_0}{x_0} = 0.5$, $\frac{x_i}{x_0} = 1$

Figure below shows that the highest effectiveness is given by the counter-flow configuration, the effectiveness lies between that for counter-flow & cross-flow for both simulation and correlation results.[8]

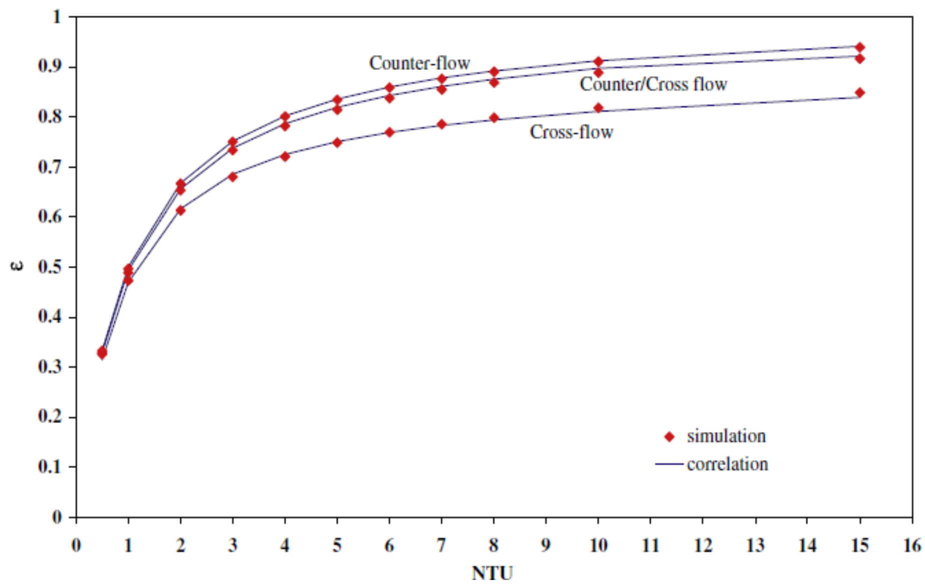

Figure 2.5: Effectiveness results of three flow configurations for simulation and correlation parameters.

The literature goes on to specify the overall effectiveness for the Run-around coil heat recovery system, using simulation and correlation results. The related equations used are given below:

Simulation:

$$\varepsilon_o = \frac{C_{A,S}(T_{A,in,S} - T_{A,out,S})}{C_{\min}(T_{A,in,S} - T_{A,in,E})} = \frac{C_{A,E}(T_{A,out,E} - T_{A,in,E})}{C_{\min}(T_{A,in,S} - T_{A,in,E})}$$

Expression for overall effectiveness of the system:

$$\frac{1}{\varepsilon_o} = \frac{1}{\varepsilon_E} + \frac{1}{\varepsilon_S} - Cr, \quad \text{for} \quad C_A \le C_L$$
$$\frac{1}{\varepsilon_o} = \frac{1}{Cr}\left(\frac{1}{\varepsilon_E} + \frac{1}{\varepsilon_S} - 1\right), \quad \text{for} \quad C_A > C_L$$

Zeng et al. [12] developed a correlation for the heat recovery system with two identical heat exchangers and the same air mass flow rates in supply and exhaust heat exchangers $m_{A,E} = m_{A,S}$ The effectiveness of the counter/cross flow configuration lies between that for pure counter-flow and cross-flow and the proposed correlation is also given, it does not account for changes in the entrance ratio. [8] The third equation predicts a constant effectiveness of 0.72.

$$\varepsilon_{cross} = 1 - exp\left[\left(\frac{1}{Cr}\right)(NTU)^{0.22}\{exp[-Cr(NTU)^{0.78}] - 1\}\right]$$

$$\varepsilon_{counter} = \frac{1 - exp(-NTU(1 - Cr))}{1 - Crexp(-NTU(1 - Cr))}$$

$$\varepsilon_{counter/cross} = \left[\left(\frac{y_0}{x_0}\right)\left(1 + \frac{NTU}{200}\right)\varepsilon_{cross} + \left(1 - \frac{y_0}{x_0}\right)\varepsilon_{counter}\right], \quad \text{for} \quad 0 < x_i/x_0 \leq 0.25$$

<u>Initial Cost</u>

According to ASHRAEJournal a heat recovery system was analyzed at a high efficiency building designed for Montana State University. The heat recovery savings of a flat plate heat exchanger where compared to those for a run-around coil, using a typical year of hourly weather data. The result was that the glycol based run-around coil offered slightly better savings, despite having lower peak heat recovery effectiveness. The run-around coil offered an effectiveness of 60%. The flat plate system effectiveness (i.e. 80%) could not overcome the fan energy cost associated with the flat plate systems. Therefore the low fan cost, hence the low initial cost makes the glycol-based run-around coil a better selection. [22]

2.2.4 Two-Stage Regenerative Heat Exchanger

Introduction

The set-up for this heat exchanger can be better understood by a diagram that illustrates how the heat transfer process takes place. Figure 2.7 shows two diagrams a) and b), where both diagrams have a matrix A and matrix B, with a valve opening to each matrix.

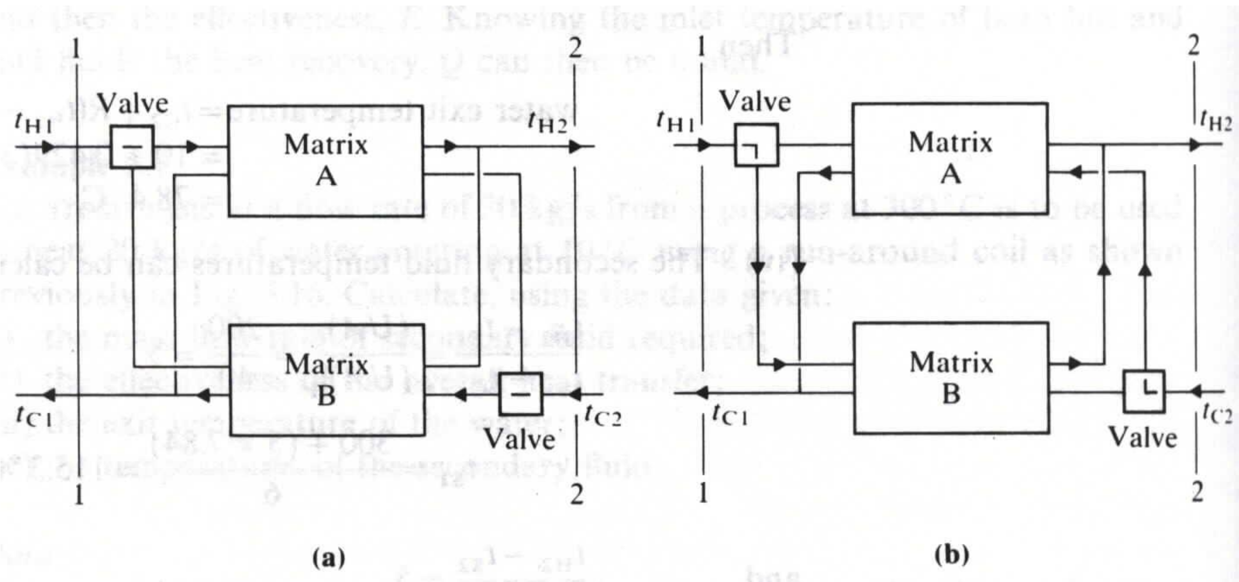

Figure 2.6: Schematic illustration of the Two-Stage Regenerative Heat Exchanger.

In diagram a), the valve opening to Matrix A allows the matrix to store heat from the hot fluid at temperature t_{H1}. Matrix B is already heated up so it transfers heat to the cold fluid which flows in through the valve at temperature t_{C2}.

In diagram b), the hot fluid valve opens to Matrix B allowing the matrix to store up heat. The cold fluid valve now opens to Matrix A allowing the fluid to take up heat stored from diagram a).

This sequence is regulated by the opening and closing of the valves, which allows the matrices to exchange the storing and transferring of heat.

Effectiveness

David G. Wilson presented results of regenerator preliminary designs for a 40kWe engine; this was presented in the form of a table. Only the relevant data for this project were extracted from this preliminary design, the data is given in the table below: [13]

Effectiveness	0.9	0.95	0.975
Number of Transfer Units	10	23	50
Pressure drop, Hot side	2.5%	2.5%	2.5%
Pressure drop, Cold side	0.8%	0.9%	0.93%

Table 2.3: Results of regenerator preliminary designs for a 40kWe engine.

Initial Cost

The construction cost of regenerator core is much less than for recuperators, because the regenerator accepts much smaller passages than the recuperator due to the following reasons:

- The flow in regenerators reverses frequently, so that if the passages are fine enough for the core face to act as a filter, the dirt that is collected is blown off.
- The fine passages do not have to be brazed or otherwise joined to a header, as in a recuperator.

System Pressure drop

N. Natajaran & K. Pitchandi found that the pressure drop in a periodic regenerative heat exchanger to be given by: [14]

$$\mathrm{Re} = \frac{G \times hydraulic\ Dia\ of\ the\ Matrix}{\mu}$$

$$\Delta P = \frac{G^2}{2\rho_1} \cdot \left[\left(1+\sigma^2\right)\left(\frac{\rho_1}{\rho_2}-1\right) + f\,\frac{A}{A_c}\,\frac{\rho_1}{\rho_m} \right]$$

A more detailed evaluation of the periodic regenerative pressure drop was done by Kwanwoo Nam & Sangkwon Jeong, where they conducted measurements of cryogenic regenerator characteristics under oscillating flow and pulsation pressure. The results where given in the form of two graphs, with the mean pressure being between 17bar and 19 bar.[21]

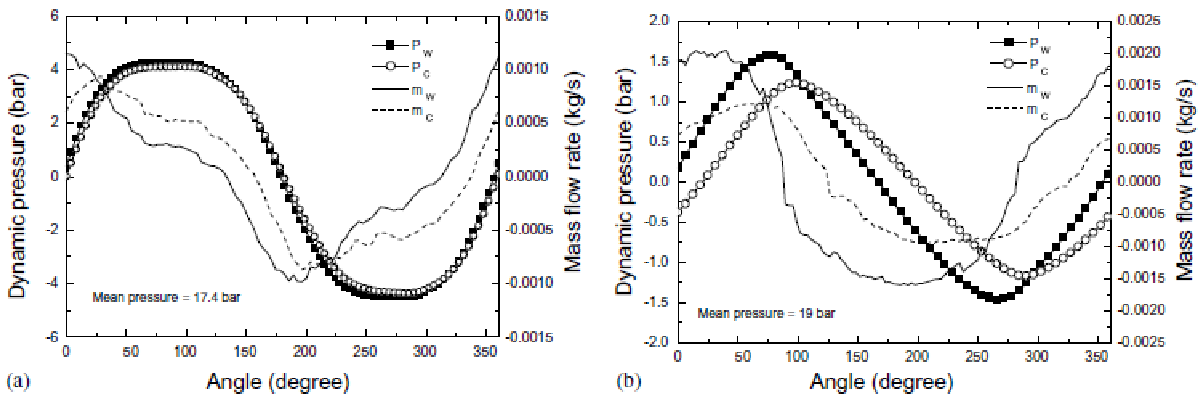

Figure 2.7: Pressure and mass flow rates at low and high frequencies. (a) 4.6Hz (b) 60Hz

2.2.5 Rotary Regenerative Heat Exchanger

Introduction

The figure above illustrates the rotary regenerative heat exchanger which transfers heat from a hot gas to a cold one via a rotating cylinder of densely packed metal sheets, called elements. These elements are packed in containers and slowly rotate through one gas stream and into the other. A hot gas flows over the surface of the metallic elements, raising their temperature. As the rotor turns, at around 1RPM, the heated elements move into the cool gas stream, increasing its temperature accordingly.

Effectiveness

The thermal ***effectiveness*** of a rotary heat exchanger is dependent of the pressure difference on the face of the rotor.[15] Karel Hemzal derives the pressure differences and dependence of leakage air flow values on the pressure difference in his paper on rotary heat exchanger efficiency influenced by air tightness. The effectiveness of the exchanger is given by the equation below: (See figure 2.10 below for flow configuration)

$$\emptyset = (t_{22} - t_{21})/(t_{11} - t_{21})$$

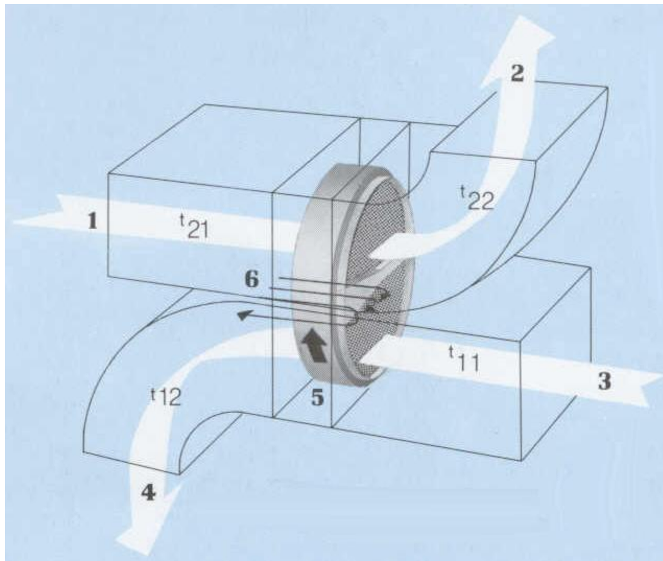

Figure 2.8: Flow configuration in a rotary regenerative heat exchanger.

Ease of Condensation

In a rotary regenerative heat exchanger the condensate film flow of a rotating heat transfer surface is provided by the action of the centrifugal force $\omega^2 R \sin\alpha$. [19]

Sparrow & Hartnet [20] estimated a correlation for the heat transfer of a rotating surface at condensation to be:

$$\frac{h_{rot}}{h_g} = \left[\frac{\omega^2 R sin\ \alpha}{g}\right]^{1/4}$$

Pressure drop

Below is a performance chart for determination of pressure drop and heat recovery figure for different wheel sizes. [24]

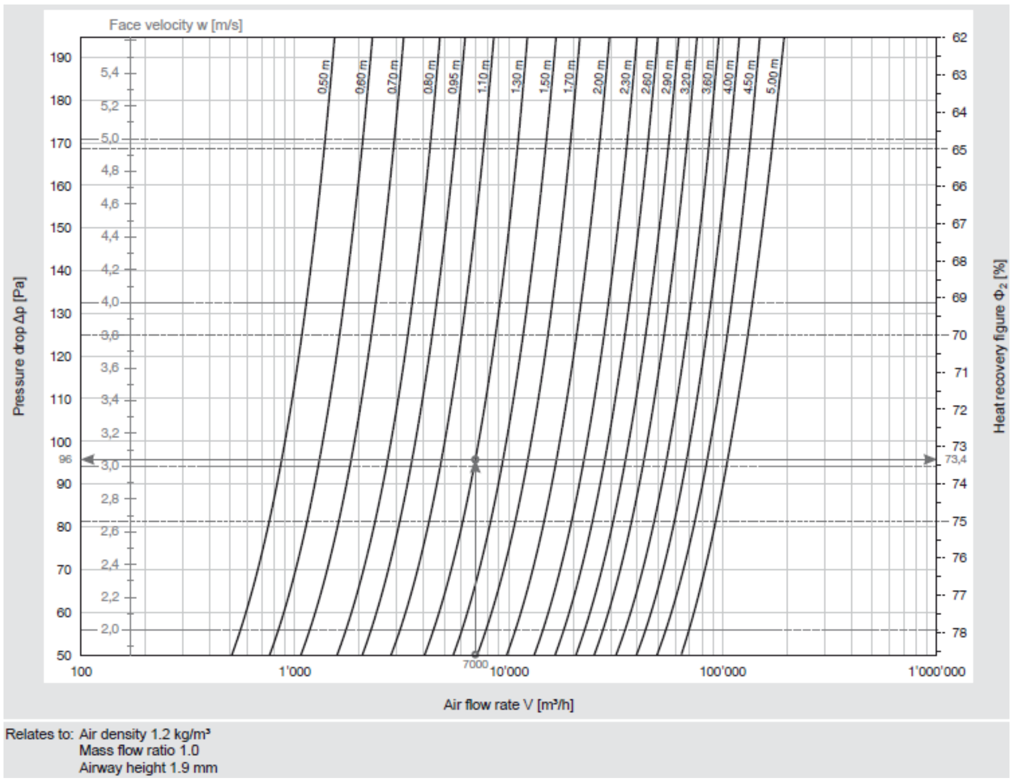

Figure 2.9: Performance chart for determination of pressure drop and heat recovery figure for different wheel sizes.

CHAPTER 3

METHODOLOGY

3 METHODOLOGY

The methodology employed in fulfilling the requirements of this project is a procedure involving the data analysis of a product mixture ($H_2 + \ Light\ Hydrocarbons$). The procedure also involves the following milestones: Construction of a rating table for different heat recovery systems, selection of the most suitable heat recovery system and lastly the design of the selected heat recovery

3.1 Rating Table & Selection criteria

The construction of the rating table will focus on a set of criteria that are of importance to the operation of the heat recovery system. The criteria used in constructing the rating table are:

- Percent heat recovery
- Pressure drop
- Initial Cost
- Ease of hydrocarbon condensation

The above mentioned criteria will be used to qualitatively evaluate the heat recovery systems mentioned in the literature section. The heat recovery system that proves to be the most suitable will be selected and designed.

3.2 Design Methodology

Process Specification

Problem Specification: Layout of the challenges and specific design requirements that need to be fulfilled.

Operation conditions: The maximum temperatures and pressures at which the heat exchanger will be operating.

Heat Exchanger Properties: Flow arrangement, fluid side selection material and surface selection, fin-geometry in case of an extended surface heat exchanger, construction and fabrication methodology.

Thermal and Hydraulic Design

The design takes the surface characteristics and geometrical properties of the exchanger and combines it with the thermo-physical properties of the fluids. The $\varepsilon - NTU$ method is used to analyze the effectiveness of the heat exchanger, to fulfill the objectives of the project. The outlet temperature of the cold stream fluid is also determined to evaluate the amount of external heat that has to be added to the hydrogen gas before it is fed back into the reactor (shown in the schematic in fig. 1.1).

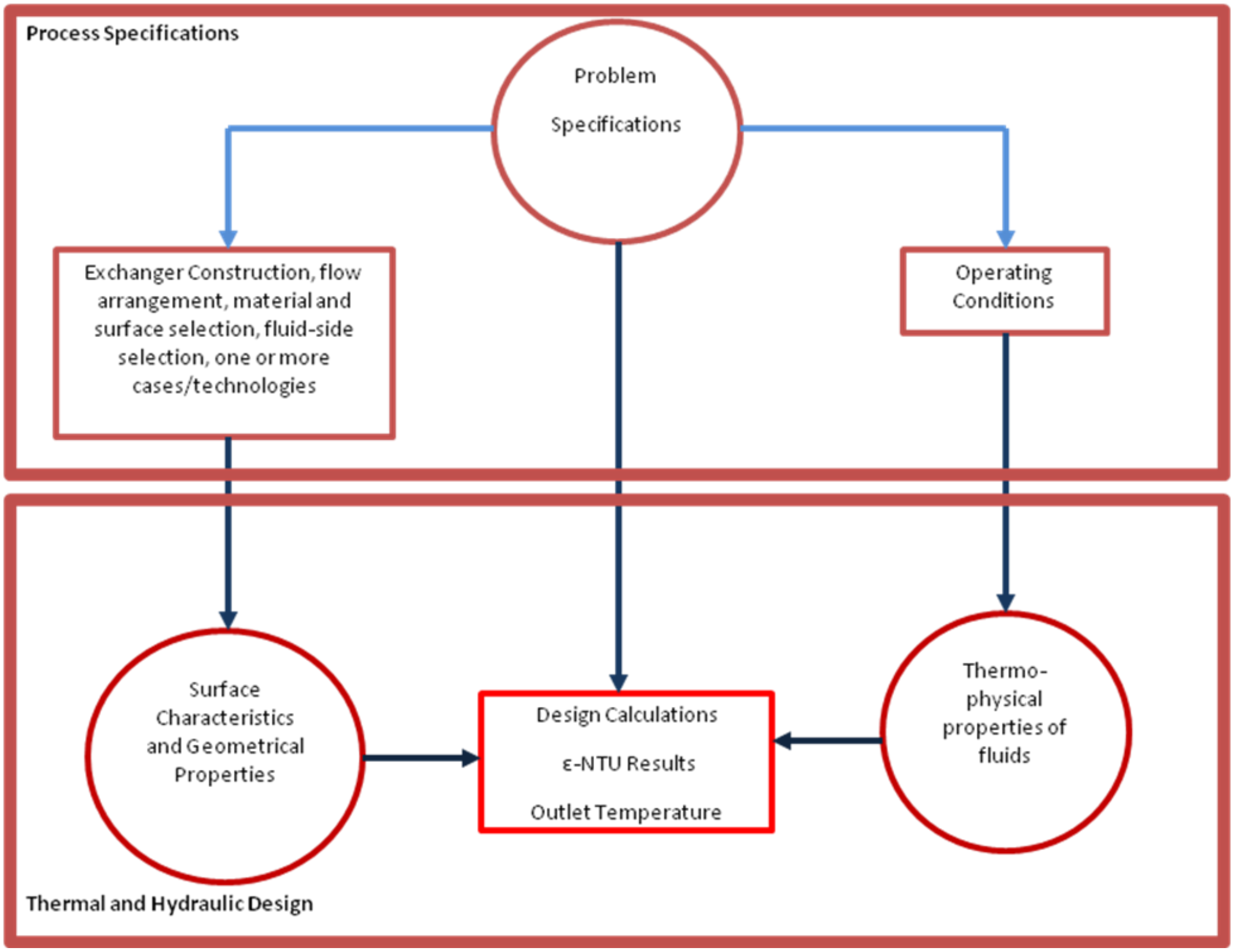

Figure 3.1: Flow-chart of the Design Methodology.

CHAPTER 4
RESULTS AND DISCUSSION

4 RESULTS & DISCUSSION

The qualitative evaluation of the heat recovery systems is done based on the criteria mentioned in the report. Table 4.3 shows the rating table, which serves the purpose of selecting the most suitable heat recovery system for a bio oil conversion system. The selected heat recovery system will be designed using assumed operating conditions (high pressure and temperatures), the typical product compositions from the reactor (given in the literature section) and the Thermo-physical properties of the fluid streams.

4.1 Rating table

The rating table helps with a qualitative evaluation of the different heat recovery systems mentioned in the literature review section. The criteria used in doing this evaluation have also been mentioned in the report and are given in table 3 in the top row (i.e. Percent heat recovery, Pressure drop, Initial cost, Ease of Separation). The table uses a rating scale of 1 to 5, the values defined in the following manner: 1-Poor, 2-fairly poor, 3-good, 4-very good, 5-excellent.

4.1.1 Percent Heat Recovery

The qualitative evaluation of the heat recovery systems using this criterion has meant that more research be done on the individual thermal effectiveness of the systems. It was found that both the Compact Recuperative and Two-Stage regenerative heat exchangers are capable of reaching the highest effectiveness of 98% (see table 4.1). Therefore as far as this criterion is concerned the two exchangers would be the most suitable for the bio-oil conversion system. The table below shows the typical individual heat exchanger effectiveness.

Heat exchangers	% heat recovery
Compact Recuperative heat exchanger	Up to 98%
Shell-&-tube heat exchanger	65% - 75%
Run-around coil heat exchanger	Approx. 72%
Two-Stage Regenerative heat exchanger	90% - 98%
Rotary Regenerative heat exchanger	80% - 87%

Table 4.1: Typical values for individual heat exchanger percent heat recovery

4.1.2 Pressure drop

The pressure drop evaluation between the regenerative heat exchangers (i.e. Rotary & Two-stage) indicated that the two-stage regenerative heat exchanger has a significantly larger system pressure drop. The rotary regenerative heat exchanger will be a preferred option between the two, due to its low pressure drop characteristics. Literature revealed that compact recuperative heat exchanger has a pressure drop characteristic that varies, reaching very high pressures (i.e. up to 400Mbar). The table below shows the typical individual heat exchanger pressure drops.

Heat exchangers	Pressure drop
Compact Recuperative heat exchanger	About 50bar
Shell-&-tube heat exchanger	4.8bar – 20.68bar
Run-around coil heat exchanger	2.49µbar – 0.87µbar
Two-Stage Regenerative heat exchanger	17bar – 19bar
Rotary Regenerative heat exchanger	<0.34 bar

Table 4.2: Typical values for individual heat exchanger pressure drop

4.1.3 Initial Cost

Literature on the initial cost of the heat recovery systems being evaluated suggests that the compact recuperative heat exchanger is the one suited for a bio oil conversion system, due to the exchanger being economical for small mean temperature difference and large heat duties. The shell-&-tube heat exchangers are economical for large mean temperature difference and small heat duties.

4.1.4 Ease of hydrocarbon Condensation

Due to the limited literature on the subject, this criterion hasn’t been fully explored. The literature that implements the criterion suggests that the Rotary regenerative heat exchanger has a better ease of hydrocarbon condensation than Stationary regenerative heat exchanger. Therefore on the rating table Rotary regenerative heat exchangers will be rated higher than the two-stage regenerative heat exchanger.

The rating table below only shows the ratio of each rating of the individual heat exchangers. The ratio is calculated by dividing each individual rate in each column with the highest rate in that column.

Heat Exchanger	% Heat Recovery	Pressure Drop	Initial Cost	High Temperature Operation	High Pressure Operation	Ease of Separation of Condensates	Rating Total
Compact Recuperative Heat Exchanger	0.8	0.2	0.6	0.4	0.4	0.6	3.0
Run around coil with Shell-&-tube Heat Exchanger	0.6	0.4	1	0.8	1.0	1	4.8
Run-Around Coil with Rod-Baffle Heat Exchanger	0.8	1	0.6	0.8	1.0	0.8	5
Two-Stage Regenerative Heat Exchanger	1	0.4	0.4	1.0	0.8	0.5	4.1
Rotary Regenerative Heat Exchanger	0.9	0.3	0.25	0.4	0.2	0.5	2.55

Table 4.3: Rating Table

Rating Scale

1 – Poor, 2 – Fairly Poor, 3 – Good, 4 – Very Good, 5 – Excellent.

4.2 Heat Recovery System Design

The Run-around coil regenerative heat exchanger is selected as the most suitable heat recovery system for this project, with reference to the rating table. The problem specifications for this design have been clearly specified in the introduction section of this report and therefore it would be redundant to specify them again in this section.

The run-around coil heat exchanger employs three fluids, two primary fluids and one secondary fluid. The system uses two individual Rod-Baffled Shell & Tube heat exchangers with the secondary fluid (SYLTHERM 800 Fluid) transferring heat between the two primary fluids in the individual heat exchangers. The selected material for the heat exchanger tubes and baffles is stainless steel; it is suitable for high pressure & temperature applications, with a thermal conductivity of 21.4W/mK at 500°C.

4.2.1 Design Calculations

These include the application of the ε-NTU Method to determine the effectiveness of the heat exchanger; this method is applied because the cold stream outlet temperature of the system is unknown. The hot stream outlet temperature can only be assumed to be approximately equal to the saturation temperature of the condensing hydrocarbon (gasoline).

The following data sheets and design calculations indicate two different run-around coils;

1. Run-around coil system that uses a gas heater to heat the hydrogen gas to the required inlet temperature to the reactor in the bio-oil conversion system.
2. Run-around coil system that uses a liquid heater to heat the heat transfer oil within the run-around coil system.

Supply Exchanger Data Sheet

SHELL & TUBE DETAILS	
TEMA Type: DEL Rod Baffle Exchanger	Number of Shells in Parallel: One
Shell Inside Diameter: 0.736 m	Number of Shells in Series: One
TUBE DETAILS	
Overall tube length: 1.5m	Number of tube-side passes: One
Tube Outside Diameter: 0.0254m	Tube Wall Thickness: 2×10^{-6}
Is the first tube pass: Counter flow	Total number of tubes: 400
Tube Pitch: 0.03175m	Tube Layout angle: 30 degrees
BAFFLE DETAILS	
Baffle Type: Rod Baffle	Number of Baffles: 12
Baffle Spacing: 0.125m	

Operating Conditions

Operating Parameters	Heat Exchanger 1: Supply Exchanger	
	Hot Stream: Product mixture (H_2 gas & HC vapors)	Cold Stream: Heat transfer oil SYLTHERM 800 Fluid
Inlet Temperature	450 °C	250 °C
Outlet Temperature	220 °C	-
Mass flow rate	3.0 kg/s	4.0kg/s
Inlet pressure	40bar	40bar
Heat Duty	1000Kw	1000kW

1. Heat capacities of both streams:

$$C_c = \left(\dot{m}c_p\right)_c = (4.0)(2.001) = 8.004kW/K$$
$$C_h = (\dot{m}c_p)_h = (3.0)(14.58) = 43.74kW/K$$

$$\therefore C_c < C_h$$

2. Determination of the cold stream outlet temperature:

$$T_{c,out} = 250 + \frac{1000}{8.004} = 375°\mathrm{C}$$

3. Heat capacity ratio:

$$C^* = \frac{8.004}{43.74} = 0.18$$

4. Effectiveness:

$$\varepsilon = \frac{C_c}{C_{min}} \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} = \frac{375 - 250}{400 - 250} = 0.83$$

5. $\varepsilon - NTU$ Expression for a counter-flow arrangement:

$$NTU_1 = \frac{1}{1 - C^*} ln\left[\frac{1 - \varepsilon C^*}{1 - \varepsilon}\right] = \frac{1}{1 - 0.18} ln\left|\frac{1 - (0.83)(0.18)}{1 - 0.83}\right| = 2$$

6. Heat transfer surface area:

$$\therefore A = \frac{(8.004kW/K)\,(2)}{(0.3kW/m^2K)} = 53.4m^2$$

The above designed rod-baffled shell-&-tube heat exchanger will be utilized together with the exhaust exchanger in a run-around coil system that has a gas heater after the exhaust heat exchanger. The gas heater will heat up the hydrogen gas to the required inlet feed temperature at the reactor.

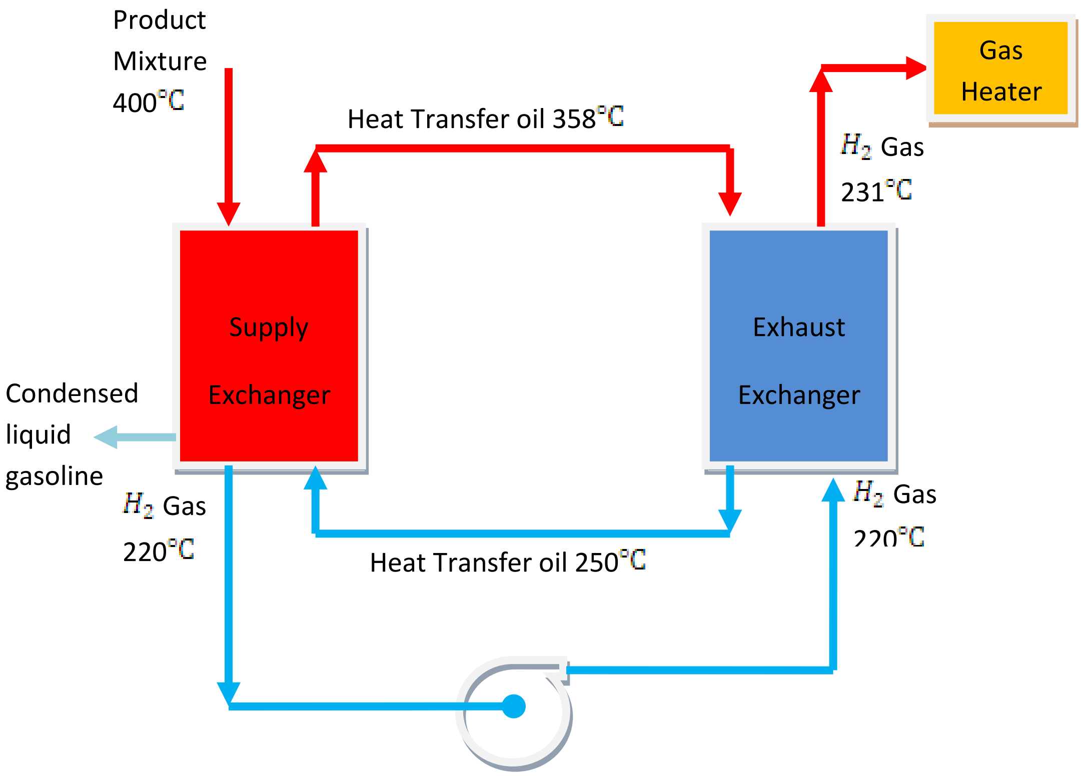

Figure4.1: Schematic diagram for the run-around coil with a gas heater heating the outlet hydrogen gas.

Exhaust Exchanger Data Sheet

SHELL & TUBE DETAILS	
TEMA Type: DEL Rod Baffle Exchanger	Number of Shells in Parallel: One
Shell Inside Diameter: 0.736 m	Number of Shells in Series: One
TUBE DETAILS	
Overall tube length: 1.5m	Number of tube-side passes: One
Tube Outside Diameter:0.0254m	Tube Wall Thickness:2×10^{-6}
Is the first tube pass: Counter-flow	Total number of tubes: 400
Tube Pitch:0.03175m	Tube Layout: 30 degrees
BAFFLE DETAILS	
Baffle Type: Rod Baffle	Number of Baffles: 12
Baffle Spacing: 0125m	

Operating Conditions

Operating Parameters	Heat Exchanger 2: Exhaust Exchanger	
	Hot Stream: Heat Transfer oil SYLTHERM 800 Fluid	Cold Stream: Hydrogen gas
Inlet Temperature	375 ˚C	220 ˚C
Outlet Temperature	250˚C	-
Mass flow rate	4.0 kg/s	6.0kg/s
Inlet pressure	40bar	40bar
Heat Duty	1000kW	1000kW

1. Heat capacities of both streams:

$$C_c = \left(\dot{m}c_p\right)_c = (6.0)(14.58) = 87.48kW/K$$
$$C_h = (\dot{m}c_p)_h = (4.0)(2.2145) = 8.858kW/K$$

$$\therefore C_c > C_h$$

2. Determination of the outlet temperature for both streams:

$$T_{c,out} = 220 + \frac{1000}{87.48} = 231°\mathrm{C}$$

3. Heat capacity ratio:

$$C^* = \frac{8.858}{87.48} = 0.1$$

4. Effectiveness:

$$\varepsilon = \frac{C_h}{C_{min}} \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} = \frac{375 - 250}{375 - 220} = 0.8$$

5. $\varepsilon - NTU$ Expression for a counter-flow arrangement:

$$NTU_2 = \frac{1}{1 - C^*} ln\left[\frac{1 - \varepsilon C^*}{1 - \varepsilon}\right] = \frac{1}{1 - 0.1} ln\left|\frac{1 - (0.8)(0.1)}{1 - 0.8}\right| = 1.7$$

6. Heat transfer surface area:

$$\therefore A = \frac{(8.858kW/K)\,(1.7)}{(0.3kW/m^2K)} = 50m^2$$

The overall effectiveness of the designed heat recovery system is 68.7%. The heat transfer surface areas of both the supply and exhaust exchangers are relatively similar. The cold stream outlet temperature is 231°C, and therefore would require an additional temperature of 169°C from the gas heater before the hydrogen can be recycled back into the reactor. The specifications of the gas heater are taken from FP/BFP' Hazardous Area Process Heater (ref. [25]).

The next design is for a run-around coil system that utilizes a liquid heater to heat up the heat transfer oil within the system in order to give a better heat recovery and to have the hydrogen gas leaving the heat recovery system at the require reactor inlet temperature.

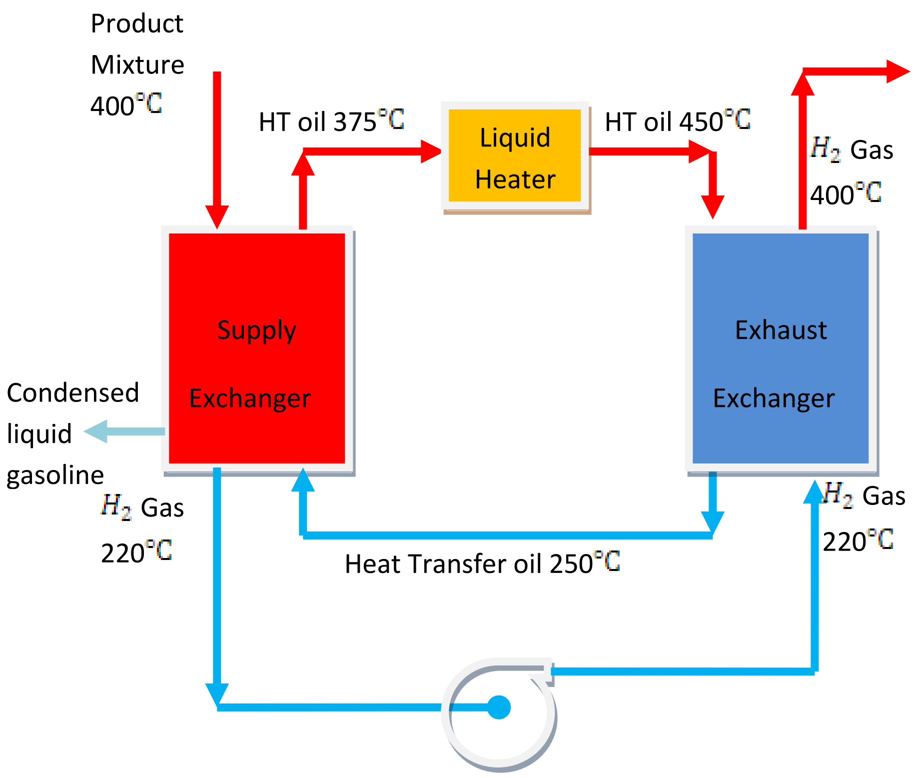

Figure 4.2: Schematic diagram of the run-around coil with a liquid heater heating the heat transfer oil.

Supply Exchanger Data Sheet	
SHELL & TUBE DETAILS	
TEMA Type: DEL Rod Baffle Exchanger	Number of Shells in Parallel: One
Shell Inside Diameter: 0.736 m	Number of Shells in Series: One
TUBE DETAILS	
Overall tube length: 1.5m	Number of tube-side passes: One
Tube Outside Diameter: 0.0254m	Tube Wall Thickness: 2×10^{-6}
Is the first tube pass: Counter flow	Total number of tubes: 400
Tube Pitch: 0.03175m	Tube Layout angle: 30 degrees
BAFFLE DETAILS	
Baffle Type: Rod Baffle	Number of Baffles: 12
Baffle Spacing: 0.125m	

Operating Conditions

Operating Parameters	**Heat Exchanger 1: Supply Exchanger**	
	Hot Stream: Product mixture (H_2 gas & HC vapors)	Cold Stream: Heat transfer oil SYLTHERM 800 Fluid
Inlet Temperature	400 ˚C	250 ˚C
Outlet Temperature	220 ˚C	-
Mass flow rate	3.0 kg/s	4.0kg/s
Inlet pressure	40bar	40bar
Heat Duty	1000kW	1000kW

1. Heat capacities of both streams:

$$C_c = \left(\dot{m}c_p\right)_c = (4.0)(2.001) = 8.004kW/K$$
$$C_h = (\dot{m}c_p)_h = (3.0)(14.58) = 43.74kW/K$$

$$\therefore C_c < C_h$$

2. Determination of the cold stream outlet temperature:

$$T_{c,out} = 250 + \frac{1000}{8.004} = 375°C$$

3. Heat capacity ratio:

$$C^* = \frac{8.004}{43.74} = 0.18$$

4. Effectiveness:

$$\varepsilon = \frac{C_c}{C_{min}} \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} = \frac{375 - 250}{400 - 250} = 0.83$$

5. $\varepsilon - NTU$ Expression for a counter-flow arrangement:

$$NTU_1 = \frac{1}{1 - C^*} ln\left[\frac{1 - \varepsilon C^*}{1 - \varepsilon}\right] = \frac{1}{1 - 0.18} ln\left|\frac{1 - (0.83)(0.18)}{1 - 0.83}\right| = 2$$

6. Heat transfer surface area:

$$\therefore A = \frac{(8.004kW/K)\,(2)}{(0.3kW/m^2K)} = 53.4m^2$$

Exhaust Exchanger Details	
SHELL & TUBE DETAILS	
TEMA Type: DEL Rod Baffle Exchanger	Number of Shells in Parallel: One
Shell Inside Diameter: 0.736 m inches	Number of Shells in Series: One
TUBE DETAILS	
Overall tube length: 1.5m	Number of tube-side passes: One
Tube Outside Diameter:0.0254m	Tube Wall Thickness: 2×10^{-6}
Is the first tube pass: Counter-flow	Total number of tubes: 400
Tube Pitch:0.03175m	Tube Layout: 30 degrees
BAFFLE DETAILS	
Baffle Type: Rod Baffle	Number of Baffles: 12
Baffle Spacing: 0125m	

Operating conditions

Operating Parameters	Heat Exchanger 2: Exhaust Exchanger	
	Hot Stream: Heat Transfer oil SYLTHERM 800 Fluid	Cold Stream: Hydrogen gas
Inlet Temperature	450 ˚C	220 ˚C
Outlet Temperature	250˚C	400˚C
Mass flow rate	4.0 kg/s	6.0kg/s
Inlet pressure	40bar	40bar
Heat Duty	16MW	16MW

1. Heat capacities of both streams:

$$C_c = \left(\dot{m}c_p\right)_c = (6.0)(14.58) = 87.48kW/K$$
$$C_h = (\dot{m}c_p)_h = (4.0)(2.342) = 9.836kW/K$$

$$\therefore C_c > C_h$$

2. Heat capacity ratio:

$$C^* = \frac{9.836}{87.48} = 0.11$$

3. Effectiveness:

$$\varepsilon = \frac{C_h}{C_{min}} \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} = \frac{450 - 250}{450 - 220} = 0.87$$

4. $\varepsilon - NTU$ Expression for a counter-flow arrangement:

$$NTU_2 = \frac{1}{1 - C^*} ln\left[\frac{1 - \varepsilon C^*}{1 - \varepsilon}\right] = \frac{1}{1 - 0.11} ln\left|\frac{1 - (0.87)(0.11)}{1 - 0.87}\right| = 2.18$$

5. Heat transfer surface area:

$$\therefore A = \frac{(9.836kW/K)\,(2.18)}{(0.3kW/m^2K)} = 50m^2$$

The design estimates an overall effectiveness of 73.8%. This run-around coil system is assumed to be able to reheat the hydrogen gas back to the reactor inlet feed temperature of $400°C$, though some adjustments on the heat duty of the exhaust exchanger had to be made. The heating of the heat transfer oil manages to raise the inlet temperature of the oil into the exhaust exchanger and this manages to give the overall run-around coil system a better percent heat recovery when compared to the first design.

Therefore based on the theoretical design and study of the heat recovery systems and without any cost analysis on the part of the heaters supplying the additional heat, the run-around coil heat recovery system utilizing the liquid heater offers a better performance than the one utilizing the gas heater.

CHAPTER 5

CONCLUSIONS & RECOMMENDATIONS

The study has managed to serve as an eye opener into the world of heat recovery systems, qualitatively evaluating different systems using the rating table to select the most suitable heat recovery system. It has also managed to fulfill the requirements of conceptually designing an efficient heat recovery system that has an increased overall effectiveness. Its hydrogen outlet temperature must be at the required feed temperature into the reactor of the bio-oil conversion system.

The application of the ε-NTU method established a design procedure for the study to evaluate the overall effectiveness of the run-around coil heat recovery system. The method was also able to evaluate the hydrogen outlet temperature.

The study took the project a step further by determining the required heat transfer area of each exchanger, to establish a relative idea of how big the unit will be. More studies similar to this one can be done, in which a sizing problem is conducted on the heat recovery system.

From the outlet temperatures determined in this project, a new study can be conducted to design and size the heat recovery system using the LMTD Method. This will be less tedious because the inlet and outlet temperatures would be known.

REFERENCES

[1]. “Compact Heat Exchangers for Condensation Applications”, C. B. Panchal, Argonne National Laboratory, Argonne July 1993

[2]. “Design of Compact Plate Fin Heat Exchanger”, Jainender Dewatwal, Department of Mechanical Engineering, National Institute of Technology, Rourkela, 2009

[3]. Website: www.scribd.com/doc/20301504/Lect-2

[4]. Website: www.sammicheal.com/ftp/process/class13.pdf

[5]. “Condensation studies using cross-corrugated polymer film compact heat exchangers”, J. R. Burns, R. J. J Jachuck, April 2000

[6]. “Circulating fluidized bed gasifier for biomass”, Guangzhou Institute of Energy Conversion, Chinese Academy of Sciences, 1994.

[7]. “Hydrogen production via catalytic steam reforming of fast pyrolysis bio-oil in a two-stage fixed bed reactor”, C. Wu. Q. Huang, M. Sui, Y. Yan, F. Wang, Research Center for Biomass Energy, State Key Laboratory of Bioreactor Engineering, East China University of Science and Technology, Shanghai 2002

[8]. “Numerical model and effectiveness correlations for a run-around heat recovery system with combined counter and cross flow exchangers”, A. Vali, C. J. Simonson, R. W. Besant, G. Mahmood, Department of Mechanical Engineering, University of Saskatchewan, July 2009.

[9] “Fundamentals of Heat Exchanger Design”, R. K. Shah, D. P. Sekulic, Wiley 2003.

[10] “Thermal Design of Heat Exchangers a numerical Approach”, E. M. Smith, Wiley 1997.

[11] “Heat Exchangers Selection, Rating and Thermal Design”, Sadik Kakac, Hongtan Liu, 2nd edition, CRC Press LLC 2002.

[12] “The effect of temperature-dependent properties on the performance of run-around heat recovery systems using aqueous-glycol coupling fluids”, Y.Y. Zeng, R.W. Besant, K.S. Rezkallah, ASHRAE Trans. 98 (l) (1992) 551–562.

[13] “Regenerative heat exchangers for microturbines and an improved type”, D. G. Wilson, Wilson Turbo-Power Inc, Proceedings of ASME Turbo Expo 2003.

[14] “Thermodynamic optimization of periodic regenerative heat exchanger”, N. Natajaran, K. Pitchandi, World Academy of Science, Engineering and Technology 56 2009

[15] “Rotary Heat Exchanger efficiency influenced by air tightness”, K. Hemzal, Czech Technical University of Prague, 2006.

[16] "Estimating Costs of Shell-and-Tube Heat Exchangers," Purohit, G.P., *Chemical Engineering*, pg 56-67, August 22, 1983.

[17] “Advantages of Brazed Heat Exchangers in the Gas Processing Industry”, K. M. Lunsford, Bryan Research & Engineering, Inc., Bryan, Texas 1996.

[18] Website: www.fao.org/docrep/T4470E/t4470e0m.htm#TopOfPage

[19] “Condensation Heat Transfer Enhancement”, V. G. Rifert, H. F. Smirnov, WITPress, Southampton, Boston, 2004.

[20] “Condensation on rotating cone” E. Sparrow, J. Hartnet, J. Heat Transfer, 83(1), 1961.

[21] “Measurement of cryogenic regenerator characteristics under oscillating flow and pulsating pressure”, Kwanwoo Nam, Sangkwon Jeong, Cryogenic Engineering Laboratory, Korea Advanced Institute of Science and Technology, 2003.

[22] “Laboratory low pressure drop designs” J. Weale, P.E., P. H. Rumsey, P.E., D. Sartor, P.E., and Lee Eng Lock, ASHRAEJournals, 2002.

[23] “Hoval Rotary Heat Exchanger for Heat Recovery in Ventilation Systems”, Handbook for Design, Installation and Operation, Art.Nr. 4 208 534 – 04 / 2009.

[24] “Heat Transfer, A Basic Approach”, M. N. Özişik, McGraw-Hill, 1985.

[25] “‘FP/BFP’ Range of Explosionproof/Flameproof Certified Process Flow Heaters”, Product datasheet, Hazardous Area Process Heat & Control Solutions.

APPENDICES

GANNT CHART

	Task Name	Duration	Start	Finish
1				
2	− **Problem Specifications**	**10 days**	**Mon 1/25/10**	**Fri 2/5/10**
3	Heat Recovery system deliverables	2 wks	Mon 1/25/10	Fri 2/5/10
4	Progress Report 1 Submission	1 day	Fri 2/19/10	Fri 2/19/10
5	− **Qualitative Evaluation of heat recovery systems**	**28 days**	**Mon 2/8/10**	**Wed 3/17/10**
6	Literature review on heat recovery systems	15 days	Mon 2/8/10	Fri 2/26/10
7	Rating table construction	10 days	Mon 3/1/10	Fri 3/12/10
8	Selection of suitable system to be designed	3 days	Mon 3/15/10	Wed 3/17/10
9	Progress Report 2 Submission & 1st Seminar	1 day	Mon 3/22/10	Mon 3/22/10
10	− **Heat Recovery System Design**	**26 days**	**Thu 3/18/10**	**Thu 4/22/10**
11	Run-around coil heat recovery system design	7 days	Thu 3/18/10	Fri 3/26/10
12	Rod-Baffled Shell-&-Tube Heat exchangers	4 days	Mon 3/29/10	Thu 4/1/10
13	Comparison between the two Designs	14 days	Mon 4/5/10	Thu 4/22/10
14	Poster Exhibition	1 day	Mon 4/12/10	Mon 4/12/10
15	Submission of Dissertation final draft	1 day	Mon 5/3/10	Mon 5/3/10
16	Oral	1 day	Mon 5/10/10	Mon 5/10/10