The copyright of this thesis vests in the author. No quotation from it or information derived from it is to be published without full acknowledgement of the source. The thesis is to be used for private study or non-commercial research purposes only.

Published by the University of Cape Town (UCT) in terms of the non-exclusive license granted to UCT by the author.
MEC5061Z

Dissertation towards partial fulfilment of the Degree of Master of Science in Sustainable Energy Engineering

Feasibility Study of Heat Pumps for Waste Heat Recovery in Industry

Author: Devin de Waal

(DWLDEV001)

Date Submitted: 8 February 2012
Disclaimer

I know the meaning of plagiarism and declare that all the work in the document, save for that which is properly acknowledged, is my own.

______________________________
Devin de Waal

8 February 2012

University of Cape Town
Acknowledgements

I would like to thank the following people for their assistance and support throughout this thesis without which it would not have been possible.

- My supervisor, Andrew Hibbard, for agreeing to supervise my dissertation and for providing valuable guidance
- Douglas February and Peter Hofmann for providing the data to make this dissertation possible
- The ERC staff for offering helpful advice
- Ann Steiner for all her help and encouragement
- My family and friends for all the support and encouragement
Abstract

Heat pumps are used extensively in the residential and commercial sectors in South Africa for water heating but so far there are no published installations of industrial heat pumps for process heating. Internationally, however, they are used successfully in the food and beverage, pulp and paper and chemical sectors to name a few. The industrial sector is particularly energy intensive and thermal energy makes up the majority of the total energy consumption. A high percentage of this is low-grade waste heat that has the potential to be upgraded to a higher quality for re-use through the use of an industrial heat pump. With South Africa’s current tight electricity supply, massive price increases forecast and the increasing awareness of climate change, industrial heat pumps integrated for waste heat recovery can achieve the large-scale energy savings required while being economically viable.

A case study was thus carried out at an applicable local industry (brewery) to assess the feasibility of implementing the heat pump for waste heat recovery. Through analysis, the focus was narrowed down from a site wide audit, to a departmental breakdown and then eventually to a specific process; the wort boiler. Three different alternatives were investigated and the performance and economic viability compared; a simple waste heat recovery solution involving a vapour condenser (VC), a mechanical vapour recompression (MVR) heat pump and a thermal vapour recompression (TVR) heat pump.

It was found that the MVR system yielded the greatest energy savings, followed by the VC and then the TVR system. All three systems had positive rates of return, with the VC and TVR systems being tied for first place. This was due to the high investment cost of the MVR system combined with a greater electricity to steam price ratio. The VC system was only valid for this specific scenario though, as there was a lower temperature sink for the excess waste heat. In other studies this might not be the case. When extrapolating the data to other similar industries, heat pumps in this specific sub-sector alone could go a long way in meeting demand side management targets and reducing South Africa’s overall energy intensity.
Table of Contents

Acknowledgements................................................................................................................................. ii
List of Figures ........................................................................................................................................ vii
List of Tables ........................................................................................................................................ viii
Nomenclature ........................................................................................................................................ ix

1 Introduction....................................................................................................................................
  1.1 Background ............................................................................................................................. 1
  1.2 Research Objectives ................................................................................................................ 3
  1.3 Chapter Overview ................................................................................................................... 3

2 Literature Review ................................................................................................................
  2.1 Heat Pump Theoretical Overview ................................................................................................
    2.1.1 Basic principles................................................................................................................ 4
    2.1.2 Vapour-Compression ...................................................................................................... 8
    2.1.3 Absorption ..................................................................................................................... 11
    2.1.4 Components .................................................................................................................. 12
    2.1.5 Working fluids ............................................................................................................... 16
    2.1.6 Heat sources .................................................................................................................. 20
    2.1.7 Operational modes ....................................................................................................... 20
    2.1.8 Performance characteristics ......................................................................................... 21
  2.2 Waste Heat Recovery Methods ............................................................................................... 23
    2.2.2 Waste Heat Recovery Options and Technologies ......................................................... 23
    2.2.3 Factors Affecting Waste Heat Recovery Feasibility ....................................................... 26
  2.3 Energy Use in Industry ............................................................................................................ 31
  2.4 International Surveys, Case Studies and Best Practices ...................................................... 32
    2.4.1 Surveys.......................................................................................................................... 32
    2.4.2 Case Studies .................................................................................................................. 34

3 Methodology ................................................................................................................................
  3.1 Research Overview ............................................................................................................... 35
  3.2 Design Method...................................................................................................................... 36
    3.2.1 Site selection ................................................................................................................ 37
    3.2.2 Data collection ............................................................................................................. 37
    3.2.3 System design ............................................................................................................. 38
    3.2.4 Economic analysis ....................................................................................................... 38
3.3 Design Tools .......................................................................................................................... 39
  3.3.1 Pinch Analysis ................................................................................................................. 40
  3.3.2 Design, calculation and simulation software tools .................................................... 41
4 Results ....................................................................................................................................... 42
  4.1 Preliminary Analysis .......................................................................................................... 42
    4.1.1 Company selection ........................................................................................................ 42
    4.1.2 Site focus ....................................................................................................................... 42
  4.2 Data Used in the Analysis .................................................................................................. 42
    4.2.1 Company and process description .............................................................................. 42
    4.2.1 Data supplied by the company ..................................................................................... 44
    4.2.3 Other data used in the analysis ..................................................................................... 45
  4.3 Energy Consumption in the Present State ......................................................................... 45
    4.3.1 Site-Wide Fuel and Electricity Consumption ............................................................. 45
    4.3.2 Brewhouse Process Heat and Cooling Demand ......................................................... 47
    4.3.3 Wort boiler heating ....................................................................................................... 52
    4.3.4 Comparison with Benchmark Reference Data .............................................................. 53
  4.4 Description and Design of Alternatives Proposed .............................................................. 55
    4.4.1 Vapour condenser ......................................................................................................... 55
    4.4.2 Mechanical Vapour Recompression ............................................................................. 56
    4.4.3 Thermal Vapour Recompression ................................................................................... 58
    4.4.4 Combination heat pump and waste heat recovery ...................................................... 59
  4.5 Comparative Study of the Proposed Alternatives .............................................................. 60
    4.5.1 Thermal performance ................................................................................................... 60
    4.5.2 Emissions saving potential ............................................................................................ 61
    4.5.2 Economic analysis ......................................................................................................... 62
  4.6 South African Energy Savings Potential ............................................................................ 64
5 Discussion of Results .................................................................................................................. 65
  5.1 Preliminary Analysis .......................................................................................................... 65
  5.2 Data Used in the Analysis .................................................................................................. 65
  5.3 Energy Consumption in the Present State ......................................................................... 65
  5.4 Description and Design of Alternatives Proposed .............................................................. 66
  5.5 Comparative Study of the Proposed Alternatives .............................................................. 66
  5.6 South African Energy Savings Potential ............................................................................ 67
# List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Heat pump and heat engine operating between temperature levels $T_z$ and $T_1$</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>Schematic and T-s diagram for the ideal vapour compression refrigeration cycle</td>
<td>5</td>
</tr>
<tr>
<td>3</td>
<td>The P-h diagram of an ideal vapour compression refrigeration cycle</td>
<td>6</td>
</tr>
<tr>
<td>4</td>
<td>Schematic of an NH$_3$H$_2$O absorption cycle</td>
<td>7</td>
</tr>
<tr>
<td>5</td>
<td>Closed-cycle compression schematic</td>
<td>8</td>
</tr>
<tr>
<td>6</td>
<td>Schematic of mechanical vapour recompression</td>
<td>9</td>
</tr>
<tr>
<td>7</td>
<td>Ejector cycle</td>
<td>10</td>
</tr>
<tr>
<td>8</td>
<td>Simplified energy balance for absorption heat pump</td>
<td>11</td>
</tr>
<tr>
<td>9</td>
<td>Simplified energy balance for heat transformer</td>
<td>12</td>
</tr>
<tr>
<td>10</td>
<td>Influence of temperature difference on required heat exchanger area</td>
<td>29</td>
</tr>
<tr>
<td>11</td>
<td>Energy flows in an industrial heat supply system</td>
<td>31</td>
</tr>
<tr>
<td>12</td>
<td>Standard process model with one incoming and one outgoing stream</td>
<td>32</td>
</tr>
<tr>
<td>13</td>
<td>Distribution of the heat demand per industrial branch (top) and usage (bottom)</td>
<td>33</td>
</tr>
<tr>
<td>14</td>
<td>EINSTEIN’s 10 steps of the auditing procedure</td>
<td>36</td>
</tr>
<tr>
<td>15</td>
<td>Pinch curves (a) Overlapping of the HCC and CCC (b) design of the GCC</td>
<td>40</td>
</tr>
<tr>
<td>16</td>
<td>Summary of typical brewing process</td>
<td>43</td>
</tr>
<tr>
<td>17</td>
<td>Chart showing distribution of electricity demand</td>
<td>45</td>
</tr>
<tr>
<td>18</td>
<td>Chart showing distribution of heat demand</td>
<td>46</td>
</tr>
<tr>
<td>19</td>
<td>Chart showing distribution of total energy demand</td>
<td>46</td>
</tr>
<tr>
<td>20</td>
<td>Monthly heating and cooling demand for period April 2010 to January 2011</td>
<td>47</td>
</tr>
<tr>
<td>21</td>
<td>Distribution of heat demand by temperature levels</td>
<td>49</td>
</tr>
<tr>
<td>22</td>
<td>Distribution of heat demand to show external heat supplied vs heat recovery</td>
<td>49</td>
</tr>
<tr>
<td>23</td>
<td>Distribution of cooling demand by temperature levels</td>
<td>50</td>
</tr>
<tr>
<td>24</td>
<td>Distribution of cooling demand to show external cooling supplied vs heat recovery</td>
<td>51</td>
</tr>
<tr>
<td>25</td>
<td>Hot and cold composite curves</td>
<td>51</td>
</tr>
<tr>
<td>26</td>
<td>Remaining yearly energy demand and energy availability</td>
<td>51</td>
</tr>
<tr>
<td>27</td>
<td>Schematic of existing system</td>
<td>55</td>
</tr>
<tr>
<td>28</td>
<td>Schematic of wort boiling vapour condenser</td>
<td>55</td>
</tr>
<tr>
<td>29</td>
<td>Schematic of mechanical vapour recompression</td>
<td>57</td>
</tr>
<tr>
<td>30</td>
<td>Schematic of thermal vapour recompression</td>
<td>58</td>
</tr>
<tr>
<td>31</td>
<td>Overview of energy consumption and savings for each of the three alternatives</td>
<td>61</td>
</tr>
<tr>
<td>32</td>
<td>Cumulative net savings of each of the alternatives</td>
<td>63</td>
</tr>
</tbody>
</table>
List of Tables

Table 1: Performance and economic comparison of industrial heat pump types ........................................ 22
Table 2: Comparison of refrigerants for one stage simple vapour compression cycle ................. 22
Table 3: Temperature classification of waste heat sources and recovery opportunities ................. 28
Table 4: Range of heat transfer coefficients for sensible heat transfer in tubular exchangers ......... 30
Table 5: Number of installations and heating capacity for the IHPs reported .................................. 34
Table 6: Examples of heat pump applications ................................................................................. 34
Table 7: Useful process heat demand by process ........................................................................... 48
Table 8: Useful process cooling demand by process ........................................................................ 48
Table 9: Process heat demand and supply by temperature levels .................................................... 48
Table 10: Process cooling demand and supply by temperature levels ............................................. 50
Table 11: Table showing recorded energy usage in the wort boiler ............................................. 52
Table 12: Table showing EINSTEIN calculated energy usage in the wort boiler ............................ 53
Table 13: Table showing volume and energy released in the boil-off phase .................................. 53
Table 14: Table showing overall energy benchmark figures ............................................................ 54
Table 15: Table showing process specific benchmark figures ............................................................ 54
Table 16: EINSTEIN calculated vapour condenser heat exchanger data ........................................ 56
Table 17: EINSTEIN calculated vapour condenser heat transferred ................................................ 56
Table 18: EINSTEIN calculated mechanical vapour recompression system data ............................ 57
Table 19: EINSTEIN calculated mechanical vapour recompression heat capacity ....................... 58
Table 20: EINSTEIN calculated thermal vapour recompression system data ................................ 59
Table 21: EINSTEIN calculated thermal vapour recompression heat capacity ................................ 59
Table 22: Combined heat pump and heat exchanger recovered heat .................................................. 60
Table 23: Table showing annual supply heat consumption and savings for each of the alternatives .60
Table 24: Table showing annual electricity consumption and savings for each of the alternatives ... 60
Table 25: Table showing total annual energy consumption and savings for each of the alternatives .61
Table 26: Table showing relative energy savings for the three alternatives .................................... 61
Table 27: Table showing CO₂ savings for each of the alternatives .................................................. 62
Table 28: Specific investment costs .................................................................................................. 62
Table 29: Life cycle costs and savings of the base case and three alternatives .............................. 63
Table 30: Feasibility indicators of the three alternatives ................................................................. 63
Table 31: National savings for the three alternatives ....................................................................... 64
Table 32: Extract from IRP2010 - Energy Efficiency Demand Side Management .............................. 64
# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q</td>
<td>Heat</td>
</tr>
<tr>
<td>Q__E</td>
<td>Cooling capacity (evaporator)</td>
</tr>
<tr>
<td>Q__C</td>
<td>Heating capacity (condenser)</td>
</tr>
<tr>
<td>W</td>
<td>Work</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat</td>
</tr>
<tr>
<td>\eta</td>
<td>Efficiency</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>IRP</td>
<td>Integrated resource plan</td>
</tr>
<tr>
<td>IEA</td>
<td>International energy association</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone depletion potential</td>
</tr>
<tr>
<td>GWP</td>
<td>Global warming potential</td>
</tr>
<tr>
<td>IHP</td>
<td>Industrial heat pump</td>
</tr>
<tr>
<td>FET</td>
<td>Final energy thermal</td>
</tr>
<tr>
<td>USH</td>
<td>Useful supply heat</td>
</tr>
<tr>
<td>UPH</td>
<td>Useful process heat</td>
</tr>
<tr>
<td>CCC</td>
<td>Cold composite curve</td>
</tr>
<tr>
<td>HCC</td>
<td>Hot composite curve</td>
</tr>
<tr>
<td>GCC</td>
<td>Grand composite curve</td>
</tr>
<tr>
<td>CCC</td>
<td>Closed cycle compression</td>
</tr>
<tr>
<td>MVR</td>
<td>Mechanical vapour recompression</td>
</tr>
<tr>
<td>TVR</td>
<td>Thermal vapour recompression</td>
</tr>
<tr>
<td>AHP</td>
<td>Absorption heat pump</td>
</tr>
<tr>
<td>HT</td>
<td>Heat transformer</td>
</tr>
<tr>
<td>VC</td>
<td>Vapour condenser</td>
</tr>
<tr>
<td>IRR</td>
<td>Internal rate of return</td>
</tr>
</tbody>
</table>
1 Introduction

1.1 Background

Heat pumps are being used successfully in the residential and commercial sectors in South Africa for water heating applications, but thus far there are no published installations of heat pumps in the industrial sector in South Africa.

In the residential sector, 39 671 GWh (DoE, 2009) of electricity is consumed yearly of which water heating accounts for approximately 36% (Harris, Kilfoil, & Uken, 2007). Recent Eskom initiatives have introduced a subsidy for both solar water heaters and heat pumps to help achieve a government target of 10 000 GWh from renewable energy by 2013 (Eskom, 2011). Due to the small size of the installations and South Africa's high solar radiation levels, solar water heaters are the preferred choice to meet these targets.

In the commercial sector, 28 833 GWh (DoE, 2009) of electricity is consumed yearly of which heat pump water heaters have penetrated 16% of the water heater market (Harris et al., 2007). These installations range from hotels to universities, prisons, old age homes, swimming pools, schools and hospitals. The market has a scope for higher penetration due to the suitability of heat pumps for large scale, low temperature heating applications. The water heater market is seen as established with many case studies to draw experience from and validate savings.

In the industrial sector, 116 631 GWh (DoE, 2009) of electricity is consumed yearly with a total final energy consumption of 241 884 GWh. There are, however, no published installations of heat pumps for hot water heating, space heating or process heating. The Department of Energy published an Energy Efficiency Baseline Study (DoE, 2002) listing all the possible energy saving opportunities in the various sectors. For the year 2011, the potential savings from thermal measures were estimated to be 8 806 GWh (92% of total savings) in the iron and steel sector, 2 083 GWh (87%) in the pulp and paper sector, 556 GWh (17%) in the mining sector, 7 861 GWh (85%) in the chemical and petrochemical sector, 1 333 GWh (71%) in the non-metallic minerals sector, 833 GWh (59%) in the non-ferrous metals sector, 417 GWh (83%) in the textile sector and 3 278 GWh (92%) in the food and beverage sector. These savings combined represent over 10% of the energy consumption in the industrial sector and far surpass the total Energy Efficiency Demand Side Management estimate of 3 020 GWh electricity savings in 2011 as stated in the IRP2010 (DoE, 2011).

The lack of installations in the industrial sector could be attributed to perceptions of unattractive economic feasibility, a low electricity / primary energy price, lack of knowledge of available technologies and few local installations to draw experience from.

Internationally, however, the situation is rather different. A report by the IEA Heat Pump Programme (IEA, 1995) shows that there are over 4 600 industrial heat pump installations across the United States, Canada, Japan, United Kingdom, Norway and other European countries with an average installation size of over 7.5 MW. The installations are predominantly in the food and beverage, pulp and paper and chemical sectors with the metal, lumber, textile and petroleum sectors comprising a small share.
In the chemical sector the majority of heat pumps are used for evaporation plants, waste heat recovery and distillation columns. In the food and beverage industry they are used for evaporation, wort boiling, distillation, drying and waste heat recovery. In the pulp and paper industry they are used mainly for evaporation. The main application across all sectors is either drying or general heat recovery.

The majority of the installed systems are the closed cycle compression heat pump type but the total installed heating capacity is mainly achieved by the mechanical vapour recompression heat pump type. Other types include the thermal vapour recompression, absorption heat pump and heat transformer.

With South Africa’s GDP growth forecast at 3% in 2012 rising to 3.7% in 2013 (Fin24 2011) and an electricity growth rate forecast of between 2.7% - 3.2% (DoE, 2011), combined with a tight electricity supply until the end of 2013/2014, there is a great need for demand side management initiatives that have a big impact on reducing the current consumption.

While the demand needs to be reduced to prevent rolling blackouts, it also makes economic sense to do so. Historically, South African energy and electricity prices in particular have been very low. However, with the Nersa approved electricity price increases of 24.8% in 2010/2011, 25.8% in 2011/2012 and 25.9% in 2012/2013 (Eskom, 2011), the opportunity is becoming ever more apparent to make energy reduction measures a primary concern.

There is also an increasing awareness of climate change and the environmental impacts of greenhouse gas emissions. South Africa’s electricity supply and energy use is very carbon intensive and thus there is a need to reduce CO₂ emissions so as to transition to a low carbon economy. There are increasing financial incentives for this whereby either penalties will be imposed or carbon trading markets are used.

Industrial heat pumps with a focus on integrating them for waste heat recovery can achieve large-scale energy savings across a number of industrial sectors. These energy savings will culminate in both financial and carbon savings while easing the currently tight supply/demand crisis. It is therefore believed that a feasibility analysis of the performance and economic viability of industrial heat pumps for waste heat recovery will be valuable for industry and governmental decision makers.
1.2 Research Objectives

The objective of this study is to provide evidence supporting this thesis statement:

*An industrial heat pump system used for waste heat recovery is currently a good investment for South African industry and will reduce the company's total energy consumption.*

The objective will be achieved by evaluating:

- Which industries heat pumps are applicable to
- The energy use in a specific industry according to process and energy type
- The proportion of the total use for thermal purposes
- The proportion that waste heat recovery and heat pumps can displace energy use
- The performance of industrial heat pumps under local conditions
- The economic feasibility of waste heat recovery and heat pump integration

1.3 Chapter Overview

The remainder of this report comprises the literature review, methodology, results, discussion, conclusion and recommendations.

The literature review provides an in-depth review of the literature relating to (i) heat pump and waste heat recovery principles, existing technologies, operational characteristics and factors affecting their use, (ii) which systems are appropriate for this study, (iii) the energy use in industry and (iv) international case studies and surveys. The methodology chapter discusses the methods used to achieve the objectives including the research approach, design method and tools used. The results chapter presents the research results and analysis of those results for the industry that heat pumps are applicable to, the energy consumption in a specific case study and the performance and economic analysis of the alternatives proposed. The discussion chapter highlights the analysis of the results obtained during the study with the conclusion chapter summarising conclusions drawn from the study. Recommendations are then made for further potential research in the subject where it is required.
2 Literature Review

This chapter provides an in-depth review of the literature relating to the following topics:

- Heat pump principles, existing technologies, operational characteristics and factors affecting its use
- Waste heat recovery principles, existing technologies, operational characteristics and factors affecting its use
- Which systems are appropriate for this study
- Energy use in industry to understand where energy is used and which industries are appropriate for this study
- International case studies to determine feasibility of this study

2.1 Heat Pump Theoretical Overview

Heat pumps are devices used to raise the temperature of low-grade heat energy to a more useful level using a relatively small amount of high-grade energy.

Refrigerators and heat pumps are essentially the same device but differ in their objectives only. A refrigerator maintains a refrigerated space at a low temperature by removing heat from it. Discharging this heat to a higher temperature medium is a necessary part of the operation and not the purpose. The objective of a heat pump, however, is to discharge heat into a required heating space by absorbing heat from a lower temperature source.

There are two main cycles on which heat pumps operate; either the vapour compression cycle with a compressor which requires the input of mechanical drive energy or the absorption cycle which, instead of a mechanical compressor, requires thermal drive energy. There are other thermodynamic cycles such as adsorption systems, Stirling cycle, single phase cycle, solid vapour sorption systems and electromagnetic processes but these have either not reached technical maturity or economic competitiveness to have entered into the marketplace. Therefore the thermodynamics and technology overview will be based on the vapour-compression and absorption cycles.

2.1.1 Basic principles

According to the second law of thermodynamics, heat flows naturally in the direction of decreasing temperature. The reverse process cannot occur by itself and requires a special device; either a refrigerator, air conditioner or heat pump. A heat pump is essentially a heat engine operating in reverse.
A heat engine produces work $W$ by extracting heat $Q_2$ from the temperature source $T_2$ and delivering heat $Q_1$ to the temperature sink $T_1$. A heat pump operates in the reverse and delivers heat $Q_2$ at the temperature $T_2$ by extracting heat $Q_1$ from $T_1$ and requiring a work input $W$. From the first law of thermodynamics the relation between heat and work is shown by:

$$Q_2 = Q_1 + W$$

The second law of thermodynamics states that the work output produced by the heat engine cannot be greater than the work input required by a heat pump when operating between the same temperature levels. This results in the following relationship between temperatures and transferred heat:

$$\frac{Q_1}{T_1} = \frac{Q_2}{T_2}$$

The efficiency of this process is defined as the ratio of the heat output to the work input. The performance of a heat pump can be measured in terms of its coefficient of performance (COP) as defined below:

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_{in}}{W_{net,in}}$$

### 2.1.1.1 Vapour Compression Cycle

Many of the impossibilities of the ideal Carnot heat pump cycle can be taken into consideration. By vaporising the refrigerant completely before it is compressed and by replacing the turbine with a throttling device, the compression and expansion stage can operate in a real world environment. This cycle is called the vapour-compression cycle and is shown schematically below.

![Schematic and T-s diagram for the ideal vapour compression refrigeration cycle](image)

*Cengel & Boles, 2006*
This cycle is the most widely used for refrigerators, air-conditioning systems and heat pumps. It consists of four processes:

- **1-2** Isentropic compression in compressor
- **2-3** Constant pressure heat rejection in condenser
- **3-4** Throttling in expansion device
- **4-1** Constant pressure heat absorption in evaporator

The refrigerant enters the compressor at state 1 as a saturated vapour and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases to well above the temperature of the medium being heated. The energy required to drive the compressor is the power consumption, W.

The refrigerant enters the condenser as superheated vapour at state 2, transfers heat to the heat sink, thus increasing its temperature, and leaves as a saturated liquid at state 3. The total capacity of heat transfer in the condenser is called the heating capacity, \( Q_C \) (heat from condenser, also \( Q_{H} \)).

The saturated liquid then passes through an expansion valve and is throttled to the evaporator pressure. Since the ideal vapour-compression cycle is not an internally reversible cycle because it contains an irreversible (throttling) process, the refrigerant enters the evaporator at 4 instead of 4'.

The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture and evaporates completely by absorbing heat from the heat source. The total capacity to absorb heat is called the cooling capacity, \( Q_e \) (heat from evaporator, also \( Q_{L} \)). From here it re-enters the compressor as a saturated vapour, completing the cycle.

On the T-s diagram shown above, the area under the process curve represents the heat transfer. The area under the curve 4-1 is the heat absorbed by the refrigerant in the evaporator \( (Q_L) \) and the area under the curve 2-3 is the heat rejected in the condenser \( (Q_H) \). The three main parameters of the heat pump system are the cooling capacity, the heating capacity and the power consumption. The other main parameters are:

- Condensing temperature: \( T_c \) (°C)
- Evaporation temperature: \( T_e \) (°C)
- Heat sink temperature: \( T_{heat\_sink} \) (°C)
- Heat source temperature: \( T_{heat\_source} \) (°C)
- Refrigerant mass flow rate: \( m_r \) (kg/s)

Another diagram used in the analysis of vapour-compression cycles is the P-h diagram (shown below).

![Figure 3: The P-h diagram of an ideal vapour compression refrigeration cycle (Cengel & Boles, 2006)](image-url)
All four components are steady-flow devices and thus all four processes that make up the cycle can be analysed as steady-flow processes. The kinetic and potential energy changes are small relative to the heat transfer and work terms and can be ignored. The steady-flow equation on a unit mass basis becomes:

\[ q_{in} - q_{out} + w_{in} - w_{out} = h_e - h_c \]

The condenser and evaporator do not involve any work and the compressor can be approximated as adiabatic. This leads to the COP \(_{HP}\) as:

\[ COP_{HP} = \frac{q_H}{w_{net, \ in}} = \frac{h_2 - h_3}{h_2 - h_1} \]

The actual vapour-compression cycle deviates slightly from this due to fluid friction, heat loss and allowances for the operating conditions of the components.

2.1.1.2 Absorption cycle

Another heat pump cycle that is economically attractive when there is a large source of inexpensive thermal energy available is the absorption cycle. This cycle involves the absorption of a refrigerant by a transport medium. A popular refrigeration system is the ammonia-water system, where ammonia \((\text{NH}_3)\) serves as the refrigerant and water \((\text{H}_2\text{O})\) serves as the transport medium. The basic principles will be explained using this \(\text{NH}_3\)-\(\text{H}_2\text{O}\) system as shown below.

![Figure 4: Schematic of an NH₃-H₂O absorption cycle (Cengel & Boles, 2006)](image)
This system is very similar to the vapour compression system except for the compressor being replaced by a complex absorption mechanism consisting of an absorber, pump, generator, regenerator, expansion valve and rectifier. This mechanism merely serves the purpose of raising the temperature of the refrigerant. Since the operation of the rest of the system remains the same, only that appearing in the box will be explained.

Ammonia vapour leaves the evaporator to enter the absorber where it dissolves in water to form \( \text{NH}_3\cdot\text{H}_2\text{O} \). This is an exothermic reaction where the amount that can react is inversely proportionate to the temperature. Heat thus needs to be removed at this step to maximise the amount of ammonia dissolved in water. This liquid solution which is rich in \( \text{NH}_3 \) is pumped to the generator.

At the generator, heat is transferred to the solution from an external source, vaporising the solution. The vapour, rich in \( \text{NH}_3 \), passes through a rectifier to separate and return the water to the generator. The high pressure, pure \( \text{NH}_3 \) vapour moves to the condenser and continues through the cycle.

The hot and weak \( \text{NH}_3\cdot\text{H}_2\text{O} \) solution passes through a regenerator to transfer some heat to the incoming solution and then is throttled to the absorber pressure, where it returns. Since this system pumps a liquid instead of compressing a gas, the work input is negligible compared to the heat transferred.

2.1.2 Vapour-Compression

There are a number of different types of heat pump cycles in use. These can be categorised in different ways such as mechanically or heat driven, compression or absorption or closed or open cycles. This section deals with the vapour compression type and under this category falls three sub-types, namely: Closed cycle, mechanical vapour recompression and thermal vapour recompression.

2.1.2.1 Closed-cycle compression

The closed cycle compression heat pump basic principle has already been mentioned above that heat is generated through compression. In this case, the energy input in the form of work for compression is converted to heat energy. This energy is then added to the energy taken in from the surroundings and delivered to a heat sink. The entire process utilises a constant volume of working fluid and operates in a continuous closed-cycle. The heat energy is transferred by the use of heat exchangers between the source, the process and the sink. This is illustrated in the diagram below:

![Schematic of closed cycle compression](image-url)

*Figure 5: Schematic of closed cycle compression*

*Source: (Laue & Heinloth, 2010)*
Closed cycle compression heat pumps are mainly used to recover waste heat at relatively low temperatures and to upgrade it for process preheating to temperatures between 50 and 100°C. The current maximum output temperature is 120°C (Laue & Heinloth, 2010). These systems typically have a COP value in the range of 2 – 10.

2.1.2.2 Mechanical vapour recompression

This type of heat pump operates on the same thermodynamic cycle as the closed-cycle vapour compression but instead of employing a separate refrigerant, the process vapour itself is compressed and its temperature is raised by a mechanical compressor. The process vapour thus acts as both a working fluid and waste heat stream. The high pressure and temperature vapour can either be re-used directly or passed through a heat exchanger to transfer heat to another process stream. The different types are shown below:

Figure 6: Schematic of mechanical vapour recompression

(a) Open cycle heat pump
(b) Semi-open cycle heat pump
(c) Alternative semi-open cycle heat pump
Source: (Laue & Heinloth, 2010)

a) This is a conventional open cycle consisting of a compressor operating directly on waste vapour and delivering high temperature/pressure vapour in the process.
b) This is a semi-open cycle whereby waste vapour is compressed and then condensed in a heat exchanger to transfer heat to another medium.
c) This is another semi-open cycle where the heat source is used to boil a liquid in a heat exchanger. The vapour is then compressed and delivers heat either directly or through the use of another heat exchanger.
These systems are the most common type of industrial heat pumps and operate with a typical COP value of between 10 and 30 (Everest, 2009), much higher than closed-cycle systems. The investment costs are low compared to closed cycle heat pumps due to the reduced use of heat exchangers and refrigerants. Due to the fact that many industrial heat distribution systems are low-pressure steam grids, the direct compression of steam is simple and has certain advantages, particularly in distillation and evaporation plants. Current mechanical vapour recompression use heat source temperatures from 70 to 80°C and deliver heat between 110 and 150°C, with the maximum up to 200°C (Berntsson & Feng, 1997). Since most systems use process steam, water is the most common working fluid, although other process vapours can be used.

2.1.2.3 Thermal vapour recompression
Thermal vapour recompression systems achieve heat pumping with the aid of an ejector and high pressure vapour. Unlike mechanical vapour recompression systems, this heat pump is driven by thermal, not mechanical energy. The cycle is also known as the steam ejector cycle and is similar to the conventional vapour compression cycle with the compressor being replaced by a liquid feed pump, boiler and ejector pump (Meyer, 2006). This is shown below:

![Ejector cycle diagram](image)

Liquid refrigerant is vaporised in a conventional boiler with the addition of heat, increasing the temperature and pressure. This high pressure refrigerant vapour passes through line 1 and enters the primary nozzle of the ejector. The primary steam accelerates and expands through a convergent-divergent nozzle to increase the kinetic energy and reduce the pressure. This partial vacuum entrains refrigerant vapour from the evaporator (line 2) into this low-pressure, high velocity jet, where mixing occurs. The diffuser portion then reconverts this kinetic energy of the mixture back into a pressure between that of the motive steam and low-pressure suction steam.

From the diffuser exit, the mixed flow is fed directly into a condenser (line 3) where it transfers heat and is condensed. The condensate then returns to the evaporator via an expansion valve in line 6 as well as to the boiler via a feed pump in line 4 to repeat the cycle.
The COP for this heat pump is defined as the relation between the heat released during condensation and the heat input of the motive vapour. Typical COP values for this cycle are in the range of 1.5 to 3 (Berntsson & Feng, 1997). The advantage of this cycle is that it has no moving parts so is relatively maintenance free but it is suited to conditions where there is a large difference between fuel and electricity prices.

### 2.1.3 Absorption

Absorption heat pumps are thermally driven, which means that heat rather than mechanical energy is used to drive the cycle. The heat pump cycles are based on the fact that the boiling point for a mixture is higher than the corresponding boiling point of a pure, volatile working fluid. The working fluid therefore needs to be a mixture of a volatile component and non-volatile one. The most common mixture used in industrial applications is a lithium bromide solution in water (LiBr\(_{aq}\)) (Santos, 2008).

The absorption cycle has two possible configurations which are suitable for different purposes; absorption heat pump (Type I) and heat transformer (Type II). The difference between the cycles is the pressure level in the heat exchangers which influences the temperature level of the heat flows. The standard cycle is known as a heat amplifier and the alternative configuration (Type II) is known as a temperature amplifier. The basic principle of its operation has been explained previously, but now the operating characteristics of these two types are explained below.

#### 2.1.3.1 Absorption heat pump - Type-I

Waste heat at a low temperature is delivered to the evaporator and prime heat at a high temperature is delivered to the generator. An amount of heat equivalent to the sum of the high and low temperature heat inputs can be recovered at an intermediate temperature via the condenser and absorber. This is illustrated in the diagram below.

![Energy Balance Diagram](image)

This cycle is similar to the operation of the thermal vapour recompression system, in which high pressure steam is used to increase low pressure waste vapour to a higher temperature and pressure. However, the absorption type can have temperature lifts in the range of 50 to 150°C as opposed to the thermal vapor recompression lifts of 10 to 20°C. The COP for this heat pump configuration is in the range of 1.6 - 1.7, excluding boiler efficiency, and 1.2 - 1.4, including boiler efficiency (Laue & Heinloth, 2010).
2.1.3.2 Heat transformer - Type-II

This is an alternate configuration to the absorption heat pump type whereby a medium temperature waste heat stream is split into a higher temperature stream and a lower temperature stream. The higher temperature stream is the useful heat output and the lower temperature one is rejected at close to ambient conditions. This configuration is achieved using the same components as the standard absorption heat pump, but by adjusting the operating pressures and working fluid concentrations of it. The thermal transfer is illustrated in the diagram below.

![Diagram of heat transformer](image)

**Figure 9: Simplified energy balance for heat transformer.**
Adapted from: (McMullan, 2003)

These heat transformers can achieve a delivery temperature up to 150°C, typically with a lift of 50°C. The COP varies under these conditions from about 0.45 to 0.48 (Laue & Heinloth, 2010).

2.1.4 Components

In this section the basic components of a heat pumping system will be explained. In vapour compression heat pumps there are essentially four components:

- Compressor
- Condenser
- Expansion valve
- Evaporator

Absorption heat pump systems use these components but instead the compressor is replaced with the following components:

- Absorber
- Generator
- Regenerator
- Rectifier

These are shown below.
2.1.4.1 Compressor

The function of a compressor is to receive gas at a low pressure and deliver at a higher pressure and hence temperature. In a closed-cycle system it removes the gas produced by the evaporator to maintain a constant pressure and then delivers it to a condenser. In an open cycle system it receives the gas from a process stream or exhaust gas and delivers it either back to the process stream or to an evaporator. The compressor can either be electric motor driven or diesel engine driven.

There are several different types of compressors (Spinato, 2011) which can be divided into two operating principles:

- Positive displacement compressors
  - Screw compressors
  - Rotary compressors
  - Scroll compressors
  - Rolling piston compressors
- Dynamic compressors
  - Turbo compressors

Positive displacement compressors work on the principle that a certain volume of gas is enclosed in a space that is continuously being reduced by the compressing device (piston, scroll, screw etc) inside the compressor. This reduction in volume increases the pressure of the vapour while the compressor is operating.

Dynamic compressors work on the principle that a gas is compressed by accelerating it with an impeller. The pressure is further increased in the diffuser, where the kinetic energy is transformed into a pressure. These types are generally used in very large capacities, with high flow rates.

Compressors can be installed in single or multi-stage configuration and can be connected to each other in either series or parallel. Series will increase the pressure differential for a certain flow rate and parallel will increase the flow rate for a certain pressure differential.

2.1.4.2 Condenser

The function of the condenser is to transfer heat from the refrigerant to the heat sink. This is achieved by transferring the hot discharge gas from the compressor to a slightly sub-cooled liquid flow. This operation can be divided into three parts which can either all be carried out inside the condenser or in separate heat exchangers. These are shown below:

- De-superheating
- Condensation
- Sub-cooling

The first part of the condenser de-superheats the gas to the saturation temperature. This cooling typically represents 15-25% of the total heat rejection. The phase of the gas remains the same during heat transfer and the temperature of the refrigerant gas decreases.
When the refrigerant reaches its saturation temperature the latent heat is rejected at a constant temperature. This condensation process represents the majority (70-80%) of the total heat rejection.

The fully condensed refrigerant is then sub-cooled a few degrees to ensure that pure liquid enters the expansion valve. The phase of the gas also remains the same in this stage, with the temperature decreasing. This represents about 2-5% of the total heat rejection.

The energy is transferred to the heat sink whose temperature increases to approach that of the condensing temperature. The temperatures of the two working fluids will converge but never be equal due to a temperature difference between the condenser and heat sink being required to facilitate heat transfer. This difference is usually in the range of 2-5°C.

2.1.4.3 Expansion valve
The expansion valve is located in the liquid line between the exit of the condenser and the inlet of the evaporator. It serves two purposes:

- To maintain the pressure difference, created by the work of the compressor, between the high pressure condenser side and the low pressure evaporator side.
- To regulate the amount of refrigerant that enters the evaporator.

Expansion valves do not directly control the evaporation temperature but rather regulate the superheating by adjusting the mass flow rate of refrigerant into the evaporator. The evaporation temperature depends on the capacity of the compressor and the characteristics and efficiency of the evaporator.

There are several different types of expansion valves depending on the demand for control and the type of evaporator:

- Thermal expansion valves
- Manual valves
- Capillary valves
- Automatic valves
- Electronic expansion valves

2.1.4.4 Evaporator
In the evaporator the refrigerant evaporates by absorbing energy from the heat source. The heat source can be a gas or liquid, depending on the system. This process can be divided into two parts, as shown below:

- Evaporation
- Superheating

The sub-cooled liquid refrigerant at high pressure from the condenser is expanded through the expansion valve, the pressure and therefore the saturation temperature decreases. This mixture of
liquid and gas enters the evaporator and starts to evaporate as energy is absorbed from the heat source. The temperature level of the refrigerant inside the evaporator will remain relatively constant as it corresponds to a certain pressure level. In reality there will be friction which causes a pressure drop and a corresponding decrease in the saturation temperature.

After evaporation, when the entire refrigerant mixture has become saturated vapour, the temperature of the vapour will increase to become slightly superheated. This is to ensure that dry gas enters the compressor.

The total heat absorbed by the refrigerant consists of the latent heat of evaporation plus the sensible heat of superheating. A good evaporator should provide a stable evaporation process with a small temperature difference between the refrigerant and the heat source. This low temperature difference means a higher evaporation temperature and hence pressure is possible. Thus the pressure difference between the condenser and evaporator is reduced, decreasing the energy use in the compressor and increasing the COP.

### 2.1.4.5 Absorber

The absorber is the vessel whereby the refrigerant can dissolve into the absorbent (transport medium). This process involves simultaneous mass and heat transfer. Mass transfer involves the mixing of the fluids and heat transfer involves the removal of the heat generated by the exothermic reaction. The objective of the absorber is to facilitate a high rate of reaction by maximising these two processes.

The contact area between the gas and the liquid solution must be as large as possible and a relative motion between the phases must be created. This can be achieved using the following configurations:

- Plate column
- Packed column
- Spray column
- Bubble column
- Surface absorber or film absorber

The rate of reaction is inversely proportional to the temperature so a cooling circuit needs to be incorporated. This also ensures that the vapour pressure of the solution in the absorber is lower than that in the evaporator for correct operation.

### 2.1.4.6 Generator

A generator is used to heat up the strong solution of the refrigerant and absorbent using a heat source such as waste heat. By adding heat and keeping the pressure constant, the refrigerant evaporates from the transport medium. The generator is required to:

- Break up the association between the refrigerant and absorbent
- Change the temperature of the transport medium to its saturation temperature
• Vaporise liquefied refrigerant
• Raise the temperature of the incoming strong solution to that of the generator

A good generator should be able to control the quality and quantity of heat released to minimise the absorbent boiling off.

2.1.4.7 Regenerator
The regenerator simply serves as a heat exchanger between the generator and absorber to increase the temperature of the incoming strong solution and decrease the temperature of the outgoing weak solution.

2.1.4.8 Rectifier
The rectifier serves as a distillation column to separate and concentrate the refrigerant vapour from the absorbent. Its purpose is to produce a vapour of high purity at its exit. The rectifying column has many capped or perforated plates that bring the liquid and vapour solutions into contact. The rising vapour contains some absorbent which condenses and the falling liquid contains some refrigerant which evaporates. This increases the cycle efficiency as it filters the refrigerant and reduces the quantity of absorbent in the evaporator. Absorbent in the circuit will lower the heating capacity and reduce the COP.

2.1.5 Working fluids
The working fluid, or refrigerant, must satisfy some requirements. These are broadly:

• The chemical, physical and thermodynamic properties must be appropriate for the system and the working conditions at a reasonable cost.
• The refrigerant should not cause any risk of injuries, fire or property damage in the case of leakage.

The ideal properties for a working fluid are:

• Good heat transfer properties
• High latent heat
• Appropriate pressures for the operating temperature
• Chemical stability
• Low toxicity
• Low fire risk

Other requirements that the working fluid should fulfil are:

• Environmentally friendly
• Satisfactory oil solubility/miscibility
• Easy leak detection
• Low cost
2.1.5.1 Classification

Refrigerants are classified according to standards specified by the international organisation ASHRAE. Refrigerants are represented by a letter R followed by a two or three digit number and, in some cases, one or two letters. The designation Rxyz is determined by the chemical composition of the molecule. The main groups will be outlined below but the full specifications can be found from the ASHRAE standards manual (ASHRAE, 2011).

- \( x = 0 \) Methane series  E.g. R12 and R22
- \( x = 1 \) Ethane series  E.g. R144, R124 and R134a
- \( x = 2 \) Propane series  E.g. R290 (propane)
- \( x = 4 \) Zeotropic mixtures  E.g. R407A and R407C
- \( x = 5 \) Azeotropic mixtures  E.g. R502 and R507
- \( x = 6 \) Organic compounds  E.g. R600 (butane)
- \( x = 7 \) Inorganic compounds  E.g. R718 (water), R717 (ammonia) and R744 (CO\(_2\))
- \( x = 11 \) Unsaturated ethane compounds  R1150 (ethylene)
- \( x = 12 \) Unsaturated propane compounds  R1270 (propylene)

The last letter, if any, can mean different things:

- Lower case letters describe the structure of the molecule. For example, R600 is butane and R600a is isobutene. These have the same chemical formula but different spatial arrangements and thus different properties
- Capital letters describe a specific mixing proportion of different compounds. For example, R407A are mixtures of refrigerants R32, R125 and R134a with a set mixing proportion.

As well as refrigerants being toxic or explosive, there are also environmental aspects which are increasingly being taken into consideration. Refrigerants can be ranked according to their impact on the stratospheric ozone layer (the Ozone Depletion Potential, ODP) or as greenhouse gases (the Global Warming Potential, GWP).

- Ozone Depletion Potential (ODP)
  The ODP is used to reflect the refrigerants impact on the ozone layer. It is a ratio of the impact on the ozone of a chemical compound compared with the impact of a similar mass of refrigerant R11. Thus the ODP of R11 is 1 by definition with ODP values of less than 1 having less impact and vice versa. The range lies from 0 for chlorine free hydrofluorocarbon's to 10 for Halon's.

- Global Warming Potential (GWP)
  The GWP is used to reflect the refrigerants impact on global warming. It is a ratio of the warming caused by a substance to the warming caused by a similar mass of carbon dioxide. Thus the GWP of carbon dioxide is 1 by definition with GWP values of less than 1 having less impact and vice versa. The range lies from 0 for water to 12 100 for hydrofluorocarbon's.
2.1.5.2 Refrigerant types

Refrigerants are divided into groups according to their chemical composition. Some refrigerants are being replaced with more environmentally friendly refrigerant alternatives due to the discovery of harmful compounds. These groups are outlined below:

- **CFC (Chlorofluorocarbons)**
  These refrigerants contain chlorine and have been banned since the start of the 90's because of their negative environmental impacts.
  Some examples are: GWP ODP
  - R11 Trichlorofluoromethane 6 730 1.0
  - R12 Dichlorofluoromethane 11 000 1.0

- **HCFC (Hydrochlorofluorocarbons)**
  These refrigerants contain less chlorine and hence a lower ODP and serve as a temporary transition refrigerant.
  Some examples are: GWP ODP
  - R22 Chlorodifluoromethane 5 160 0.055
  - R123 Dichlorofluoromethane 273 -

- **HFC (Hydro fluorocarbons)**
  These refrigerants contain no chlorine and are not harmful to the ozone layer. They do, however, have a higher impact on global warming.
  Some examples are: GWP ODP
  - R23 Trifluoromethane 12 000 0
  - R134a Tetrafluoromethane 3 830 0

- **FC (Fluorocarbons)**
  These refrigerants contain no chlorine and are not harmful to the ozone layer. They do, however, have a high GWP.
  Some examples are: GWP ODP
  - R218 Octofluoropropane 6 310 0

- **HC (Hydrocarbons)**
  Hydrocarbons are harmless to the ozone layer and have hardly any green house effect but are a limited solution due to their high flammability. Many countries have banned the use of flammable gas in the presence of public. Some examples are isobutane and propane.

- **NH$_3$ (Ammonia)**
  Ammonia, R717, is a high class refrigerant alternative since its ODP and GWP are 0 and it is non-flammable. Its smell, however, can be hazardous.

- **CO$_2$ (Carbon dioxide)**
  Carbon dioxide, R744, is non-flammable, does not cause ozone depletion, has a very low toxicity index, is available in large quantities and has a low cost. However, it is has a low efficiency and a high operating pressure which will require improvements in the
refrigeration cycle and related technology before it is adopted as a more widespread refrigerant.

2.1.5.3 Vapour compression heat pump refrigerants

- CFCs, HCFCs and HFCs
  In the past the following refrigerants were commonly used:
  - CFC R12 for low and medium temperatures (max 80°C)
  - CFC R114 for high temperatures (max 120°C)
  - HCFC R22 for low temperatures (max 55°C)

These refrigerants have since been exposed as being harmful to the environment and are not used anymore in heat pump systems. Nowadays the most common refrigerants in vapour compression heat pump systems are HFCs. These are long term refrigerants since they are chlorine free and do not contribute to ozone depletion. Some examples are R134a, R152a, R32, R125 and R143a.

- Blends
  A blend consists of a mixture of two or more pure refrigerants and can be azeotropic, azeotropic or near-azeotropic. Azeotropic mixtures evaporate and condense at a constant temperature while others over a certain temperature range.

Heat pump systems have used some blends in the past to replace CFC refrigerants with HCFC and HFC refrigerants. Nowadays blends are used to replace HCFCs with chlorine free HFCs and hydrocarbons. An example of these is R407C (23% HFC R32, 25% HFC R125, 52% HFC R134a) and R410a (50% HFC R32, 50% HFC R125).

- Natural refrigerants
  Natural refrigerants are more environmentally friendly than other types due to their ODP and GWP being zero or near zero. Hence they are long-term alternatives to CFCs. They are, however, more flammable or toxic than other refrigerants and their use requires specific safety systems. Examples are: carbon dioxide, hydrocarbons, ammonia, air and water.

2.1.5.4 Absorption heat pump refrigerants

As stated earlier, absorption refrigerants are used in pairs: one being the refrigerant and one the absorbent. There is a vast amount of refrigeration pairs that can be used such as:

- Aqua ammonia solution
- Lithium bromide brine solution
- Sodium chloride brine solution
- Calcium chloride brine solution

These refrigerant pairs need to have the same properties as stated earlier as well as:
- Be miscible within the operating temperature range of the cycle
- Have a large boiling point difference between the pure refrigerant and the mixture at the same temperature
- Have a high concentration of refrigerant within the absorbent

The following two refrigerants are commonly used in heat pump applications:

- **Aqua ammonia solution**
  This working pair is environmentally friendly, has a low cost and has good high temperature properties. It has the disadvantage of requiring a rectifier, operating at high pressures, is toxic and is corrosive to copper and its steel alloys.

- **Lithium bromide brine solution**
  Lithium bromide is not volatile and hence the need for a rectifier is not required. The refrigerant water also has a very good, high heat of vaporisation. Disadvantages are that lithium bromide is prone to crystallisation at high concentrations, is corrosive at high temperatures and is expensive.

There are many different types of refrigerants available for heat pump use and so the selection must take into account the heat pump parameters and the exposed refrigerant requirements.

### 2.1.6 Heat sources

Heat pumps rely on absorbing heat from the ambient air, water, ground or waste heat. This thesis deals with waste heat as the heating medium which will be investigated thoroughly in the next chapter. Nevertheless the following properties are required of a heat source (Berntsson, 2002):

- High temperatures and stable
- Abundant
- Have favourable thermo-physical properties
- Require low investment to exploit
- Have a high specific heat per unit volume

### 2.1.7 Operational modes

Heat pumps and refrigerators operate with similar components and thermodynamic cycles and merely differ with their objectives. Heat pumps utilise the warm part of the cycle and refrigerators utilise the cold part. Because of this, heat pumps can operate in two modes, either where only the heat is required, or where both heating and cooling is required.
2.1.7.1 Heating only
When used for heating purposes, heat pumps can be run in three different operating modes:

- Mono-mode
  These heat pump systems meet the annual heating demand alone. The system is designed with a heating capacity above that of the maximum heating load such that supply is greater than demand.

- Dual-mode
  These heat pump systems utilise an auxiliary heating system to supplement the heating requirement. They can either be designed to be operated simultaneously with another heat source (solid, liquid, gas fuels) or to meet the base-load heat requirement with the alternative heat source only used for the peak heating requirements. These heat pumps are usually sized for 20-60% of the maximum heating load but normally meet 50-95% of the maximum heating demand (Laue & Heinloth, 2010).

- Mono-energetic
  These heat pumps use an auxiliary heating system that is based on the same supply of energy used for the heat pump, e.g. electric resistance heater. This represents a compromise between investment outlay and energy efficiency.

2.1.7.2 Heating and cooling
When both heating and cooling is required simultaneously, there is an opportunity to use a single machine to produce both heat and cold. This eliminates any rejected energy at both the evaporation and condensation ends. This is known as thermal coupling. The COP for the individual heating and cooling capacity will be reduced, but the net COP will increase.

2.1.8 Performance characteristics
When identifying a feasible heat pump for installation, a detailed comparison of the performance characteristics, operating ranges and economics needs to be taken into account. The following two sections gives some examples of the performance in terms of heat pump types and refrigerant types.

2.1.8.1 Heat pump types
Each heat pump type is applicable to a specific operating range with different sink temperatures, temperature lifts, COPs and costs. These approximate operating limits of various types of industrial heat pumps are shown below with a typical cost for a 1MW installation.
Table 1: Performance and economic comparison of industrial heat pump types (Berntsson & Feng, 1997)

<table>
<thead>
<tr>
<th>IHP type</th>
<th>Maximum sink temperature (°C)</th>
<th>Maximum temperature lift (°C)</th>
<th>Typical COP</th>
<th>Typical investment cost €/kW heat output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Closed cycle compression</td>
<td>120</td>
<td>80</td>
<td>3-5</td>
<td>320-550</td>
</tr>
<tr>
<td>Mechanical vapour recompression</td>
<td>190</td>
<td>90</td>
<td>5-20</td>
<td>380-450</td>
</tr>
<tr>
<td>Thermal vapour recompression</td>
<td>150</td>
<td>40</td>
<td>1.5-2.5</td>
<td>210-270</td>
</tr>
<tr>
<td>Absorption</td>
<td>100</td>
<td>50</td>
<td>1.3-1.7</td>
<td>300-350</td>
</tr>
<tr>
<td>Heat transformer</td>
<td>150</td>
<td>60</td>
<td>0.45-0.5</td>
<td>720-830</td>
</tr>
</tbody>
</table>

2.1.8.2 Refrigerants

Many different types of refrigerants can achieve the required operating conditions. The following table gives a comparison of the most commonly used refrigerants for heat pump applications with the following parameters:

- Evaporation temperature - 34°C
- Condensation temperature - 81°C
- Heating capacity - 5000 kW

Table 2: Comparison of refrigerants for one stage simple vapour compression cycle Adapted from: (Sánchez, 2008)

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>COP</th>
<th>Specific cooling capacity kJ/kg</th>
<th>Specific heating capacity kJ/kg</th>
<th>Power consumption kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFC R12</td>
<td>5.44</td>
<td>99</td>
<td>80</td>
<td>918.6</td>
</tr>
<tr>
<td>HCFC R22</td>
<td>5.14</td>
<td>128</td>
<td>103</td>
<td>972.7</td>
</tr>
<tr>
<td>Blend R500</td>
<td>5.29</td>
<td>115</td>
<td>93</td>
<td>945.5</td>
</tr>
<tr>
<td>HFC R134a</td>
<td>5.07</td>
<td>113</td>
<td>91</td>
<td>985.2</td>
</tr>
<tr>
<td>HFC R152a</td>
<td>5.69</td>
<td>215</td>
<td>177</td>
<td>879.1</td>
</tr>
<tr>
<td>Blend R407C</td>
<td>4.02</td>
<td>104</td>
<td>78</td>
<td>1242.1</td>
</tr>
<tr>
<td>HC R290</td>
<td>4.77</td>
<td>208</td>
<td>165</td>
<td>1046.4</td>
</tr>
<tr>
<td>R 717</td>
<td>6.20</td>
<td>1064</td>
<td>892</td>
<td>807.0</td>
</tr>
</tbody>
</table>
2.2 Waste Heat Recovery Methods

Industrial waste heat refers to energy in industrial processes that is not put to practical use. This heat can arise from both equipment inefficiencies and thermodynamic limitations on equipment and processes. Sources of this heat include hot combustion gases discharged to the atmosphere, hot process mediums being discharged and heat transfer from hot equipment surfaces.

While some waste heat losses are unavoidable, industries can reduce these losses by improving equipment efficiencies or by using waste heat recovery technologies. Waste heat recovery entails capturing and re-using waste heat in industrial processes for heat or work generation. Heat recovery can reduce the operating costs of the industry by increasing their energy productivity. This section will outline the major waste heat recovery technologies and explain the classification of waste heat and how it affects recovery feasibility.

2.2.2 Waste Heat Recovery Options and Technologies

There are various methods for waste heat recovery which include: transferring waste heat between gases and liquids, transferring heat to a medium entering a furnace, generating mechanical or electrical power and the main method, which the focus of this thesis is on, is using waste heat with a heat pump for low temperature recovery. While heat pumps have already been discussed in the previous chapter, the other technologies will be briefly addressed below.

2.2.2.1 Heat exchangers

The objective of a heat exchanger is to transfer heat from one medium to another. The medium can be a gas to liquid and can be kept separate or be in direct contact. There are many different types which are highlighted below (Tucker, 1979):

- Recuperator
  
  This can be constructed of either ceramic or metallic materials and is used to recover exhaust gas waste heat in medium to high temperature applications. They can be based on radiation, convection or a combination:

  - Radiation - The basic radiation recuperator consists of two concentric lengths of ductwork. Hot waste gases pass through the inner duct which radiates heat to the wall and to the cold incoming air in the outer shell.

  - Convection - This tube-type recuperator passes hot gases through small diameter tubes contained in a larger shell. The incoming air enters the shell and is baffled around the tubes, picking up heat from the waste gas.

  - Combination - This system consists of a radiation section followed by a convection section in order to maximise heat transfer effectiveness.
• Regenerator
These work on the principle that a hot gas is passed over a solid thermal storage medium which can then provide this heat to a cold gas stream. There are two main types:
  o Furnace regenerator - This consists of two brick checker-work chambers through which hot and cold gases pass over alternatively. Hot gases pass through one chamber which absorbs heat and increases its temperature. The flow is then adjusted so that the incoming cold gas is passed over the bricks, thus gaining heat. There are two chambers so that while one is absorbing heat from the hot gas, the other is transferring heat to the cold gas.
  o Heat wheel - This consists of a rotating porous disc placed across two parallel ducts, one containing hot waste gas and the other containing cold gas. Heat is transferred to the high heat capacity material in the hot duct which then rotates and transfers this heat to the cold gas stream.

• Passive air pre-heaters
These are gas-to-gas heat recovery devices used in low to medium temperature applications where cross-contamination between gas streams must be prevented. They can be of two types, plate type and heat pipe:
  o Plate type - This consists of multiple parallel plates that create separate channels for hot and cold gas streams. Hot and cold flows alternate between the plates to create a significant area for heat transfer.
  o Heat pipe - This system consists of several closed pipes with the cold medium in contact with the one side and the hot medium in contact with the other. As hot gas passes over the one end of the pipe, the working fluid inside the pipe evaporates. This hot vapour has a higher pressure and moves to the other end of the pipe, where the vapour condenses and transfers heat to the cold gas.

• Finned tube heat exchangers
This is used to recover heat from low to medium temperature exhaust gas. The finned tube consists of a round tube with attached fins that maximise surface area and heat transfer rates. Liquid flows through the tubes and receives heat from hot gases flowing across the tubes.

• Waste heat boilers
These are ordinary water tube boilers in which the hot exhaust gases pass over a number of parallel tubes containing water. The water is vaporised in the tubes and collected in a steam drum from which it is drawn off for use as heating or process steam.
2.2.2.2 Power generation

While it is more common for waste heat to be re-used within the process, there is a possibility to use the waste heat for electricity generation. This can be done by either generating power through mechanical work or by direct electrical conversion devices. This is summarised below:

- Steam Rankine cycle
  This is frequently used for power generation and involves using heat to generate steam to drive a turbine. This is more suited to higher temperature applications as the low temperature waste heat may not provide sufficient energy to superheat the steam, which is a requirement to prevent steam condensation and erosion of the turbine blades.

- Organic Rankine cycle
  This cycle operation is similar to the steam Rankine cycle but uses an organic working fluid instead of steam. The working fluid has a lower boiling point and higher vapour pressure than water which allows the cycle to operate with a lower waste heat temperature.

- Kalina cycle
  This is a variation of the Rankine cycle and uses a mixture of ammonia and water as the working fluid. Due to this fact, the temperature profile during boiling and condensation are different. Unlike single fluid cycles, as heat is transferred to the working medium the temperature of the mixture of water and ammonia increases during evaporation. This allows better thermal matching with the waste heat source and with the cooling medium in the condenser. Hence greater efficiencies can be achieved.

- Direct electrical conversion devices
  These devices generate electricity directly from heat and do not involve the conversion to mechanical energy first. These include thermoelectric, thermionic, piezoelectric and thermophotovoltaic generators which use a temperature difference to generate a voltage. They are not widely used in industrial waste heat recovery applications.

2.2.2.3 Low temperature energy recovery

Most industrial waste heat falls into the low temperature range (38-200°C) (DoE, 2008). Economics often limits the feasibility of low temperature waste heat recovery but there are various areas where it can be recovered. The following challenges are inherent in low temperature waste heat recovery:

- Corrosion of the heat exchanger surface
  Water vapour contained in the exhaust gas will cool and condense and deposit corrosive solids and liquids on the heat exchanger surface. Thus the heat exchanger must be designed to withstand these corrosive deposits.

- Large heat exchanger surfaces required
  Since low temperature waste heat involves a smaller temperature gradient between two streams, larger surface areas are required for effective heat transfer.
• Low temperature uses
  There needs to be a use for this low temperature heat within the plant for heat recovery in this range to make sense.

There are two possibilities to recover this low temperature waste heat:

• Low temperature heat exchangers
  These involve the cooling of the exhaust gas such that the water vapour condenses and releases its latent heat to a medium being heated. There are various devices available such as deep economisers, indirect contact condensation recovery, direct contact condensation recovery and transport membrane condensors.

• Heat pumps
  These use an external energy input to upgrade the quality of the waste heat to achieve a desired end-use temperature. They have been investigated previously and hence will not be explained further.

2.2.3 Factors Affecting Waste Heat Recovery Feasibility

Evaluating the feasibility of waste heat recovery requires characterising the waste heat source and sink temperatures. Important parameters that must be determined are:

• Heat quantity
• Heat temperature / quality
• Waste stream composition
• Operating schedules, availability and other logistics

These parameters allow for the analysis of the quantity and quality of the waste heat stream and provide insight into the design limitations.

2.2.3.1 Heat quantity

The quantity of heat is a measure of how much energy is contained in the waste heat stream, while quality is a measure of the usefulness of the waste heat. The quantity of waste heat contained in a waste stream is a function of the mass flow rate, temperature and specific heat.

\[ Q = m \times C_p \times \Delta T \]

Where:

• \( Q \) is the heat content
• \( m \) is the mass flow rate
• \( C_p \) is the specific heat
• \( \Delta T \) is the temperature difference
Although the quantity of waste heat available is an important parameter, it should only be used together with the waste heat quality as an effective way of measuring waste heat recovery potential.

2.2.3.2 Temperature / quality

The temperature of the waste heat stream is a key factor in determining the recovery feasibility. The magnitude of the temperature difference between the heat source and sink is an important determinant of the waste heat’s utility or “quality”. The source and sink temperature difference influences:

- The rate at which heat is transferred across the heat exchanger
- The maximum theoretical efficiency of converting the thermal energy to another form
- The selection of materials in the heat exchanger design

Waste heat recovery opportunities can be categorised into the following temperature ranges with their respective advantages and disadvantages:

- **High:** $T > 650^\circ C$
  - **Advantages**
    - High quality energy available for a diverse range of end-uses with varying temperature requirements.
    - High efficiency power generation
    - High heat transfer rate per unit area
  - **Disadvantages**
    - High temperatures create increased thermal stresses on heat exchanger materials
    - Increased chemical activity / corrosion

- **Medium:** $232^\circ C < T < 650^\circ C$
  - **Advantages**
    - More compatible with heat exchanger materials
    - Practical for power generation

- **Low:** $T < 232^\circ C$
  - **Advantages**
    - Large quantities of low temperature heat contained in numerous product streams
  - **Disadvantages**
    - Few end uses for low temperature heat
    - Low efficiency power generation
    - For combustion exhausts there is acidic condensation and exchanger corrosion

This report will primarily deal with the low temperature range to be used for upgrading via a heat pump. The following table gives examples of industrial processes within these ranges and possible recovery methods.
Table 3: Temperature classification of waste heat sources and recovery opportunities (US DoE, 2008)

<table>
<thead>
<tr>
<th>Temp range</th>
<th>Example sources</th>
<th>Temp (°C)</th>
<th>Recovery methods</th>
</tr>
</thead>
<tbody>
<tr>
<td>High (&gt;650°C)</td>
<td>Nickel refining furnace</td>
<td>1370 - 1650</td>
<td>Combustion air preheat</td>
</tr>
<tr>
<td></td>
<td>Steel electric arc furnace</td>
<td>1370 - 1650</td>
<td>Steam generation for process heating</td>
</tr>
<tr>
<td></td>
<td>Basic oxygen furnace</td>
<td>1200</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Copper refining furnace</td>
<td>760 - 820</td>
<td>Furnace load preheating</td>
</tr>
<tr>
<td></td>
<td>Hydrogen plants</td>
<td>650 - 980</td>
<td>Transfer to medium/low temperature processes</td>
</tr>
<tr>
<td></td>
<td>Fume incinerators</td>
<td>650 - 1430</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Glass melting furnace</td>
<td>1300 - 1540</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Coke oven</td>
<td>650 - 1000</td>
<td></td>
</tr>
<tr>
<td>Medium (230 - 650°C)</td>
<td>Steam boiler exhaust</td>
<td>230 - 480</td>
<td>Combustion air preheat</td>
</tr>
<tr>
<td></td>
<td>Gas turbine exhaust</td>
<td>370 - 540</td>
<td>Steam/power generation</td>
</tr>
<tr>
<td></td>
<td>Heat treating furnace</td>
<td>430 - 650</td>
<td>Feed water preheating</td>
</tr>
<tr>
<td></td>
<td>Drying and baking ovens</td>
<td>230 - 590</td>
<td>Transfer to low temperature processes</td>
</tr>
<tr>
<td></td>
<td>Cement kiln</td>
<td>450 - 620</td>
<td></td>
</tr>
<tr>
<td>Low (&lt;230°C)</td>
<td>Exhaust gases</td>
<td>70 - 230</td>
<td>Space heating</td>
</tr>
<tr>
<td></td>
<td>Process steam condensate</td>
<td>50 - 90</td>
<td>Domestic hot water heating</td>
</tr>
<tr>
<td></td>
<td>Cooling water</td>
<td>30 - 230</td>
<td>Upgrade via heat pump</td>
</tr>
<tr>
<td></td>
<td>Drying and baking ovens</td>
<td>90 - 230</td>
<td>Organic Rankine cycle</td>
</tr>
<tr>
<td></td>
<td>Hot processed liquids</td>
<td>30 - 230</td>
<td></td>
</tr>
</tbody>
</table>

The following three factors are affected by the waste heat temperature:

- Heat exchanger area requirements
  The temperature difference between the heat source and sink influences the rate of heat transfer between them, which significantly influences recovery feasibility. The expression for heat transfer can be generalised by the following equation:

\[
Q = UA \Delta T
\]

Where:
- \(Q\) is the heat transfer rate
- \(U\) is the heat transfer coefficient
- \(A\) is the surface area for heat exchange
- \(\Delta T\) is the temperature difference between the two streams

The following graph shows an example of the correlation between the heat exchanger area and the temperature difference for varying heat transfer coefficients.
The figure demonstrates the relative heat exchanger area required to transfer heat from a hot gas at varying temperatures to liquid water. As can be seen, there is an inflection point at lower temperatures where the area required for heat transfer increases drastically. The shape of the curve and area required will vary depending on the heat transfer fluids, heat transfer coefficient and desired heat transfer rate.

- Maximum efficiency for power generation
  The theoretical efficiency limit for power generation depends on the heat source temperature. Maximum efficiency is based on the Carnot efficiency:

  \[ \eta = 1 - \frac{T_L}{T_H} \]

  Where \( T_H \) is the waste heat temperature and \( T_L \) is the heat sink temperature.

  Since the temperature of the waste heat has a dramatic impact on the feasibility of heat recovery, an assessment of heat recovery opportunities should consider both waste heat quantity and quality. To account for the variations in the waste heat temperature the work potential can be used in the analysis. This represents the maximum possible work that can be extracted from a heat engine operating between the waste heat temperature and ambient temperature. This is represented by multiplying the Carnot efficiency by the waste heat quantity:

  \[ WP = \eta \times Q = (1 - \frac{T_L}{T_H}) \times Q \]

- Temperature and material selection
  The temperature of the waste heat source has implications on the material selection in the heat exchangers and recovery systems. Corrosion and oxidation reactions accelerate at increased temperatures and if the waste heat source contains corrosive substances, the heat recovery surface can become damaged. Therefore advanced alloys or composite materials must be used at higher temperatures.
2.2.3.3 Waste stream composition

The chemical composition of the waste stream does not directly affect the quality or quantity of the available heat but rather affects the recovery process and material selection. The composition and phase will determine factors such as thermal conductivity and heat capacity which influence heat exchanger effectiveness. This impacts on the heat exchanger design, material selection and cost.

Heat transfer rates in heat exchangers are dependent on the phase and composition of waste heat streams, as well as by the deposition of fouling substances on the heat exchanger. More dense fluids have a higher heat transfer coefficient which enables higher heat transfer rates per unit area for a given temperature difference. The following table illustrates this:

Table 4: General range of heat transfer coefficients for sensible heat transfer in tubular exchangers (US DoE, 2008)

<table>
<thead>
<tr>
<th>Fluid conditions</th>
<th>Heat transfer coefficient W/(m².K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water, liquid</td>
<td>$5 \times 10^3 - 1 \times 10^4$</td>
</tr>
<tr>
<td>Light organics, liquid</td>
<td>$1.5 \times 10^3 - 2 \times 10^4$</td>
</tr>
<tr>
<td>Gas (P=1000 kPa)</td>
<td>$2.5 \times 10^2 - 4 \times 10^2$</td>
</tr>
<tr>
<td>Gas (P=100-200 kPa)</td>
<td>$8 \times 10 - 1.2 \times 10^2$</td>
</tr>
</tbody>
</table>

Another consideration is the interaction between chemicals in the exhaust stream and the heat exchanger materials. Fouling is common and causes a reduction in heat exchanger effectiveness or system failure. Deposition of substances on the heat exchanger surface can reduce heat transfer rates and inhibit the fluid flow.

2.2.3.4 Other factors

Several other factors can deem whether or not heat recovery is feasible in a given application.

- Minimum allowable temperature
  Some exhaust gases contain water vapour that, when cooled below the dew point temperature, will condense and deposit corrosive substances on the heat exchanger surface. Therefore heat exchangers are generally designed to maintain the exhaust temperature above the condensation point.

- Economies of scale
  Small scale operations are less likely to install heat recovery equipment due to insufficient capital being available and payback periods may be too long.

- Operating schedules
  If a waste heat source is only available for a limited time, the heat exchanger may be exposed to high and low temperatures and be subjected to fatigue due to thermal cycling. Also the schedule for the heat source should match that of the heat load.
2.3 Energy Use in Industry

Energy in industry is mainly consumed in the form of electricity, fuels such as fossil fuels and biogas, and externally supplied heat or cold. This energy can be expressed in terms of final energy and primary energy; primary energy being the total amount of energy necessary for generating the final energy supply, taking into account losses through transport and through the change in its form from processing, and final energy being the amount of energy contained in the energy source entering the industry, independent of its form. The final energy can be used for either thermal or non-thermal uses. This report is concerned with the thermal part of the energy used (FET) where the predominant use is in process heating and cooling. Other uses are for space heating, space cooling and sanitary hot water demand. The non-thermal use encompasses all electrical energy consumption for lighting, machinery and other equipment except for air-conditioning, refrigeration and heating. The energy within the supply system can be expressed in terms of the useful supply heat (USH) and useful process heat (UPH); useful supply heat being the heat generated in the heat supply system that is distributed to the heat consuming processes in the form of steam, hot air, hot water etc, and useful process heat being the heat delivered to a process after all conversion and distribution losses.

The following diagram illustrates these energy flows:

![Energy flows in an industrial heat supply system](image_url)

Processes in industry generally require both heating of a fluid stream (hot water, hot air) and heating of some reservoir (liquid bath, oven). The reservoir heating can be divided into pre-heating before the start of the operation, and into maintenance of temperature due to thermal losses during operation. The following diagram illustrates this:
The total heat demand of a process can be split into the three components mentioned above:

a) Circulation heat: The heat related with the mass flow of the medium entering the process. This is the heat needed to heat up the entering medium to the process temperature and can be defined for continuous and batch processes.

b) Initial heating at start-up: The heat needed to bring the process mass that remains within the process equipment to the process temperature after a process interruption.

c) Maintenance heat: The heat needed to maintain the process temperature constant. It is made up of the thermal losses through the process boundary to the ambient air and to the latent supply heat for evaporation or chemical processes.

### 2.4 International Surveys, Case Studies and Best Practices

There have been numerous studies relating to the technical and economic viability of using heat pumps for waste heat recovery in industry. One part of these studies has included surveys across different industries to establish where heat is used in industry and at what temperature levels. The second part relates to actual installations of waste heat recovery systems and heat pump integration in specific industries.

#### 2.4.1 Surveys

A study conducted by the European Centre and Laboratories for Energy Efficiency Research (Dupont & Sapora, 2009) investigated the heat recovery potential for French industries with specific focus on the opportunities for heat pump systems. They targeted both industrial branches (industrial sectors) and energy usages (specific processes) with temperature levels in the range of 0-200°C. They discovered that almost 70% of the heat between 0°C and 200°C is needed by only 8 industrial branches with food, dairy and sugar industries requiring over 45% of the heat demand under 100°C. They also discovered that for all industrial branches almost 90% of the heat between 0°C and 200°C
is used by only 6 energy usages with these usages predominantly in the pulp and paper and food industries. They then categorised each industrial branch and each energy usage into 10°C temperature increments for the majority of the heat demand which fell between the temperature range of 60°C and 140°C. They estimated that of the 33 TWh/yr that the most promising industrial sectors and usages used between the temperature range of 60-140°C, roughly 20 TWh/yr could be saved through the use of heat pumps. The following graph summarises these findings and illustrates which industries and processes require further study into the feasibility of using heat pumps for waste heat recovery.

![Graph showing distribution of heat demand per industrial branch and usage](image-url)

**Figure 13: Distribution of the heat demand per industrial branch (top) and usage (bottom) (Dupont & Sapora, 2009)**
Another study conducted by the Heat Pump and Thermal Storage Technology Centre of Japan (HPTCJ, 2010) investigated the CO₂ reduction potential in 11 countries. They investigated the possible introduction of current-technology heat pumps into the food and beverage fields at a thermal temperature range of between 0°C and 100°C. They first estimated the energy consumption structure for each country based on the energy consumption per industrial category (sub classification) and by the energy type in the food and beverage fields of the countries surveyed. The top 5 users were in the meat, dairy, grain, bakery and malt beverages categories. They then estimated the fuel consumption that could be replaced by heat pumps in the range of 0-100°C for each industrial sub-classification and combined it with the figures obtained to estimate the total CO₂ reduction through the use of heat pumps in the food and beverage field. They estimated 40 million t-CO₂/year could be reduced across the 11 countries which accounts for 1.3% of the total CO₂ emissions across all industry in all countries.

2.4.2 Case Studies

In 1995 the IEA Heat Pump Centre (IEA, 1995) published results of the installed capacity of industrial heat pumps, the heat pump type and in participating countries. Of the over 4600 installed heat pumps, 110 reported their heating capacity. From the table below it is clearly seen that closed cycle compression and mechanical vapour recompression are the most common with the latter having the largest installed capacity.

<table>
<thead>
<tr>
<th>HP type</th>
<th>No. of reported installations specifying heat capacity</th>
<th>Total installed heating capacity (MW)</th>
<th>Average size (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CCC</td>
<td>54</td>
<td>48</td>
<td>0.9</td>
</tr>
<tr>
<td>MVR</td>
<td>53</td>
<td>794</td>
<td>15.0</td>
</tr>
<tr>
<td>TVR</td>
<td>3</td>
<td>7</td>
<td>2.3</td>
</tr>
<tr>
<td>AHP</td>
<td>1</td>
<td>2</td>
<td>2.0</td>
</tr>
<tr>
<td>HT</td>
<td>3</td>
<td>8</td>
<td>2.7</td>
</tr>
</tbody>
</table>

The following table gives some examples of actual installations in various industries where these installations took place. The COP and payback period are included.

<table>
<thead>
<tr>
<th>Heat source</th>
<th>Heat sink</th>
<th>Output of system</th>
<th>Average COP</th>
<th>Installed cost</th>
<th>Payback period</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poultry</td>
<td>Refrigeration system</td>
<td>Service hot water</td>
<td>900 kW</td>
<td>4.4</td>
<td>$229 000</td>
</tr>
<tr>
<td>Sugar</td>
<td>Warm air from storage tank</td>
<td>Cooled air entering storage tank</td>
<td>154 kW</td>
<td>2.9</td>
<td>$131 000</td>
</tr>
<tr>
<td>Paper</td>
<td>Weak cooking liquor</td>
<td>Heavy black liquor</td>
<td>149.4 MW</td>
<td>36</td>
<td>$8 170 000</td>
</tr>
<tr>
<td>Brewery</td>
<td>Refrigeration system</td>
<td>Service hot water</td>
<td>513 kW</td>
<td>6.1</td>
<td>$160 000</td>
</tr>
<tr>
<td>Dairy</td>
<td>Refrigeration system</td>
<td>Service hot water</td>
<td>2831 kW</td>
<td>4.2</td>
<td>$1 250 000</td>
</tr>
</tbody>
</table>
3 Methodology

The objective of this study is to determine:

- Which industries heat pumps are applicable to
- The energy use in a specific industry according to process and energy type
- The proportion of the total use for thermal purposes
- The proportion that waste heat recovery and heat pumps can displace energy use
- The performance of industrial heat pumps under local conditions
- The economic feasibility of waste heat recovery and heat pump integration

This chapter describes the method followed to achieve these objectives and evaluate the feasibility of using industrial heat pumps for waste heat recovery in industry. The chapter begins with an overview of the research approach and then describes the design method and tools used to obtain the results.

3.1 Research Overview

There are two possible approaches that could be taken when analysing the feasibility of heat pumps for waste heat recovery; these are a case study of a particular industry or a survey across multiple industries.

A case study requires the detailed energy breakdown of each process such that a heat pump can be integrated into the processes. It will be able to evaluate and compare the specific performance of a number of scenarios. A heat pump's basic specifications can be chosen with a complete thermal and economic analysis done for the industry concerned. The downside is that the opportunity to extrapolate the savings to establish a national potential is limited.

A survey requires a breakdown of the energy requirements and temperature levels across a number of industries. From this, an average heat pump performance will be applied to estimate the national energy savings potential. The downside to this approach is that the accuracy cannot be guaranteed due to the highly specific nature of heat pump installations.

Due to the inaccuracy in the survey approach, and the ability to do an in-depth review of the local performance and economic analysis of the case study, the approach used to achieve the objectives as stated above takes the form of a case study.

This industry and specific site will be chosen based on the technological characteristics of heat pumps together with consideration of heat flows and temperature limits in industry. Assessment criteria will be developed to formalise this selection together with the aid of international case studies and surveys. This will ensure that the industry chosen is one that fits the profile of heat pump integration. It will also open up the possibility of extrapolating the potential savings in industry through the use of this technology.

Once a particular site is chosen, the process of collecting all necessary energy data will begin. This will primarily be based on a standard thermal energy audit questionnaire such that the data can
easily be input into simulation software. Energy data that is needed for the system design such as energy flows and temperature levels will be collected.

After the data has been collected it will be entered into the chosen simulation software for analysis. Costs of the systems and components will also be entered such that both a technical and an economical solution can be developed. This software will be used to design an optimised heat exchanger network to maximise waste heat potential as well as to size and correctly place the heat recovery heat pump in the network.

The economic feasibility of the proposed solutions will be established by using financial indicators such as payback period, net present value and internal rate of return.

3.2 Design Method

For optimising the thermal energy supply in industry, a holistic approach is needed that includes both demand reduction by heat recovery and process integration and an integral combination of heat and cold supply technologies under certain economic constraints. A range of methods have been developed to ease the process by which to analyse the complex thermal energy supply and demand system. These methods involve an auditing procedure in which data is collected, analysed and compared to give an output in the form of possible solutions. This procedure can be seen below:

![Diagram of EINSTEIN's 10 steps of the auditing procedure](Schnitzer & IEE, 2009)
3.2.1 Site selection

Since heat pump technology has rather low temperature limits on both the condenser and evaporator side, the industrial sectors dealt with will be those with a high fraction of low and medium temperature heat demand. International experiences and case studies are also examined to determine where this technology works in industry. Once suitable industrial sectors are found, a specific site that is both willing to participate and easily accessible will be investigated further. If this site has the potential for this technology, a full thermal audit will be undertaken.

3.2.2 Data collection

Once the site is chosen, the audit will follow the steps as mentioned above. Pre-audit data acquisition and analysis will justify the need for a full thermal energy audit. The full audit will require data from the industry concerned which will analyse the status quo from which design improvements can be based. A questionnaire (data sheet) will be used to obtain data relating to the following: (data sheet withheld due to confidentiality concerns)

- Total energy use
- Thermal energy use by process and energy type
- Heating and cooling energy use and temperature levels
- Process diagram and thermal energy network
- Heat and cold generation equipment
- Heat and cold distribution equipment
- Operating schedules
- Economic parameters

Both the available heat source and the heat demand needs to be well known in order for a feasible heat recovery solution to be developed. In analysing waste heat potential, not only the amount (quantity) of energy in each of the subsystems needs to be considered, but also the temperature level (quality) of the energy on both the supply and demand side. This makes the analysis much more complex but ultimately necessary for the design of an energy efficient solution. Since heat pumps are limited to low and medium temperatures, the design of a supply system that makes maximum use of the low temperature sources is a necessary precondition for the use of these technologies. Heat integration is therefore strongly dependent on the temperature levels of the demand and supply heat.

The heat demand profile for each process also needs to be known. The time dependence of the heat demand and waste heat availability needs to be known and can be divided as follows:

- Operation: The time during which a constant temperature needs to be maintained
- Start-up: The time when initial heating at start-up begins
- Circulation: Schedule for incoming flows
- Evacuation: Schedule for outgoing flows
While this data will help with the ideal integration of heat pumps into the heat supply system, other factors including economic and practical reasons need to then be considered. Fuel and electricity costs and the proposed rates of increases need to be known for the simulation. The operation and maintenance costs of the machines used for the processes and the generation supply equipment will add to the accuracy of the economic analysis. Company policy around financing and specific discount rates will be used in this analysis.

3.2.3 System design

The system design entails using the data obtained from the thermal energy audit to develop a feasible system to reduce the heat supply and ultimately energy usage. The simulation software, EINSTEIN, and pinch analysis will be used for this process. The design will first establish the baseline energy consumption figures to compare various scenarios against. These scenarios include:

- An optimised heat exchanger network to economically pair waste heat sources and heat sinks. The size, type and placement of these heat exchangers will be calculated based on the best technical and economical solution.
- The integration of a heat pump into the system to upgrade waste heat from a particular heat source to a heat sink. The size, type and placement of the heat pump will be calculated based on the best technical and economical solution.
- Both the optimisation of the heat exchanger network together with the integration of heat pumps to have a complete waste heat recovery solution based on technical and economic aspects.

3.2.4 Economic analysis

The installation of energy efficiency measures requires an initial investment on certain equipment in order for longer term savings to be achieved. The industry needs to be able to evaluate whether these savings can justify the initial investment. Thus an economic analysis will provide clarity on this using the system cost, economic parameters of the business and various feasibility indicators.

3.2.4.1 System cost

While this technology is used extensively internationally, there is very little information relating to the cost breakdown of these systems. The system is also site-specific so it is difficult to obtain estimates on the system costs. The costs used for this analysis will thus be based on both the total costs for international installations of similar sizes as well from the cost estimates provided by contacting local suppliers.

3.2.4.2 Economic parameters

In conjunction with the system costs, the following parameters are used in the economic analysis to calculate the feasibility indicators:
Project life: The project life will be chosen based on the guarantee period of the equipment being used.

Discount rate: The discount rate is the percentage that cash flows in the future are discounted to value their present worth. The company standard discount rate will be used.

Interest rate: The interest rate is linked to the discount rate and will affect the total investment cost on the borrowed money for the initial investment. The company's interest rate that is attainable will be used.

Energy price: Both the fuel and the electricity price for the company as well as rates of increases need to be established. The prices will be obtained from the company with the rates of increases from NERSA.

Incentives: Various energy efficiency funding schemes offered by Eskom and international funding organisations will be investigated to determine if any rebates are available.

### 3.2.4.3 Feasibility indicators

Once the system costs and economic parameters are known, the economic feasibility of the project needs to be evaluated to permit the company to make valuable investment decisions. For this, various feasibility indicators are calculated:

Payback period: This is the length of time required by the project to recover the initial investment through positive cash flows it generates. In other words the time at which the capital investment is equal to the cumulative savings.

Internal rate of return: This is the discount rate that makes the net present value of all cash flows from the investment equal to zero. This indicates the rate of return on this investment so as to be compared to alternative investments.

Net present value: This is the sum of the present values of the individual cash flows of the project. It is used to compare the base case of the business-as-usual scenario with that of the system installed. If the value is higher on the installed system case, this indicates more positive cash flows and hence a feasible investment.

### 3.3 Design Tools

While the following methods have been presented above, this section goes more in-depth with the tools used for these methods. There are many different optimisation strategies for heat recovery systems using multi objective optimisation approaches, non-linear optimisation methods or mixed linear integer programming models. All of these strategies, however, are based on the formulation of composite curves using pinch analysis. These optimisation and integration methods as well as the simulation software that will be used will be presented here.
3.3.1 Pinch Analysis

This is a methodology for minimising energy consumption by optimising process operating conditions, energy supply methods and heat recovery systems (Dumbliauskaite & Becker, 2010). The heat and cold demand of the entire process is displayed in one diagram that shows both the energy demand (heating and cooling) and temperature levels. The analysis presents the energy that can be theoretically saved by the integration of new supply systems and heat recovery techniques.

The pinch theory separates the process into a cold part with surplus heat energy with a need for cooling and a hot part that needs to be heated. This is done by combining the temperature enthalpy curves of all streams that require heating (cold composite curve) and all streams that require cooling (hot composite curve) into one temperature enthalpy diagram. The curves are drawn in such a way that the cold streams are at a lower temperature than the hot streams everywhere in the diagram. This is done by moving the curves along the enthalpy axis since the enthalpy difference represents a relative not absolute measurement. The curves are separated by the point of lowest difference in temperature, $\Delta T_{\text{min}}$. This temperature is chosen based on the minimum $\Delta T$ over the heat exchanger network and represents the “thermodynamic bottleneck” of the process, the “pinch”. This can be seen in figure a below:

![Pinch curves (a) Overlapping of the HCC and CCC (b) design of the GCC (Schnitzer & IEE, 2009)](image)

The pinch temperature cuts the system in half with the area below the pinch requiring cooling and above the pinch requiring heating. Therefore there should ideally be no external heating below the pinch and no external cooling above the pinch as well as no heat exchanger across the pinch. The minimum temperature difference is determined by economical optimisation as a lower $\Delta T_{\text{min}}$ decreases the energy costs but increases the heat exchanger costs (Becker & Maréchal, 2009).

Another way of demonstrating the heat demand of processes in a system is the grand composite curve (GCC). This is constructed by moving the hot composite curve (HCC) and the cold composite curve (CCC) towards each other, such that they touch at the pinch. The horizontal difference between the curves is now drawn into a new temperature-enthalpy diagram. This can be seen in figure b above. If the heat flux increases with increasing temperature, the process functions as a heat sink and conversely if the heat flux increases with lowering temperature, the process can act as a heat source. This can identify the ideal external energy sources required for different streams. By identifying which heat sources can transfer heat to heat sinks of the processes, only the remaining heat demand where no waste heat is available, requires external energy. Also the temperature at which the external source needs to be applied can be seen.
3.3.2 Design, calculation and simulation software tools

There are many software tools available to specifically evaluate the heat recovery potential and design heat exchanger networks. There are also separate tools to aid in the heat generation system design and the heat distribution network design. These tools can only optimise the specific system they are meant to simulate and therefore do not reflect the entire network accurately. There is one software tool, however, that can simultaneously optimise the process, design heat exchanger networks and integrate renewable energy supplies. This tool is called EINSTEIN (The Expert System for Intelligent Supply of Thermal Energy in Industry). This program is used for these reasons as well as the fact that numerous case studies involving the program are given to prove its accuracy in real world simulations as well as a complete user manual and forum for help with its use. This toolkit guides the user through a methodology that designs energy efficiency solutions for a production process based on energy savings and renewable energy sources. It is an open source software designed by Intelligent Energy Europe that allows for a quick energy and economical evaluation of energy efficient systems and technologies. This was used in conjunction with a basic Excel model to check if the results lie in the correct range and also to assist in reporting the results. The EINSTEIN toolkit consists of six modules to guide the user from the initial data acquisition to the final report.

1) Data acquisition and analysis module:
   This module includes a questionnaire for data acquisition and a procedure for estimating non-available data and for consistency checking. Benchmarking of processes can also be done for pre-evaluation.

2) Process optimisation module:
   This module shows the variety of options available to improve the efficiencies of the processes and the equipment installed. It also contains a database of the best available technologies and process optimisation measures for different unit operations.

3) Heat recovery module:
   This module helps with the design and optimisation of an appropriate heat exchanger network for heat recovery and process integration. Pinch analysis is used on the energy demand and availability of processes to analyse process streams and waste heat/cooling demand. Thus the potential for heat recovery can be identified and estimated.

4) Energy supply and renewables module:
   This module helps to select and design the most appropriate energy supply equipment and heat or cold distribution networks. Standard heat and cold supply systems as well as heat pumps, combined heat and power, solar thermal systems and biomass can be considered.

5) Evaluation module:
   This module calculates the energy performance of the designed supply systems using the total cost assessment for economic and financial evaluation. It includes all parameters influencing economic performance of energy efficiency measures.

6) Reporting module:
   This module presents the final audit report that summarises the results of the analysis on both the present state and with the energy saving alternatives. It presents both the performance and economical evaluation.
4 Results

This chapter discloses the results and the analysis of the industry that heat pumps are applicable to, the energy consumption in a specific case study and the performance and economic analysis of the alternatives proposed.

4.1 Preliminary Analysis

4.1.1 Company selection

As mentioned in the literature review regarding international best practices, the following industrial sectors are suitable for the integration of heat pump technology:

- Food & beverage
- Chemical, plastic, rubber and glass
- Metal and metal products
- Pulp and paper
- Textiles

A complete list of these industries that are in South Africa was compiled and can be seen in Appendix A. Companies were then contacted to establish the willingness and relevance to undertake a thermal energy audit, with the potential to implement this technology. There were found to be a number of companies suitable for the study but due to either their unwillingness, uncertainty of what this technology could bring or too many procedural hurdles, they were disregarded. The sector finally chosen was the food and beverage sector with the specific company being Brewery X (name withheld due to confidentiality concerns). They were chosen due to their suitability, ease of access to data and willingness to undertake the energy audit.

4.1.2 Site focus

Within this company there were three levels at which the audit and analysis took place. The first being a high-level, site-wide energy balance showing which specific department to focus on. The next being a more detailed analysis of the specific processes and energy flows within this department. The final being on a specific process within this department where there is a large source of waste heat that heat pumps can be applied to. This is shown in the results below.

4.2 Data Used in the Analysis

The data used to create a base case of the energy consumption as well as that used to design the alternative solutions is discussed below.

4.2.1 Company and process description

The site chosen is a brewery in South Africa. They have a total production of over 3 million hectolitres annually. The brewery consists of a number of departments that incorporate the typical
baking process. This process can be seen summarised below with the full process flow shown in Appendix B.

The brewing process uses malted barley, hops, water and yeast to produce beer. The first step of the process is the transport of the malt grains from storage which are then milled to break open the kernel. This grist is then mixed with hot water to form mash. It then enters the mash tun where it is first heated using hot water and subsequently steam is applied in three stages. This activates the enzymes in the malt which converts the naturally occurring starch into sugar.

The next step involves separating the sugar laden liquid of the mash, called wort, from the mash. This occurs in the lauter tun and involves a filtering and sparging process whereby hot sparge water is sprayed onto the grains to rinse off the remaining extract. The next step involves this wort entering the wort boiler where it is boiled-off at an evaporation rate of 4% for about 1 hour. This boiling performs a number of functions so as to cultivate the colour and flavour of the wort.

Once the boiling is complete, it is filtered before entering the wort cooler where it is cooled to the required pitching temperature. It is then put into a fermentation vessel with yeast where alcohol
and carbon dioxide are produced. This process releases heat that needs to be dissipated and hence it requires cooling. Fermentation lasts between 1 and 10 days.

After fermentation the yeast is removed through a separation process before entering a storage tank where the beer is allowed to mature and stabilise at a certain conditioning temperature. It is then filtered and sent to a Bright beer tank before packaging.

From these tanks it is then pasteurised to remove the remaining harmful bacteria and bottled and packaged to be sent for delivery.

To achieve the heating and cooling in the process above, the following equipment and circuits are used:

- **Steam supply**
  Steam is supplied at a pressure of 10 bar from an electrode boiler. The efficiency of the electrode boiler is stated as being 98%.

- **Hot water supply**
  Hot water at 85°C is supplied by the W85 circuit. This water is heated using an existing heat exchanger to recover the heat of the wort that exits the wort boiler.

- **Chilled liquor supply**
  Chilled water is supplied at 5°C via 2 chillers operating with a coefficient of performance of 3.9.

- **Refrigerant supply**
  There are 2 refrigerant circuits operating with ammonia as the refrigerant; one at -8°C and another at -1°C. Each circuit is supplied by 3 compressors with the -1°C circuit operating with a COP of 3.9 and the -8°C circuit operating with a COP of 3.5

### 4.2.1 Data supplied by the company

The base data used for the analysis was obtained from a previous energy efficiency audit completed at the end of February 2011. This data was presented in the form of a spreadsheet with both high level data of site wide metering and departmental sub-metering data over the period 10/04/2010 to 19/02/2011. The data was recorded in daily intervals and includes water, steam, fuel and electricity consumption as well as production output. Departmental water, steam and electricity consumption was recorded as well as the supply by the generation equipment. Energy balances pertaining to electricity and heat have been constructed.

The data required for this project has been extracted from this audit and was used in the Einstein data sheet. This data sheet covers seven different areas where data is required; from general company information, overall energy usage and economic data to process specific data, generation
and distribution of heat and cold and existing heat recovery equipment. The data from this is used as the input data for the simulation modelling and further analysis.

4.2.3 Other data used in the analysis

All other external data that has been used in the analysis such as electricity tariffs, benchmarks and growth forecasts are mentioned in the section concerned.

4.3 Energy Consumption in the Present State

This section presents the results of the company's electricity and steam consumption. A site wide analysis is done first, from which a more detailed focus is taken on a potential department.

4.3.1 Site-Wide Fuel and Electricity Consumption

The following tables present the site wide electricity and steam consumption disaggregated into the various departments. For the purpose of this study, all steam is assumed to come from an electricity source.

The following charts show the distribution of the electricity and steam consumption for the various departments to ascertain which are the most electricity and steam intensive departments.

![Site Electricity Balance](image)

Figure 17: Chart showing distribution of electricity demand
The following chart shows the total of the electricity and steam energy requirement per department.

Since this thesis is focussed on waste heat recovery, the site heat balance which records the steam consumption is more important than the electricity balance. The brewhouses and pasteurisers are the most energy intensive, using 22% and 20% of the total energy consumption respectively. After analysing the literature and consultation with the engineering managers, it was discovered that the pasteuriser energy consumption is higher than benchmark data. This was, however, primarily due to energy management issues such as maintenance of steam traps and process optimisation, rather than a waste heat recovery solution. Hence this is beyond the scope of this thesis and was disregarded.

Figure 18: Chart showing distribution of heat demand

Figure 19: Chart showing distribution of total energy demand
On the other hand, the brewhouses energy consumption lends itself to waste heat recovery with many fluid streams requiring both heating and cooling throughout several processes. Steam is directly used in the process for heating and while most of it is re-used, there is a large percentage that is discarded to the atmosphere. Hence the focus of this thesis will be on the brewhouse energy consumption which will be presented in the next section.

4.3.2 Brewhouse Process Heat and Cooling Demand

This section presents the results of the brewing process energy use. Both the heating and cooling demand need to be taken into account for a complete energy balance. The first graph shows the total monthly heat and cooling energy demand to see the seasonal distribution of the demand. The chilled liquor energy demand is a calculated value based on the measurement of the chilled water daily volume. The brewhouse energy demand is a recorded value of the steam usage which has been converted to an energy value using the brewery conversion factor of 704 kWh / tonne.

![Figure 20: Monthly heating and cooling energy for brewing](image)

The graph shows the minimum monthly demand being 1 907 MWh with the demand rising before the holiday season to 3 156 MWh. The average monthly brewing demand is 2 358 MWh. The next step is to disaggregate this demand into specific processes within the brewhouse.

As can be seen in the process flow diagram in Appendix B, there are four heating and three cooling processes within the brewhouse. Hot water is generated in a heat exchanger from cooling the wort that comes from the wort boiler. This heated water is then used to heat the incoming mash in the first phase of the mash tun and also to heat the sparge water and flush water in the lauter tun. Steam is used to heat the second phase of the mash tun as well as for the wort boiling process. Chilled water is used to cool the wort coming from the wort boiler which is then further cooled in the fermentation process. A post centrifugal chiller then further chills the wort before it goes into storage.
With the process temperature and flow data supplied in the report, a simulation model was constructed in both Excel and EINSTEIN to calculate the energy consumption per process. Firstly, the average energy consumption per brew was calculated and then multiplied by the number of brews per year. The average number of brews for brewhouse 1 and brewhouse 2 were 1244 and 1803 respectively. The following two tables present the EINSTEIN results of this heat demand and cooling demand per process.

### Table 7: Useful process heat demand by process

<table>
<thead>
<tr>
<th>Process</th>
<th>Brewhouse 1 (kWh)</th>
<th>Brewhouse 2 (kWh)</th>
<th>Total (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mash tun phase 1</td>
<td>2124.33</td>
<td>1593.25</td>
<td>19.2</td>
</tr>
<tr>
<td>Mash tun phase 2</td>
<td>688.16</td>
<td>516.12</td>
<td>6.2</td>
</tr>
<tr>
<td>Lauter tun</td>
<td>2952.20</td>
<td>2214.15</td>
<td>26.8</td>
</tr>
<tr>
<td>Wort boiling</td>
<td>5271.24</td>
<td>3953.43</td>
<td>47.8</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>11035.93</strong></td>
<td><strong>8276.95</strong></td>
<td><strong>100.0</strong></td>
</tr>
</tbody>
</table>

### Table 8: Useful process cooling demand by process

<table>
<thead>
<tr>
<th>Process</th>
<th>Brewhouse 1 (kWh)</th>
<th>Brewhouse 2 (kWh)</th>
<th>Total (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort cooling</td>
<td>9168.23</td>
<td>6876.18</td>
<td>89.0</td>
</tr>
<tr>
<td>Fermentation</td>
<td>309.04</td>
<td>231.78</td>
<td>3.0</td>
</tr>
<tr>
<td>Post centrifuge chiller</td>
<td>824.11</td>
<td>618.08</td>
<td>8.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>10301.39</strong></td>
<td><strong>7726.04</strong></td>
<td><strong>100.00</strong></td>
</tr>
</tbody>
</table>

The total process heat demand is analysed not only in quantity but also in quality, i.e. by temperature levels. This is important in the subsequent steps for the design of energy efficient supply solutions. The following figures present this demand by temperature levels.

### Table 9: Process heat demand and supply by temperature levels

<table>
<thead>
<tr>
<th>Temperature level</th>
<th>Process heat consumption by process temperature</th>
<th>Total heat supply by central heat supply temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MWh</td>
<td>% of total</td>
</tr>
<tr>
<td>&lt; 60 °C</td>
<td>9905.41</td>
<td>35.28</td>
</tr>
<tr>
<td>60 - 80 °C</td>
<td>6718.07</td>
<td>23.92</td>
</tr>
<tr>
<td>80 - 100 °C</td>
<td>11456.70</td>
<td>40.80</td>
</tr>
<tr>
<td>100 - 120 °C</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>&gt; 120 °C</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>28080.19</strong></td>
<td><strong>100.00</strong></td>
</tr>
</tbody>
</table>
In the above table and figure we can see that the total heat supplied by the central supply system is less than the process heat consumption. This is due to the existing heat recovery system installed consisting of heat exchangers across the exiting wort boiler line and the W85 line. 5 745 MWh and 7 674 MWh of heat is transferred to the incoming wort for the first phase of the mash tun and the lauter tun respectively. This is the entire heating needs of these processes and is represented in the chart below.
The following figures show the cooling demand by temperature levels.

Table 10: Process cooling demand and supply by temperature levels

<table>
<thead>
<tr>
<th>Temperature level</th>
<th>Process cooling consumption by process temperature</th>
<th>Total cooling supply by central cooling supply temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MWh</td>
<td>% of total</td>
</tr>
<tr>
<td>&lt; -20 °C</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>-20 - -10 °C</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>-10 - 5 °C</td>
<td>1872.16</td>
<td>7.00</td>
</tr>
<tr>
<td>5 - 15 °C</td>
<td>2674.50</td>
<td>10.00</td>
</tr>
<tr>
<td>15 - 25 °C</td>
<td>2674.50</td>
<td>10.00</td>
</tr>
<tr>
<td>25 - 40 °C</td>
<td>4011.73</td>
<td>15.00</td>
</tr>
<tr>
<td>40 - 60 °C</td>
<td>5349.00</td>
<td>20.00</td>
</tr>
<tr>
<td>60 - 80 °C</td>
<td>5349.00</td>
<td>20.00</td>
</tr>
<tr>
<td>80 - 100 °C</td>
<td>4814.10</td>
<td>18.00</td>
</tr>
<tr>
<td>&gt; 100 °C</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>Total</td>
<td>26744.98</td>
<td>100.00</td>
</tr>
</tbody>
</table>

Figure 23: Distribution of cooling demand by temperature levels

As can be seen again in the above table and figure, the total cooling supplied by the central cooling system is less than the process cooling consumption. This is as a result of the same heat exchangers as mentioned above that remove heat in the wort cooling process. A total of 13 419 MWh of heat is removed from the cooling process in these heat exchangers. The following figure illustrates this.
With the heating and cooling demand calculated above, it is now possible to construct the composite curves to perform the pinch analysis. This can be seen in the figures below.

![Figure 24: Distribution of cooling demand to show external cooling supplied vs heat recovery](image)

![Figure 25: Hot and cold composite curves](image)

![Figure 26: Remaining yearly energy demand and energy availability](image)
From the composite curve above, we can see that the pinch temperature occurs at the far right of the hot composite curve with the cold composite curve extending beyond this. This indicates that external heat demand is only required for that portion of the cold composite curve that doesn't lie under the hot composite curve (above 100°C). There is an excess of waste heat (below 100°C) that can be supplied to the heat demand through simple heat exchangers. Most of this heat (25°C-78°C) is already used in existing heat exchangers to heat the W85 circuit and hence the heat between 78°C and 100°C is the section that needs to be focussed on. Since the hot composite curve extends well beyond the cold composite curve on the left, a large external cooling demand is required.

The remaining yearly energy demand also illustrates this with the remaining heat demand only being required above 80°C with a large portion of heat being available below 100°C.

Therefore either a simple heat exchanger could be integrated to re-use the heat between 80°C and 100°C for heating or a heat pump could be suitably integrated with the source coming from the 100°C waste heat and being upgraded to 110°C to meet that demand.

4.3.3 Wort boiler heating

The wort boiling process consumes the majority of the heat demand (47.8%) and does not use any available waste heat due to the high temperature levels required (over 100°C). The energy from the wort leaving the boiler is transferred to the W85 water that is used to heat the first phase of the mash tun and the lauter tun. There is, however, no heat recovery systems installed for the wort boiling process whereby wort is boiled off directly to the atmosphere. This makes it the ideal process to design a waste heat recovery solution. Wort boiling can be divided into two phases; wort pre-heating whereby the wort is heated from the lauter tun to 100°C in the kettle, and wort boiling whereby the wort in the kettle is boiled off at an evaporation rate of 4%. The following table shows these figures for both brewhouses.

Table 11: Table showing recorded energy usage in the wort boiler

<table>
<thead>
<tr>
<th>Company X - recorded</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boil-off kg per brew</td>
<td>5019.8</td>
<td>3939.3</td>
<td>8959.1</td>
</tr>
<tr>
<td>Pre-heat kg per brew</td>
<td>5228.4</td>
<td>3020.7</td>
<td>8249.1</td>
</tr>
<tr>
<td>Boil-off kWh per brew (USH)</td>
<td>3534.0</td>
<td>2773.2</td>
<td>6307.2</td>
</tr>
<tr>
<td>Pre-heat kWh per brew (USH)</td>
<td>3680.8</td>
<td>2126.6</td>
<td>5807.4</td>
</tr>
<tr>
<td>Boil-off kWh per brew (UPH)</td>
<td>2993.8</td>
<td>2349.3</td>
<td>5343.1</td>
</tr>
<tr>
<td>Pre-heat kWh per brew (UPH)</td>
<td>3118.1</td>
<td>1801.5</td>
<td>4919.7</td>
</tr>
</tbody>
</table>

The steam usage was recorded in terms of mass per brew and was converted to an energy figure using a conversion factor. The useful supply heat (USH) was converted using the agreed brewery figure for the heat input to the steam of 704 kWh / tonne, while the useful process heat (UPH) of 596 kWh / tonne used the heat content of the steam at the wort boiler supply pressure of 2.5 bar. The following table shows the energy usage as calculated in the EINSTEIN model.
Table 12: Table showing EINSTEIN calculated energy usage in the wort boiler

<table>
<thead>
<tr>
<th>EINSTEIN - calculated</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Boil-off kWh per brew</strong></td>
<td>2708.8</td>
<td>2031.6</td>
<td>4740.3</td>
</tr>
<tr>
<td><strong>Pre-heat kWh per brew</strong></td>
<td>3069.3</td>
<td>2302.0</td>
<td>5371.3</td>
</tr>
<tr>
<td><strong>Boil-off Yearly MWh</strong></td>
<td>3369.7</td>
<td>3662.9</td>
<td>7032.6</td>
</tr>
<tr>
<td><strong>Pre-heat Yearly MWh</strong></td>
<td>3818.2</td>
<td>4150.5</td>
<td>7968.7</td>
</tr>
</tbody>
</table>

We can see the actual recorded data vs the figures calculated in the EINSTEIN model. The boil-off energy consumption figures indicate a difference of 10% and 13% less than actual in brewhouse 1 and 2 respectively. These differences could be associated to losses in the system. The pre-heat energy consumption figures indicate a difference of 2% less and 27% more than actual in brewhouse 1 and 2 respectively. While the brewhouse 1 figure could be associated to losses, the brewhouse 2 figure cannot be true. There is a suspected recording error in the brewhouse 2 steam meter which could be the cause of this discrepancy. The EINSTEIN figures are used in calculating the proposed solution shown below.

The following table shows the boil-off volume and energy released per brew. The boil-off volume was calculated based on the 4% evaporation rate as mentioned above, with a heat of vaporisation of wort being 2257 kJ/kg. The energy shown is that which can ideally be recovered using one of the alternative solutions shown below.

Table 13: Table showing volume and energy released in the boil-off phase

<table>
<thead>
<tr>
<th>Wort boiloff</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Volume per brew (l)</strong></td>
<td>3666.7</td>
<td>2750.0</td>
<td>6416.7</td>
</tr>
<tr>
<td><strong>Energy per brew (kWh)</strong></td>
<td>2202.4</td>
<td>1651.8</td>
<td>3854.1</td>
</tr>
<tr>
<td><strong>Yearly (MWh)</strong></td>
<td>2739.7</td>
<td>2978.1</td>
<td>5717.9</td>
</tr>
</tbody>
</table>

4.3.4 Comparison with Benchmark Reference Data

While the above data shows the total absolute energy consumption of the whole site, the brewhouses and the wort boiling process, it is also useful to calculate energy intensity figures. This allows comparative analysis with both previous site figures and international case studies to see if there have been any improvements as well to see where the brewery intensity lies globally. The figures can also be compared against published best available techniques data to see how much further improvement can be made. The following table shows these figures.
Table 14: Table showing overall energy benchmark figures

<table>
<thead>
<tr>
<th>Site wide energy benchmark</th>
<th>Average European breweries</th>
<th>German breweries</th>
<th>Brewery X</th>
<th>BAT</th>
<th>BREF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific heat requirement</td>
<td>kWh/hl</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11-12</td>
<td>10.87</td>
<td>8.8</td>
<td>7.5</td>
<td>4.5</td>
<td></td>
</tr>
<tr>
<td>Specific electricity requirement</td>
<td>kWh/hl</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>36-40</td>
<td>28.47</td>
<td>24.4</td>
<td>23.6</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Total energy requirement</td>
<td>kWh/hl</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>47.52</td>
<td>39.34</td>
<td>33.2</td>
<td>31.1</td>
<td>19.5</td>
<td></td>
</tr>
</tbody>
</table>

Source: (Scheller, Michel, & Funk, 2008), (Blumelhuber, Doemens Brewing, & Malting Academy, 2008)

From the above figures we can see that Brewery X has lower energy intensity figures than both the average European brewery and German breweries. The figures are only slightly off the Best Available Techniques (BAT) figures that are published based on standard energy efficiency improvements in breweries involving process optimisation and energy management measures. The Best Available Techniques Reference Guide (BREF) shows what is possible when all energy efficiency improvements are implemented. This includes waste heat recovery and vapour recompression systems and can be seen as the current lowest benchmark for energy intensity.

The following tables show the energy intensity of specific processes.

Table 15: Table showing process specific benchmark figures

<table>
<thead>
<tr>
<th>Process</th>
<th>Atmospheric boiling</th>
<th>Dynamic low-pressure boiling</th>
<th>Atmospheric boiling with internal boiler</th>
<th>Brewery X</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>7.5% total evaporation kWh/hl</td>
<td>4.5% total evaporation kWh/hl</td>
<td>3% total evaporation kWh/hl</td>
<td>4% total evaporation kWh/hl</td>
</tr>
<tr>
<td>Mashing</td>
<td>2.21</td>
<td>2.21</td>
<td>2.21</td>
<td>2.22</td>
</tr>
<tr>
<td>Wort pre-heating</td>
<td>3.38</td>
<td>3.29</td>
<td>3.24</td>
<td>3.63</td>
</tr>
<tr>
<td>Wort boiling</td>
<td>5.03</td>
<td>3.02</td>
<td>2.01</td>
<td>4.13</td>
</tr>
<tr>
<td>Hot service water</td>
<td>0.28</td>
<td>0.28</td>
<td>0.25</td>
<td>-</td>
</tr>
<tr>
<td>CIP</td>
<td>0.28</td>
<td>0.28</td>
<td>0.23</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td>11.18</td>
<td>9.08</td>
<td>7.94</td>
<td>9.75</td>
</tr>
</tbody>
</table>

Source: (Scheller et al., 2008)

The table above shows benchmark figures taken from the above-mentioned Energy Efficiency in the Brewhouse report. Figures are given for atmospheric boiling, dynamic low-pressure boiling and atmospheric boiling with an internal boiler for 5 different processes. The boiling at Brewery X takes place at atmospheric pressure with an evaporation rate given as 4%. The mashing figure above is calculated from the EINSTEIN simulation model since there were no direct measurements available, while the wort pre-heating and wort boiling are calculated from direct measurements of steam flow and brew volume. There were no calculations done on hot service water and CIP as it was beyond the scope of this thesis.
4.4 Description and Design of Alternatives Proposed

This section describes the alternatives proposed for heat recovery in the wort boiler. A description, schematic diagram and thermal performance will be given for each of the proposals. The first proposal is simply a waste heat recovery solution, the next two are solutions that involve different heat pumps for waste heat recovery and the last is a combination of heat pumps and straight waste heat recovery. The next section will then compare these proposals to the base case and each other. While the two boilers have different heating systems, calculations were only done on one and approximated to the other. A schematic of the existing system is shown below:

![Schematic of existing system](image)

**4.4.1 Vapour condenser**

This solution proposes the recovery of high grade heat energy from the kettle vapours using either spray condensers or heat exchangers. The heat from the vapour can be used to pre-heat the incoming wort while the heat from the vapour condensate can be used to produce hot water for CIP, other process heating or in other applications in the brewery. A schematic diagram of this design is shown below.

![Schematic of wort boiling vapour condenser](image)
External supply steam is used to boil-off the wort at the rate required. Instead of this wort vapour being released to the atmosphere (1), it is instead directed to a vapour condenser (2) where it condenses and transfers heat to an external fluid. This fluid (normally water) is pumped to a storage vessel where it is stored until the start of the next brew. When the next brew begins, the hot water being stored is pumped through a heat exchanger (4) to transfer heat to the incoming wort (3). This pre-heats the wort to near boiling temperature such that a small amount of external steam is required for wort pre-heating. External steam is still required for the wort boil-off stage. After boiling is completed, the wort leaves to the whirlpool from which it is later cooled using existing heat exchangers and the cooling medium.

With the EINSTEIN model constructed for the base case scenario, a new proposal was now created with the vapour condenser alternative. The following table shows the heat exchanger data after the simulation was run.

### Table 16: EINSTEIN calculated vapour condenser heat exchanger data

<table>
<thead>
<tr>
<th>Vapour condenser</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Heat source</strong></td>
<td>Wort vapor boil-off</td>
</tr>
<tr>
<td><strong>Heat sink</strong></td>
<td>Wort pre-heat</td>
</tr>
<tr>
<td>T1 hot medium</td>
<td>101 °C</td>
</tr>
<tr>
<td>T2 hot medium</td>
<td>100.9 °C</td>
</tr>
<tr>
<td>T3 cold medium</td>
<td>72 °C</td>
</tr>
<tr>
<td>T4 cold medium</td>
<td>88.62 °C</td>
</tr>
<tr>
<td><strong>Heat transfer rate</strong></td>
<td>1815.24 kW</td>
</tr>
<tr>
<td>LMTD</td>
<td>19.5 °C</td>
</tr>
<tr>
<td>UA value</td>
<td>146.6 kW/K</td>
</tr>
<tr>
<td>Surface area</td>
<td>80.7 m²</td>
</tr>
</tbody>
</table>

The following table shows the energy transferred in this heat exchanger.

### Table 17: EINSTEIN calculated vapour condenser heat transferred

<table>
<thead>
<tr>
<th>Vapour condenser</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Heat transferred per brew</strong></td>
<td>kWh</td>
<td>1815.24</td>
<td>1482.44</td>
</tr>
<tr>
<td><strong>Heat transferred annually</strong></td>
<td>MWh</td>
<td>2258.16</td>
<td>2672.85</td>
</tr>
</tbody>
</table>

This heat transferred represents 82.4% of the total heat released during the boil-off phase and 61.9% of the heat required for the pre-heat phase.

### 4.4.2 Mechanical Vapour Recompression

In a mechanical vapour recompression system, the wort is boiled using an external steam supply. Once boiling, the vapour is captured and compressed to a higher pressure where it may be re-used in the process. A schematic diagram of this design is shown below.
Using an external heat supply to start with, the wort is boiled and expands in the kettle at 100°C. Vapour from the kettle is drawn in by a compressor (1) and compressed above atmospheric pressure (2). The temperature increase of this compressed vapour is enough such that the vapour can be reused for heating the wort in the boil-off phase. Hence no external steam supply will be required while the mechanical vapour compressor is in operation. The generated useful heat contains more energy than the electricity required to compress the steam and thus will reduce the total steam requirements substantially while only increasing the electricity requirements marginally.

The new proposal for the mechanical vapour recompression alternative was constructed. The following table presents the details of this system.

Table 18: EINSTEIN calculated mechanical vapour recompression system data

<table>
<thead>
<tr>
<th>Mechanical vapour recompression</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Heat source</strong></td>
<td>Wort vapour boil-off</td>
</tr>
<tr>
<td><strong>Heat sink</strong></td>
<td>Wort boil-off</td>
</tr>
<tr>
<td><strong>T1</strong></td>
<td>101 °C</td>
</tr>
<tr>
<td><strong>T2</strong></td>
<td>127.41 °C</td>
</tr>
<tr>
<td><strong>Temperature lift</strong></td>
<td>26.41 °C</td>
</tr>
<tr>
<td><strong>P1</strong></td>
<td>101.325 kPa</td>
</tr>
<tr>
<td><strong>P2</strong></td>
<td>250 kPa</td>
</tr>
<tr>
<td><strong>Compression ratio</strong></td>
<td>2.5</td>
</tr>
<tr>
<td><strong>Isentropic efficiency</strong></td>
<td>0.7</td>
</tr>
<tr>
<td><strong>Electric motor efficiency</strong></td>
<td>0.85</td>
</tr>
</tbody>
</table>
The following table shows the heating capacity of this system.

Table 19: EINSTEIN calculated mechanical vapour recompression heat capacity

<table>
<thead>
<tr>
<th>Mechanical vapour recompression</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort vapour flow rate (kg/s)</td>
<td>0.98</td>
<td>0.91</td>
<td>1.89</td>
</tr>
<tr>
<td>Work input (kW)</td>
<td>242.65</td>
<td>227.48</td>
<td>470.13</td>
</tr>
<tr>
<td>Power consumption (kW)</td>
<td>407.81</td>
<td>382.32</td>
<td>790.13</td>
</tr>
<tr>
<td>Electricity consumption per brew (kWh)</td>
<td>407.81</td>
<td>305.86</td>
<td>713.67</td>
</tr>
<tr>
<td>Electricity consumption annually (MWh)</td>
<td>507.32</td>
<td>551.46</td>
<td>1058.78</td>
</tr>
<tr>
<td>Heating capacity per brew (kWh)</td>
<td>2610.17</td>
<td>1957.63</td>
<td>4567.80</td>
</tr>
<tr>
<td>Heating capacity annually (MWh)</td>
<td>3247.06</td>
<td>3529.61</td>
<td>6776.66</td>
</tr>
</tbody>
</table>

The system as mentioned above has a heating coefficient of performance (COP) of 6.4 and represents 96.4% of the heat required for boil-off.

### 4.4.3 Thermal Vapour Recompression

In a thermal vapour recompression system, the wort is first boiled using an external steam supply. Once boiling, a portion of the evaporated water vapour is compressed by high pressure steam and re-used in the boiling process. A schematic diagram of this design is shown below.

Using an external heat supply to start with, the wort is boiled and expands in the kettle at 100°C. A portion of the vapours is condensed for hot water generation and a portion (1) is sucked into the steam jet compressor being forced by an external steam supply of at least 6 bar (2). The discharge
steam and vapour mix (3) is then used to heat the boiler. A disadvantage is that the condensate cannot be sent back to the steam plant as it is contaminated by the vapours given off by the wort.

The new proposal for the thermal vapour recompression alternative was constructed. The following table presents the details of this system.

**Table 20: EINSTEIN calculated thermal vapour recompression system data**

<table>
<thead>
<tr>
<th>Thermal vapour recompression</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source</td>
<td>Wort vapour boil-off</td>
</tr>
<tr>
<td>Heat sink</td>
<td>Wort boil-off</td>
</tr>
<tr>
<td>T1</td>
<td>101 °C</td>
</tr>
<tr>
<td>T2</td>
<td>175.38 °C</td>
</tr>
<tr>
<td>T3</td>
<td>127.41 °C</td>
</tr>
<tr>
<td>Temperature lift</td>
<td>26.41 °C</td>
</tr>
<tr>
<td>P1</td>
<td>101.325 kPa</td>
</tr>
<tr>
<td>P2</td>
<td>900 kPa</td>
</tr>
<tr>
<td>P3</td>
<td>250 kPa</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>2.5</td>
</tr>
<tr>
<td>Motive steam to recovered vapour ratio</td>
<td>0.48</td>
</tr>
</tbody>
</table>

The following table shows the heating capacity of this system.

**Table 21: EINSTEIN calculated thermal vapour recompression heat capacity**

<table>
<thead>
<tr>
<th>Thermal vapour recompression</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort vapour flow rate</td>
<td>kg/s</td>
<td>0.88</td>
<td>0.82</td>
</tr>
<tr>
<td>Motive steam flow rate</td>
<td>kg/s</td>
<td>0.42</td>
<td>0.39</td>
</tr>
<tr>
<td>Motive steam energy consumption per brew</td>
<td>kWh</td>
<td>1065.57</td>
<td>799.18</td>
</tr>
<tr>
<td>Motive steam energy consumption annually</td>
<td>MWh</td>
<td>1325.57</td>
<td>1440.92</td>
</tr>
<tr>
<td>Heating capacity per brew</td>
<td>kWh</td>
<td>2787.83</td>
<td>2090.87</td>
</tr>
<tr>
<td>Heating capacity annually</td>
<td>MWh</td>
<td>3468.06</td>
<td>3769.84</td>
</tr>
</tbody>
</table>

The system as mentioned above has a heating coefficient of performance (COP) of 2.6 and represents 103% of the heat required for boil-off.

**4.4.4 Combination heat pump and waste heat recovery**

In a combined heat pump and waste heat recovery system, the wort vapour is still raised to the required boil-off heating temperature using vapour recompression as shown above. The difference, however, is that once the recompressed wort vapour has been used to supply heat for the boil-off process, it is passed through a further heat exchanger to pre-heat the incoming wort. Since this is a batch process, there will need to be a storage vessel to account for the time offset.
The new proposal for the combined mechanical vapour recompression and heat exchanger system was constructed. The following table presents the recovered energy of this system.

Table 22: Combined heat pump and heat exchanger recovered heat

<table>
<thead>
<tr>
<th>Combined heat pump and heat exchanger</th>
<th>Brewhouse 1</th>
<th>Brewhouse 2</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat pump capacity per brew</td>
<td>kWh</td>
<td>2610.17</td>
<td>1957.62</td>
</tr>
<tr>
<td>Heat transferred per brew</td>
<td>kWh</td>
<td>84.88</td>
<td>63.66</td>
</tr>
</tbody>
</table>

While the heat pump will recover the same amount of heat as calculated above, the heat exchanger will only be able to recover 3% of the heat required for the pre-heat phase. This proposal is therefore deemed unsuitable and will not be carried further to the comparative stage.

4.5 Comparative Study of the Proposed Alternatives

The description and design for each of the separate alternatives has been presented in the previous section. This section aims to compare each of these alternatives with respect to both the thermal performance and an economic analysis such that a final solution can be suggested.

4.5.1 Thermal performance

The following three tables show the useful supply heat (steam), electricity, and total energy usage in the wort boilers for the present state (base), vapour condenser (VC), mechanical vapour recompression (MVR) and thermal vapour recompression (TVR).

Table 23: Table showing annual supply heat consumption and savings for each of the alternatives

<table>
<thead>
<tr>
<th>Useful supply heat (steam)</th>
<th>Base</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort boiler 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Boil-off</td>
<td>MWh</td>
<td>3370</td>
<td>123</td>
<td>1227</td>
</tr>
<tr>
<td>Pre-heat</td>
<td>MWh</td>
<td>3818</td>
<td>3818</td>
<td></td>
</tr>
<tr>
<td>Wort boiler 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Boil-off</td>
<td>MWh</td>
<td>3663</td>
<td>133</td>
<td>1334</td>
</tr>
<tr>
<td>Pre-heat</td>
<td>MWh</td>
<td>4150</td>
<td>4150</td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>MWh</td>
<td>15001</td>
<td>10070</td>
<td>8225</td>
</tr>
<tr>
<td>Savings</td>
<td>MWh</td>
<td>-</td>
<td>4931</td>
<td>6777</td>
</tr>
</tbody>
</table>

Table 24: Table showing annual electricity consumption and savings for each of the alternatives

<table>
<thead>
<tr>
<th>Electricity</th>
<th>Base</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort boilers</td>
<td>MWh</td>
<td>819</td>
<td>1878</td>
<td>819</td>
</tr>
<tr>
<td>Savings</td>
<td>MWh</td>
<td>-</td>
<td>-1059</td>
<td>0</td>
</tr>
</tbody>
</table>
Table 25: Table showing total annual energy consumption and savings for each of the alternatives

<table>
<thead>
<tr>
<th>Energy</th>
<th>Base</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam MWh</td>
<td>15001</td>
<td>10070</td>
<td>8225</td>
<td>10530</td>
</tr>
<tr>
<td>Electricity MWh</td>
<td>819</td>
<td>819</td>
<td>1878</td>
<td>819</td>
</tr>
<tr>
<td>Total MWh</td>
<td>15821</td>
<td>10890</td>
<td>10103</td>
<td>11349</td>
</tr>
<tr>
<td>Savings %</td>
<td>-</td>
<td>31.2%</td>
<td>36.1%</td>
<td>28.3%</td>
</tr>
</tbody>
</table>

The following gives a graphical overview of the thermal performance.

![Chart showing energy consumption and savings for the three alternatives](image)

Figure 31: Overview of energy consumption and savings for each of the three alternatives

While the consumption and savings above have been shown for the wort boilers, the following table presents the relative savings for the wort boiler, the whole brewhouse and the whole plant.

Table 26: Table showing relative energy savings for the three alternatives

<table>
<thead>
<tr>
<th>Energy savings</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort boilers total kWh/hl</td>
<td>1.50</td>
<td>1.74</td>
<td>1.36</td>
</tr>
<tr>
<td>Wort boilers total %</td>
<td>31.2%</td>
<td>36.1%</td>
<td>28.3%</td>
</tr>
<tr>
<td>Brewhouses total %</td>
<td>21.9%</td>
<td>25.4%</td>
<td>19.9%</td>
</tr>
<tr>
<td>Brewery X total %</td>
<td>4.8%</td>
<td>5.6%</td>
<td>4.4%</td>
</tr>
</tbody>
</table>

4.5.2 Emissions saving potential

This section presents the CO₂ emissions reduction potential of these alternatives. The emissions factors used, as given by Brewery X, are shown below.

- Electricity - 0.848 kgCO₂/kWh
- Steam - 0.762 kgCO₂/kWh
Table 27: Table showing CO₂ savings for each of the alternatives

<table>
<thead>
<tr>
<th>CO₂ savings</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>tCO₂/year</td>
<td>3759</td>
<td>4268</td>
<td>3408</td>
</tr>
<tr>
<td>%</td>
<td>4.7%</td>
<td>5.3%</td>
<td>4.2%</td>
</tr>
</tbody>
</table>

### 4.5.2 Economic analysis

This section presents the results of the economic feasibility analysis carried out with respect to the 3 alternatives discussed above. The indicators used to assess feasibility are net present value (NPV), internal rate of return (IRR) and payback period. The process of calculation is now shown together with the key data required to obtain these feasibility indicators. A project lifespan of 10 years is assumed.

- **Investment cost**
  This is the turn-key cost of the alternative. It is calculated using the heating capacity of the system and an estimated cost per kW given by international experience. The cost for the vapour condenser was established using the EINSTEIN database of heat exchanger and storage costs while the specific costs for the heat pumps were taken from the industrial heat pumps utilisation guide by Leonardo Energy (Soroka, 2007) and the industrial heat pumps report by IEA (IEA, 1995). These costs are shown in the table below.

  Table 28: Specific investment costs

<table>
<thead>
<tr>
<th>Investment cost</th>
<th>R/kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>VC</td>
<td>1 967</td>
</tr>
<tr>
<td>MVR</td>
<td>2 405</td>
</tr>
<tr>
<td>TVR</td>
<td>1 312</td>
</tr>
</tbody>
</table>

- **Energy cost**
  Due to confidentiality concerns, the actual steam and electricity prices were not used. The City of Cape Town large power user medium voltage tariff was used to get an electricity price of 45.4c/kWh (City of Cape Town, 2012) while an estimated relative steam price of R200/tonne was used. These figures eliminate the specific savings potential but the relative savings can still be seen. The increase in these prices for 2010 to 2014 were taken from figures published by Eskom (25%, 25.8%, 25.9%, 25%), while the price increase for the period 2015 to 2021 (4%) was taken from the estimated electricity price path in the South African Integrated Resource Plan (IRP, 2011).

  The energy consumption figures for 2010/2011 have been presented above, while the beer growth figure is taken from the SAB Global Beer Market Trends (SAB, 2011) figure of 3.3%.

- **Maintenance cost**
  The maintenance cost for each alternative system is estimated to be 1% of the installation cost per year. The maintenance cost for the current system is assumed to be incorporated into the energy price.
With these figures, the life cycle cost of the base case as well as each of the alternatives was calculated. Thereafter, the economical savings potential over the 10 year project lifespan could be calculated. The table below summarises these figures for the wort boiler.

Table 29: Life cycle costs and savings of the base case and three alternatives

<table>
<thead>
<tr>
<th>10 year period (R)</th>
<th>Base</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Investment cost</td>
<td>0</td>
<td>7 181 010</td>
<td>11 181 390</td>
<td>6 426 840</td>
</tr>
<tr>
<td>Energy cost</td>
<td>Steam 109 659 152</td>
<td>73 613 733</td>
<td>60 122 101</td>
<td>76 973 374</td>
</tr>
<tr>
<td></td>
<td>Electricity 9 569 564</td>
<td>9 569 564</td>
<td>21 938 113</td>
<td>9 569 564</td>
</tr>
<tr>
<td>Maintenance cost</td>
<td>0</td>
<td>718 101</td>
<td>1 118 139</td>
<td>642 684</td>
</tr>
<tr>
<td>Life cycle cost</td>
<td>119 228 716</td>
<td>91 082 408</td>
<td>94 359 742</td>
<td>93 612 463</td>
</tr>
<tr>
<td>Economic savings potential</td>
<td>-</td>
<td>28 146 308</td>
<td>24 868 974</td>
<td>25 616 253</td>
</tr>
</tbody>
</table>

The following graph shows the cumulative net savings for each of the alternatives to graphically represent these figures.

![Cumulative savings graph](image)

Figure 32: Cumulative net savings of each of the alternatives

The feasibility indicators are shown below. A 100% equity investment is assumed at the start of the project with a discount rate of 10%, as used by Brewery X, being used.

Table 30: Feasibility indicators of the three alternatives

<table>
<thead>
<tr>
<th></th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Net present value</td>
<td>R 9 978 563</td>
<td>R 4 446 143</td>
<td>R 9 179 799</td>
</tr>
<tr>
<td>Payback</td>
<td>3.7 years</td>
<td>5.8 years</td>
<td>3.7 years</td>
</tr>
<tr>
<td>IRR</td>
<td>29%</td>
<td>15%</td>
<td>29%</td>
</tr>
</tbody>
</table>
4.6 South African Energy Savings Potential

With the assumption that the energy consumption ratio is similar for all breweries across South Africa, Brewery X’s results were extrapolated to indicate the national energy savings potential. The estimated total beer market in South Africa is 31.4 million hectolitres (SAB, 2011) of beer produced annually. Using this figure and the relative energy savings for the three alternatives shown above, the following table illustrates the potential energy savings across all breweries in South Africa.

Table 31: National savings for the three alternatives

<table>
<thead>
<tr>
<th>Energy savings</th>
<th>VC</th>
<th>MVR</th>
<th>TVR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wort boilers</td>
<td>kWh/hl</td>
<td>1.50</td>
<td>1.74</td>
</tr>
<tr>
<td>Total savings</td>
<td>GWh</td>
<td>47.06</td>
<td>54.57</td>
</tr>
</tbody>
</table>

The following table is an extract from the IRP2010 showing assumed energy efficiency demand side management figures.

Table 32: Extract from IRP2010 - Energy Efficiency Demand Side Management
Source: IRP (2010)

<table>
<thead>
<tr>
<th>Year</th>
<th>Energy (GWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Heat pumps</td>
</tr>
<tr>
<td>2010</td>
<td>14</td>
</tr>
<tr>
<td>2011</td>
<td>142</td>
</tr>
<tr>
<td>2012</td>
<td>445</td>
</tr>
<tr>
<td>2013</td>
<td>1,137</td>
</tr>
<tr>
<td>2014</td>
<td>1,866</td>
</tr>
<tr>
<td>2015</td>
<td>2,104</td>
</tr>
<tr>
<td>2016</td>
<td>2,341</td>
</tr>
<tr>
<td>2017</td>
<td>2,579</td>
</tr>
<tr>
<td>2018</td>
<td>2,579</td>
</tr>
<tr>
<td>2019</td>
<td>2,579</td>
</tr>
<tr>
<td>2020</td>
<td>2,579</td>
</tr>
</tbody>
</table>

If the mechanical vapour recompression heat pump savings potential was achieved across the country, this single sub-sector alone could represent 12.3% of the 2012 demand side management forecast for heat pumps and 1.2% of the total demand side management initiatives.
Appendix C: EINSTEIN Screenshots

Sample screenshot showing input data.
Sample screenshot showing intermediate results.

### Final energy consumption for thermal use (FET) by equipment

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Fuel type</th>
<th>MWh</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1  Electrode boiler</td>
<td>Electricity</td>
<td>42.0</td>
<td>84.0</td>
</tr>
<tr>
<td>2  WBS supply</td>
<td>Electricity</td>
<td>2.0</td>
<td>4.0</td>
</tr>
<tr>
<td>3  Cooling water</td>
<td>Electricity</td>
<td>44.2</td>
<td>88.4</td>
</tr>
<tr>
<td>4  Chiller</td>
<td>Electricity</td>
<td>10.0</td>
<td>20.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>57.2</td>
<td>100.0</td>
</tr>
</tbody>
</table>

### Useful supply heat (USH) by equipment

<table>
<thead>
<tr>
<th>Equipment</th>
<th>MWh</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1  Electrode boiler</td>
<td>93.4</td>
<td>100.0</td>
</tr>
<tr>
<td>2  WBS supply</td>
<td>6.6</td>
<td>10.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>100.0</td>
<td>100.0</td>
</tr>
</tbody>
</table>

**Notes:**
- OK costs for equipment Chiller could not be calculated.
- OK costs for equipment Cooling water could not be calculated.