A Knowledge-Based System for Low-Grade Waste Heat Recovery in the Process Industries

A thesis submitted by

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Abstract

The ever-increasing price of energy, combined with increasingly stringent legislation to reduce greenhouse gas emissions, is driving the UK process industries toward increasing energy efficiency. Significant gains can be made in this sector, as up to 11.4TWh per annum (4% of total energy use) of the UK process industries' energy consumption is lost as *recoverable* waste heat. Substantial recovery of this waste heat would have economic benefits of the order of £100s of million/year, and environmental benefits of the order of 100s of thousands of tonnes of carbon dioxide equivalent per year.

This thesis describes the development of a knowledge-based system for the selection and preliminary design of equipment for low-grade waste heat recovery in the process industries. The system addresses two of the key barriers to low-grade waste heat recovery in the UK. Firstly, it provides a readily accessible and zero cost tool to replace expensive, time-consuming expert consultancy in the initial stages of waste heat recovery projects, and, secondly, it educates users regarding the range and benefits of novel waste heat recovery technologies.

The system requires an input of easy-to-access data from the user. Based on this data, it then selects the most appropriate technologies for waste heat recovery for the case study in question from a database including various types of heat exchanger, vapour compression heat pumps, mechanical vapour recompression and organic Rankine cycles. It also generates a preliminary design including equipment size, efficiency/effectiveness, capital cost, cost savings, payback time and potential reductions in carbon emissions. This provides sufficient information to allow the user to make an educated decision regarding whether or not waste heat recovery is suitable for their needs.

The knowledge-base of the system was built using a decision tree method that has been proven to be successful in the building of decision making tools for various engineering applications. The software is programmed using the Java language

which allows widespread free dissemination to computers running all common operating systems.

The system was tested using case studies based on data from both existing publications and collaborating companies. The results were validated against published results, common modelling software results and the views of expert consultants. Broadly, in terms of equipment specification and cost, the knowledge-based system produced the same results as the other methods. Furthermore, the preliminary designs generated were generally within 5% of the final figures from the other sources.

In certain cases, the knowledge-based system suggested alternative technologies that were more viable (economically and/or practically) than those considered by the authors of published case studies. In all cases, system operating time (data input, and processing of results) was of the order of minutes, whereas studies by consultants or the use of existing modelling packages would be significantly more time-consuming (of the order of hours or days). Hence, the system can be used as a rapid optioneering tool for investigation of waste heat recovery technologies, requiring substantially less time than current available methods.

Keywords: Knowledge-Based System; Low-Grade Waste Heat; Waste Heat Recovery; Equipment Selection Methodology; Heat Exchanger; Heat Pump; Organic Rankine Cycle; Process Industries

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Nomenclature and Abbreviations

Symbols

μ	cp	Viscosity
A	(m^2)	Surface area
C	(£/kWh)	Cost factor
Cp	$(kW.kg^{-1}K^{-1})$	Specific heat capacity
E	$(tCO_2eq./kWh)$	Emission factor
F_T		Heat exchanger correction factor
h	Hr	Hours of operation
h	$(kW.m^{-1}.K^{-1})$	Film heat transfer coefficient
H	kJ/kg	Specific enthalpy
j_f		Friction factor
k	$(kW.m^{-1}.K^{-1})$	Thermal conductivity
m	(kg/s)	Mass flow rate
N	kWh	Number of units (of electricity)
n		Number of stages (compressor)
P	kW	Power generated
P	bar	Pressure
Pwo	bar	Partial pressure of water vapour
Q	(kW)	Heating duty
Q_{max}	kW	Theoretical maximum heating duty
Re		Reynold's number
R_f	$(K.m^2.kW^{-1})$	Heat transfer resistance due to fouling
T	$(K, {}^oC)$	Temperature
U	$(kW.m^{-2}.K^{-1})$	Overall heat transfer coefficient
W	(kW)	Work (of compressor, for example)
x_w	(m)	Heat exchanger wall thickness
γ		Heat capacity ratio
ΔH_{EVAP}	$(kW.kg^{-1})$	Latent heat of evaporation

ΔP	bar	Pressure difference
ΔT	(K, ^oC)	Temperature change
η		Efficiency

Subscripts

AM	Arithmetic mean temperature difference
carrier	Working fluid (in run-around-coil type systems)
comp	Compressor
cond	Condenser
current	Current heating method (of sink); Current heating utility
dew	Dew point
drive	Motor
elec	Electricity (cost or emissions factor)
evap	Evaporator
isen	Isentropic process
levy	Cost saving of government levy
LM	Log meant temperature difference
min,GHG	Minimum COP for a heat pump project to show net savings in greenhouse gas emissions
min,profit	Minimum COP for a heat pump project to be profitable
plate	Plate heat exchanger
ref	Refrigerant/working fluid
sat	(partial pressure) at saturation temperature
turb	Turbine

Abbreviations

ASHRAE	American society of heating, refrigerating and air conditioning engineers
CFC	Chloro-fluoro-carbon

CHP	Combined heat and power
COP	Coefficient of performance
DECC	Department for energy and climate change (UK)
GG	Gas-gas
GHG	Greenhouse gas
GL	Gas-liquid
GUI	Graphical user interface
GWP	Global warming potential
HC	Hydro-carbon
HCFC	Hydro-chloro-fluoro-carbon
HFC	Hydro-fluoro-carbon
HP	Heat pump
IEA	International energy agency
IEA	International energy agency
KBS	Knowledge-based system
LL	Liquid-liquid
LMTD	Log meant temperature difference
MED	Multiple-effect desalination
MVR	Mechanical vapour recompression
n/c	Non-condensable
ODP	Ozone depletion potential
ORC	Organic Rankine cycle
OS	Operating system
PB	Payback time
PCHE	Printed circuit heat exchanger
R&D	Research and development
RO	Reverse osmosis
SRC	Steam Rankine cycle
tCO_2eq	Tonnes of carbon dioxide equivalent
TEMA	Tubular heat exchangers manufacturers association

TOPSIS	Technique for order of preference by similarity to ideal solution
TVR	Thermal vapour recompression
VC	Vapour compression
WHR	Waste heat recovery
WOCA	Write once compile anywhere
WORA	Write once run anywhere
WORE	Write once run everywhere

Chapter 1

This chapter covers the motivation for this work via a discussion of UK industrial energy use and the vast potential for low-grade waste heat recovery in the UK process industries. The project aims and objectives are also given, along with a description of the structure of this thesis.

1. Introduction

1.1. UK Industrial Energy Use

Energy use in UK industry is becoming increasingly scrutinized for a variety of reasons. Firstly, the rising cost of both electricity and fossil fuel resources (as depicted in Figure 1.1) is leading to ever-increasing utility expenditure which can be a severe constraint in the current uncertain financial climate.

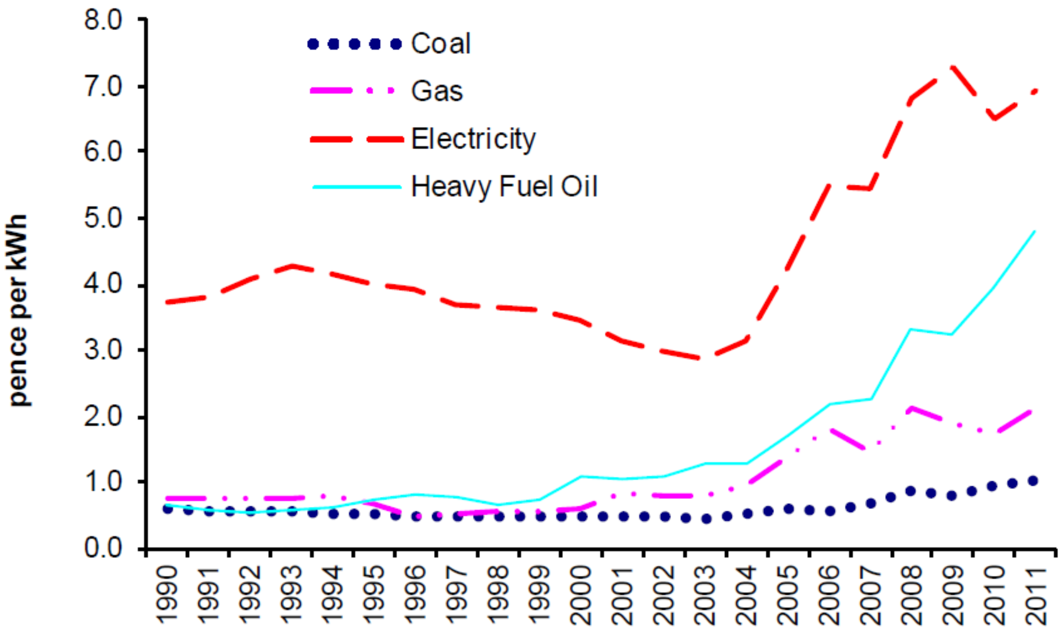

Figure 1.1. Utility prices for the process industries (DECC (a), 2013)

Figure 1.1 shows that the price of industrial utilities have sharply increased in the last ten years, with electricity and natural gas prices more than doubling in the period 2002 to 2011. Hence, an ever-increasing monetary incentive exists for reducing utility consumption.

Secondly, self-imposed government legislation set out in the Climate Change Act of 2008 requires the reduction of greenhouse gas emissions to achieve ambitious targets of a 34% reduction by 2020, and 80% by 2050 (based on 1990 levels). Figure 1.2 below shows trends in greenhouse gas emissions across the whole of the UK from 1900 to 2010 (with predictions to 2050), while Figure 1.3 shows the contribution of the process industries to the overall emissions.

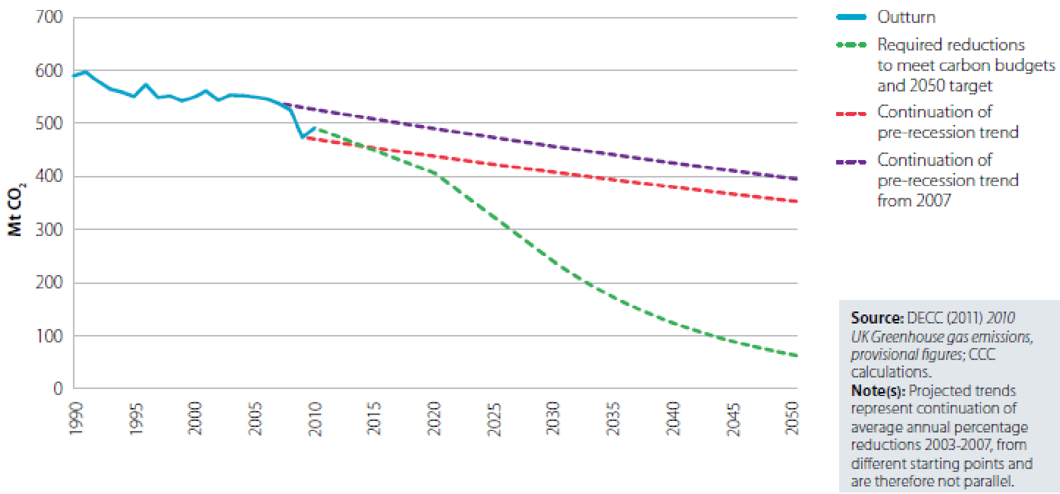

Figure 1.2. UK greenhouse gas emissions from 1990 to 2010 (& predictions to 2050) (Committee on Climate Change, 2011)

Figure 1.2 shows that the UK is not on course to achieve climate change targets, according to both predicted trend lines. Hence, wholesale changes are required across all sectors in order to make a step change in greenhouse gas emissions in line with the targets set out in the Climate Change Act.

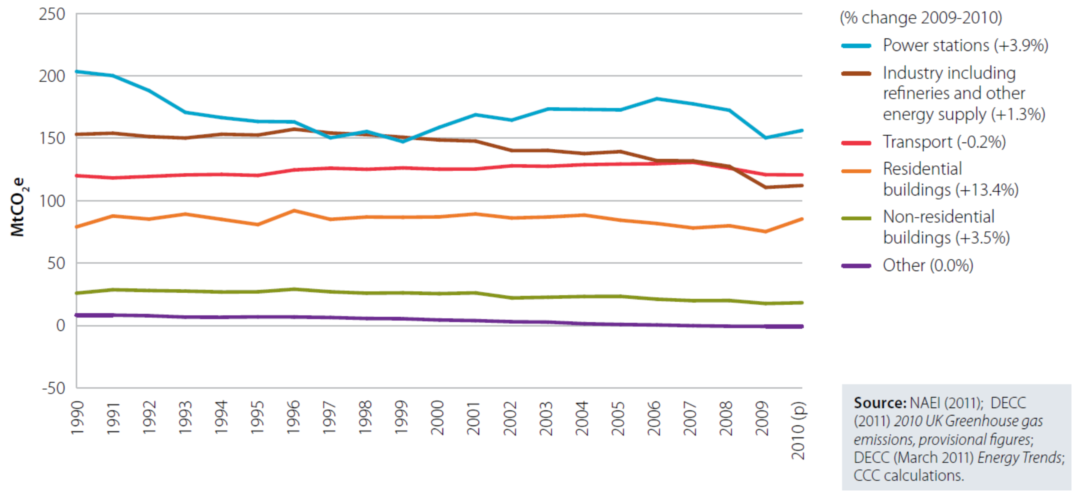

Figure 1.3. UK carbon emissions by sector, 1990-2010 (Committee on Climate Change, 2011)

Figure 1.3 shows that the industrial sector contributes approximately 20% to the overall greenhouse gas emissions in the UK, the third biggest contributor (behind power stations and transport). Hence, this sector is expected to come under increased scrutiny in the coming years. Note: emissions in this sector have dropped by almost a third since 1990; the explanation for this lies in the decline of heavy industry (by up to 60% in the years 1980-2010 (The Guardian, 2011)) rather than wholesale changes in industrial energy efficiency.

Other drivers for reducing energy consumption include exhaustion of fossil fuel resources, corporate sustainability drives and pollution reduction (both heat and gaseous pollution).

One way of increasing industrial energy efficiency and decreasing utility consumption is via recovery of waste heat, and this topic is investigated in this thesis (particularly low-grade waste heat which is considered the most difficult to recover).

1.2. Potential for Industrial Low-Grade Waste Heat Recovery

Low-grade waste heat is defined as any process stream (liquid or gaseous) currently emitted to the environment at a temperature below 260°C (The Watt Committee, 1994).

Low-grade waste heat recovery has great potential for reducing industrial energy consumption, and increasing energy efficiency. Reay and Morrell (2007) studied the potential for waste heat recovery in the process industries and found that an estimated 11.4TWh of *recoverable* waste heat is currently emitted to the environment across all sectors. This data is broken down by sector in Figure 1.4 below. Note: the term *recoverable* here refers to waste heat which may be recovered using current technologies including direct re-use of heat, heat transfer via heat exchanger, heat pumps (open and closed cycle) and power cycles.

A study of the same topic by McKenna and Norman (2010) using a spatial modelling technique found the process industry waste heat recovery potential to be 14.4TWh, a reasonably good agreement with the data by Reay and Morell (2007).

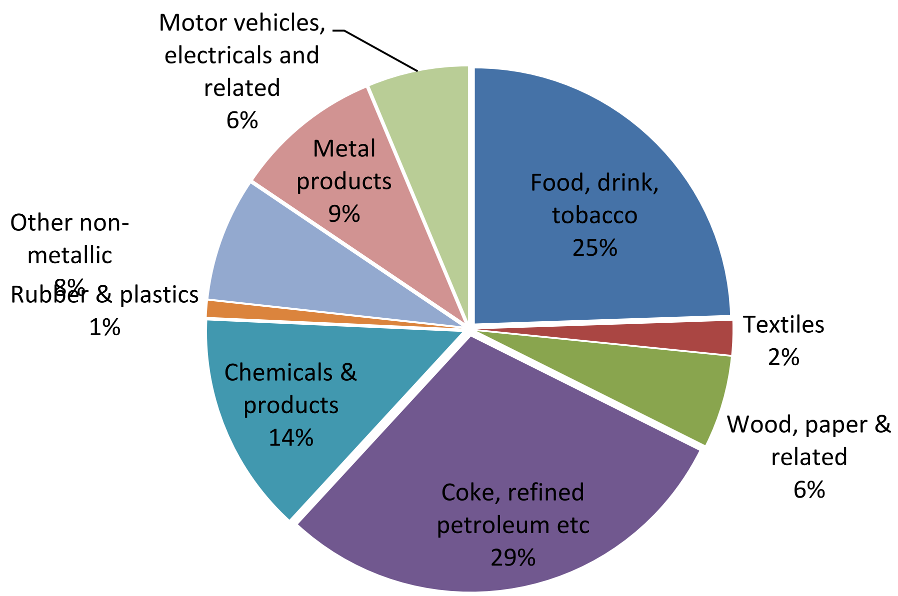

Figure 1.4. Waste heat recovery potential by sector (adapted from Reay and Morrell, 2007)

Figure 1.4 shows that the sectors with the largest proportion of recoverable waste heat are "coke and refined petroleum *etc*" (29%), "food, drink and tobacco" (25%), "chemicals and products" (14%) and "metal products" (8%).

Figure 1.5 shows the breakdown of energy usage across all of the sectors studied by Reay and Morell. Note that data from the petroleum industry ("coke and refined petroleum *etc*") is excluded from this data-set as it was unobtainable. The figure shows that the majority of the energy usage is in "low temperature processes" which are defined in the study as less than 300^{o}C. Hence, it can be inferred that the majority of waste heat available across these sectors will be low-grade. Furthermore, drying/separation processes typically operate below 200^{o}C, motors emit low-grade heat from cooling circuits and refrigeration condensers emit low-grade heat, hence 66% of the energy use has the potential to emit low-grade waste heat. Therefore, it can be concluded that the *majority* of the 11.4TWh of *recoverable* waste heat is of low-grade.

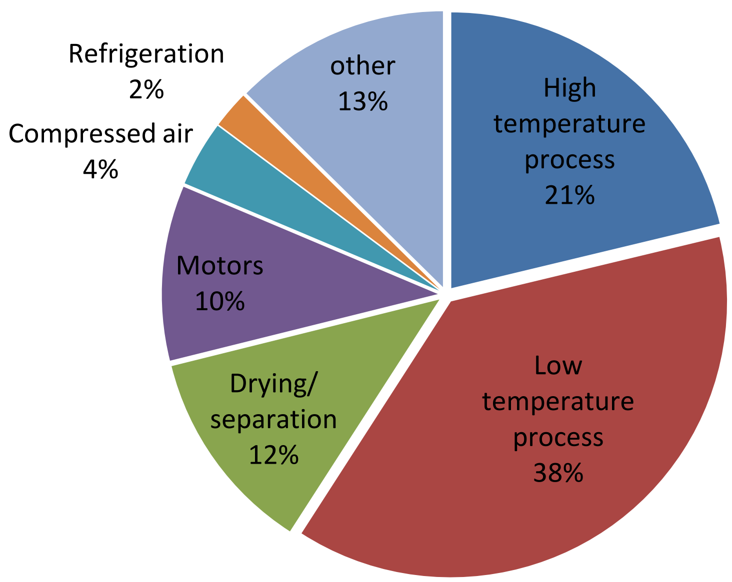

Figure 1.5. Energy consumption by process (adapted from Reay and Morrell, 2007)

1.2.1. Typical Low-Grade Heat Sources and Sinks

Low-grade waste heat is emitted from a variety of processes via both liquid and gaseous effluents. Table 1.1 below (David Reay and Associates, 1994) shows a list of generic heat sources that may exist across all sectors of the process industries.

Table 1.1. Generic heat sources (David Reay and Associates, 1994)

Source of heat	Nature of heat source		
	Gas	**Liquid**	**Vapour**
Air Compressor	X	X	
Boiler	X	X	X
Distillation		X	X
Drying	X		X
Evaporation	X	X	
Furnaces	X	X	
Gas turbine	X	X	
Kilns	X		X
Ovens	X		X
Pasteurisers	X	X	
Prime movers	X	X	X
Refrigeration	X	X	
Sterilisation	X	X	
Ventilation	X		X
Washing	X	X	X

Uses, or sinks, for the waste heat also vary considerably. The easiest and most economical method of recovery is by heat transfer from the heat source to a heat sink of suitably lower temperature, facilitated by a heat exchanger (or direct re-use, where possible). However, with low-grade waste heat recovery, it is often difficult to “*match”* the heat source to a lower temperature heat sink. Hence, often novel methods of recovery must be considered. For example, vapour compression heat pumps can be used to upgrade the waste heat to heat a sink of higher temperature, absorption heat pumps may be used to convert low-grade waste heat into coolth and power cycles may be used to convert the waste heat into electricity (these technologies are discussed further in Chapter 3). Therefore, significant expertise is required to analyse all available uses for low-grade waste heat recovery.

1.2.2. Potential Benefits of Waste Heat Recovery

DECC, 2012 (b), state that the total energy consumption by the process industries in 2012 was 290TWh. Hence, if all of the theoretically *recoverable* waste heat (11.4TWh) was to be recovered, this would represent a total energy saving of up to 4% (depending on the methods of recovery for each individual case study). In order to quantify the economic and environmental effect of this, three theoretical scenarios can be envisaged (originally discussed with regards to the food industry in Law *et al*, 2012):

1. All of the available waste heat is recovered via heat exchangers to replace a current gas heating duty of 75% efficiency
2. All of the available waste heat is recovered by heat pumps with a COP of 3.5 (generally the minimum target of a well-designed system) to replace a current gas heating duty of 75% efficiency
3. All of the available waste heat is recovered by organic Rankine cycle machines with a thermal efficiency of 12% (generally the minimum target of a well-designed system)

Calculations based on heat balances, current utility cost factors (DECC, 2013 (b)) and greenhouse gas emission factors (Carbon Trust, 2012) create the economic and environmental benefits shown in Table 1.2.

Table 1.2. Potential economic and environmental benefits of waste heat recovery in the UK process industries

	Potential Cost Saving (£million/year)	Potential Greenhouse Gas Emission Reductions (million $tCO_2eq./year$)
Scenario 1	459	2.79
Scenario 2	108	0.510
Scenario 3	111	0.718

Table 1.2 shows that, under the best scenario of heat transfer via a heat exchanger between source and sink, low-grade waste heat recovery in the UK process industries has the potential to save £459m and 2.79 $MtCO_2eq$ per annum. Under the worst case scenario (heat pump heat recovery), the cost saving is £108m per annum and the greenhouse gas reductions 0.510 million tCO_2eq per annum. Hence, waste heat recovery has huge potential economic and environmental benefits to the UK process industries.

In reality, a combination of the three solutions would be required in order to recover all of the waste heat available, as well as other solutions such as absorption heat pumps (for cooling) and mechanical vapour recompression (all discussed in Chapter 3 of this thesis). Furthermore, the scenario assumptions will vary between case studies. Hence, the scenarios listed above (and a combination of the three) are not definitive. However, the data shows that the potential cost savings are in the region of hundreds of millions of pounds per annum, and the potential greenhouse gas reductions are in the region of (at least) hundreds of thousands of tonnes of CO_2eq per annum.

1.2.3. Barriers to Low-Grade Waste Heat Recovery

Low-grade waste heat is traditionally the most difficult to recover due to difficulties in finding *"matching"* (i.e. lower temperature) heat sinks. Hence, the traditional method of heat exchanger waste heat recovery, as featured in common heat integration methods such as Pinch technology (Linnhoff and Hindmarsh, 1983), is often not possible. This leads to the need for more novel and complex solutions such as heat pumps and power cycles.

This creates a barrier, as there is a substantial knowledge-gap with regards to these more specialised waste heat recovery technologies in the UK process industries.

This was highlighted by Sinclair (2001). Here, an industrial survey was undertaken to gauge the attitude of the process industries with regards to heat exchangers and heat pumps for waste heat recovery. As shown in Figure 1.6 below, the attitude towards heat exchangers and heat pumps varied significantly. While 87% of the study "support" heat exchangers, only 66% "support" heat pumps. Also, a combined 34% of the study are "unsure" about heat pumps or consider them to be a "risky" investment. This highlights a general gap in knowledge with regards to the more novel waste heat recovery methods, as heat pumps have been proven to be a sound economic investment on numerous occasions (for example Department of Energy, 1981, and Star Refrigeration, 2013, and see Section 3.2 for a full discussion of heat pumps for waste heat recovery).

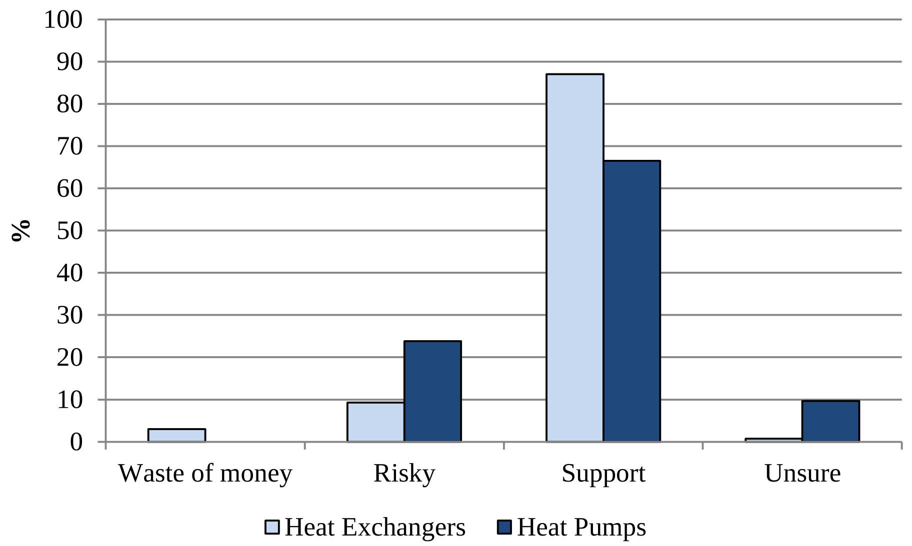

Figure 1.6. Process industry attitudes to waste heat recovery equipment (adapted from Sinclair, 2001)

This lack of knowledge, and the fact that industrial engineers often do not have time to investigate all waste heat recovery options, results in the need for expert consultancy from the initial stages of waste heat recovery projects. This is highlighted by the Good Practice Guide to Waste Heat Recovery (David Reay and Associates, 1994) where consultants are noted as “ideal for a preliminary assessment of the feasibility of the installation” and as having “good knowledge of equipment required”. Such consultancy can cost a great deal with no guarantees of positive results. Hence, this can be detrimental to the uptake of waste heat recovery projects.

1.3. Project Aims and Objectives

The project aim is to create a knowledge-based system (KBS) to encourage the uptake of waste heat recovery projects in the process industries by addressing the barriers to waste heat recovery discussed above. The system is intended to act as a consultancy tool for use in the initial stages of waste heat recovery system design. Hence, it must be able to select and design various types of equipment for use in waste heat recovery, and provide the user with sufficient data for a decision to be made regarding whether waste heat recovery is suitable at their plant. The system must have an educational element and present reasons for the decisions to the user.

Therefore, the system is to address two of the key barriers as follows:

i. Cost of consultancy: KBS provides a free alternative to outside consultancy during the initial stages of low-grade waste-heat recovery projects.
ii. Awareness of best-available/novel technologies: they are highlighted when suitable.

The system is also expected to be significantly faster than the use of traditional modelling techniques and/or expert consultants (of the order of minutes for the

KBS, as opposed to days for consultancies). Hence, the system may also be used as a rapid optioneering tool for screening of waste heat recovery technologies.

1.3.1. Scope of knowledge-based system

The scope of the system is: to select and design the most appropriate process waste heat recovery technologies for individual case studies. The target end-users are industrial engineers who are assumed to have the following characteristics:

- Limited knowledge of waste heat recovery techniques
- No previous experience of process waste heat recovery projects
- Limited time to investigate all waste heat recovery options
- Education in heat transfer engineering to undergraduate degree standard
- Interested in waste heat recovery and therefore aware of other useful tools. For example, energy auditing tools such as EINSTEIN (Brunner, 2010) which may be used prior to this system to identify potential waste heat sources and sinks

Therefore, the scope of the system is to achieve the objective stated above whilst accommodating the needs of the end user. This creates the following set of design constraints:

1. Must be simple and intuitive to use: to aid users with no previous experience of process waste heat recovery
2. Must make use of easy-to-access data: this will aid users with limited time in the collection of data for use in the software
3. Must explain selection/design logic to the user: this will educate the user in the methods employed by the system thereby reducing/avoiding user confusion or mistrust
4. Must allow easy dissemination into the industrial domain: different users are likely to run various operating systems (Apple OS, Linux, Windows etc) meaning the software must be multi-platform compatible
5. Must allow a comparison of various technologies: this will educate users as to the benefits of each type of technology (when appropriate)
6. Must give accurate results: results from this software must be comparable with other modelling tools (to be validated by case studies)
7. Must include a variety of waste heat recovery techniques: this will allow a wide range of possible process conditions to be accommodated

8. Must include technologically viable results: results must be meaningful on an industrial scale. Technologies requiring significant further R&D should not be included
9. Must include economically viable results: only technologies which have been proven to achieve economically viable results will be considered. Technologies incurring typical pay back times of greater than 5 years (under economic conditions at the time of writing) will be considered non-economical

A number of related methods/objectives were excluded from the scope of the system. They are as follows.

- Energy audit analysis. This was excluded as software is already available for such tasks, for example the EINSTEIN energy auditing tool (Brunner, 2010) as discussed in Section 2.3.
- Pinch analysis. This was excluded as such methods are well established, as discussed in Section 2.21. Software is readily available for such tasks such as the Aspen Energy Analyzer tool for Aspen HYSYS/Aspen Plus (Aspen Tech, 2010) and HERO (Kemp, 2007).
- Integration of renewables and/or combined heat and power systems (CHP). This was excluded as the objective of this work is to aid waste heat recovery only. Furthermore, software is available for such tasks such as, again, the EINSTEIN energy auditing tool (Brunner, 2010).

1.4. Structure of the Thesis

This thesis is presented in seven chapters, each of which considers a major aspect of the work and the development of the knowledge-based system for low-grade waste heat recovery.

The first chapter provides an introduction to the work, listing the drivers and barriers to low-grade waste heat recovery in the UK process industries, which in turn leads to the motivation for the work and the scope of the knowledge-based system.

Chapter two presents a review of literature relevant to the work completed in this thesis. This includes literature regarding tools and methods for the design of waste heat recovery systems and knowledge-based/expert systems written for process industry applications.

Chapter three covers a discussion of the state-of-the-art in waste heat recovery technology and the selection of suitable equipment for inclusion in the knowledge-based system. The advantages and disadvantages of each technology are analysed and reasons provided for the inclusion/exclusion in the system knowledge-base.

Chapter four discusses the logic, decision pathways and design equations of the knowledge-based system. Justification of the methods employed and schematics of the system logic are presented.

Chapter five introduces the programming and compilation of the knowledge-based system using a suitable computer language. The choice of programming language is discussed, along with screenshots of the graphical user interface produced.

Chapter six covers the testing of the system via case studies which are provided from both published literature and original data from industrial partners. The results of the knowledge-based system are compared to both published data and those of industry standard modelling tools in order to judge the validity and accuracy of the data.

Finally, in chapter seven, conclusions are drawn based on the findings and recommendations for future work are presented.

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Chapter 2

This chapter reviews current literature relevant to the work completed in this thesis. This includes literature regarding tools and methods for the design of waste heat recovery systems (including heat exchangers and thermodynamic cycles) and knowledge-based/expert systems written for process industry applications.

2. Literature Review

2.1. State of the Art in Waste Heat Recovery Techniques

The state of the art in waste heat recovery techniques is covered in Chapter 3 where a review of current literature, analysis of equipment data sheets and accompanying calculations is provided to justify the inclusion/exclusion of each technology in the software knowledge base.

2.2. Tools and Methods for Waste Heat Recovery System Design

Various tools and methods are available to aid the design of waste heat recovery systems and heat exchanger networks. They range from pinch technology methods to methods for exploring optimum designs of heat exchangers and thermodynamic cycles.

2.2.1. Pinch Technology

Heat integration was originally suggested by Linnhoff *et al* (1979) using composite curves to create an overall view of the heat demand of a process and highlight opportunities for heat recovery between hot and cold streams (heat sources and sinks). The method was later extended to include the concept of the pinch (Linnhoff and Hindmarsh, 1983), allowing energy targets to be realised in practice according to a number of rules. This has led to heat integration being commonly referred to as “Pinch Technology”.

The method requires the generation of hot and cold composite curves and the determination of a pinch point to allow realistic (with regards to size and cost of heat exchangers) recovery of heat between sources and sinks. A typical composite curve, showing the pinch point, heating target and cooling target is shown below in Figure 2.1 (Smith, 2000).

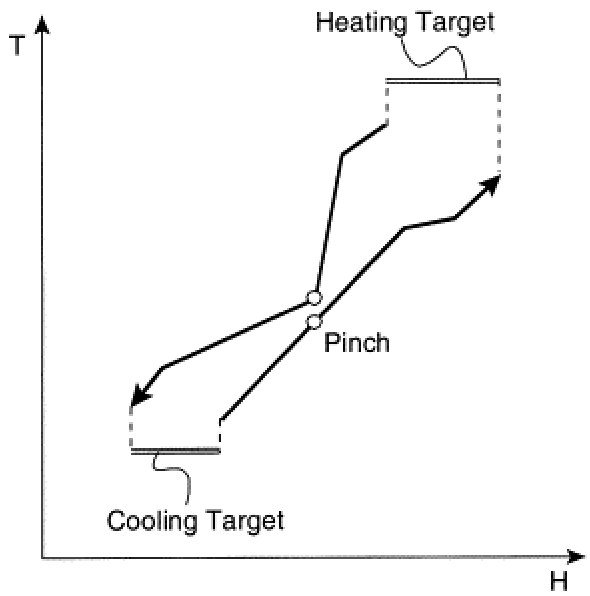


Figure 2.1. Composite curves and pinch point (Smith, 2000)

The composite curves are used to *match* heat sources and sinks according to certain rules employed by the method such as “no heat transfer across the pinch” (*i.e.* streams on right hand side of pinch may not be matched with streams on left hand side, and vice-versa). The external heating and cooling duties required are then shown by the overlap at the extremes of the plot.

This method can be used for individual processes (for example in multi-product distillation where multiple heat sources and sinks exist) or can be extended for site-wide heat integration. The matching of heat sources and sinks is decided via mathematical methods. For example, graph theory was originally used by Linhoff *et al* (1979) to devise the minimise the amount of heat exchangers required in the network, thereby finding the optimum source-sink matches to minimise the capital expenditure of heat integration. More recently, complex algorithms, such as artificial neural networks (Smith *et al*, 2010) and the genetic algorithm (Ravagnani *et al*, 2005), have been used to find the best matching sources and sinks. Such methods are advantageous as they can take into account more variables, such as distance between source/sink, capital expenditure, cost savings and energy/exergy analysis. However, the complexity of the mathematics involved may impede clear explanation of these techniques, making it difficult to gain the trust of engineers in the industrial domain.

In current practice, a grid diagram of matched sources and sinks is typically produced to show the suggested retrofitted network, as per the example shown in Figure 2.2 below (Smith *et al*, 2010). In Figure 2.2, the red horizontal lines represent heat sources (numbered 1-8), the blue horizontal lines represent heat sinks (numbered 11-13), the black circles attached by black vertical lines represent *new* heat exchangers for waste heat recovery, and the blue and red circles represent existing heat exchangers for heating/cooling utilities.

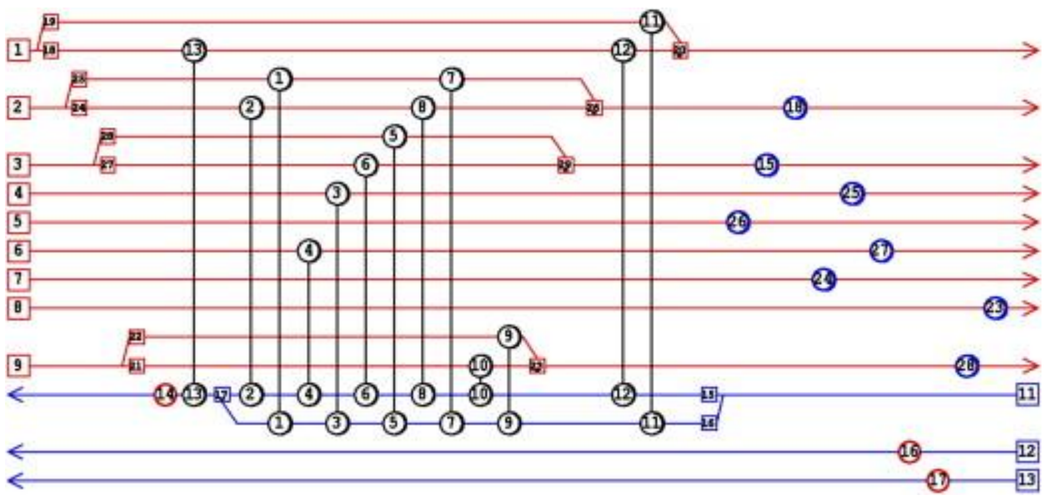


Figure 2.2. Example of a heat exchanger network map (Smith *et al*, 2010)

The original pinch method was designed for relatively simple continuous, single process situations, but has since been modified to consider batch processing (*e.g*. Kemp, 1990) site-wide integration (Varbanov *et al*, 2012), "*cross-border*" integration (between processing sites or district heating networks, Kapil *et al*, 2012) and integration of more complex waste heat recovery methods, such as heat pumps (Benstead & Sharman, 1990).

Other techniques have been devised from the original heat integration methodology to cover efficient use of other plant commodities such as water (Wang and Smith, 1994) and integration of renewable energy (Muster-Slawitsch *et al*, 2011). This is not further reviewed here, as it is outside the scope of this thesis.

Kemp, 1990, investigated the use of pinch heat integration methods in batch processing. Here, the processes were divided into time intervals allowing calculations for heat integration targets at specific time periods. This highlights possibilities for optimal heat exchanger networks, heat storage and batch rescheduling, as well as displaying the time dependence of the utility demand. This concept was further developed by Adonyi *et al*, 2003, where an S-graph approach was taken to consider both heat integration and batch scheduling simultaneously rather than consecutively. This approach was shown to improve the optimum solution compared to the original methodology. However, complex mathematical algorithms (combinatorics and combinatorial algorithms) were required. Therefore, it is less likely that such a method would be accepted so readily into the industrial domain due to concerns with explanation/trust with local engineers. Furthermore, the resolution of batch scheduling may be a concern. If this is too small (*i.e.* second or minute scale) it is likely that the results will be too difficult to practically implement.

Benstead & Sharman, 1990, considered heat pumps in process heat integration to allow matching of heat sources to heat sinks of greater temperature. Standard pinch technology methods were combined with heat pump thermodynamic models to identify opportunities for heat recovery incorporating the possible temperature lifts between sources and sinks in the grand composite curve. The method was shown to be successful in producing economic optimum heat exchanger/heat pump networks, although greater computer processing power is required (compared to standard heat exchanger network techniques). However, the work has not led to any significant uptake of industrial heat pump installations in the UK since publication 23 years ago and an apparent scepticism in industry remains (as discussed in Section 1.2.3).

Varbanov *et al*, 2012, investigated the concept of site-wide process heat integration. A new method was suggested based on carrying the minimum approach temperature (ΔT_{min}, pinch point) for each process (or major unit operation). This method was compared to the traditional method of one constant minimum approach temperature (simply an extension of traditional pinch technology to a greater number of sources/sinks). The new method was found to achieve a 30% greater decrease in cold utility requirement and an 18% greater decrease in hot utility requirement, although only one case study was presented. However, drawbacks exist, such as the greater complexity of the method and the need for greater computational power. In a discussion of future work, the authors stated that such a method could unlock potential for pinch methods to replace shell-and-tube heat exchangers with more efficient options, such as plate heat exchangers. In this respect, the work presented in this thesis may be complimentary, and the methods created could be used in conjunction with the work of Varbanov to help identify the situations in which one could use a variety of different heat exchangers. This in turn could lead to the selection of a suitable ΔT_{min} based on which heat exchanger options are compatible with the source/sink in question.

Kapil *et al* (2012) used an extension of pinch technology to study the feasibility of integrating process waste heat with district heating networks. It was shown that the concept was feasible and that waste heat (at a temperature of greater than 105^{o}C) could be economically transported to a district heating hub up to 86.5km away. The system also significantly decreased the cost of thermal energy from the district heating system and negated the need for a second CHP boiler during normal operation. However, the methods presented were based on the assumption of an existing district heating network being in place. As such schemes are scarce in the

UK, the results presented here are not immediately relevant. Firstly, district heating facilities would have to be widely installed in the UK, but they are currently hindered by a number of economic, commercial and infrastructural barriers (as discussed by Davies and Woods, 2009).

Numerous commercial software tools have been developed in an attempt to ease the dissemination of pinch methods into the industrial domain and increase industrial energy efficiency. Tools are available with different levels of complexity and cost. For example, Hero by Chepro ltd (Chepro ltd, 1994) was developed to produce only heating profiles, energy targets and the grand composite curves, without including heat exchanger network design (Kemp, 2007). The main drawback of such tools is that a certain degree of user expertise is still required in order to design the subsequent heat exchanger network.

More sophisticated methods include Aspen Energy Optimisation (Aspentech, 2013), which is used in conjunction with the Aspen HYSYS (Aspentech [2], 2013) process modelling suite to optimise heat exchanger networks according to user-defined energy targets. This tool is very comprehensive and accurate economic analysis can also be performed (capital costs, cost savings, payback time *etc*). However, there are also several drawbacks. One drawback is that the software does not take into account heat source/sink nature and bases all heat exchanger networks (and resulting energy/economic analysis) on shell-and-tube heat exchangers, which are not always the most suitable (or appropriate design). Secondly, this software is very expensive to purchase and expert training is required. This would represent a barrier to many sectors of the process industries. Finally, this software cannot consider heat pumps or other novel heat recovery techniques (unless the user builds such systems into the flow-sheet themselves, which requires significant expertise).

Another example of commercial software tools for this purpose is the recent "Expert System for an Intelligent Supply of Thermal Energy in Industry and other Large-Scale Applications", or EINSTEIN (Brunner *et al*, 2010) which combines methods of pinch technology and renewable energy integration for process industry (and large building) applications. This tool is again useful in accurately analysing the benefits of heat integration (and renewable integration), but has the familiar drawback of not being able to select/design the specific heat exchanger or thermodynamic cycle most suitable for heat recovery between the sources and sinks.

Overall, pinch technology and its implementation has been very successful since the introduction by Linhoff *et al* (1979) and research is ongoing to improve and extend the methods as computational methods/power improves.

However, one key drawback of all the pinch technology methods discussed is the lack of selection/design of equipment. Pinch methods are still based upon the use of shell-and-tube heat exchangers which are not the best design for many heat transfer duties. Furthermore, while methods incorporating novel recovery methods such as heat pumps do not provide a substantial cycle design. Therefore these methods would be complimentary to the knowledge-based system (KBS) developed in this work *i.e*. the pinch methods could be used to *identify* opportunities for heat exchanger/heat pump waste heat recovery, whilst the KBS could be used for specific heat exchanger/heat pump selection and design based on each individual case study. The extremes of the hot composite curve (*i.e.* where no sink can be matched) could also be assessed for utilisation in an organic Rankine cycle, for example.

2.2.2. Heat Exchanger Selection/Design

Heat exchanger selection and design has been studied for many years. A number of novel heat exchangers have been developed for various duties to replace the shell-and-tube heat exchanger, which is often described as the “work horse of the industry” (Klemes *et al*, 2008). However, many of these works focus more on optimization and design of specific heat exchangers, rather than advising the user on when to select which units. As a result, the shell and tube heat exchanger is by far the most common in the process industries although other types have found certain niche applications, such as the gasketted plate heat exchanger in the food industry (due to ease of cleaning) and the printed circuit heat exchanger in offshore oil and gas facilities (due to compact size and weight). A full discussion of the merits of each type of heat exchanger is found in Section 3.1, but the focus in this section is the literature underpinning the selection and design of heat exchangers.

The Best Practice Programme (2001) provided a comprehensive list of heat exchangers and their operating limits. This data is extremely useful in gaining an understanding of when one must not use a certain unit. For example, the report states that gasketted plate heat exchangers have a temperature limit of 175^{o}C (a figure confirmed by manufacturer data, see Section 3.1). Guidance is also offered on when one should select one unit rather than another, suggesting that installed cost should be used as the main indicator. However, this suggested method ignores other, often crucial, factors such as fouling limitations and ease of maintenance *etc*.

The problem of heat exchanger selection was addressed by Heppenstall and Halliday, 1990. Here, the authors attempted to develop an expert system for heat exchanger selection to try to negate the need for an expert contractor in heat recovery system design. The method in this work considered the following five key areas of consideration in heat exchanger selection:

- Environmental conditions: Factors such as the nature of the streams, e.g. gas or liquid, temperature, pressure *etc*.
- Material selection: Materials of construction must be compatible with heat source and sink.
- Fouling: The type of heat exchanger must be appropriate for the fouling potential of the source and sink.
- Effectiveness and pressure drop: The type of heat exchanger must meet reasonable effectiveness and pressure drop targets.
- Cost: The type of heat exchanger must not be of exuberant cost

The paper notes that an effective selection procedure must address each of the five areas stated above. The five key areas of consideration were combined in the “structure of the knowledge base”, as shown in Figure 2.3 below.

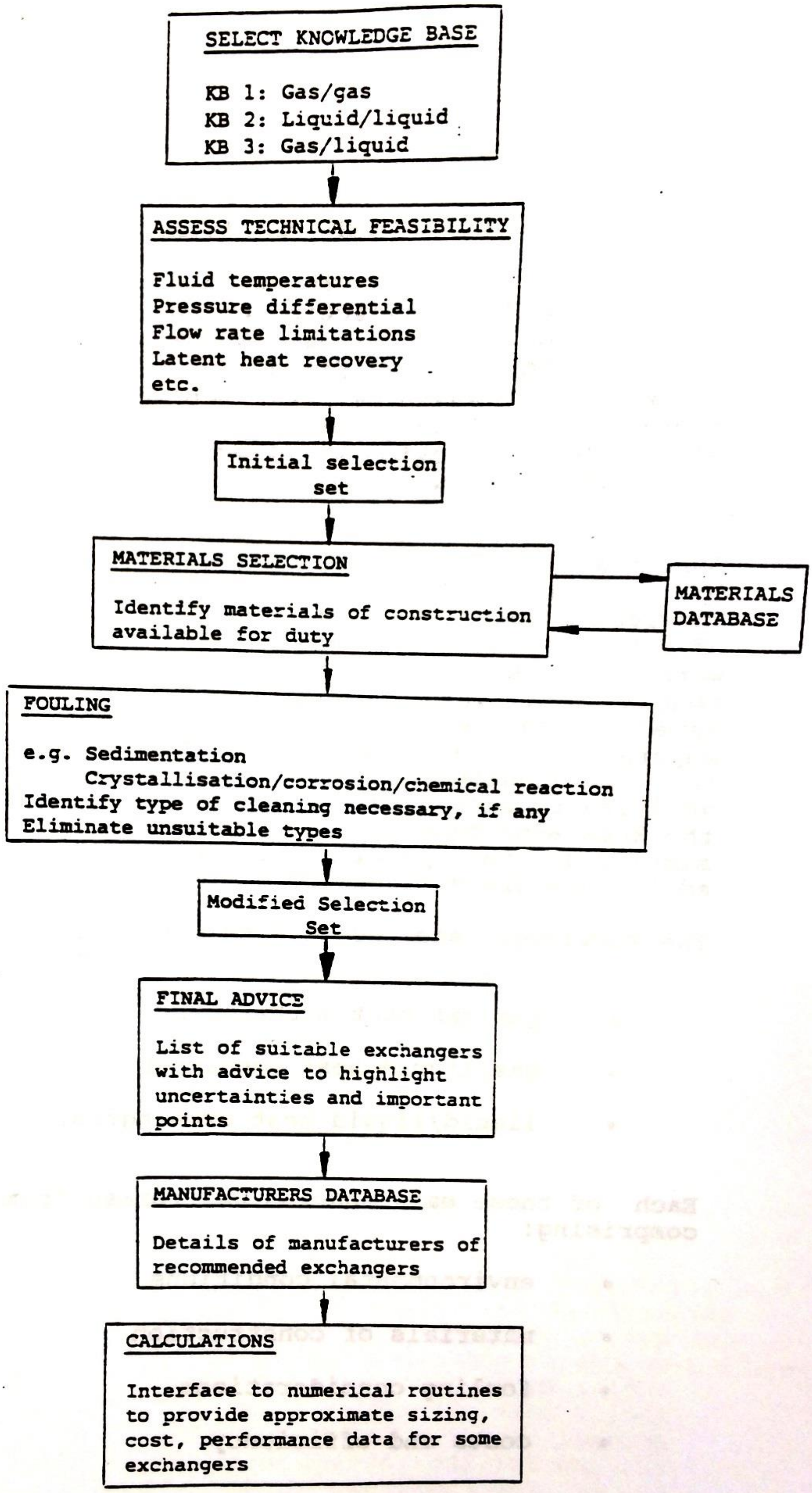


Figure 2.3. Overall structure of knowledge base of system by Heppenstall and Halliday, 1990

The authors sought clarification of their selection rules and procedures from numerous heat transfer "experts" in order to set limits for the selection of each type of unit (temperature, pressure, fouling *etc*) and to collect data such as typical materials of construction and pressure drop.

The overall structure of the knowledge base was then broken down into detailed sub-sections that select specific heat exchanger types based on various criteria and heat exchanger limitations. An example of this is shown in Figure 2.4 overleaf.

Note: Figure 2.4 has been reproduced by the author as an exact replica of the original version by Heppenstall and Halliday, 1990. This is due to the difficulty of photo-copying the original printed paper version with sufficient resolution for printing in this thesis.

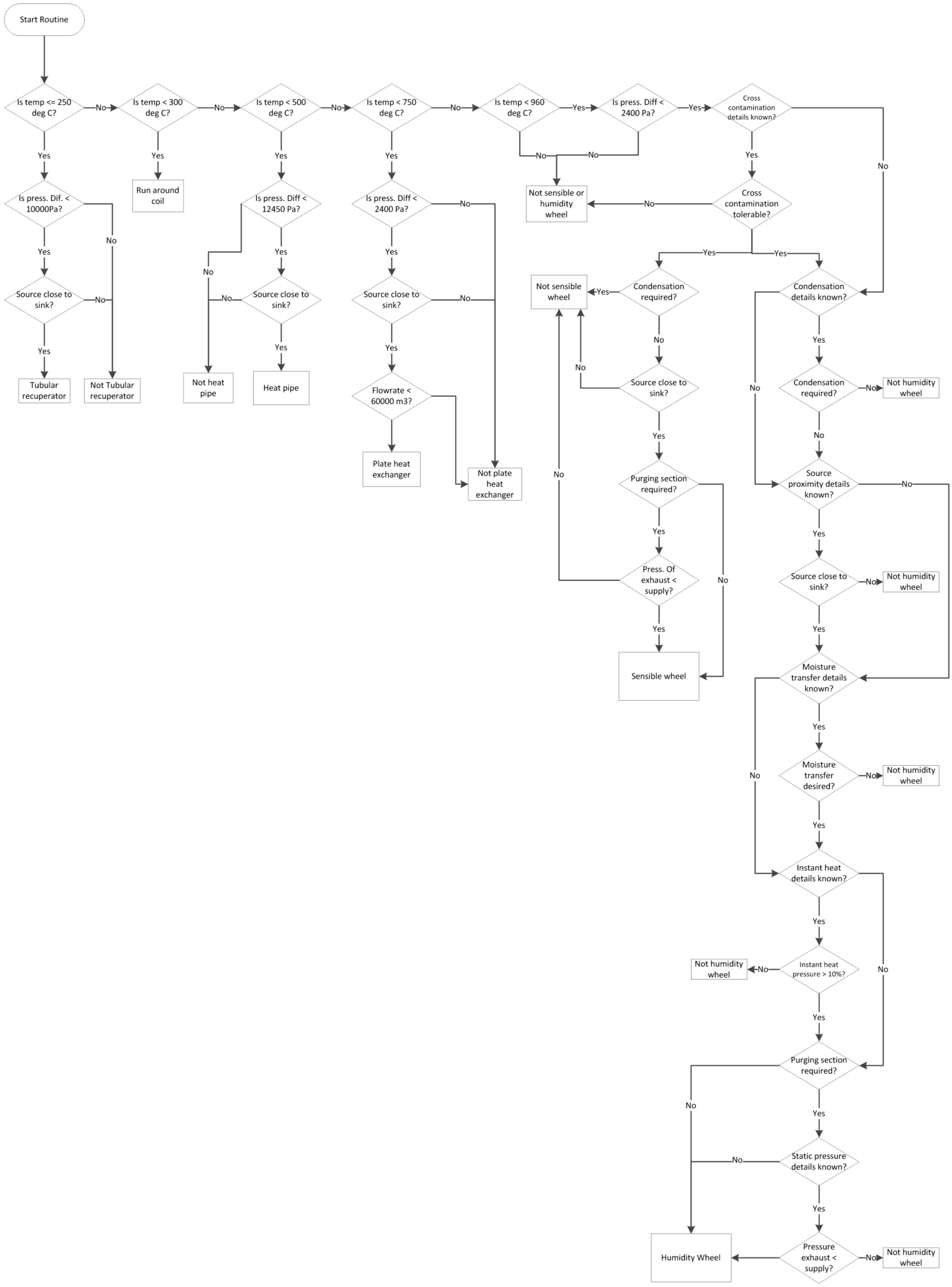


Figure 2.4. Section of gas-gas knowledge base in system (exact replica of figure taken from Heppenstall and Halliday, 1990)

The method shown in Figure 2.4 can be reduced, in simplistic terms, to a method of the form shown in Figure 2.5 below, *i.e.* if an operating limit of unit A is exceeded (for example pressure, temperature, fouling potential) do not use unit A.

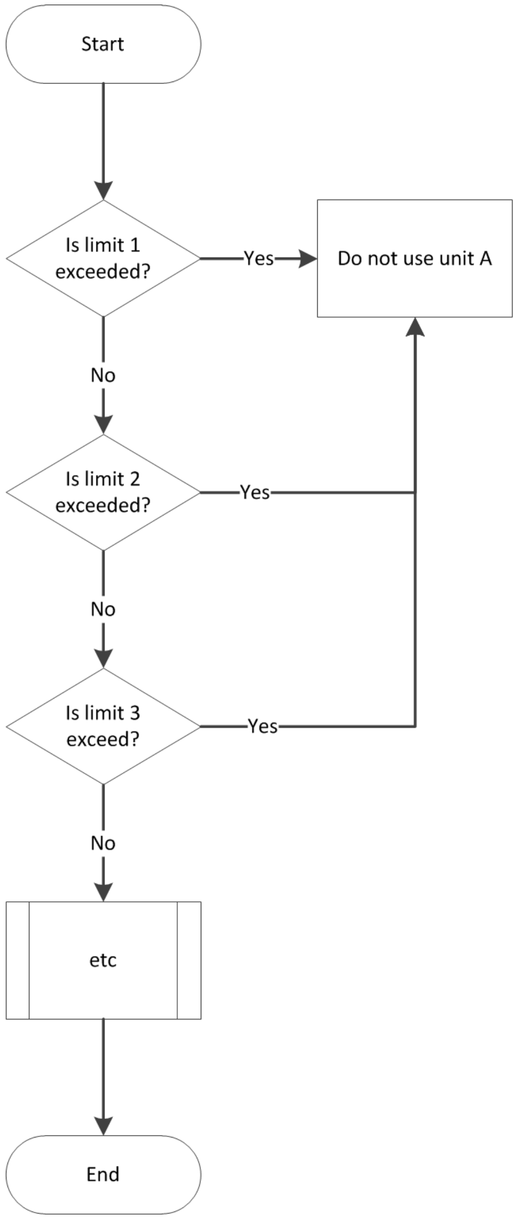


Figure 2.5. Form of knowledge-base in Heppenstall and Halliday (1990) method

The authors then considered a number of expert system "shells" in which to build the knowledge base. "Savoir", "PC-PLUS" and "CRYSTAL" were considered (none of which are still commonly used). "Savoir" was chosen as the most appropriate shell due to the simplicity of use, “good” user interface and robustness. The use of shells is still possible at the current time, however, given the advances in personal computers and programming languages in the past 25 years, it is now more common-place to simply write the code in its entirety.

Overall, the system was found to perform well during testing, producing comparable results to the opinions of experts in the field of heat exchanger selection/design. The method of formulating the system knowledge-base was thorough and beneficial to the development of the system, despite being time consuming. However, several limitations were identified as follows:

- The system was not able to produce detailed heat exchanger designs. Hence it was not possible for accurate cost estimates to be calculated which is often the deciding factor in selection procedures, particularly when it is possible to use more than one unit. The reason given was the difficulty in programming procedural routines into an expert system shell.
- The system often produced too many results, with no way to effectively compare between available heat exchangers. This is linked to the point above as a detailed design/cost estimate is required to do so.
- The system was not ready for commercial use. This was mainly due to difficulty of implementing a comprehensive testing scheme as the system was not user friendly and each tester of the system would have had to undergo a rigorous training procedure.

Many of the drawbacks listed above can be addressed using modern computer programming methods (as discussed in Section 5.1), but beyond this, the conclusions from this paper are a useful starting point for this work: they can be used as a guide for the initial stages of system development.

Many other software tools have addressed the topic of heat exchanger design. For example, Aspen Design and Rating (Aspentech [3], 2013) is a comprehensive tool for designing shell-and-tube heat exchangers, plate heat exchangers and plate-fin heat exchangers. This tool allows users to input inlet/outlet data for each stream (source and sink) and select stream components from a vast database. The software results include a detailed design of the unit including cost estimates.

However, the software again does not provide a method of heat exchanger selection. In order to gain valid results, the user must have a certain level of knowledge in this area in order to be sure that the particular type of heat exchanger is appropriate for use. A further drawback of this software is that it is very expensive and, perhaps as a consequence of this, has not reached all subsectors of the process industry.

Numerous academic papers address the subject of optimal heat exchanger design without discussing which types of unit should be selected for which duties. For example, Gut and Pinto (2004) investigated the optimal design configuration for (gasketted) plate heat exchangers. They chose six parameters (number of channels, number of passes per stream, feed connection location, hot fluid location, type of flow in the channels) and a complex screening algorithm to find an optimum heat exchanger effectiveness for a user-defined number of transfer units. The method was shown to find optimum design solutions for a given case-study of heat transfer between two organic liquids. However, the method does not take into account the need to often oversize heat exchangers in practice due in order to compensate for unanticipated fouling/blockage. Therefore, the optimal heat exchanger design would be unlikely to be implemented in the industrial domain. Furthermore, it is unlikely that a user would define the number of transfer units prior to heat exchanger design in practice.

Approaches such as this highlight the need for practicality to be taken into consideration when developing methods and tools which are to be disseminated into the industrial domain. Furthermore, this method again offers no procedure for the selection of this type of heat exchanger and assumes the reader (or person to implement the method in future) has a pre-existing knowledge of heat exchanger selection/design.

Another example is given by Yousefi *et al* (2012). Here, the optimal design of plate-fin heat exchangers is investigated using a hybrid evolutionary algorithm (essentially a customization of the standard genetic algorithm). Results show that the new method was an improvement on standard genetic algorithm methods in that more efficient heat exchangers are designed (smaller area for the same duty) with lower pressure drop, while the computational time of the method is also reduced (for execution on the same computer). However, again the methods employed in this paper are only of use to a reader with a significant background in heat transfer, and the user will have also already devised that a plate-fin heat exchanger is suitable for the specific case study in question.

2.2.3. Thermodynamic Cycles for Waste Heat Recovery

The design of thermodynamic cycles such as heat pumps (vapour compression, absorption *etc*) and power cycles (organic Rankine, Kalina *etc*) has been studied for many years and is currently gaining more interest due to rising energy prices.

A full discussion of the merits of various thermodynamic cycles for waste heat recovery is presented in Chapter 3. Here, literature covering design and selection methodologies of the various cycles is reviewed.

Many papers investigate various aspects of these cycles including working fluid selection (increasingly important due to a possible future phase-out of common hydro-fluoro-carbon working fluids), cycle configuration, exergy analysis and cycle optimization.

Literature addressing working fluid selection in organic Rankine cycles and heat pumps is addressed in Section 4.35-4.37 where the methodology of selecting working fluids for use in this work is discussed.

Nguyen *et al* (2010) studied optimal power generation from residual waste heat using organic Rankine cycles. Here, a number of parameters were considered including working fluid selection (discussed in Section 4.35-4.37) and cycle configuration (inclusion/exclusion of a recuperator). It was shown that the inclusion of a recuperator in the cycle increases the overall cycle efficiency by up to 5%. The authors conclude that this is an “effective means of improving overall cycle efficiency". However, no economic assessment was carried out to investigate the financial reward of this improvement in efficiency vs. the increased capital cost of the cycle. Therefore, the true benefit of this addition is not known.

Quoilin *et al* (2011) studied a thermo-economic optimization of organic Rankine cycles. Here, the optimization of the cycle was performed by varying the working fluid evaporator temperature. The case-study used in the model was for the recovery of a sensible heating duty, hence the effect of varying the evaporator temperature would affect both the cycle thermal efficiency and the amount of waste heat recovered, hence an optimum point exists.

The authors found two optimum points: the economic optimum and the thermodynamic optimum. The results varied for each working fluid, but a general trend existed in that the economic optimum was at a higher evaporation temperature than the thermodynamic optimum. The reason cited is that the fluid density increases at higher temperatures, hence the size of the purchase equipment is reduced. Furthermore, the fact that this would also require an evaporator of smaller heat transfer area must also have been a factor. The method is flawed, in that in reality the turbine (the highest capital cost component) would not be designed for use only at the optimal point, as one would have to account for an off-design heat source temperature. Also, one would generally seek to find an "*off the shelf*" turbine in a standard size, rather than a costly custom design. Hence, in reality, the results of the economic optimization may be used as a guide but are not definitive. Therefore, in such cases the thermodynamic optimum should be used in the initial design.

The literature regarding thermodynamic cycles is of a similar ilk to that discussed in Section 2.2.2 regarding heat exchanger selection/design, in that the majority of the papers focus on optimization of cycles rather than selecting when each type of cycle may be useful, *i.e.*, the bulk of the literature does not cover the problem addressed in this thesis.

A number of software tools are generally available to aid the design of thermodynamic cycles, including Aspen HYSYS (AspenTech [2] 2013) and IPSE Pro

(Sim Tech, 2013) amongst many others. However, again these tools require a large degree of existing expertise from the user and can often be very costly. Hence, such tools are not in direct competition with the aims of this thesis.

2.3. Knowledge-based/Expert Systems in Process Engineering

Knowledge-based systems and expert systems have been suggested for a variety of process industry applications. Examples include pipeline leak detection (Zhou *et al*, 2011), heat exchanger fouling detection (Afgan and Carvalho, 1995), food dryer selection (Lababidi *et al,* 2003) and solvent selection (Chan, 1995).

The focus of this section is on expert system/knowledge-based system literature linked to heat recovery/transfer and selection of industrial equipment as this is the most relevant to the work presented in this thesis.

Abou-Ali and Beltagui (1995) created an expert system for the selection of type of shell-and-tube heat exchangers. The work was effectively a method of automating the initial shell-and-tube design process presented in the standards by TEMA (Tubular Heat Exchangers Manufacturing Association). An expert system shell was used to create the expert system and a decision-tree type knowledge-base was developed (similar to that presented by Heppenstall and Halliday, 1990). Again, the decision tree followed the “IF criteria fulfilled, THEN action” pathway, as depicted in Figure 2.6 below.

```
IF      condition 1
AND     condition 2
THEN    hypothesis
AND     actions
```

For example,

```
IF      thermal expansion is severe
AND     bellows are allowed
AND     shell side fouling factor
        <0.00035 W/m² K
THEN    design_type
LET     design_type is fixed tube sheet
DO      tube side fouling factor of
        removable bundle is True
```

Figure 2.6. "IF-THEN" logic of expert system by Abou-Ali and Beltagui (1995)

The system had the aim of presenting a full TEMA (TEMA, 2013) shell-and-tube heat exchanger design based on user data input consisting of source/sink data (temperatures, mass flows, specific heat capacity) and user answers to various questions. However, the final system had a number of drawbacks and did not produce all of the desired results. The following results were excluded from the system as presented in the paper:

- The exchanger shell type
- Fluid allocation
- Heat exchanger geometry such as:
    - Tube diameter, thickness and length
    - Tube layout, pitch and number of passes
    - Shell diameter
    - Baffle spacing

The results presented only included the type of shell and tube bundle (e.g. fully welded shell, gasketted shell) based on relatively simple questions such as anticipated fouling concerns. Methods are presented for the full selection criteria but the authors were not able to implement them in the expert system.

The reason cited for this is the time taken to build the knowledge base. However, it is noted that the results missing from the system all stem from systematic calculations (for example, calculation of heating duty, then calculation of correction factor, calculation of number of passes, calculation of tube and shell geometry). All of the results generated were from the user-answered questions only. Therefore, a parallel exists between these results and those presented by Heppenstall and Halliday, 1990 (discussed in Section 2.2.2). Heppenstall and Halliday stated that the use of an expert system shell was limiting due its inability to process procedural routines. It is possible that such problems existed in the work by Abou-Ali and Beltagui. Therefore, this adds weight to the suggestion that for such an application, one is not advised to use expert system shells in program development. It is also noted that in the 18 years since publication of this paper, computational programming methods have improved dramatically and the authors may have had more success using modern day languages and operating systems.

Lababidi and Baker (2003) presented an expert system for food dryer selection. The problem addressed was similar to the work in this thesis in that a wide-range of potential options were available for selection, and were influenced by a wide-range of parameters. Here, the authors again used the decision tree "IF-THEN" method to select the most appropriate dryer based on user input data. The Java programming language was used in the development of the system, with ease of web-based dissemination cited as the main driver. A case study was used to test the viability of the system and results were shown to be in agreement with the views of industrialists, hence, the system was deemed a success. However, it is noted that only limited conclusions can be taken from only one published case-study.

The favourable result of Lababidi and Baker compared to that of Heppenstall and Halliday (1990) and Abou-Ali and Beltagui (1995) suggests that the use of a modern programming language such as Java has led to the creation of a more successful system than those based on expert system shells. Furthermore, the same type of "IF-THEN" decision tree knowledge-base development was employed. This suggests that revisiting previous works of Heppenstall and Halliday, and Abou-Ali and Beltagui using modern computational techniques may be worthwhile.

Chan and Tontiwachwuthikul (1995) developed an expert system for solvent selection for use in carbon capture processes. An expert system shell (G2) was used to create the system and the decision tree "IF-THEN" methodology was used to create the knowledge-base. The system was successfully tested using two case-studies, with results matching the views of industrial experts. Here, the authors stated that the use of an expert system shell was beneficial and that it would have taken significantly longer to develop the system without the use of one. However, it is worth noting that this system worked entirely by asking the user various yes/no questions, *i.e.* no procedural routines were required.

Afgan and Carvalho (1996) created a knowledge-based expert system for fouling assessment of heat exchangers. Here, a combination of heat transfer theory (in particular, effectivenesses and overall heat transfer coefficients) and process knowledge was combined to produce a warning system for the fouling of heat exchangers. The theory was used to produce an online calculation of the heat exchange effectiveness. This was used to calculate the overall heat transfer coefficient, which in turn was compared to the original, or "clean" overall heat transfer coefficient. From there, the degree of fouling could be inferred. Process knowledge was then used to create procedures dictating when the system would advise of the various levels of fouling and when the unit should be cleaned. The system was comprised of an acquisition element (to acquire online measurements), a validation analyser and trend analyser (to calculate trends in heat exchange effectiveness) and the system knowledge base which would infer the degree of fouling and when the unit should next be cleaned. This is shown in Figure 2.7 below.

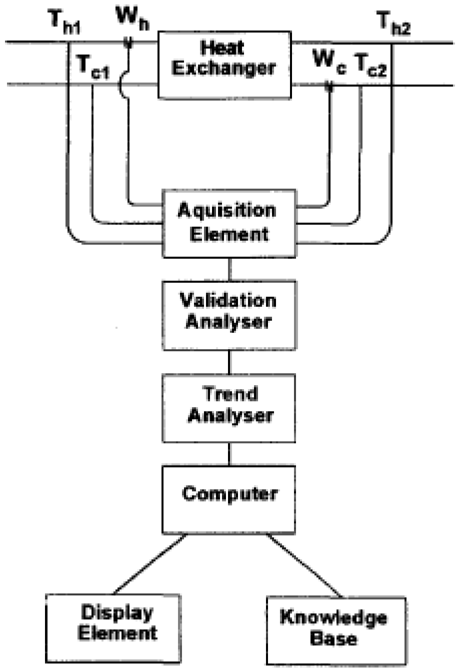


Figure 2.7. Flow diagram of expert system by Afgan and Carvalho (1995)

The system knowledge-base was again built using the "IF-THEN" methodology. The system was written using the LISP programming language .This language allowed the author freedom to program as desired, rather than follow the limitations of an expert system shell. Although not explicitly mentioned by the authors, in light of the papers reviewed above it seems likely that the use of this language rather than the use of a shell allowed the user more freedom and allowed the integration of the knowledge base, acquisition element and data analysers all into one program, creating a comprehensive package.

Overall, the authors stated that the concept was a success, although no case-study data was presented to demonstrate successful implementation of the methods.

Brunner *et al* (2010) developed an expert system for the intelligent supply of thermal energy in industry ("EINSTEIN" - previously mentioned in Section 2.2.2). The program was designed to guide the user through an energy audit procedure before assessing energy demand at the site. Pinch heat integration methods are then used to "match" waste heat sources and sinks and economic/environmental analysis is performed. This part of the program was not novel and was a based on the original works by Linhoff *et al* (1979). The user could then select a variety of alternative energy sources to replace existing equipment on site. For example, one could select a solar thermal system to replace an existing gas boiler. Economic and environmental analysis was also performed, allowing the user to make an informed decision about whether or not the results were viable on an individual case-study basis.

While this tool is useful, it does not represent a significant improvement on the pinch integration methods suggested many years ago. Also, the tool integrates renewable energy, such as solar thermal, at the users discretion, meaning that the system only performs the relevant economic and environmental calculations, *i.e.* it does not use programmed *knowledge* to suggest renewable energy systems to the user. There are no published case studies available to prove the worth of the system, however, the system is of interest here from a programming perspective: the Python programming language is used rather than a traditional expert system shell. Again, the use of a traditional expert system shell has been avoided in favour of writing the program from scratch where procedural routines are required.

2.4. Chapter Conclusions

The following conclusions are taken from this literature review:

- The previous literature shows that many methods and tools are available for use in the selection and design of waste heat recovery systems. In particular, a lot of time has been invested in research into pinch technology/heat integration methods for identifying opportunities for waste heat recovery by heat exchangers, heat pumps and other novel methods. However, current pinch methods are limited by the fact that they do not provide a full selection/design of the most suitable waste heat recovery equipment. Hence, the user must have significant knowledge prior to using pinch methods in order to design the waste heat recovery system (rather than revert to the shell-and –tube standard employed by these methods).

- Software has been developed for the design of specific heat exchangers or thermodynamic cycles, but does not offer advice on when particular technologies and/or cycles should be used. Furthermore, such software is often expensive, meaning that it is not generally accessible to every process industry subsector.
- Many papers focus on the optimization of certain cycles or heat exchangers, but the methods are often overly complex and ignored in industry.
- Expert systems have been developed for industrial applications with varying success for around 30 years. Early expert systems were hindered by old-fashioned computational methods, in particular the use of expert system shells which do not allow the easy implementation of procedural routines.
- Expert systems in the field of heat transfer engineering and heat exchanger selection tend to follow the decision tree “IF-THEN” method which has been proven successful.

2.4.1 Original Contribution to Knowledge

The conclusions of the literature review show that a number of methods exist to aid the recovery of low-grade waste heat in the process industries. Pinch technology has been the most successful and has widespread utilization. However, this and the other methods reported, are limited by the fact that they do not offer any advcice of *when* each different piece of equipment should be selected. Heppenstall and Halliday (1990) attempted to solve this problem with a view to constructing an expert system capable of selecting the most appropriate heat exchanger for industrial use, but were hindered by computing methods at the time.

The work presented here is intended to produce a knowledge-based system for the selection and design of waste heat recovery equipment, including a comprehensive database of options (see Chapter 3). Therefore, this work intends to build on the work of Heppenstall and Halliday (1990) from a heat exchanger selection point of view (including subsequent economic and environmental analysis), but extend the concept to allow the design of heat exchangers and the selection/design of more novel heat recovery methods such as heat pumps and organic Rankine cycles. Therefore, this work presents an original contribution to knowledge in the field of industrial waste heat recovery.

Furthermore, this project has also contributed to knowledge through the following conference and journal publications:

Law, R., Harvey, A. P., Reay, D. A. (2013) Techno-economic comparison of a high-temperature heat pump and organic Rankine cycle machine for low-grade waste heat recovery in UK industry. Int. J. Low-Carbon Tech. 8 (Special issue - Heat Powered Cycles Conference 2012). pp i47-i54.

Law, R., Harvey, A. P., Reay, D. A. (2013) Opportunities for low-grade heat recovery in the UK food processing industry. Applied Thermal Engineering 53. pp 188-196.

Law, R., Harvey, A. P., Reay, D. A. (2013) A Knowledge-based system for low-grade waste heat recovery. American Institute of Chemical Engineers: Annual Meeting. San Francisco. CA. USA. November 2013.

Law, R., Harvey, A. P., Reay, D. A. (2013) A Knowledge-based system for low-grade waste heat recovery. European Congress of Chemical Engineers 2013. The Hague. NL. April 2013.

Law, R., Harvey, A. P., Reay, D. A. (2012) Techno-economic comparison of a high-temperature heat pump and organic Rankine cycle machine for low-grade waste heat recovery in UK industry. Heat Powered Cycles 2012 Conference Proceedings. Alkmaar. NL. September 2012.[Later extended for journal publication as shown above]

Law, R., Harvey, A. P., Reay, D. A. (2011) Opportunities for low-grade heat recovery in the UK food processing industry. SUSTEM 2011 Conference Proceedings. Newcastle-Upon-Tyne. UK. October 2011. [Later extended for journal publication as shown above]

Law, R., Harvey, A. P., Reay, D. A. (2011) Steam-raising heat pump for low-grade waste heat recovery. 12th UK Heat Transfer Conference Proceedings. Leeds. UK. August 2011.

Chapter 3

This chapter covers a discussion of the state-of-the-art in waste heat recovery technology and the selection of suitable equipment for inclusion in the knowledge-based-system. The advantages and disadvantages of each technology are analysed and reasons provided for inclusion/exclusion in the system knowledge-base.

Much of this chapter is based on “Opportunities for low-grade waste heat recovery in the UK food processing sector” by Law *et al* (2013). However, it will also include a more detailed discussion, consideration of more process industry sectors and a critical evaluation of the credentials of each technology leading to its inclusion/exclusion in the system knowledge-base.

3. Technology Selection

A wide range of technologies are available for low-grade waste heat recovery, as briefly discussed in Chapters 1 and 2. In general, the various technologies fall into the following five categories:

1. Heat exchanger heat transfer: heat transfer from waste heat source to *matching* waste heat sink (as defined in Section 1.2.1and Section 2.2.1)
2. Heat pumps (heating): heat transfer from a lower temperature heat source to a higher temperature heat sink facilitated by the input of energy from an external source
3. Power generation
4. Generation of coolth
5. Waste water treatment

The selection of the most suitable technologies for the knowledge-based system was completed according to the scope of the system (Section 1.3.1) via review of literature and technical data, and accompanying calculations. In particular, points 7-9 from the system scope are considered in this Chapter and are restated as follows:

7. Must include a variety of waste heat recovery techniques: this will allow a wide range of possible process conditions to be accommodated

8. Must include technologically viable results: results must be meaningful on an industrial scale. Technologies requiring significant further R&D should not be included

9. Must include economically viable results: only technologies which have been proven to achieve economically viable results will be considered. Technologies incurring typical pay back times of greater than 5 years (under economic conditions at the time of writing) will be considered non-economical

3.1. Heat Transfer to Matching Heat Sink

3.1.1. Gas-Gas Heat Transfer

In gas-gas heat transfer a variety of heat exchangers are available to facilitate waste heat recovery. Here three scenarios are considered as follows:

- Gas (non-condensable, n/c) heat source, gas heat sink. Sensible heat transfer only. A typical example of this would be recovery of waste heat from a spray dryer exhaust (above source dew point) to preheat inlet air.
- Vapour heat source, gas heat sink. Sensible and latent heat transfer. A typical example of this would be recovery of vapour from the “Wort Boiling” process in brewing for space heating.
- Humid gas heat source, gas heat sink. Majority sensible heating, some latent heat from condensation of water vapour. A typical example of this would be waste heat recovery from an industrial hood dryer (including condensation) to pre-heat the inlet air-feed.

The three scenarios have varying design constraints and require a range of heat exchangers exhibiting different properties. Furthermore, each individual case study will have further design constraints due to varying stream properties including pressure, corrosivity and fouling considerations.

A summary of the various options considered is shown below in Table 3.1.

Table 3.1. Summary of gas-gas heat exchangers considered for inclusion in system knowledge base

Heat Exchanger	Selected for system?	Suitable for which scenario?
Run-Around-Coil	Yes	Gas-Gas (Sensible heat only); Humid Gas-Gas (Some latent heat)
Gas-Gas Plate Heat Exchanger (Air handling Unit)	Yes	Gas-Gas (Sensible heat only); Humid Gas-Gas (Some latent heat)
Rotary Regenerator	Yes	Gas-Gas (Sensible heat only)
Rotary Regenerator with moisture transfer	No	N/A
Finned-Tube	Yes	Vapour-Gas (Majority latent heat)
Shell and Tube	Yes	Gas-Gas (Sensible heat only); Humid Gas-Gas (Some latent heat)
Welded Plate	Yes	Gas-Gas (Sensible heat only); Humid Gas-Gas (Some latent heat)
Printed Circuit	No	N/A
Heat Pipe	No	N/A
Polymer	No	N/A

3.1.1.1. Gas-Gas Plate Heat Exchanger

The gas-gas plate heat exchanger (often referred to as an *air handling unit,* particularly in non-process applications) is a proven technology in the area of gas-gas waste heat recovery. Here, heat is exchanged between the source and sink across a series of metal (often stainless steel or aluminium) plates in a counter-current or cross-flow configuration. The unit may also be configured with a drip tray for condensation collection in the recovery of latent heat from humid air sources (Reay, 1979).

Figure 3.1 below shows the typical set-up of the cross-flow gas-gas plate heat exchanger (Reay, 1979).

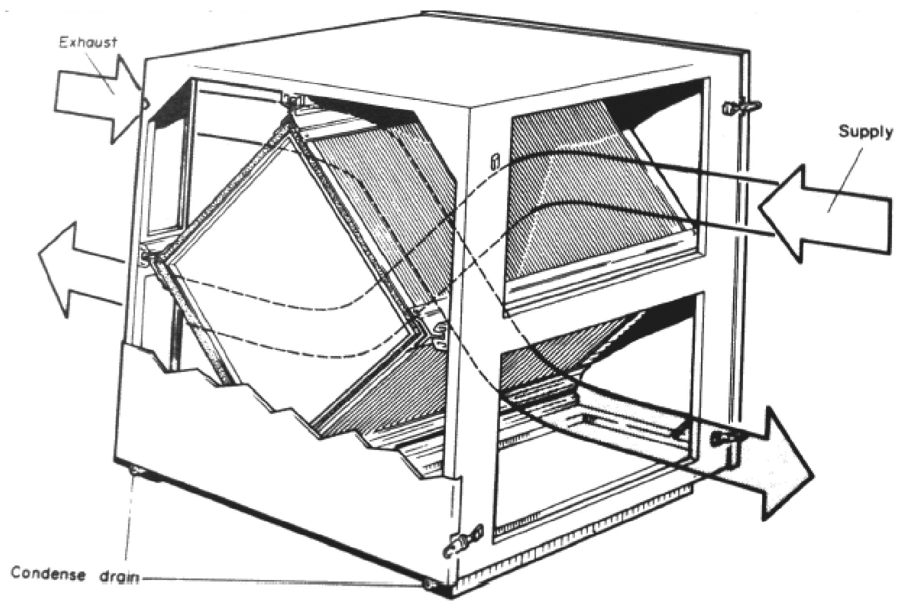


Figure 3.1. Typical gas-gas plate heat exchanger configurations (Reay, 1979)

This use of this heat exchanger has been proven to provide economical and environmentally beneficial solutions. For example, British Bakeries Ltd (published in the Energy Efficiency Office, Energy Efficiency Demonstration Scheme, Profile 235, 1987) demonstrate the use of 4 air handling units to recovery heat from both gas burner exhausts (to pre-heat burner inlet air) and the oven exhaust (to heat the air inlet to the *prover*). The data reveal annual energy savings of 6,900 GJ/year and a project payback period of 2.6 years.

Another case-study is presented by United Biscuits Ltd (published in the Energy Efficiency Office, Energy Efficiency Demonstration Scheme, Profile 76, 1982) where waste heat (including some latent heat) was recovered from an oven to provide space-heating in a packaging hall on site. Here, the gas-gas plate heat exchanger was shown to provide effective waste heat recovery. Furthermore, it is reported that the fouling build up caused by recovering latent heat was easily cleaned due to the simplicity of opening up the gasketted access panel for mechanical cleaning. Project payback time was stated as three years while the fuel consumption for space heating was reduced by 50% (16,400 GJ/year).

In summary, the gas-gas plate heat exchanger is included in the system database as it is a proven technology in the field of gas-gas heat transfer.

3.1.1.2. Rotary Regenerator

The rotary regenerator (often referred to as a *Heat Wheel*) is again a proven technology in this area. Here, air in two adjacent ducts flows through a rotating matrix spanning the ducts which achieves heat transfer as shown in Figure 3.2 (Sanaye *et al*, 2008). Note: the thermometers, electric coils and axial fans shown on the diagram are related to the experimental study carried out in the paper from which the diagram has been taken.

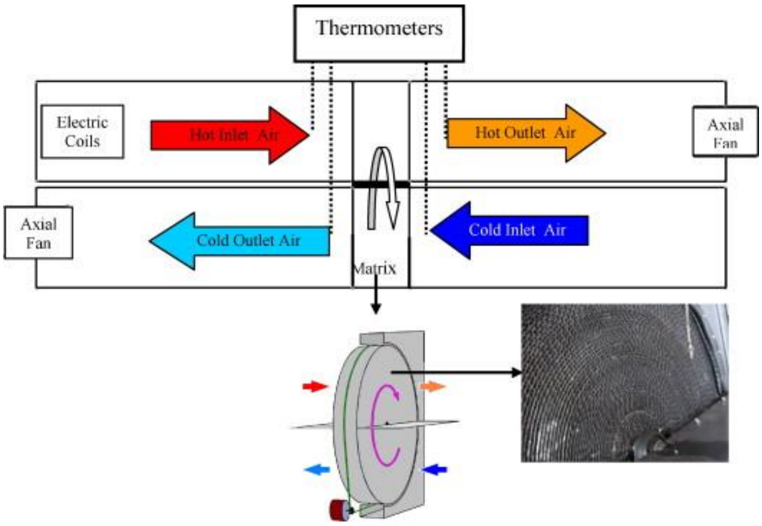


Figure 3.2. Schematic of the rotary-regenerator (Sanaye *et al*, 2008)

The rotary regenerator is often chosen as an alternative to the gas-gas plate heat exchanger due to the high effectiveness of the unit (up to 95%). However, the unit has inherent problems with cross-contamination (up to 5% (Reay, 1979)) between streams due to gas entrainment between ducts during rotation of the matrix. Light fouling can be tolerated due to the inclusion of a purge section (Shah and Sekulic, 2003).

Numerous successful installations of the rotary regenerator are reported. For example, E Bottomley and Sons Ltd (published in the Energy Efficiency Office, Energy Efficiency Demonstration Scheme, Profile 7, 1981) installed the unit to facilitate waste heat recovery between two loose-stock fibre dryers and the air inlet. A payback time in the region of 3 years was reported.

Another example is in paper drying, as published by the CADDET energy efficiency programme for the International Energy Agency (CADDET case study 316, 1998). Here, the rotary regenerator was used to pre-heat inlet air to the dryer by recovering heat from the exhaust of temperature 80-200^{o}C (depending on the varying process conditions due to varying products being processed on site). This case study demonstrated a number of advantages of this unit. Firstly, the unit was shown to achieve an efficiency of 95%, as reported in other literature. This is shown below in Figure 3.3. Note: in this figure, the cooled exhaust efficiency was maintained at around 70% by design to ensure that the stream was not cooled below the dew point to prevent the condensation of "harmful corrosive" compounds in the flue.

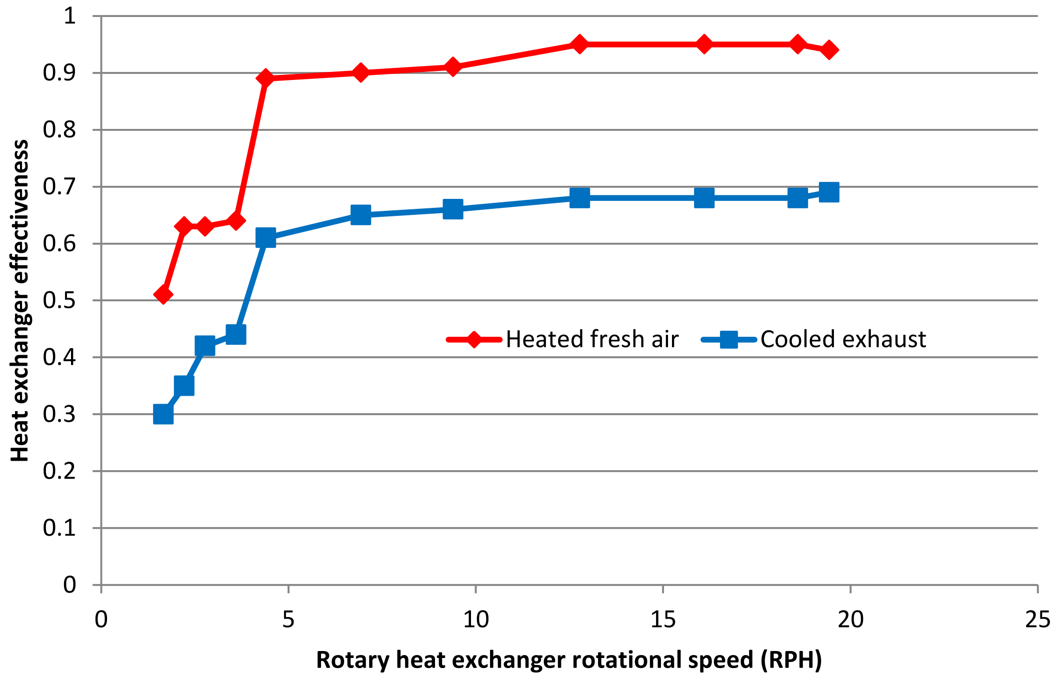


Figure 3.3. Rotary regenerator efficiency (adapted from International Energy Agency, 1998)

Secondly, the report also states that the outlet temperature of the exhaust can be controlled by altering the rotational speed (reducing the speed reduces the rate of heat transfer, thereby increasing the exhaust exit temperature). This shows that the rotary regenerator is particularly useful when temperature control is important.

The project payback time for this case study was 2.2 years and no maintenance problems were reported. Therefore, the rotary regenerator is deemed a suitable technology for inclusion in the system knowledge-base as it meets the criteria outlined in the scope of the system.

A number of manufacturers (Flakt Woods, 2013, and Air XChange, 2013) also offer a variation on the rotary regenerator that includes moisture transfer (*i.e.*

condensation of vapour from a humid stream into the regenerator matrix, which is then transferred to the heat sink stream). This is not beneficial to process waste heat recovery. Such a process is only beneficial when dealing with heat recovery and comfort/humidity control in buildings. Therefore, this variation is excluded from the knowledge base.

3.1.1.3. Run-Around-Coil

The run-around-coil heat exchanger is comprised of two separate coiled heat exchangers (often finned) which are connected by pipe work, as shown in Figure 3.4. Water or a brine solution is most commonly used as the heat transfer fluid,

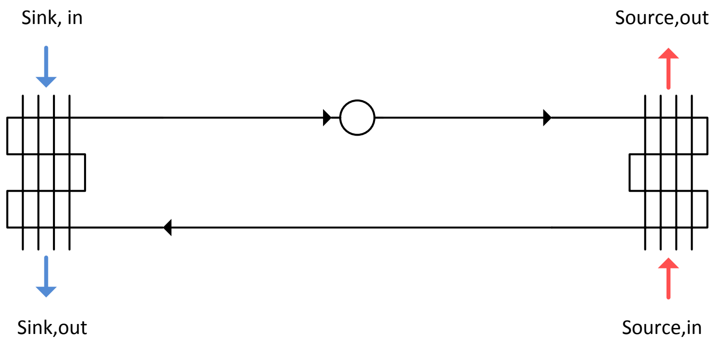


Figure 3.4. Typical run-around-coil configuration

This heat exchanger configuration inherently gives the unit two distinct advantages over the other options listed in Table 3.1 in that, firstly, the probability of cross-contamination of the two streams is close to zero and, secondly, the waste heat may be transported over large distances (dependant on the size of the pump installed and potential heat losses over the distance). However, as two approach temperatures are required (at either ends), the maximum overall heat exchanger effectiveness is around 60% (Reay, 1979). Therefore, this unit is normally only considered when the user has a need for the process advantages listed above.

Clayton Aniline Co Ltd (published by the Department of Energy, Energy Efficiency Demonstration Projects Scheme, Profile 78, 1982) have demonstrated the use of the run-around-coil to recover waste heat from a spray dryer to pre-heat inlet air. Here, the run-around-coil was selected due to the need for zero cross-contamination. A further advantage was that the exhaust outlet temperature could be controlled using simple proportional methods by altering the circulating flow rate: this was crucial in this case study to prevent the condensation of corrosive components from the exhaust stream. Energy savings in the region of 1.4 GWh/year (natural gas) are reported, which would correspond to a GHG emission reduction in the region of 150 tonnes/year. Project payback time is quoted as 2 years.

Another successful application of the run-around-coil was reported by Rockware Glass Ltd (published by the Department of Energy, Energy Efficiency Demonstration Projects Scheme, Profile 42, 1981). Here, waste heat was recovered at 55^{o}C for use in space heating. A run-around-coil was employed due to the large distance between the heat source and the space heating station. It was deemed safer and more cost effective to use a run-around coil rather than install larger duct work for the gas streams. The data show energy savings in the region of 270 GWh/year and the project payback time was approximately 1.5 years.

Therefore, from the reported data it can be concluded that the run-around-coil is proven to satisfy the scope of the knowledge-based system and is therefore included in the program.

3.1.1.4. Shell-and-Tube Heat Exchanger

The **Shell-and-Tube** heat exchanger is the most commonly used unit across the process industries and therefore must be considered for use in WHR. The unit is capable of withstanding high temperatures and pressures (>260^{o}C, >500bar) and can facilitate heat transfer between two gaseous or two liquid streams (gas-liquid generally requires an extended surface on the gas side), as well as boiling/condensation duties. Fouling can be tolerated on the tube-side due to the relative ease of removing the tube bundle for cleaning.

The merits, limitations and various configurations are well known and therefore not further discussed here. Extensive further reading can be found in Chemical Engineering Volume 1 (Coulson and Richardson, 2005), Process Heat Transfer (Kern, 1950) and in numerous literature by the Tubular Exchanger Manufacturers Association, Inc (TEMA, 2013). However, it is noted that generally this unit would not be considered for gas-gas WHR unless the other, more compact units were not suitable.

Tubular heat exchangers should only be considered in gas-gas WHR "where size is not important but access for cleaning is essential" (David Reay and Associates, 1994). Other situations where this unit should be considered include when the source temperature exceeds the temperature limit of other gas-gas heat exchangers (greater than 200^{o}C) or in custom solutions such as glass-tubular heat exchangers which proven useful in WHR from heavily fouled streams (Energy Conservation Demonstration Projects Scheme Case Study 146, 1983) due to the ease of cleaning.

Therefore, this unit is included in the KBS gas-gas heat exchanger knowledge-base to provide a solution outside of the operating range of the other, preferred, units.

3.1.1.5. Welded Plate Heat Exchanger

The welded plate heat exchanger operates according to the same heat transfer principles set out in Figure 3.7, Section 3.1.2. This unit is most commonly used in liquid-liquid heat transfer, and is therefore discussed further in Section 3.1.2. However, this unit can also facilitate gas-gas WHR although it is less commonly used than the run-around-coil, gas-gas plate and the rotary regenerator.

This unit can tolerate high temperatures and pressures (>260^{o}C and up to 40bar) (Alfa Laval, 2013) but cannot tolerate fouling as the unit is fully welded and therefore difficult to clean. Therefore, this unit can be seen as a compact alternative to the shell-and-tube heat exchanger when the heat source temperature is out of range of the more commonly preferred units and fouling is not a concern.

3.1.1.6. Finned-Tube Heat Exchanger

The finned-tube heat exchanger is considered when either the source or sink has a significantly limiting heat transfer coefficient. Therefore, it is useful in the case of a condensing vapour transferring heat to a gas stream. This unit is more commonly used in gas-liquid heat transfer and the reasons for inclusion of this heat exchanger are therefore discussed in Section 3.1.3.

3.1.1.7. Heat Pipe Heat Exchanger

The heat pipe heat exchanger is comprised of a number of heat pipes separated by a splitter plate. The source and sink flow on opposite sides of the splitter plate and heat transfer is facilitated by the heat pipes. The heat pipe itself is a passive two phase device of very high effective thermal conductivity (order of 100000 W/mK) achieved by the simultaneous evaporation and condensation of a working fluid at each end of the pipe. A schematic of a single heat pipe in this heat exchanger is shown in Figure 3.5 (Carbon Trust, 2012).

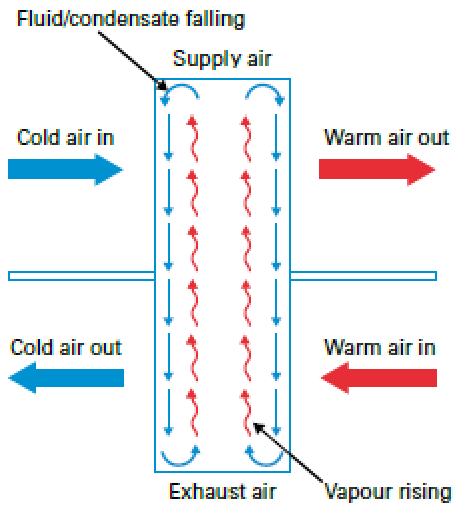


Figure 3.5. Heat pipe heat exchanger (Carbon Trust, 2012)

The advantages of this unit in gas-gas waste heat recovery are, firstly, the heat pipe structure creates a fin-like extended surface, thus helping to overcome the low convective heat transfer coefficients associate with gases, and secondly, the heat pipes allow a high rate of heat transfer and low approach temperatures (as low as 5K, Dunn and Reay, 1994).

This heat exchanger has been proven to be economical in low-grade waste heat recovery, for example T. Lucas & Co. Ltd (published by the Department of Energy, Energy Efficiency Demonstration Projects Scheme, Profile 56, 1982) demonstrated the use of this unit to recover heat from a gas burner to pre-heat a spray dryer inlet. Here, payback times are reported to be in the region of two years.

Another demonstration of this unit was in heat recovery from a paint surface coating oven by Alcan Plate Limited (published by the Department of Energy, Energy Efficiency Demonstration Projects Scheme, Profile 80, 1982). Here, heat was recovered from the oven exhaust to pre-heat inlet air. Again, favourable project economics are reported (payback time of 1.7 years).

Despite favourable published data, this unit is not selected for inclusion in this system. This unit a *custom build* and algorithms for unit design are kept in house. Spirax Sarco (Amini, 2013), consider the heat exchanger to be a *"Bespoke Unit"*.

Design procedure is therefore expected to be extremely difficult for this type of heat exchanger, as there are no published design procedures. The design task is summarised by Dunn and Reay, 1979, "*There are a considerable number of variables which can affect heat pipe performance, and limitations exist at present on maximum operating temperature*". Clearly, if data are not made public by the manufacturers, then it is extremely difficult to formulate such procedures for use in this system. Hence, the heat pipe heat exchanger is deemed out of scope.

3.1.1.8. Printed Circuit Heat Exchanger

The printed circuit heat exchanger (PCHE) is a highly compact unit developed by Heatric, a division of Meggitt Ltd. The units are comprised of a number of plates containing chemically etched flow channels joined via diffusion bonding. This creates a highly effective unit (up to 98% heat transfer effectiveness) which can withstand extreme temperature (from cryogenic temperatures to 900 ^{o}C) and pressure in excess of 600 bar (Le Pierres, 2013). This unit can be used for a variety of gas-gas, gas-liquid and evaporative/condensing duties. Figure 3.6 below shows the compact channel arrangement of the PCHE.

Figure 3.6. Compact channel arrangement of PCHE (Le Pierres, 2013)

The PCHE is excluded from the system knowledge base for two reasons. Firstly, this unit is extremely costly compared to the other units considered due to the complex multi-stage manufacturing procedure. As a result, it is most commonly used in situations where space and mass is at a premium, such as oil-and-gas platforms. The process industries have thus far neglected this unit in favour of cheaper options. Secondly, the design procedures and algorithms are kept under patent and are only available *in-house* at Heatric. Therefore, inclusion of the PCHE would require formulation of design procedure which is out of the scope of the system.

The **Polymer** heat exchanger is not considered for gas-gas waste heat recovery for the reasons explained in Section 3.1.2.8.

3.1.2. Liquid-Liquid Heat Transfer

In liquid-liquid heat transfer, the following scenarios are considered.

- Liquid heat source, liquid heat sink (no boiling). A typical example of this would be heat recovery from spent wash water to pre-heat water to a hot well.
- Liquid heat source, liquid heat sink (with boiling). A typical example of this would be waste heat recovery from a liquid effluent for use in a heat pump system.

The following heat exchangers are considered for inclusion in the equipment database to satisfy the design constraints of the two scenarios.

Table 3.2. Summary of liquid-liquid heat exchangers considered for inclusion in the system knowledge base

Heat Exchanger	Selected for system?	Suitable for which scenario?
Gasketted Plate (Plate and Frame)	Yes	Both
Brazed Plate	Yes	Both
Welded Plate	Yes	Both
Plate and Shell	Yes	Both
Shell and Tube	Yes	Both
Spiral Plate	Yes	Sensible heating only
Printed Circuit	No	N/A
Scraped Surface	No	N/A
Polymer	No	N/A

3.1.2.1. Gasketted Plate Heat Exchanger

The gasketted **p**late heat exchanger is commonly used for a variety of liquid-liquid heat transfer duties, including waste heat recovery. Figure 3.7 (Alfa Laval, 2013) depicts a typical schematic of the unit.

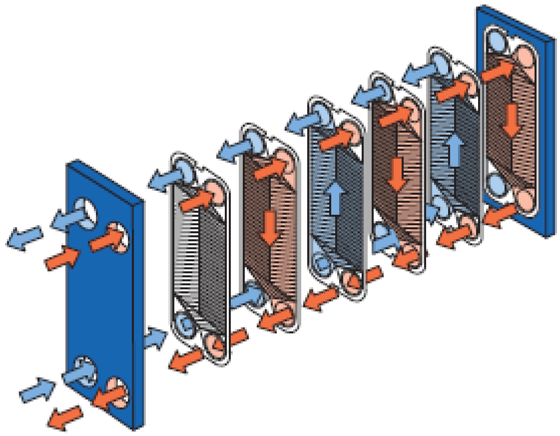


Figure 3.7. Plate heat exchanger flow configuration (Alfa Laval, 2013)

This heat exchanger consists of a series of thin corrugated plates packed together using gaskets. The hot and cold fluids flow in a counter-current configuration in the adjacent flow channels created by the plate structure. This creates large heat exchange areas in excess of 200m^2/m^3 (Alfa Laval, 2013).

The use of gaskets creates a number of advantages. First of all, the unit can be easily opened for cleaning and general maintenance, therefore light fouling can be tolerated. Secondly, the unit can be purchased *off-the-shelf* and assembled on site. This results in lower capital expenditure compared to similar units, such as the brazed and welded plate heat exchangers. However, the gaskets lead to temperature and pressure limitations of 180^{o}C and 16 bar respectively (Alfa Laval, 2013).

A case-study by Corus Steel and Spirax Sarco (Spirax Sarco, 2013) proved that the gasketted plate heat exchanger can provide economical solutions to waste heat recovery. Here, flash steam was recovered from a slab furnace cooling system to pre-heat boiler feed water. Payback times are reported as less than 12 months.

Kandilli and Koclu (2011) presented an optimisation of (gasketted) plate heat exchanger for waste heat recovery from fouled effluent to heat process water in the textiles industry. The unit was selected for use here due to its high effectiveness and tolerance to fouling. Waste water and fresh water flowrates were varied in order to find the flowrates to maximise heat exchanger effectiveness and exergy efficiency. The study was highly empirical in nature, and the numerical results and conclusions are only valid to the particular case study in question (*i.e.* no models, equations or dimensionless analysis is presented for cross-case-study implementation of the methods involved). However, the study does highlight two key points about the unit. Firstly, it is suitable for use in fouling environments. Secondly, the unit exhibits effectivenesses ranging from 86.8% to 99.1%, although this data must be taken with caution as the typical effectiveness for this unit is 95%. Therefore it is assumed that the test unit in this paper was largely oversized upon achieving an effectiveness of 99.1%. In industrial scenarios it is likely this would be greatly detrimental to the economics of the project and the heat exchanger pressure drop.

In summary, the gasketted plate heat exchanger is included in the knowledge base as it exhibits a number of advantages such as high effectiveness (due to high heat transfer coefficients), compact nature, tolerance to mild fouling and favourable project economics.

3.1.2.2. Brazed Plate Heat Exchanger

The brazed plate heat exchanger operates in an identical manner to the gasketted unit, although the plate structure is held together via brazing (most commonly copper brazing) rather than gaskets. The unit flow regime is identical to the unit shown in Figure 3.7 and described above. The advantage of this exchanger is that the brazing allows for high temperature and pressure limits of 225^{o}C and 25 bar respectively (Alfa Laval, 2013). However, it cannot tolerate fouling or solid particles in the heat source/sink (Best Practice Programme, 2000) but other advantages such as the compact size and high effectiveness remain. This unit is considered in the KBS database as a high temperature/pressure alternative to the gasketted plate heat exchanger, only with fouling limitations. The brazed plate heat exchanger is more costly that the gasketted plate heat exchanger due to associated costs of brazing compared to gaskets.

3.1.2.3. Welded Plate Heat Exchanger

The welded plate heat exchanger operates in an identical manner to the gasketted unit, although the plate structure is held together via welding rather than gaskets. The flow regime is identical to that depicted in Figure 3.7 and described above. The advantage of this unit is that the welding allows for high temperature and pressure limits of >260^{o}C and 40 bar respectively (Alfa Laval, 2013). However, this type cannot tolerate fouling or solid particles in the heat source/sink (Best Practice Programme, 2000). It is included in the KBS database as a higher temperature/pressure alternative to the brazed plate heat exchanger. It is also noted that the welded plate is more costly than the gasketted plate heat exchanger due to associated costs of welded compared to gaskets.

3.1.2.4. Plate and Shell Heat Exchanger

The plate and shell heat exchanger may be considered a hybrid of the plate heat exchanger and the shell-and-tube heat exchanger. This type features an outer shell enclosing pairs of welded circular plates. The cooling medium generally flows on the shell-side, between the plate pairs while the heat source flows between the welded plate pairs. The principal of design is to combine the high heat transfer coefficients of the plate heat exchanger with the rigid design of a shell-and-tube. This arrangement is shown below in Figure 3.8 (Best Practice Programme, 2000).

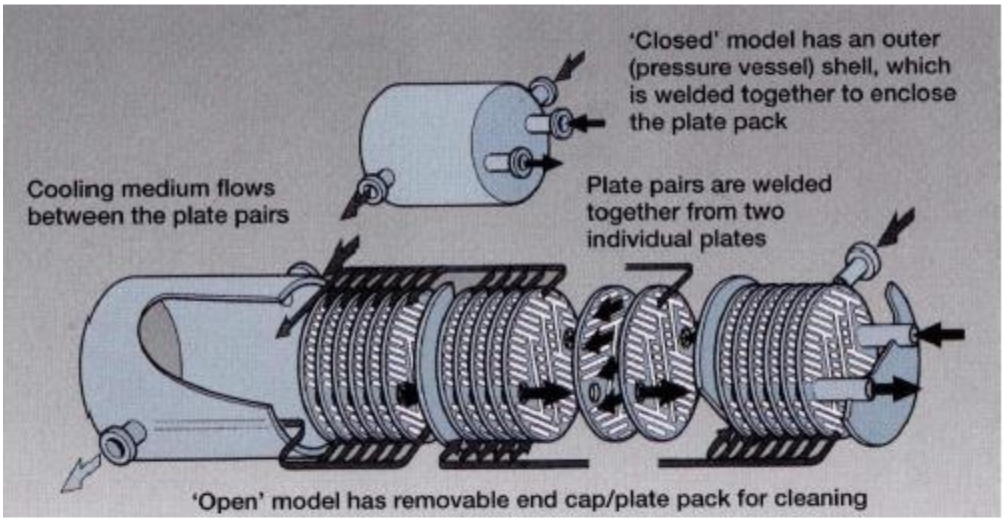


Figure 3.8. Plate and shell heat exchanger (Best Practice Programme, 2000)

The plate and shell heat exchanger is included in the KBS database as a high pressure (100 bar) and high temperature (>260 ^{o}C) alternative to the plate-type heat exchangers. It can additionally handle corrosive media (Best Practice Programme, 2000).

3.1.2.5. Shell and Tube Heat Exchanger

The shell-and-tube heat exchanger is, again, noted as the most common unit in liquid-liquid heat transfer and as a result the merits of the unit are not further discussed here. This unit is included for situations when other, more compact, units are not suitable for selection. Such scenarios include fouling fluids (this unit can tolerate fouling on the tube-side), high temperature and high pressure.

3.1.2.6. Spiral Plate Heat Exchanger

The spiral plate heat exchanger is configured as two elongated plate channels rolled around a central core. The heat source and sink flow counter-currently in adjacent plates, as shown in Figure 3.9 (Alfa Laval, 2013).

Figure 3.9. Spiral plate heat exchanger (Alfa Laval, 2013)

The smooth and curved channels of the unit tend to reduce fouling. Fouling is further reduced as any local fouling will result in a reduction in the channel cross sectional area which in turn increases the fluid velocity, creating a scouring effect to clean the channel (Best Practice Programme, 2000). Therefore, the spiral heat exchanger is generally considered for use when dealing with highly fouled, or slurry type media. Heavily fouling fluids can be accommodated on *both sides* of this heat exchanger.

In summary, this unit is included in the KBS database for selection when both the heat source and heat sink have high fouling tendency.

3.1.2.7. Scraped Surface Heat Exchanger

The scraped surface heat exchanger has been specifically developed for use with fluids of complex rheology. This can include concentrated slurries, highly viscous fluids or non-Newtonian fluids. The unit is essentially a double pipe heat exchanger where the heating or cooling media flow in the outer pipe. The fluid of complex rheology flows in the inner pipe, in which a blade rotates to remove (or *scrape*) any solids from the heat exchanger wall. This prevents fouling build up and ensures uniform heat transfer throughout the unit. This flow configuration is depicted in Figure 3.10 below (adapted from RheoHeat, 2013).

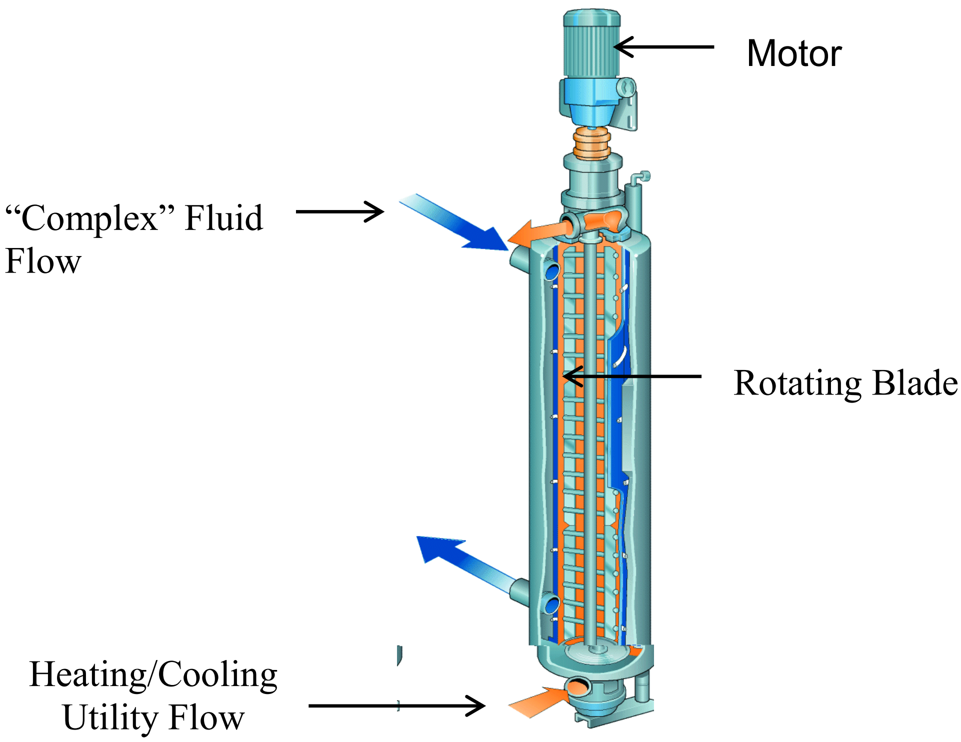
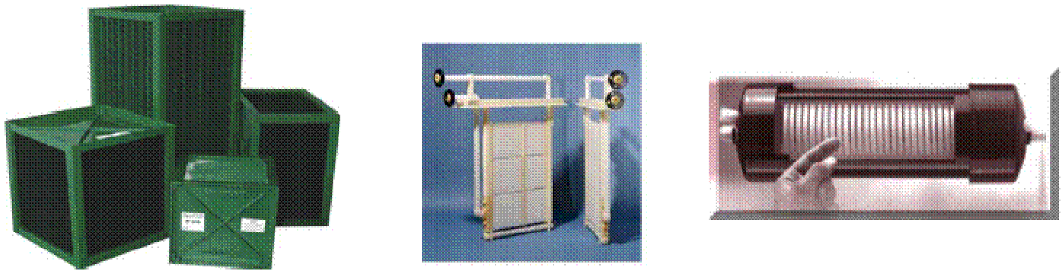


Figure 3.10. Scraped surface heat exchanger (adapted from RheoHeat, 2013)

A large range of scraped-surface heat exchangers are available from Alfa Laval (2013) including the Contherm for liquid-liquid duties, the Convap for evaporation/concentration duties and low-shear options. Design procedures and algorithms are not available for this unit. For this reason, and the fact that it is highly uncommon that one would recover waste heat from a highly viscous and potentially valuable product, this unit is considered a bespoke, custom solution. Therefore, it is not included in the KBS database.

3.1.2.8. Polymer Heat Exchanger

The polymer heat exchanger has been developed in a number of configurations for specialist applications ranging from biotechnology environments to heat transfer in aggressive/corrosive fluids (Zaheed and Jachuck, 2004). The most common configurations of the polymer heat exchanger are plate (similar to a plate-fin structure), coil and shell-and-tube. The three common configurations are shown in Figure 3.11 below (adapted from Zaheed and Jachuck, 2004).



(a) (b) (c)

3.11. Polymer heat exchanger configurations: (a) Plate, (b) Coil and (c) Shell and Tube (adapted from Zaheed and Jackuck, 2004)

The coiled configuration is a submerged heat exchanger, while the other two are of standard flow configurations. Various polymers can be used for construction and is chosen according to application.

Polymer heat exchangers are not widely accepted by the process industries as highlighted by the review paper by Zaheed and Jackhuck, 2004, "*the use of polymers in industrial heat exchangers has remained a niche market for some time. Their acceptance in the process industries is not yet widespread*". Hence, the inclusion of this heat exchanger in the system knowledge base would be a violation of the scope of the system, particularly point 8. Therefore, the polymer heat exchanger is not included in the KBS database.

The **Printed Circuit** heat exchanger is not considered for liquid-liquid waste heat recovery for the reasons explained in Section 3.1.1.8.

3.1.3. Gas-Liquid Heat Transfer

In gas-liquid heat transfer a variety of heat exchangers are again available to facilitate waste heat recovery. Here we consider seven scenarios as follows:

- Liquid heat source, gas heat sink. Sensible heating only. A typical example of this would be waste heat recovery from a liquid effluent for space heating.
- Gas (n/c) heat source, liquid heat sink (no boiling). Sensible heating only. A typical example of this would be recovery of boiler flue gas (without condensation of water vapour) to pre-heat inlet water. Note: many process plants choose such an option, as liquid effluents can be costly to dispose of, and may have to be pre-treated
- Vapour heat source, liquid heat sink (no boiling). Majority of latent heat recovered. A typical example of this would be recovery of flash steam to heat hot well storage water.
- Humid gas heat source, liquid heat sink (no boiling). Majority involve sensible heating, some latent heat. A typical example of this would be recovery of boiler flue gas to pre-heat inlet water (with provision for the corrosive products of condensation).
- Gas (n/c) heat source, liquid heat sink (with boiling). Sensible heating from source, latent heat in sink. A typical example of this would be recovery of waste heat from a dryer exhaust (without condensation) for use in a heat pump (possibly to pre-heat the inlet air). Note: many process plants choose such an option as liquid effluents can be costly to dispose of.
- Vapour heat source, liquid heat sink (with boiling). Majority latent heat recovered with latent heating required in sink. A typical example of this would be recovery of flash steam for use in an organic Rankine cycle machine.

- Humid heat source, liquid heat sink (with boiling). Majority sensible heating with some latent heat in source, boiling of liquid heat sink. A typical example of this would be recovery of waste heat from a dryer exhaust (with condensation) for use in a heat pump.

Again, a number of different heat exchangers are required to satisfy the varying design constraints present in each of the seven scenarios. Table 3.3 below summarises the heat exchangers considered.

Table 3.3. Summary of gas-liquid heat exchangers considered for inclusion in system knowledge base

Heat Exchanger	**Selected for system?**	**Suitable for which scenario?**
Gasketted Plate (Plate and Frame)	Yes	Vapour heat source, liquid heat sink; Vapour heat source, boiling liquid heat sink
Brazed Plate	Yes	Vapour heat source, liquid heat sink; Vapour heat source, boiling liquid heat sink
Welded Plate	Yes	Vapour heat source, liquid heat sink; Vapour heat source, boiling liquid heat sink
Plate and Shell	Yes	Vapour heat source, liquid heat sink; Vapour heat source, boiling liquid heat sink
Shell and Tube	Yes	Vapour heat source, liquid heat sink; Vapour heat source, boiling liquid heat

		sink
Finned-Tube	Yes	Gas (n/c) heat source, liquid heat sink; Humid Gas heat source, liquid heat sink, liquid heat source, gas heat sink; Gas (n/c) heat source, boiling liquid heat sink; Humid gas heat source, boiling liquid heat sink
Plate-Fin	No	N/A
Polymer	No	N/A
Spray Recuperator	No	N/A

3.1.3.1. Plate Heat Exchangers

The **Gasketted Plate**, **Brazed Plate**, **Welded Plate**, **Plate-and-Shell**, and **Shell-and-Tube** heat exchangers are all described in Section 3.1.2. Each of these units are also suitable for use in duties involving condensing vapours exchanging heat with liquids (with optional boiling). For the reasons explained in Section 3.1.2, they are included in the KBS equipment database.

3.1.3.2. Finned-Tube Heat Exchanger

The fined-tube heat exchanger is an extended surface heat exchanger designed specifically for duties where one of the fluids has a significantly smaller film heat transfer coefficient than the other. Hence it is commonly utilised in gas-liquid heat transfer as gas film coefficients are typically around a tenth of the value for liquids (Coulson and Richardson, 2005). A typical tube with radial fins is shown below in Figure 3.12 (Coulson and Richardson, 2005).

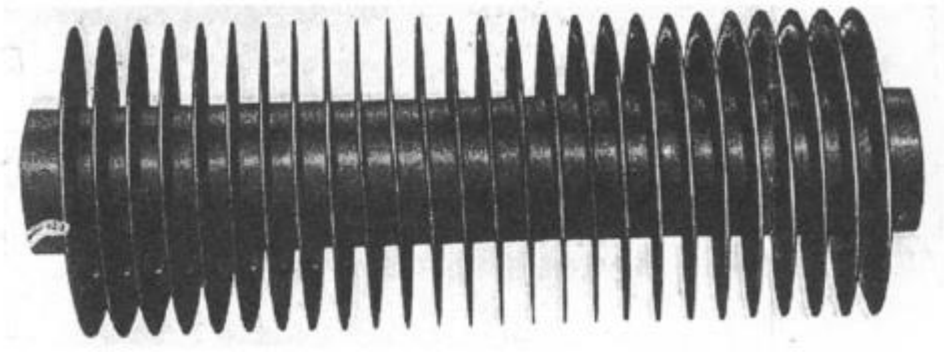


Figure 3.12. Radial finned-tube (Coulson and Richardson, 2005)

The finned-tube heat exchanger is commonly packaged in a shell-and-tube type configuration, and hence shares similarly high temperature and pressure limitations (> 260 ^{o}C and > 500 bar). The unit can also tolerate fouling due to the ability to remove the tube bundle for cleaning while protective coatings can be applied to the tubes to withstand any corrosivity concerns, particularly in duties such as economisers and condensing economisers.

Therefore, the finned-tube heat exchanger is seen as an ideal unit for gas-liquid duties where an extended surface is necessary due to its operational flexibility and is included in the KBS database. Furthermore, this unit is viewed as the industrial standard in extended surface heat transfer (Sinnott, 2005).

3.1.3.3. Plate-Fin Heat Exchanger

The plate-fin heat exchanger is a compact alternative to the finned-tube type in extended surface heat transfer. The unit is assembled from flat sheets and corrugated fins which are stacked and joined by brazing or diffusion bonding. This creates a strong physical structure capable of withstanding extreme temperature (cryogenic to 650 ^{o}C) and high pressure (greater than 200 bar) (Best Practice Programme, 2000). Figure 3.13 below shows the typical heat exchanger configuration.

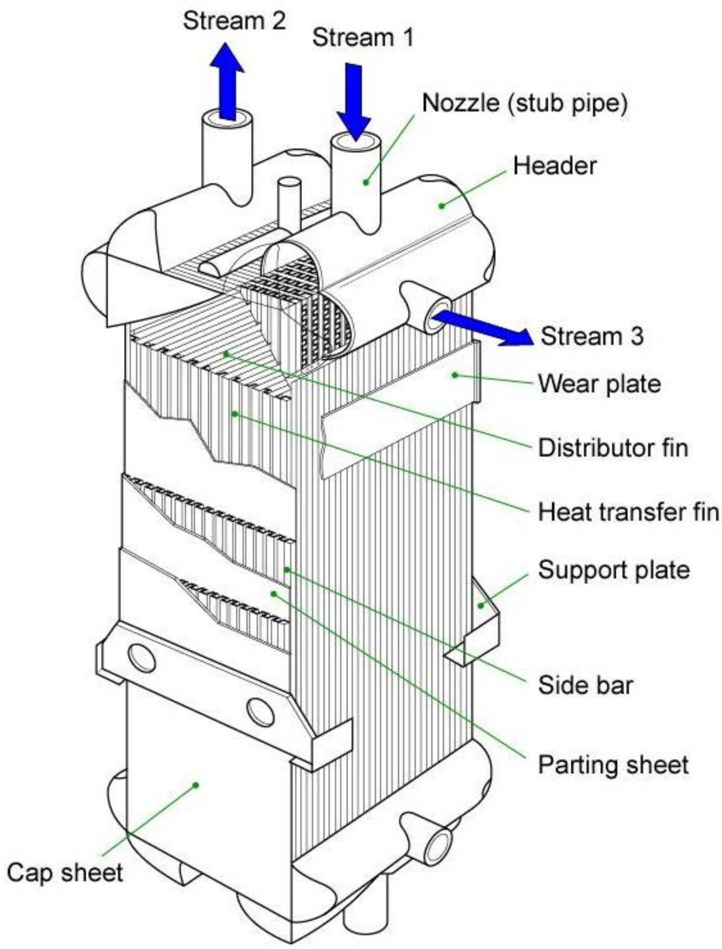


Figure 3.13. Plate-fin heat exchanger configuration (Best Practice Programme, 2000)

This unit is most commonly used in cryogenic applications, due to the tolerance to extremely low temperatures and high pressure. Other applications include offshore oil/gas platforms (due to the high surface area to volume ratio), fuel cells and high temperature heat recovery (such as gas turbine recuperators) (Best Practice Programme 2000; ALPEMA, 2013).

The high cost of the manufacturing process of this unit, and the bespoke design procedure, has led to only specialist application of this unit and it is not considered for general process industry duties. Furthermore, the unit is prone to fouling and relatively high pressure drops which is of further detriment. Therefore, it is not suitable for generic low-grade waste heat recovery and is excluded from the KBS database of technologies.

The **Polymer** heat exchanger is not considered for gas-liquid waste heat recovery for the reasons explained in Section 3.1.2.8.

3.1.3.4. Spray Recuperator

The spray recuperator is a type of direct contact heat exchanger designed for gas-liquid duties. The use of such a unit was suggested many years ago by Lyle (1947) to recover latent heat (and water) from evaporative processes for use as site wash water. The general configuration of the spray recuperator is a nozzle which sprays a fine mist of liquid (commonly water) into a stream of vapour (commonly steam). This is often done in a counter-current configuration with a packed bed to increase surface area. The resulting hot water then flows out of the unit.

However, no common design algorithms exist for the design of this heat exchanger and the number of parameters (including physical size of unit, nozzle design, vapour velocity, liquid velocity) creates a very complex design problem. Hence, no standard design equations or algorithms exist. Therefore, the spray recuperator is considered to be a custom build solution and out of scope of the KBS.

3.2. Heat Pumps

Heat pumps are an important technology allowing the upgrade of low-grade waste heat to a more useful temperature (according to temperature lift limitations). Heat pumps should be considered when no *matching* heat sources are available for heat transfer as they required a significantly higher capital expenditure than a simple heat exchanger system (Law, 2013). Various types of heat pumps are available to perform this task, as summarised in Table 3.4

Table 3.4. Summary of heat pump configurations considered for inclusion in system knowledge base

Name	**Brief Description**	**Selected for**

		System?
Vapour Compression Heat Pump	A reverse Rankine cycle where work is required to drive a compressor (most commonly using an electric drive) creating a temperature lift between the evaporator and condenser ends of the cycle	Yes
Absorption Heat Pump	Heat-driven heat pump most commonly used to provide cooling. Discussed in Section 3.4	No
Adsorption Heat Pump	Heat-driven heat pump most commonly used to provide cooling. Discussed in Section 3.4	No
Mechanical Vapour Recompression (MVR)	An open cycle compression heat pump where the vapour leaving an evaporative/distillation process is compressed and used as a heat source in the evaporator/reboiler	Yes
Thermal Vapour Recompression (TVR)	Similar to MVR only replacing the mechanical compressor with a thermal compressor (most commonly motive steam)	No

3.2.1. Vapour Compression Heat Pump

The (closed cycle) vapour compression (VC) heat pump acts as a reverse Rankine cycle where work is input to a compressor (most commonly from an electric drive) to create a temperature lift between the two heat exchangers of the cycle, as shown in Figure 3.14 below.

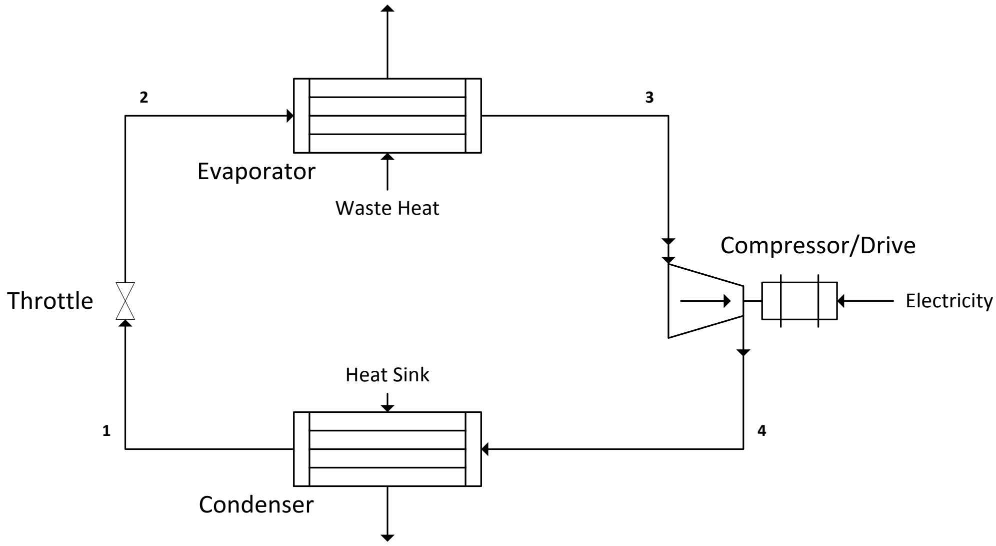


Figure 3.14. Standard vapour compression heat pump

Figure 3.14 shows the four basic components of the VC heat pump. The evaporator utilises waste heat to vaporise the working fluid at the low temperature/pressure end of the cycle. This low pressure vapour is then compressed to high pressure upon work input from the compressor/drive. The working fluid is then condensed at this higher pressure (and temperature) in the condenser, thereby heating the heat sink to a higher temperature than possible if one was to use a heat exchanger to directly transfer waste heat from the source to the sink. Finally, a throttle valve is then used to reduce the working fluid pressure and complete the cycle.

Heat pump performance is determined by the coefficient of performance (COP) which is defined as the heating duty provided by the compressor (Q_C, kW) divided by the work of the drive (W_{drive}, kW) as shown below.

$$COP = \frac{Q_C}{W_{drive}} \tag{3.1}$$

The COP determines both the economical and environmental impact of the heat pump. For example, if a heat pump is to be installed to replace a current gas heating duty, the COP must be greater than a) the ratio of the cost of electricity to gas, in order to be profitable and b) the ratio of the associated GHG emissions of grid electricity to gas in order to incur reductions in greenhouse gas emissions. The efficiency of the current heating system should also be considered, as should any government incentives for heat pump usage (such as the renewable heat incentive (HM Government, 2013)).

Therefore, the minimum COP in order to be profitable ($COP_{min,profit}$) and the minimum COP for GHG reductions ($COP_{min,GHG}$) can be defined as follows.

$$COP_{min,profit} = \frac{C_{elec}}{(C_{current}/\eta_{current}) + C_{Levy}} \tag{3.2}$$

$$COP_{min,GHG} = \frac{E_{elec}}{(E_{current}/\eta_{current})} \tag{3.3}$$

Note: C_{elec} (£/kWh) denotes the associated cost of grid electricity (to drive the motor); $C_{current}$ (£/kWh) denotes the cost of the utility *currently* used to heat the sink; $\eta_{current}$ denotes the efficiency of the current method of heating the sink; E_{elec} (tCO_2eq/kWh) denotes the associated emissions of grid electricity; $E_{current}$ (tCO_2eq/kWh) denotes the associated emissions of the current heating utility.

For a *typical* base case of using an electric drive VC heat pump to replace a current gas heating duty of 80% efficiency, the minimum required COP is as follows (assuming no government economic incentive).

Table 3.5. Minimum VC heat pump COP required for an economical and GHG reducing project

Cost Grid Electricity (June 2013) (£/kWh) [1]	0.0725
Cost Natural Gas (June 2013) (£/kWh) [1]	0.0237
Associated Emissions Grid Electricity (tCO_2eq/kWh) [2]	0.000525
Associated Emissions Natural Gas (tCO_2eq/kWh) [2]	0.000184
$COP_{min,profit}$	2.45
$COP_{min,GHG}$	2.29

[1]DECC, 2013 (c). [2]Carbon Trust,2012.

Table 3.5 shows that the minimum COP required for a VC heat pump waste heat recovery installation to be economically *and* environmentally favourable is 2.45.

However, this COP only represents the break-even point and therefore it is noted that a higher COP is favoured to incur lower project payback times.

A number of published case studies show successful installations of VC heat pumps achieving larger COP than noted above. For example, Midlands Counties Dairy Ltd (published by the Department of Energy, Energy Conservation Demonstration Projects Scheme, Profile 34, 1981) demonstrate the use of a heat pump to recover waste heat at 53 ^{o}C from a bottle sterilisation unit. The heat sink in this case was the water feed to the hot well which was heated to 70^{o}C. The project payback time was 2 years, while the COP was 5.40 which is significantly greater than the minimum required COP required for profit as stated above which would ensure similar payback periods would be expected in a similar modern installation.

One concern in comparing this data to modern day application is the change in working fluid legislation since the Montreal Protocol, 1989. The nature of the working fluid is not reported in this instance but a reasonable assumption is that a CFC or HCFC working fluid was employed such as R-114. However, numerous papers have been published with regards to new working fluids to replace (H)CFC's without detriment to the coefficient of performance. For example, Devotta (1995) discusses the feasibility of HFCs which show no reduction in COP, and the IEA Heat Pump Centre (IEA Heat Pump Centre, 2013) now recognises HFCs as the industry standard in heat pump working fluids. Other working fluids are also available, for example ammonia (Pearson, 2011). A full working fluid discussion is provided in Section 4.3.5.

Another interesting heat pump case study is provided by Smith, 1983. Here two heat pump units were installed in a dairy, both providing simultaneous useful heating and cooling. The first heat pump in the series recovered heat from the water treatment tank to produce cold water at the evaporator end, whilst preheating process water at the condenser end. This circuit produces an overall COP of 5.91. The second heat pump recovers heat from the chiller circuit at the evaporator end (again a useful cooling duty) whilst further heating the process water (from circuit 1) at the condenser end. The COP of circuit two is 5.09. A diagram of this system is shown below in Figure 3.15.

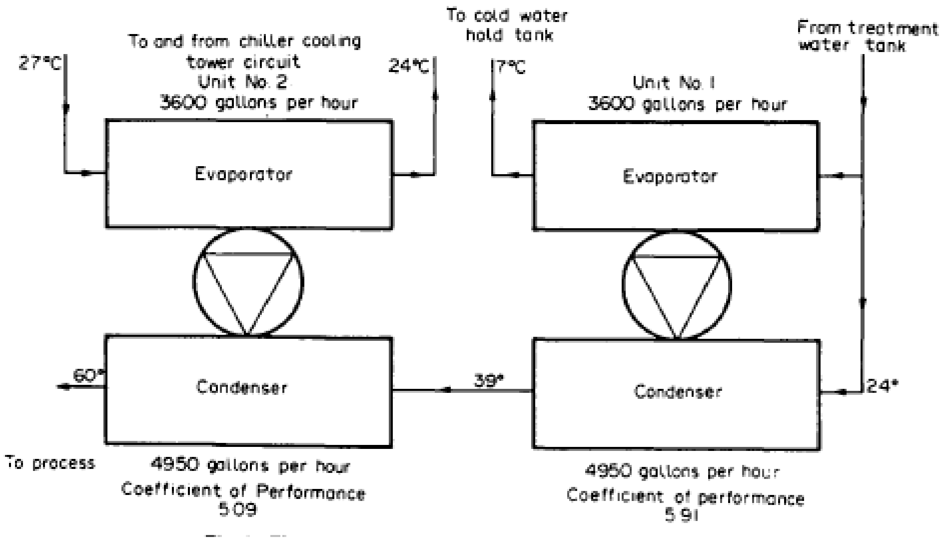


Figure 3.15. VC heat pump providing simultaneous heating and cooling (Smith, 1983)

The case study reports a project payback time of 2.7 years. No working fluid data are presented in the paper but it is a fair assumption that (H)CFC working fluids were utilised given the date of the publication. However, as stated above, a similar COP would be expected upon utilisation of modern working fluids such as HFCs. Therefore, the COP and economics reported in this study are assumed to still be valid.

Star Refrigeration, 2013, offer a modular heat pump solution using ammonia as a working fluid and a screw compressor. This unit has been demonstrated in a food processing factory (Star Refrigeration, 2010) to produce 1.25MW of heating and 3.20MW of cooling. Here, glycol solution was cooled in the evaporator from 0^{o}C to -5^{o}C for use in the plant refrigeration circuit while hot wash water was heated to 60^{o}C. This produces a combined COP of 6.25. The payback time is not presented in the published data, but it is assumed a system of such high COP would incur a low payback time. Associated GHG emissions are said to have been reduced by 199 tCO_2eq./year, a significant saving.

Despite the favourable economic, energetic and environmental results shown in the published data, UK industrialists are still not convinced of the benefits of heat pumps. This is summed up in the study by Sinclair, 2002 (as discussed in Section 1.2.3) where results show that 36% of UK engineers are “unsure” of the benefits of heat pumps in WHR or believe it to be a “risky” investment. It is hoped that the inclusion of VC heat pumps in the KBS presented in this thesis will encourage the uptake of such projects in industry and change widespread opinion.

In summary, VC Heat Pumps are selected for use in the system as they provide a waste heat recovery option by upgrading low-grade waste heat when a *matching* heat sink is not available at the original source temperature. Furthermore, published data confirms the economic and energetic validity of this technology in low-grade waste heat recovery and it is hoped that the inclusion in this software will encourage further installations.

3.2.2. Mechanical Vapour Recompression

Mechanical Vapour Recompression (MVR) is an open cycle variation of the traditional VC heat pump. Here, low-pressure vapour from an evaporative process is re-compressed via a mechanical compressor/drive to a higher pressure and then used as a heating medium in the process, thereby negating the need for an external utility at steady-state operation. A typical set-up of the MVR system is shown in Figure 3.16 (Lazzarin, 1993) with an external circulation heat exchanger, however other systems often employ internal heating coils. Note: A diesel engine is depicted as the drive in this system but it is more common to use an electric drive.

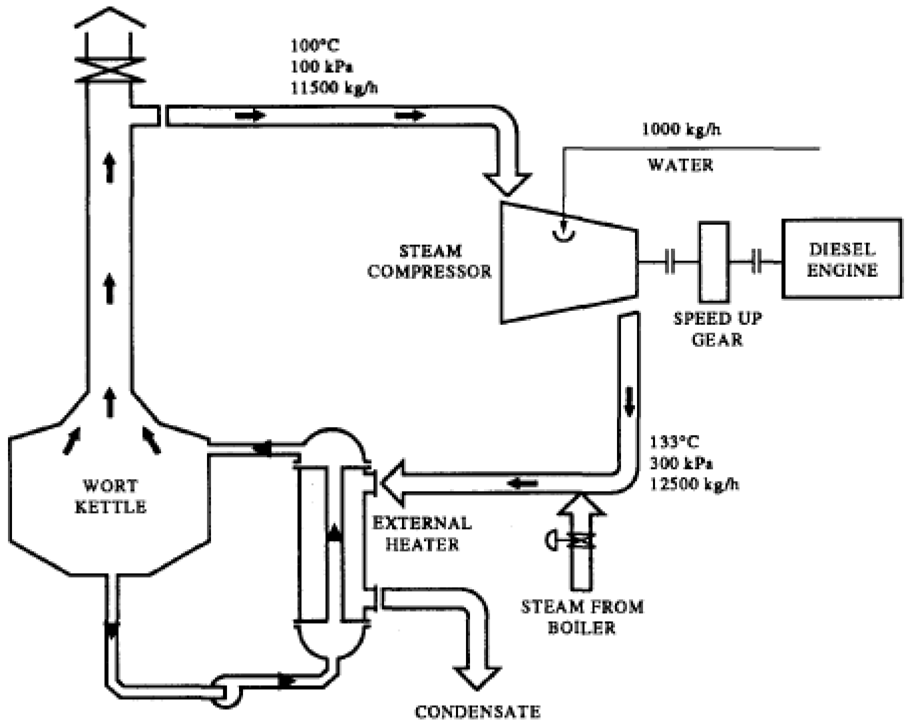


Figure 3.16. Typical MVR installation (Lazzarin, 1993)

The main components of an MVR system are noted as the compressor/drive and the heat exchanger, although this may be pre-existing. The COP is again used as a measure of performance and Equations 3.1-3.3 above, and data in Table 3.5 from Section 3.2.1 apply to MVR.

Two common scenarios are considered by MVR as follows:

- Water evaporative systems such as wort boiling (brewing) and processing of concentrated juices
- Organic fluid evaporative systems such as petrochemical distillation columns

In this work, only water evaporative systems are considered. Organic fluid systems are too wide ranging and a large chemical database would be required. This would probably require the use of an external chemical database, which in turn would require a license for use. This is a violation of the system scope which states that the software should be easy and free to disseminate into the industrial domain. The properties of water, however, can be programmed into the system with relative ease. Furthermore, steam-water evaporative systems are a lot more common than organic systems, particularly in the UK due to the prominent food processing sector and diminishing petrochemicals sectors.

Successful MVR projects have been demonstrated across a broad range of industries. Reported COPs generally far exceed the minimum COP (3.45) required to achieve a profit outlined in Section 3.2.1. For example, Staveley Chemicals Ltd (published in the Energy Efficiency Demonstration Scheme, Expanded Project Profile 259, 1989) installed an MVR system to provide the heating duty in a by-product evaporator. Here, payback time is reported as 4 years. However, it is noted that the plant experienced technical difficulties during commissioning as this was one of the first MVR plants installed in UK industry. Hence, the report states that a similar installation should in fact incur a payback time of 2.4 years. The COP of the system is not provided but can be calculated based on the data given as 22.1. This is significantly larger than the minimum COP currently required for a profitable system in the UK, hence a similar *new* installation would be anticipated to show a good economic performance and achieve a payback time in line with that outlined in the scope of the KBS presented here.

Wu *et al* (2013), investigated a novel MVR system for use in desalination. Here, the main investigation focused on the design of a novel evaporator-condenser heat exchanger to prevent scaling, which is a common problem in MVR desalination systems. However, the MVR methodology (excluding the novel heat exchanger equipment) is common to all types of system, with vapour simply leaving the evaporator at low pressure before returning to the heat exchanger at high pressure via the compressor. Therefore, the data on the COP achieved is of interest to this study. The data shows that a COP of around 18 can be achieved for a compression ratio of around 1.12 which results in a temperature difference in the heat exchanger of around 10^{o}C. Hence, the data shows that MVR systems can achieve a high COP, which would lead to favourable economic results and GHG emission-reductions for moderate temperature/pressure lifts, which could theoretically use relatively cheap, fan-like, compressors. Furthermore, the heat exchanger temperature difference of 10^oC is rather conservative and would not require the use of novel, close-approach temperature heat exchangers.

Despite the promising published data, MVR is not used on a large scale in UK industry. A number of opportunities are available, for example Brotherton, 2012, suggests the use of MVR in whiskey distillation; a concept that can also be applied to wort boiling in beer production. It is hoped that the inclusion of MVR technology in systems such as the KBS presented in this thesis will highlight its benefits and encourage industrial interest.

In summary, MVR is included in the system as it can provide a waste heat recovery for many evaporative processes used throughout the processing industries. The high COP achievable in MVR suggests that low payback times and high associated GHG reductions are highly likely with this technology but exact values are not reported in the literature.

3.2.3. Thermal Vapour Recompression

Thermal Vapour Recompression (TVR) works on a similar principle to MVR, but requires only a thermocompressor rather than a mechanical compressor to provide the temperature/pressure lift. This principle is depicted in Figure 3.17 below (GEA, 2013).

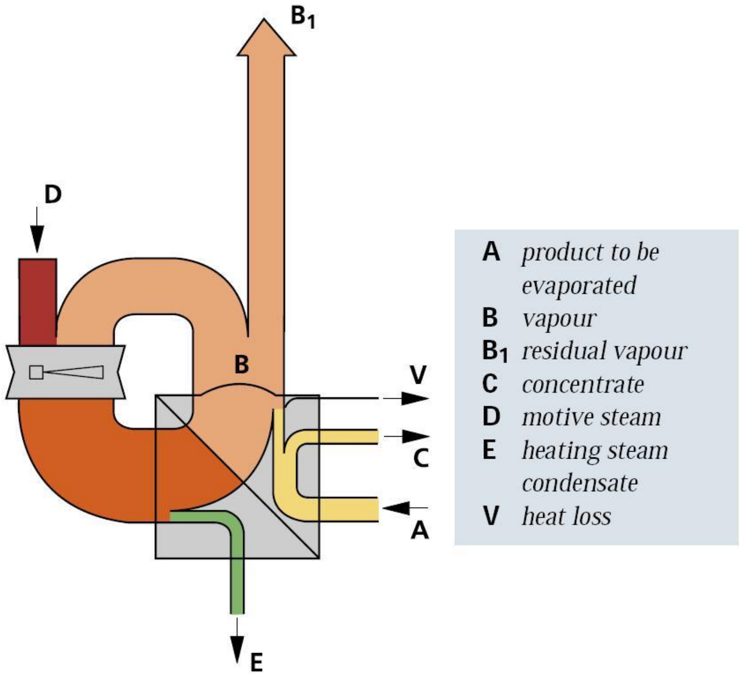


Figure 3.17. Thermal vapour recompression (GEA, 2013)

The advantage of TVR over MVR is that no moving parts are required, hence maintenance costs are assumed to be lower. However, TVR has two key disadvantages as follows:

- Motive steam is required to drive the thermocompressor. It cannot be assumed that all plants will produce such high pressure steam. Furthermore, it is assumed that any current boiler will have been designed according to the current steam demand. Any further steam demand will cause the boiler to operate outside of the design parameters and cause a decrease in efficiency
- Motive steam will be mixed with the evaporator vapour. This may cause a problem if the condensed vapour is the final product of the process (such as product dilution and/or contamination)

As one cannot assume that motive steam is readily available at all process sites, and that dilution of the evaporator vapour is acceptable, TVR is excluded from the KBS database. Furthermore, data from Brotherton, 2012, states that MVR incurs greater energy savings than TVR (80% vs 35%). Therefore, as they are generally used for the same purpose, MVR is the preferred technology.

3.3. Power Generation

Waste heat driven power generation is an increasingly attractive proposition for process plants due to the rising cost of electricity, as discussed in Section 1.1. A number of methods are available for waste-heat driven power generation as summarised in Table 3.6 below.

Table 3.6. Summary of power generation options considered for the system knowledge base

Name	Brief Description	Selected for System?
Steam Rankine Cycle	Common steam Rankine cycle used for electricity generation. Heat input required to raise steam which in turn drives a (series of) turbine(s).	No
Organic Rankine Cycle	Variation on the steam Rankine cycle using more volatile organic working fluids (rather than steam).	Yes
Kalina Cycle	Variation on the Rankine cycle using ammonia/water mixtures as the working fluid. A more complex cycle with more unit operations.	No
Thermoelectric Device	Solid state semi-conductor material which generates a voltage when a temperature	No

	difference is established between two opposite ends.	

3.3.1. Thermoelectric Device

Thermoelectric devices are solid state semi-conductor devices which generate a voltage when a temperature difference is established between two ends of the system. They are often referred to as *Peltier effect* devices due to the discovery of this phenomenon by J.C.A. Peltier in 1834. A schematic of a thermoelectric device is shown below in Figure 3.18, taken from Niu *et al* (2011).

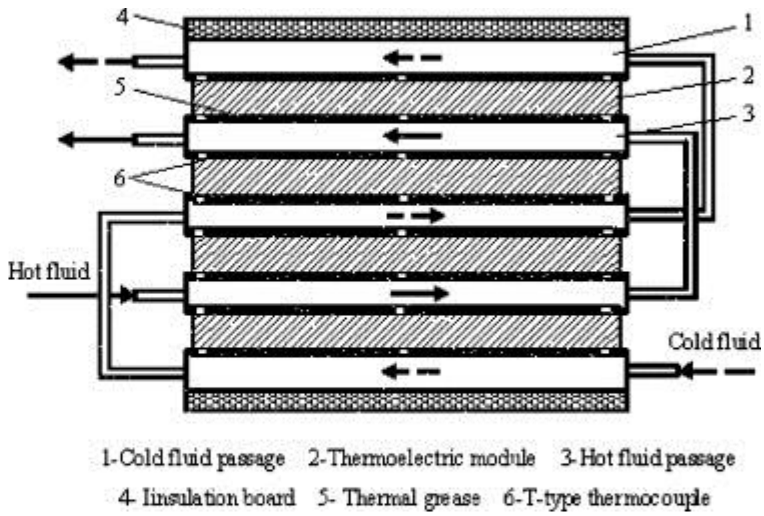


Figure 3.18. Thermoelectric device (Nie *et al,* 2011)

The most common application of the thermoelectric device is in refrigeration, whereby a current is introduced to induce a temperature difference between the two ends of the system (*i.e.* the reverse of thermoelectric power generation). Examples of this are numerous (such as Brown and Rabb, 1965, Lindenblad, 1958 and Thermovonics Co Ltd, 1993) and the technology is now commercially available for use in domestic refrigeration.

Riffat and Ma (2003) noted the main advantages of the thermoelectric device compared to other methods of power generation as follows (note: advantages relating to thermoelectric refrigeration are excluded):

1. Thermoelectric devices have no moving parts and, therefore, need substantially less maintenance [than power cycles]
2. Life testing has shown the capability of thermoelectricity devices to exceed 100,000 hours of steady-state operation
3. Thermoelectric devices contain no chlorofluorocarbons or other materials that may require periodic replenishment (Note: this paper was published 10 years ago. Chlorofluorocarbons are no longer considered in refrigeration or power cycles. However, this argument still applies to "*modern*" working fluids such as HFCs which may be harmful to the environment due to their high global warming potential)
4. Thermoelectric devices can function in environments that are too severe, too sensitive or too small for conventional refrigeration [or power cycles]
5. Thermoelectric devices are not position-dependent

The main drawback of thermoelectric devices, however, is low efficiency. Rowe and Min (1998) report typical efficiency for "High Power (hundreds of Watts to megawatts)" of 4.4%. This efficiency is similar to the reported efficiency of 4.5% of the HZ-20 module, commercially supplied by Hi-Z technology (2013). The HZ-20 figures are quoted for a continuous source temperature of 250^{o}C. An organic Rankine cycle, for example, operating at a similar heat source temperature is reported to have an efficiency of 15% (Wang *et al*, 2012). This highlights the advantage of power cycles over thermoelectric devices.

Furthermore, the economics of thermoelectric devices are not favourable. Hi-Z technology quote the HZ-20 model as available for $125.00 (approx £82.00 based on June 2013 exchange rate of $1 = £0.66) excluding other essential components such as insulating wafers ($7.00 per unit), thermal grease ($45.00 per 20 grams) and heat sink components ($100-140.00 per unit). The total cost would therefore be approximately \$250.00 (£164.00) per unit excluding installation costs. The HZ-20 model produces approximately 19W per unit.

The number of units (N, $kWh.year^{-1}unit^{-1}$) of power generated per annum per HZ-20 unit is calculated using eqn. 3.4 below (based on 8000 hours of operation per year, h).

$$N = P * h \tag{3.4}$$

$$N = 0.019 * 8000 = 162.5\ kWh.year^{-1}unit^{-1} \tag{3.5}$$

Based on the current cost of electricity in the UK of £0.0725/kWh (DECC (c), 2013) each unit would incur cost savings in utility bills of £11.78/year. Hence, the payback time per unit would be in the region of 14 years, excluding installation and maintenance costs.

Therefore, thermoelectric devices are excluded from the knowledge-based system database as they are out of the scope of the system as they do not provide an economically viable option for waste heat recovery.

However, research into these devices is ongoing and they do have certain niche applications. For example, Spirax Sarco (Miller, 2013) report the development of a steam dryness sensor powered by a thermoelectric generator. The thermoelectric device is advantageous in this case study as the sensor in question only requires a small power input (order of 10W). The thermoelectric device is ideal for such applications, as it can comfortably supply this magnitude of power in remote plant locations without the need for regular maintenance. It is expected that this is to be the future niche area of the thermoelectric device, rather than larger scale (order of 100kW+) waste heat driven power generation, unless there are significant breakthroughs in cost, materials and/or efficiency.

3.3.2. Steam and Organic Rankine Cycle

The steam Rankine cycle (SRC) is excluded from the system in favour of the organic Rankine cycle (ORC). The ORC has a number of advantages over the SRC when utilising low-grade waste heat, as discussed by Tchanche *et al* (2011). Table 3.7 below provides a comparison between the two cycles.

Table 3.7. Comparison of steam and organic working fluid properties (adapted from Tchanche *et al,* 2011)

	Steam cycle	**Organic Rankine cycle**
Fluid	Water	Organic compound
Critical pressure	High	Low
Critical temperature	High	Low
Boiling point	High	Low
Condensing pressure	Low	Acceptable
Specific heat	High	Low
Viscosity	Low	Relatively high
Flammability	No	Yes (fluid dependant)
Toxicity	No	Yes (fluid dependant)
Environmental impact	No	High (fluid dependant)
Availability	Available	Supply problem (fluid dependant)
Cost	Cheap	Expensive

Table 3.7 shows that the ORC has an advantage over the SRC when utilising low temperature heat sources as the available fluids are more volatile leading to higher vapour pressures in the low-grade temperature range. Also, many organic working

fluids are classified as “dry” or “isentropic” meaning that dry turbine outlets can be achieved with minimal (or zero) superheat. In the case of the steam Rankine cycle, a large degree of superheat is required to ensure a dry turbine outlet which would not be possible when utilising a low-temperature finite waste heat source. Figure 3.19 (Tchanche *et al*, 2011) shows the T-S for steam and various common working fluids highlighting the difference in nature.

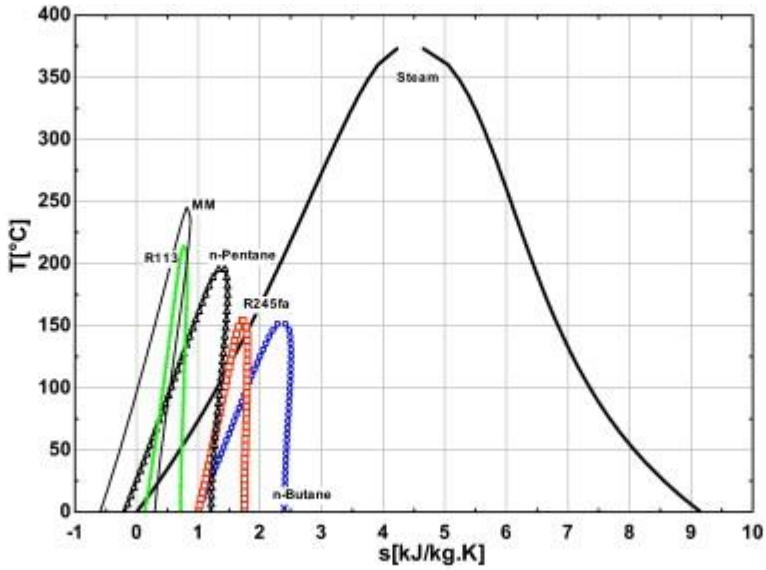


Figure 3.19. Temperature-entropy plot for steam and several organic working fluids (Tchanche *et al*, 2011)

A number of ORC configurations are suggested in literature, but all systems contain the four key components (the working fluid pump, the pre-heater/evaporator, the turbine/generator and the condenser) as set out in the conventional ORC shown in Figure 3.20.

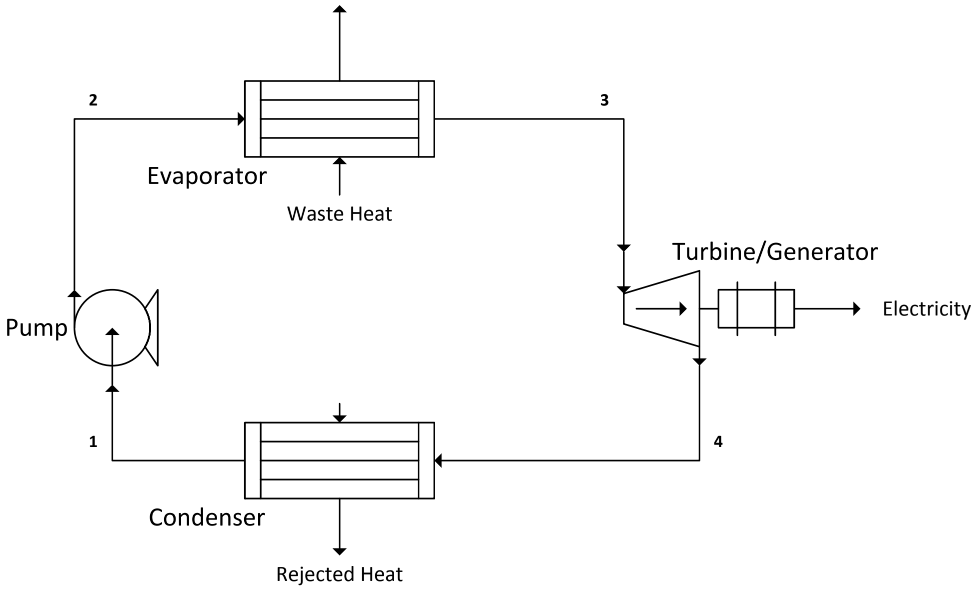


Figure 3.20. Conventional organic Rankine cycle

Other configurations of ORC are possible, including the recuperative cycle and the superheated cycle. The recuperative cycle employs an internal heat exchanger to recover sensible heat from the turbine outlet (labelled 4 on Figure 3.20) to pre-heat the evaporator inlet (labelled 2 on Figure 3.20). The superheated cycle includes a super heater prior to the turbine inlet. However, the properties of organic working fluids (as previously discussed) are such that these additions are not advantageous overall. Further discussion on ORC configuration is provided in Section 4.4.4.

The ORC has been shown to have great potential for low-grade waste heat recovery in a number of modelling studies. For example, Aneke *et al* (2012) investigated the feasibility of an organic Rankine cycle in utilising waste heat from two low-grade waste heat sources in food processing. Here, a thermal efficiency of 16% was reported when utilising a single gaseous heat source with an inlet temperature of 164^{o} C. However, it is noted that the model only allows a 4^{o} C rise in the cooling water (heat sink). This leads to a high required heat sink flow rate of 60kg/s which may not be feasible at the site in question (the author does not comment on this). Therefore, if a lower heat sink flow rate was required, the temperature rise would be larger which in turn would reduce the turbine pressure ratio and the net power output of the cycle.

This case study is further explored in the PhD thesis by Aneke (2012). Here a full economic and environmental analysis (including sensitivity analysis) of the proposed ORC system is presented. The payback period reported is dependent on a number of factors (forecast electricity cost, forecast carbon taxes) but the study shows that typical payback periods of around three years could be expected. This is acceptable for the KBS database.

Another ORC case study example is presented by Law *et al* (2013) in which a comparison of an ORC and a high temperature heat pump is presented for waste heat recovery at an inorganic chemicals site. Here the heat source was the humid exhaust from a spray dryer with a dew point of 90^{o} C. The results show a cycle thermal efficiency of 12.8% and a potential payback time in the region of 3.5 years (excluding any forecast carbon taxes). This data is, again, in line with the requirements for the programme database of technologies.

Reported installations of ORCs are scarce despite the promise highlighted in modelling studies. In the UK only one published example of ORC utilisation is reported (DRD Power, 2011). The company has recently trialled a 200kWe unit at a chemical industry site in the North East of England. Only limited data have been released and no details are revealed regarding working fluid or cycle configuration. However, cycle thermal efficiency of around 15% is reported and the typical payback period is quoted as "*around 3 years*" while the overall trial was deemed "*successful*". The published data are similar to those reported in the modelling studies and confirm the feasibility of organic Rankine cycles for industrial waste heat recovery.

Therefore the ORC is deemed a suitable technology for inclusion in the system knowledge base as it satisfies the criteria set out in the scope of the system (Section 1.3.1). It is hoped that the inclusion of the ORC in the system will increase industrial awareness of waste-heat driven ORC technology and encourage further installations in the UK process industries.

3.3.3. Kalina Cycle

The Kalina Cycle is a variation on the Rankine cycle using a water-ammonia mixture as a working fluid. The basic Rankine-type cycle has been adapted to include a separation column and an absorber, as shown in Figure 3.21 below (adapted from Rotunds, 2013).

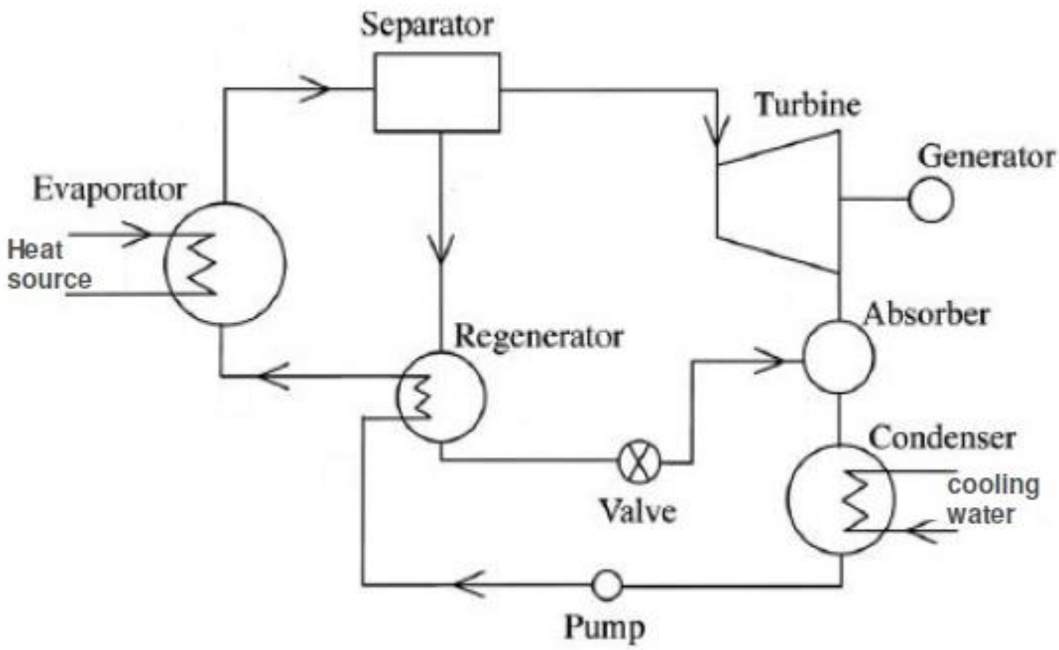


Figure 3.21. Kalina cycle (adapted from Rotunds, 2013)

The Kalina cycle has two advantages over the (organic) Rankine cycle. Firstly, the boiling point of the working fluid mixture is not isothermal, which allows a greater degree of heat recovery as, in a counter-current arrangement, the heat source can be cooled to a lower temperature. Secondly, the ammonia concentration of the working fluid at the condenser may be varied according to seasonal temperature variation in the heat sink (most commonly air or water-cooled) (Mlcak, 1996). Such advantages have led to claims of an increase in power output of up to 20% (compared to organic Rankine cycle).

However, other studies claim that this increase is significantly smaller. For example, DiPippo, 2004, provides a theoretical second law assessment of binary power plants utilizing low-grade geothermal fluids (heat source 130 ^{o}C). Here, the results show an increase of only 3% in the net power output. Bombarda *et al* (2010), found an increase in net power output of less than 1% for in a comparison of organic Rankine cycle and the Kalina cycle for waste heat recovery from a diesel engine and conclude that the increase in net power output does not compensate for the increased complexity of the cycle.

Singh and Kaushik, 2013, present a theoretical optimisation of a Kalina cycle to recover waste heat from the flue gas of coal fired power plant. Here the heat source temperature was 134^{o}C and the optimised thermal efficiency is reported as 12.95%. In the paper, both the turbine and working fluid pump are assumed to be isentropic, while no generator or motor efficiencies are included in the model. Hence, the predicted thermal efficiency would be considerably lower in reality. It would therefore be expected that an ORC could achieve a similar thermal efficiency at such a source temperature, as discussed in Section 3.2.2.

The Kalina cycle is yet to be implemented on a large scale, with less than 5 case studies published (none of which are in the UK). This suggests that the technology is not an established method of WHR and therefore not in agreement with the scope of the KBS (in particular, point 8).

In summary, the Kalina cycle is a significantly more complex cycle than the ORC with limited (often less than 3%) improvements in net power output. The increased cycle complexity will lead to higher capital and maintenance costs which are not justifiably compensated by the relatively small potential increase in net power output. Furthermore, the Kalina cycle cannot be considered an established method of WHR in the process industries. Therefore, the Kalina cycle is excluded from the KBS equipment database.

3.4. Generation of Coolth

Industrial refrigeration is most commonly achieved using vapour compression systems, akin to the vapour compression heat pumps described in Section 3.2.1 albeit at a lower operating temperature. Such systems have been proven to achieve high COPs of greater than 4 (ETSU, 2001).

The most common waste heat driven refrigeration systems are summarised in Table 3.8.

Table 3.8. Summary of coolth generation options considered for the system knowledge base

Name	Brief Description	Selected for System?
Absorption Refrigeration	A temperature lift is achieved between the cycle evaporator and condenser via a (waste) heat driven absorber/desorber (regenerator) system. The pressure difference is provided by a liquid phase pump.	No
Adsorption Refrigeration	Similar to the absorption cycle, only utilising a solid adsorbent rather than a liquid absorbent	No
ORC-Coupled VC Refrigeration	Expander from organic Rankine cycle is used to drive the compressor in a standard VC refrigeration system	No

3.4.1. Absorption Refrigeration

The absorption heat pump is a heat driven alternative to the vapour compression heat pump. The absorption cycle utilises two fluids in a mixture: the refrigerant and the absorber. The most common pairs are water-lithium bromide and ammonia-water with the latter preferred when refrigeration is required at sub-zero (< 0^{o}C) temperatures due to the lower freeze point of ammonia (-77.7 as opposed to 0^{o}C). A standard cycle schematic is shown in Figure 3.22 below (adapted from Herold and Radermacher *et al*, 1996).

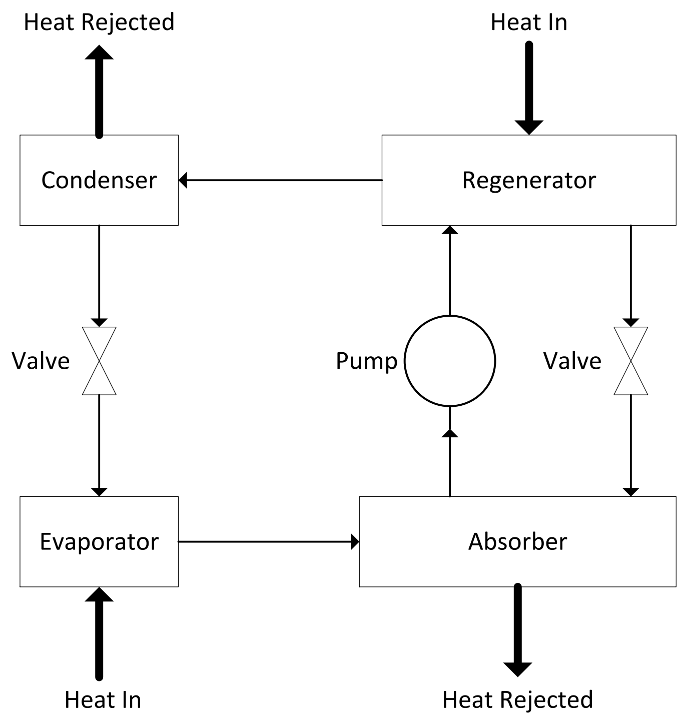


Figure 3.22. Single-effect absorption cycle (adapted from Herold and Radermacher *et al*, 2996)

One half of the absorption heat pump behaves identically to the vapour compression heat pump, with the high pressure refrigerant vapour being condensed in the condenser, before being expanded and evaporated in the evaporator. The difference is shown on the right hand side of Figure 3.22 where the combination of the regenerator, absorber, pump and a further valve operates in a heat driven process to replace the work-driven compressor of the VC heat pump.

Here, the evaporated refrigerant is absorbed back into a carrier liquid before being pumped to high pressure and entering the regenerator. Heat is required to evaporate the refrigerant from mixture which in turn travels to the condenser, whilst the refrigerant-lean liquid returns to the absorber via an expansion valve.

Here, the high pressure required in the condenser end of the cycle is provided by a liquid pump rather than a vapour compressor which requires a relatively negligible work input. Therein lies the advantage of this cycle over the standard VC heat pump/refrigerator and this cycle would be recommended for use where the electricity supply is unreliable or there is a substantial quantity of waste heat in the correct temperature range for utilisation in the regenerator. Hence, the absorption refrigeration cycle is often used in tandem with domestic/commercially-sized combined heat and power systems (CHP) to produce coolth for air conditioning during the summer months (for example Tassou *et al*, 2007, and Minciuc *et al*, 2003).

This heat pump may be utilised according to the following two scenarios in industrial low-grade waste heat recovery:

- As a refrigeration unit: heat to evaporator provides refrigeration effect, waste heat is used to drive the regenerator, condenser heat is expelled is to cooling water/air
- As a heat pump: heat source provides the heat to the evaporator, an external heating utility such as steam or gas is used to drive the regenerator, useful heat is expelled from the condenser to an identified heat sink

The main drawback of the absorption heat pump is the low COP achieved and the high capital cost. Typical COPs range from 0.5-1.2 depending on the temperature lift and the generator temperature, although this can increase with the introduction of double and triple effect systems (at greater capital expenditure) (ETSU, 1999). Also, in the case of absorption chillers, it may be seen as unreliable to rely on a waste heat source to drive a refrigeration system as often the flow of heat sources may be interrupted while refrigeration is generally required constantly (particularly in the food industry).

The economics of absorption heat pumps/chillers is summed up in a case study by ETSU 1999. Here, an absorption heat pump driven by waste heat at 115^{o}C was compared to a conventional vapour compression refrigeration system. The COP is given as 0.68 and 4.5 for the absorption and conventional cycles respectively, with a brine outlet temperature of 7^{o}C. The total capital cost for the absorption chiller is given as £150,000 while the savings in running costs between the two cycles is around £10,000. Hence, the payback period for a retrofit system (as assumed in waste heat recovery system design) would be 15 years. Such high pay back times are in violation of the scope of the KBS, particularly point 9. Hence, absorption refrigeration systems are excluded from the system knowledge-base.

However, it is noted that absorption systems can provide a useful solution for refrigeration circuits, particularly when coupled with CHP systems although the economics may only be suitable in new build scenarios. They have also provided a useful solution in gas turbine power units by providing compressor inlet air cooling using waste heat from the exhaust (for example Habeebullah *et al,* 1998, and Najjar, 1996).

3.4.2. Adsorption Refrigeration

The adsorption cycle operates on a similar principle to the absorption cycle only utilizing a solid adsorbant rather than a liquid absorbant. Suggested adsorbant-refrigeration pairs include activated carbon-ammonia, silica gel-water and zeolite-water (Wang *et al*, 2009). The typical system schematic is shown below in Figure 3.23 (Wang *et al*, 2012).

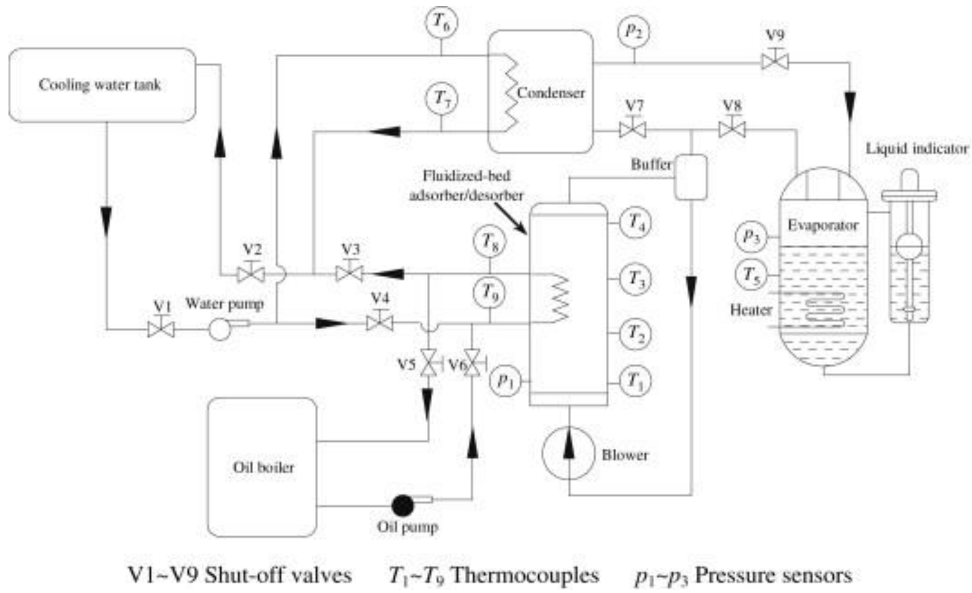


Figure 3.23. Fluidised bed adsorption refrigeration system (Wang *et al*, 2012)

One key difference between the absorption and adsorption cycle, as highlighted by Figure 3.23, is that the adsorption and desorption (regeneration) processes occur in the same unit operation by incremental heating, cooling and evacuation of the adsorbant chamber (in this case in a fluidised bed configuration). This has led to many papers attempting to optimise the design of the adsorber/desorber, for example the fluidised bed shown above and the plate-type system suggested by Critoph and Metcalf, 2004, below.

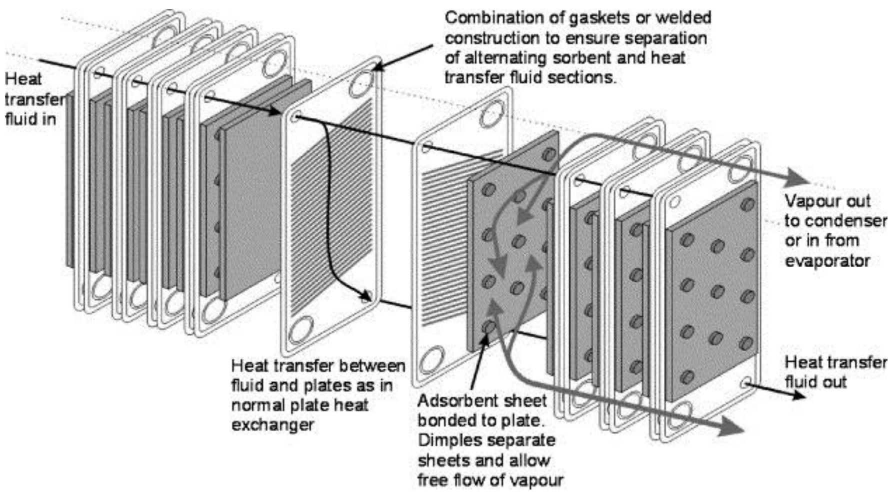


Figure 3.24. Plate-type adsorber/desorber (Critoph and Metcalf, 2004)

Adsorption refrigeration is excluded from the KBS database as the technology requires a lot of further research before it is to become widely commercially accepted. Issues must be addressed such as poor heat and mass transfer in the adsorber/desorber and adsorbant deterioation. This is summed up in the review paper by Wang, 2010, in which it is stated "*it can be predicted that adsorption refrigeration will not be used as popularly as the conventional absorption and vapour compression refrigeration in the near future if these problems are not completely resolved*". Therefore, this technology is in violation of the scope of the KBS, particularly point 8.

3.4.3. ORC-Coupled VC Refrigeration

A number of authors have suggested the use of an ORC to drive the compressors in a conventional vapour compression refrigeration system, as shown below in Figure 3.25 (Wang *et al*, 2011).

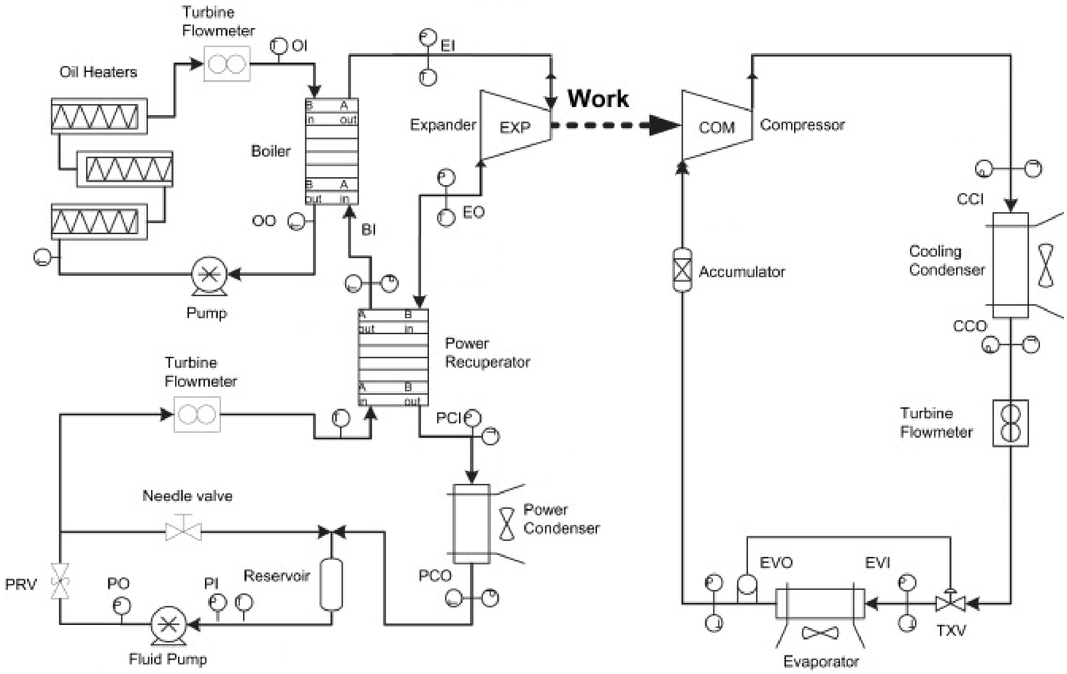


Figure 3.25. Schematic of ORC-VC refrigeration system (Wang *et al*, 2011)

Both Aphornratana and Sriveerakul (2010), and Li *et al* (2013) have modeled this system using a variety of working fluids and shown favourable results. The former found that when using R134a heat sources as low as 60^{o}C may be utilized while cooling temperatures as low as -10^{o}C are possible with COP ranging from 0.1 to 0.6. The latter used n-Butane and achieved a COP of 0.47 for a boiler exit temperature of 90^{o}C (which would correspond to a liquid heat source in region of 120^{o}C, or low-grade flash steam) and a cooling temperature of 5^{o}C.

While such papers show that this technology is feasible, this variation on the ORC is not included in the system knowledge-base. For a retro-fit case, it would be extremely complex to design such a system to "*drop-in*" to the existing plant refrigeration circuit. Hence, this must be seen as a bespoke solution to waste heat recovery which may only be possible in the case of a new build. Therefore, this is outside the scope of the system.

However, it is noted that standard ORC cycles are included in the system knowledge-base and the design results include the work produced and the electrical power generated. The user is free to investigate the use of the work produced to drive current refrigeration compressors (if feasible on the site in question) or the user may choose to offset the electricity generated by the ORC against the current VC refrigeration system.

3.5. Waste Water Treatment

Waste-heat driven water purification is of growing interest, particularly in areas where supplies of fresh water are limited or in processes producing large amounts of waste water (which may be costly to dispose of).

A number of heat-driven techniques are available for waste-water treatment and desalination. Two of the most commonly utilized techniques are discussed here, as summarised in Table 3.9 below.

Table 3.9. Summary of waste water treatment techniques considered for inclusion in the system knowledge base

Name	**Brief Description**	**Selected for System?**
Multiple-effect-desalination	Waste heat drives the initial evaporation stage while each of the proceeding stages are held at lower pressure and are heated by vapour from the previous stage	No
ORC-driven reverse osmosis	Waste heat drives an ORC which drives the high pressure pump in a reverse osmosis unit. Water molecules are transferred over a membrane hence separating fresh water from brine.	No

3.5.1. Multiple-Effect-Desalination

Multiple-effect-desalination (MED) is the most widely utilized method of waste heat driven water purification techniques which is operated as a common multiple-effect-evaporation process, as shown below in Figure 3.26 (Van der Bruggen and Vandecasteele, 2002).

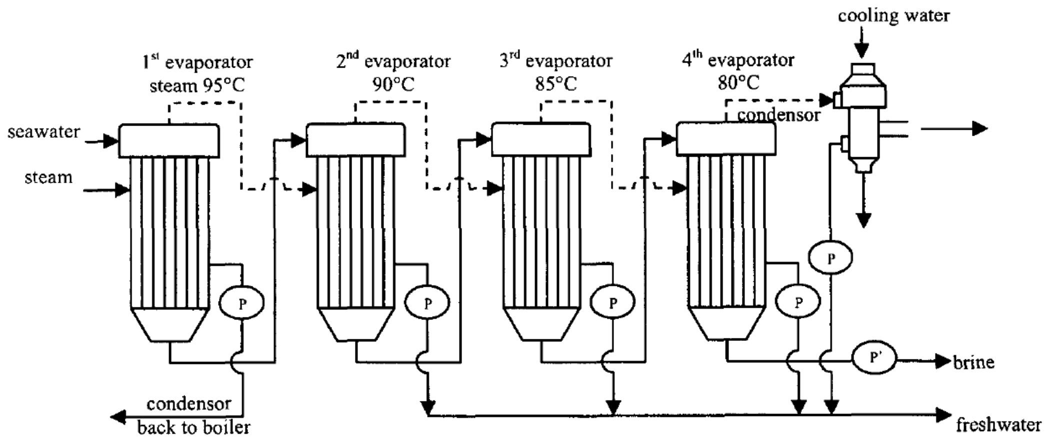


Figure 3.26. Waste heat driven MED (Van der Bruggen and Vandecasteele, 2002)

Here, steam is required in the first effect to evaporate fresh water from the brackish water. In a waste heat driven system, this may be provided by waste flash steam if available at around 2 bar (temperature must be in region of 120^{o}C) or more commonly via steam from a waste heat boiler. The vapour from each effect is then used to drive evaporation in the next effect which is held at lower pressure. Hence, a highly concentrated brine is produced following the final stage, while fresh water is condensed in each stage.

However, many problems are associated with this process including the corrosion of columns and heat transfer surfaces from the brackish water. Therefore, custom designs are required for each individual case depending on the temperature of waste heat available, the salinity of the brackish water and the space available on site (this limits the number of effects possible). Hence, the design of such a system for a plant retrofit (as assumed in the waste heat recovery case studies for the KBS) is a complex procedure, requiring a number of iterations and site visits. For this reason, MED is out of the scope of the KBS and excluded from the equipment database.

3.5.2. ORC-Driven Reverse Osmosis

Reverse osmosis (RO) is the most commonly utilized water purification technique, as highlighted by Figure 3.27 (Fritzmann *et al*, 2007).

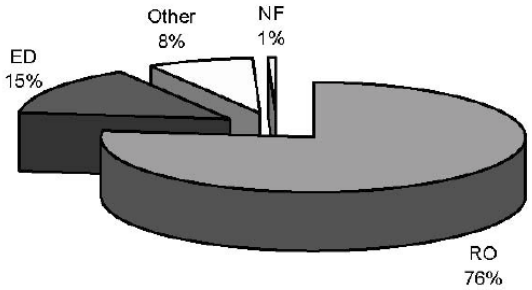


Figure 3.27. Market share of the different desalination techniques for brackish water (Fritzmann et al, 2007)

Reverse osmosis operates by pumping water to high pressure through a membrane. This forces water molecules through the membrane, thereby separating fresh water from a highly concentrated brine solution. Hence, this is not a heat driven process.

However, many have suggested the coupling of RO to an ORC machine. For example, Li *et al*, 2013 (see Figure 3.28 below) and Penate and Garcia-Rodriguez (2012). Therefore, the ORC-RO unit can be seen as an indirect waste-heat driven waste-water treatment solution.

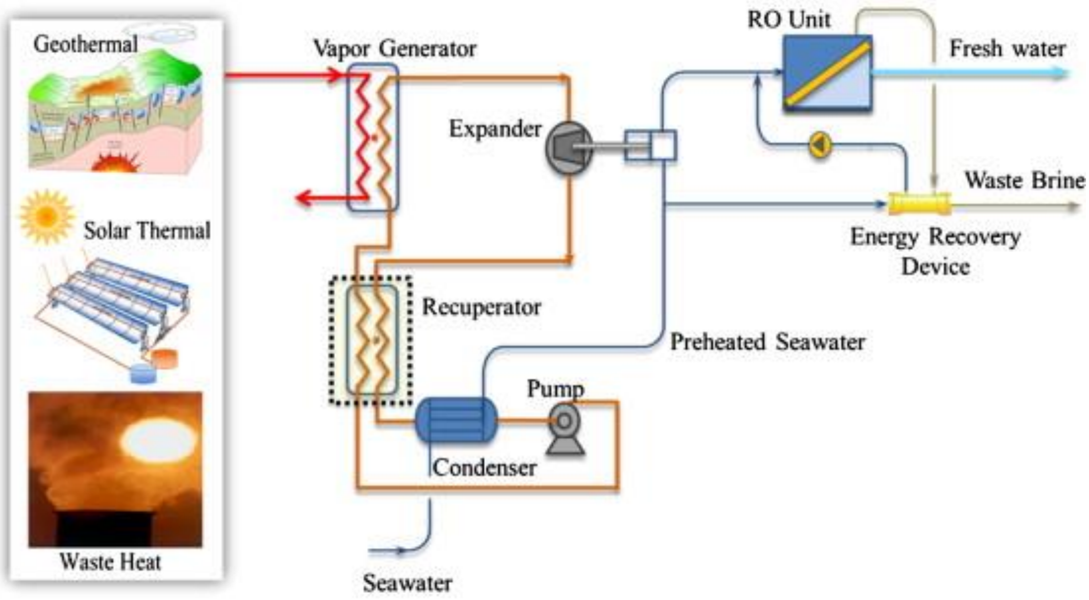


Figure 3.28. ORC-RO system schematic (Penate and Garcia-Rodriguez, 2012)

This particular variation on the ORC system is not included in the KBS as reverse osmosis is considered outside the scope of the system. The indirect nature of the waste heat recovery to power an RO unit is considered a custom solution and not currently widely accepted (despite a number of research papers investigating this concept).

However, as the standard ORC cycle is included, the results of ORC selection and design may be interpreted by the user towards use in a RO plant. For example, the work generated is a result of the ORC design module. Therefore, the user is free to utilise this work as appropriate for the site in question whether it be for electricity generation (this result is also stated) or for use in RO.

3.6. Chapter Conclusions

The state of the art in waste heat recovery technologies has been discussed and the most appropriate methods are chosen for inclusion in the equipment data base according to the scope of the KBS.

Technologies have been selected to cover a broad range of expected case-study scenarios. In brief, the technologies included cover the five categories of waste heat recovery options, as follows:

1. Heat exchanger heat transfer: a number of heat exchangers are included that cover all common scenarios
2. Heat pumps (heating): vapour compression heat pumps are considered where a heat sink is available within a suitable temperature lift of the heat source. A number of working fluids are incorporated into the database to suit the constraints of individual plants, considering factors such as various heat source/sink temperatures, and health and safety requirements (as discussed in Section 4.3.5).
3. Power generation: organic Rankine cycles are included for waste-heat power generation when no appropriate heat sinks are available. A number of working fluids are incorporated into the database to suit the constraints of individual plants, considering factors such as various heat source/sink temperatures, and health and safety requirements (as discussed in Section 4.3.7)
4. Generation of coolth: absorption and adsorption units were deemed outside the scope of the system for the reasons discussed. However, the user may choose to drive the existing VC refrigeration compressors using work from an ORC or offset the power consumption with that generated by an ORC
5. Waste water treatment: multiple effect desalination systems are deemed a bespoke solution and thus outside the scope of the system for the reasons discussed. However, the user may choose to drive an existing RO unit using work from an ORC or offset the power consumption with that generated by an ORC

Tables 3.10-3.12 overleaf provides a summary of each selected technology, including the operating limitations. Table 3.10 summarises the heat exchangers, Table 3.11 summarises the heat pumps and Table 3.12 summarises organic Rankine cycles.

Table 3.10. Summary of the heat exchangers included in system knowledge base (data adapted from various sources)

Technology	Max. temperature	Max. pressure	Typical materials of construction	Phases	Access for cleaning?	Corrosion resistance	Fouling considerations	Max Viscosity	Max Solid Particle Size	Cross contamination considerations
	°C	bar						CP	mm	
Brazed plate	225	30	Stainless steel, titanium; copper brazing	Liquid, liquid boiling, condensing vapour	No: fully brazed	Good: via coatings	Cannot accommodate solid particles in feeds	1000	N/A	No issue
Finned-tube	>260; Fin-side must not exceed 200	Shell 300; Tubes 1400	Stainless steel, titanium; fins aluminium or copper; shell may be in carbon steel; many others	Tube-side: Liquid, liquid boiling, condensing vapour Shell-side: Gas, humid gas	Yes: on tube side (remove bundle to clean)	Good: via coatings	Can accommodate fouling fluids on the tube-side	3000	15	No issue
Gas-gas plate	150 (aluminium); >260 (stainless steel)	16 Max. pressure difference between streams: 1.05	Stainless steel, aluminium	Gas, humid-gas condensation can be tolerated via use of drip tray	Yes: via gaskets	Poor: not commonly coated to prevent corrosion	Can accommodate light fouling on both sides	N/A (gas phase only)	N/A	No issue
Gasketted plate	180	16	Stainless steel, titanium	Liquid, liquid boiling, condensing vapour	Yes: via gaskets	Good: via coatings	Can accommodate light fouling on both sides	1000	2	No issue
Plate and shell	>260	100	Stainless steel, titanium; shell may be in carbon steel	Liquid, liquid boiling, condensing vapour	No: fully welded	Good: via coatings	Cannot accommodate solid particles in feeds	8	1	No issue
Rotary regenerator	>260	Normally around ambient. Max. pressure	Aluminium, Ceramics, Polymers	Gas	Yes, although can be configured to promote self-cleaning	Good: can be manufactured by a variety of materials	Can tolerate light fouling as can be configured to promote self-cleaning	N/A (gas phase only)	N/A	Up to 5%

		differential between streams: 1.06								
Run-around-coil	200; Fins must not exceed 200	75	Stainless steel; aluminium or copper fins	Gas, humid-gas condensation can be tolerated via use of a drip-tray	Yes, via gaskets	Good: via coatings	Can tolerate light fouling as coils can be mechanically cleaned	N/A (gas phase only)	N/A	No issue
Shell and tube	>260	Shell 300; Tubes 1400	Stainless steel, titanium; shell may be in carbon steel amongst many others	Liquid, boiling liquid, condensing vapour, gas	Yes: on tube side (remove bundle to clean)	Good: via coatings	Can accommodate fouling fluids on the tube-side	3000	15	No issue
Spiral plate	>260	30	Carbon steel, stainless steel, titanium	Liquid	Yes: via gaskets, although flow regime encourages scouring of fouling layer	Good: via coatings	Can accommodate fouling fluids on both sides	>1000	20	No issue
Welded plate	>260	40 fully welded; 16 on gasketted side if semi-welded	Stainless steel, titanium	Liquid, liquid boiling, condensing vapour, gas (welded side)	Yes: may be partially welded to allow access on one side via gaskets	Good: via coatings	Can only accommodate solid particles/fouling on the gasketted-side if unit is semi-welded	1000	N/A	No Issue

Note: Data is adapted from various sources listed throughout Section 3.1.

Table 3.11. Summary of heat pumps included the system knowledge base

Technology	Max. Condenser Temperature	Typical Pressure Ratio	Typical Corresponding Temperature Lift	Phases	Design Notes
	°C				
Vapour compression closed cycle	140 (working fluid dependant)[1]	2.8 (based on centrifugal compressor)[2]	Around 40	All (dependant on heat exchangers)	N/A
Mechanical vapour recompression (open cycle)	Generally around 120	1.25 - 2.5 (dependant on compressor type)[3]	5-25	Water vapour source; boiling water heat sink	Existing units most commonly use internal heating coils or an external heat exchanger; New MVR design may require a new heat exchanger or may aim to re-use existing unit

Note: Data is adapted from various sources listed throughout Section 3.2; [1]This is based on current standard refrigerants (working fluid discussion provided in Section 4.3.5); [2]Centrifugal compressors are considered as they provide a compromise between cost and pressure ratio; [3]In MVR, fan-type compressors are preferred if a small temperature lift can be tolerated. Else, centrifugal compressors may be considered.

Table 3.12. Summary of electricity generation methods included in system knowledge base

Technology	Minimum Source Temperature	Max. Evaporator Temperature	Typical Min. Condenser Temperature	Heat Source Phases	Process Limitations	Typical Heat Sinks
	°C	°C	°C			
Organic Rankine cycle	73[1]	140	11[2]	All (dependant on heat exchangers)	Heat source must be continuous, not intermittent	Water, Air

Note: Data is adapted from various sources listed throughout Section 3.3; [1]This is based on the lowest reported heat source temperature for a successful ORC installation as stated by Brasz (2011) and Renewable Energy World (2013) amongst others. [2]This is based on a cooling water heat sink with a minimum temperature of 1°C (to avoid freezing) with a 5°C temperature rise and a 5°C heat exchanger approach temperature. This value would vary with different heat sinks and heat exchanger effectiveness.

Chapter 4

This chapter discusses the logic, decision pathways and design equations of the knowledge-based system. Justification of the methods employed and schematics of the system logic are presented.

4. System Equations, Logic and Methods

The system knowledge-base will define the decision making and design processes of the system and is therefore key to the success of the project. Much of the data and knowledge required was acquired in the study of the state of the art in waste heat recovery technology found in Chapter 3 (summarized in Tables 3.10 - 3.12), while methods for formulating the knowledge-base are discussed in Chapter 2.

The system is to operate according to the schematic shown in Figure 4.1 below.

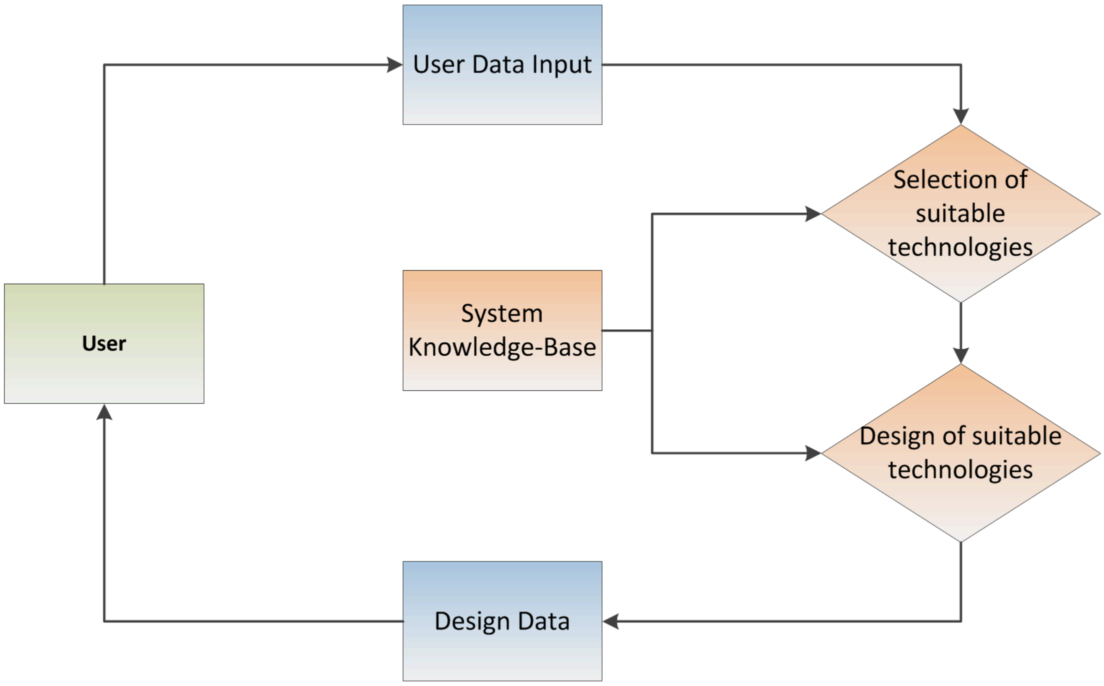


Figure 4.1. Overview of KBS operation

Figure 4.1 shows the various stages of operation in the KBS. First of all, the user is required to input data to the system. Secondly, the KBS will use the system knowledge-base to select which technologies may be suitable for use in the case study. Thirdly, the KBS will use the knowledge-base to produce a "*first design*" of each of the suitable technologies. Finally, this data is fed back to the user and should be substantial enough for a "yes" or "no" answer on which technologies fit the criteria of that individual case study.

Therefore, four key steps are required in developing the system knowledge-base as follows:

1. Selection of a suitable decision making method for use in building the system knowledge-base. This determines how the data acquired in Chapter 3 will be exploited by the KBS in order to make intelligent and accurate decisions.
2. Specifying the system data-input requirement.
3. Creation of the technology selection knowledge base. This will be used by the KBS to select which technologies are appropriate for each case-study.
4. Creation of technology design methodologies. This will determine how the KBS automatically designs each of the appropriate technologies. Key results such as equipment size, cost, payback time and greenhouse gas emissions must be calculated which may then be analysed by the user in order to make the *final* decision about waste heat recovery.

4.1. Selection of Decision Making Method

In selecting the most appropriate method of decision making for the KBS, the definition of the target end-user and scope of the knowledge-based system (see Section 1.3.1) must be adhered to, particularly with respect to the points below:

Target end-user characteristics:

- *Limited knowledge of waste heat recovery techniques.*
- *No previous experience of process waste heat recovery projects.*
- *Limited time to investigate all waste heat recovery options.*

Scope of knowledge-based system:

- Point 4. *Must make use of easy-to-access data: this will aid user with limited time in the collection of data for use in the software*.
- Point 5*. Must explain selection/design logic to the user: this will educate the user in the methods employed by the system thereby reducing/avoiding user confusion or mistrust*.

A number of methods are available for use in decision making software tools. Here, we consider two types of method: decision tree type methods and mathematical/algorithm based methods.

4.1.1. Mathematical/Algorithm-Based Methods

Mathematical methods use algorithms to find the best available solutions to multi-variable decision problems. For example, the Technique for Order of Preference by Similarity to Ideal Solution (TOPSIS) ranks each possible solution according to distance from an *ideal* solution.

These methods have four distinct disadvantages for use in this case and are therefore deemed inappropriate.

First of all, in some case there may be only one possible solution. For example, in a situation where only a waste heat source is present and no heat sinks are identified, then the only option available from the system will be to design an organic Rankine cycle (assuming the heat source is of sufficiently high temperature). Hence, multiple solutions do not exist and the system would automatically choose the ORC. Crucially, this would be done without providing any rational explanation to the user as the method is strictly mathematical (see point four for discussion of this problem).

Secondly, an *ideal* solution is difficult to define in low-grade waste heat recovery and it would probably require that each case study to have a different *ideal* solution. This would be almost impossible to program.

Thirdly, many of the "variables" in the problem are non-numerical and subjective in nature. For example, plant preferences with regard to toxic/flammable working fluids for heat pumps/ORC are highly subjective. Therefore, accounting for such variables in a mathematical method would be difficult.

Finally, the most significant factor against using numerical methods for this problem is that the entire selection process is lost within the mathematics of the method. Hence, it would be very difficult to provide a rational explanation of the programme to a user with no background in the programming of such methods. This is a violation of the scope of the system, particularly point 5 (as shown above, and in Section 1.3.1).

4.1.2. Decision Tree Methods

The decision tree method uses IF-THEN logic to formulate decisions. A number of such decisions can be built up in series or parallel in order to build multi-parameter decision criteria. This method has been successfully utilised in similar previous works. For example Heppenstall and Halliday (1990) used this method to produce a heat exchanger selection expert system, Abou-Ali and Beltagui (1995) used this method to create an expert system for the selection of shell and tube-bundle types in TEMA design of shell-and-tube heat exchangers, and Lababidi and Baker (2003) used this method to create an expert system for the selection of food drying equipment. This is further discussed in Section 2.3.

The use of this method has two distinct advantages. Firstly, it has been proven to be successful in similar projects as previously discussed (briefly above, and in Section 2.3). This is because it is a good match to the standard procedure used in the selection of waste heat recovery equipment: generally, the initial selection is based on equipment technological limitations while "*final*" selection will be based on user-defined criteria, most commonly economical data such as project payback time. Hence, the decision tree method is suited to the initial selection process. Procedural routines can then be introduced to the code to produce a *"first design"* of each selected technology in order to acquire the required design data (including process economic data *etc*), with which the user can then make the *"final"* decision.

The second advantage is related to the educational aims of the software. Point 5 of the system scope (as listed above in Section 4.1) states that a degree of explanation must be given to the user. Logic displayed in decision tree format is easy and intuitive to follow for the user, thereby providing an appropriate degree of explanation. This will also prevent user mistrust as it does not require a knowledge of complex mathematical procedures (as described in Sections 4.1.1-4.1.3) to understand the system logic.

For these reasons, the decision tree method of creating the system knowledge base is chosen.

4.2. User Data Input to the System

The specification of the data input to the system is crucial as it must strike a balance between the requirements for accurate results to be calculated and the expected time constraints of the target end-user of the KBS as pointed out in Section 1.3.1 (revisited in Section 4.1 above). The scope of the system also states:

- Point 2. Must make use of easy-to-access data: this will aid users with limited time in the collection of data for use in the software.

Table 4.1, below, summarises commonly required heat source/sink data required for accurate design of heat exchangers, heat pumps and/or organic Rankine cycles, while Table 4.2 below summarises qualitative heat source/sink data that must be considered.

Table 4.1. Heat source/sink numerical data required for design of waste heat recovery systems

	Symbol (units)	Included
Source phase	N/A	Yes
Sink phase	N/A	Yes
Source mass flow rate	m (kg/s)	Yes
Sink mass flow rate	m (kg/s)	Yes
Source temperature	T (^oC)	Yes
Source target temperature	T (^oC)	Yes
Sink temperature	T (^oC)	Yes
Sink target temperature	T (^oC)	Yes
Source specific heat capacity	C_p (kJ/kg.K)	Yes
Sink specific heat capacity	C_p (kJ/kg.K)	Yes
Source pressure	P (kPa)	Yes
Sink pressure	P (kPa)	Yes
Source density	ρ (kg/m^3)	Yes
Sink density	ρ (kg/m^3)	Yes
Source viscosity	μ (kg/m.s)	Yes
Sink viscosity	μ (kg/m.s)	Yes
Source thermal conductivity	k (kW/m.K)	No
Sink thermal conductivity	k (kW/m.K)	No
Source film heat transfer coefficient	h (kW/m^2.K)	No
Sink film heat transfer coefficient	h (kW/m^2.K)	No

Table 4.2. Heat source/sink qualitative data required for design of waste heat recovery systems

	Notes	Included
Source solid content and nature	Data required here includes the mass fraction of solids in the stream, the nature of the solids and the average particle diameter	Yes
Sink solid content and nature		Yes

Source fouling tendency	This is heavily linked to the source solid content although other types of fouling are noted such as scaling. It is difficult to definitively quantify fouling for every case study and this remains subjective	Yes
Sink fouling tendency		Yes
Source corrosivity	This is linked to both the source/sink properties and the materials of construction	Yes
Sink corrosivity		Yes
Source material compatibility	This is heavily linked to the corrosivity of the two fluids. Some heat exchangers may only be constructed from certain materials, hence this influences heat exchanger selection	Yes
Sink material compatibility		Yes
Source access for maintenance/cleaning	This is heavily linked to both corrosivity and fouling characteristics, *i.e.* if fluid(s) is (are) fouling and/or corrosive then access will be required	Yes
Sink access for maintenance/cleaning		Yes

In addition to that stated in Tables 4.1 and 4.2, a number of further data is required in order to carry out economic and environmental calculations during system design. This is summarised in Table 4.3 below.

Table 4.3. Plant data required for design of waste heat recovery systems

	Symbol (units)	**Included**
Current method of heating sink	N/A	Yes
Efficiency of current method of heating sink	η (%)	Yes
Utility costs	N/A (£/kWh)	Yes
Utility associated emissions	N/A (tCO_2eq/kWh)	Yes
Plant hours of operation	N/A (h/year)	Yes

The specified data input has been chosen to strike a balance between accuracy of selection/design procedures and the knowledge/time constraints of the target end-user. The following justification is presented for each included/excluded parameter:

- Source/sink phase(s): this is crucial for selecting a suitable heat exchanger as discussed in Section 3.1 and is therefore included.
- Source/sink mass flow: required to calculate heating duty and therefore must be included.
- Source/sink specific heat capacity: required to calculate heating duty and therefore must be included.
- Source/sink inlet temperature: required to calculate heating duty and therefore must be included. Also, temperature is a limiting factor in many types of heat exchanger.
- Source/sink target temperature: required to calculate heating duty and therefore must be included. Also, sink target temperature is a limiting factor in heat pump working fluid selection. Note that a routine will be used to set the source/sink outlet temperatures for cases where the heat balance is unequal.
- Source/sink pressure: this is a limiting factor in many types of heat exchanger and therefore must be included.
- Source/sink density: required to calculate the Reynolds number which in turn may be used to estimate key design parameters such as pressure drop. Also needed to calculate volumetric and subsequent velocity (also required in Reynolds number and pressure drop calculations). Therefore this must be included.
- Source/sink viscosity: required to calculate the Reynolds number. Also, this is a limiting factor in many types of heat exchanger and influences heat transfer coefficients. Therefore, this must be included.
- *Typical* Source/sink film heat transfer coefficients: this is required to calculate required heat transfer areas. However, this data is not easily accessible and may be difficult to calculate or find for users with limited heat transfer knowledge. Therefore, inclusion of this term would violate point 2 of the system scope and it is not included. Values for overall heat transfer coefficients from literature will be used in the system knowledge base as this

is accurate enough for a first estimate of system design. The production of final, optimal, designs is outside of the scope of the KBS.

- Source/sink thermal conductivity: this data is used in calculations to find accurate estimates of the film heat transfer coefficients. However, as stated above, these values will be taken from literature and therefore the thermal conductivity will not need to be input by the user. Furthermore, this data is not easily accessible and may be time consuming to find. Hence it would violate point 2 of the scope of the system.
- Source/sink solid content and nature: this information is a limiting factor in the selection of many types of heat exchanger and must be included. However, this data generally isn't standardised between heat exchanger manufacturers and literature and therefore is subjective. In this case, it was decided to adopt the methodology from Best Practice Programme (2000) in which heat exchanger operation limits were given based on average particle diameter and "particle type" which may be selected from "shear sensitive", "shear insensitive" and "fibre". This source was chosen as it contains a comprehensive list of criteria for heat exchanger selection for a wide range of units and the data covers the majority of the heat exchangers chosen for selection in the knowledge base. The data for the heat exchangers missing from this source will be determined from various other sources that will be adapted for use in this methodology.
- Source/sink fouling tendency: this information is a limiting factor in the selection of many types of heat exchanger and must be included. This is often directly linked to the solid content, although other types of fouling must also be considered, such as scaling. Again, this is a highly subjective area of heat exchanger design. In brief, if the heat transfer fluids are fouling it is required that the heat exchanger must be cleaned at regular intervals in order to maintain an acceptable overall heat transfer coefficient. Hence, if a fluid is fouling, a fully welded heat exchanger would not be suitable for use, for example. Due to the subjective nature of this data, it is decided that a user question will be included to capture this data, "*Is access for cleaning due to fouling anticipated*?" It is assumed that this data would be known to

the user as fouling would occur in a number of other units associated with the stream.

- Source/sink corrosivity: this information is a limiting factor in the selection of materials of construction, which in turn is a limiting factor in the selection of many types of heat exchanger. A number of factors influence this, such as pH and stream composition. Specific data input for this factor would be difficult as this again is rather subjective. Therefore, it is more suitable to instead list a number of *typical* heat exchanger materials and for the user to select which are compatible with the fluid using “yes” or “no” answers. Also, a question is included on whether an anti-corrosion layer on a metallic heat exchanger would suffice as an anti-corrosion measure, as this is available on a number of units. This data should be easily accessible to the user and should decrease the complexity of the decision-making process.
- Source/sink material compatibility: as discussed above
- Source/sink access for cleaning/maintenance: this is linked to both the fouling and corrosion data as discussed above. Here, the user is simply required to answer “yes” or “no” as to whether access for cleaning/maintenance is anticipated due to corrosion or fouling concerns. The user should have easy access to this data as it will be similar for all other unit operations linked to the fluid. Also, this method should decrease the complexity of attempts to infer whether or not access will be required based on subjective questions/data regarding fouling and/or corrosion.
- Current method of heating sink: Required to calculate the cost and greenhouse gas reductions due to waste heat recovery, two key results from the KBS. Here, options include “natural gas”, “electricity”, “steam” and “other”. The latter is included to account for any relatively obscure heating utilities (such as biomass), while the others are the most common process plant utilities.
- Efficiency of current heating method: required to accurately calculate the utility saving from implementing waste heat recovery, which in turn directly influences the economic and environmental benefits. This data should also be easily accessible. Therefore, it is included.

- Utility costs: required to accurately calculate the economic benefits of implementing waste heat recovery for the specific plant in question. This data should also be easily accessible.
- Utility associated emissions: required to accurately calculate the greenhouse gas reductions associated with waste heat recovery. This is automatically programmed into the software for electricity and natural gas as this data is widely available (for example, the Carbon Trust, 2013). However, for the case that a relatively obscure heating utility is included (*i.e.* “other” is selected as the current heating utility) then the user is required to input the data. It is assumed that this data is easily accessible if the plant in question has invested time and money in installing a novel heating utility system.
- Plant hours of operation: Required to accurately calculating the economic and environmental benefits of waste heat recovery. This should be easy to access based on plant downtime data.
- Data regarding the cooling utility in heat pump design: In heat pump design, simultaneous useful heating and cooling is considered, as this has been proven to significantly boost the COP and subsequent economics of heat pump projects (as discussed in Section 3.2.1). Therefore, data is also requested for the current cooling utility in heat pump system selection/design where appropriate. This includes the same questions as requested for the heating utility including the current utility (for example, cooling water or a refrigeration circuit), efficiency of current method (COP of refrigeration circuit, for example) and the cost of the current utility.

The data input to the system is required to undergo a “data check” procedure in order to identify and solve any errors prior to the selection/design processes described below. This data check is designed to identify any unrealistic values (negative pressure, density for example) and thermodynamic anomalies (temperature cross between source/sink data, for example). Examples of the “data check” rules are displayed in Appendix I.

4.3. Technology Selection Knowledge Base

The selection of technologies by the KBS is to be completed on two levels, as per a consultant engineer would tackle such a problem. The first level, or "*Initial Selection Procedure"* is relatively simple and addresses the availability of potential heat sinks and the aims of the plant in question. This is described in Section 4.3.1.

The second level is more complex and is based on the technological limitations of the equipment chosen for inclusion in the knowledge-base. For ease of display, this is split into the five general categories of heat recovery technology included in the system knowledge base: gas-gas heat exchangers, gas-liquid heat exchangers, liquid-liquid heat exchangers, heat pumps (MVR and closed cycle) and organic Rankine cycles. Each is described in Sections 4.3.2 - 4.3.7 respectively.

4.3.1. Initial Selection Procedure

As described above, the initial selection procedure is designed to address the availability of potential heat sinks and the aims of the plant in question. From an educational perspective, this step will also act as an introduction to the various technologies available for low-grade waste heat recovery.

Here, relatively simple statements are considered, for example:

- **IF** a "matching" heat sink is NOT available **THEN** one may not use a heat exchanger.
- **IF** a heat sink within a reasonable temperature lift (~40K) is NOT available **THEN** one may not use a heat pump.
- **IF** the plant has no interest in electricity generation **THEN** one may not use an organic Rankine cycle

Such statements can be extended to consider the various types of heat exchangers included in the system, for example, thereby creating logic such as:

- **IF** a “matching” heat sink is available **AND** heat source is in the liquid phase **AND** heat sink is in the liquid phase **THEN** one may use a liquid-liquid heat exchanger.

(Note: the selection of which types of liquid-liquid heat exchanger are suitable is then considered in level 2 of the selection logic)

Figure 4.2 overleaf shows the level 1 technology selection logic displayed in decision tree format. Figure 4.3 shows a continuation of the heat exchanger level 1 selection logic in order to select which heat exchanger category is required based on the phase of the heat source and sink.

Figure 4.2. Level one of the technology selection knowledge base

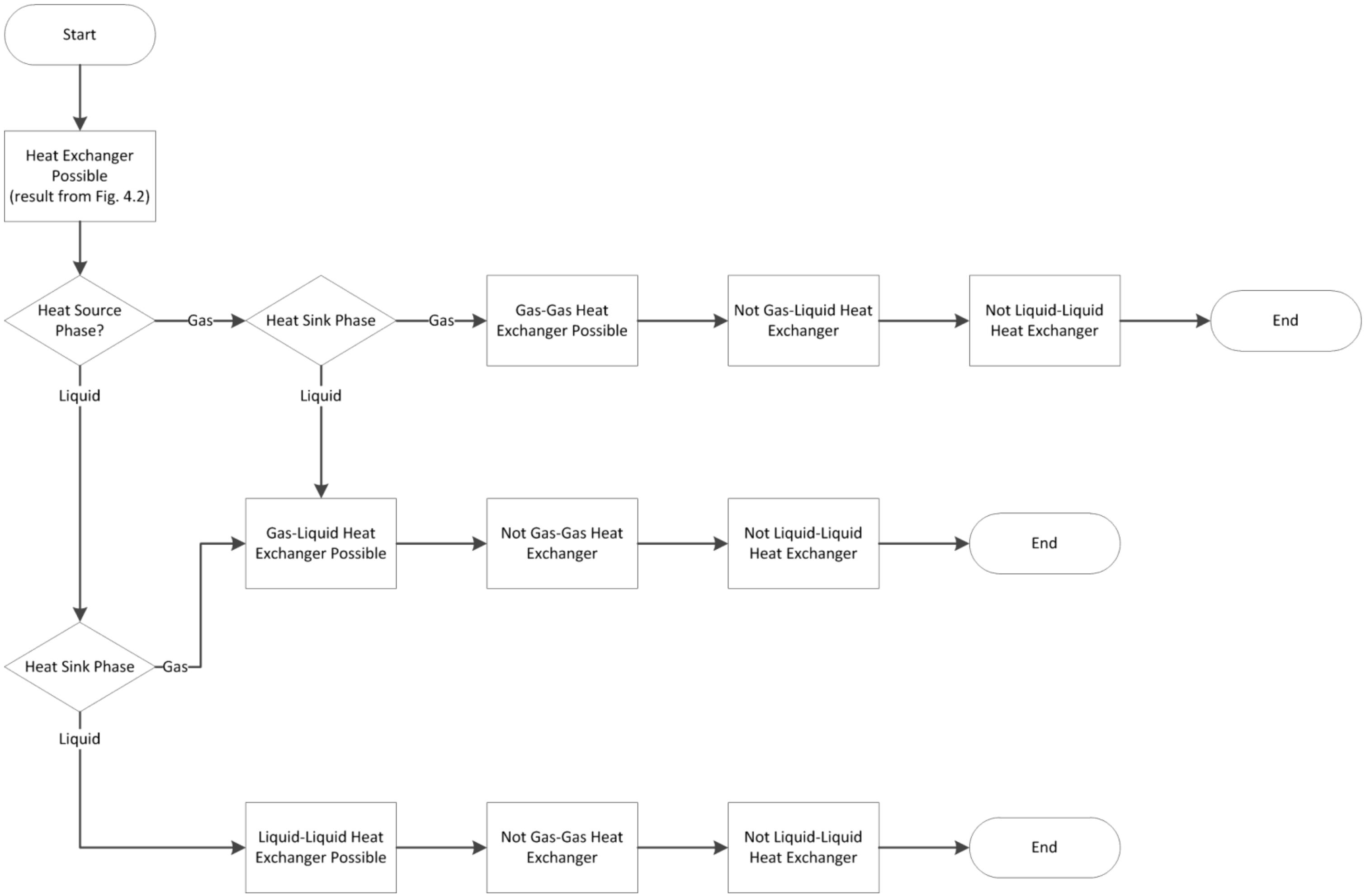


Figure 4.3. Level one of the technology selection knowledge base (heat exchangers only)

When the level 1 technology selection procedure is complete, first stage in waste heat recovery system design has been performed, i.e. the general categories of possible waste heat recovery system have been identified.

At this stage, there is an argument for the system selecting only the simplest (or cheapest) method of waste heat recovery for each case study. For example, this logic would suggest that if a "matching" heat sink is available and a heat exchanger may be used for waste heat recovery, then more complex solutions such as organic Rankine cycles should not be considered. This logic is suggested in the paper "Opportunities for low grade heat recovery in the UK food processing industry" by the author (Law *et al*, 2013), and summarised in Figure 4.4 below.

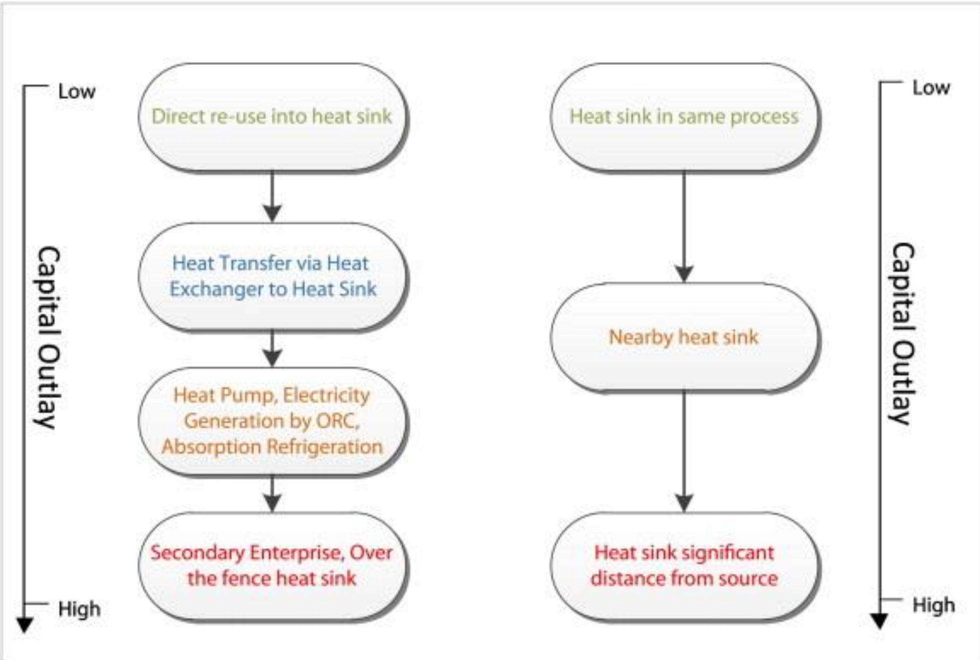


Figure 4.4. Typical technology selection logic in waste heat recovery system design (Law *et al*, 2013)

Note that in the paper, the authors state that the selection process should be directly correlated to the capital outlay. Hence, if a cheaper option is available (for example a heat exchanger) then one should not consider the more expensive options (for example an ORC).

Here, for the case of this knowledge-based system, the educational aims of the software must be considered. By including and generating results for all possible options, the user will be able to identify the key differences in each type of system. For example, for a case where a liquid heat source is identified and a liquid heat sink is identified, the obvious solution would be to design a liquid-liquid heat exchanger although it would theoretically be possible to also use the waste heat to drive an organic Rankine cycle. Therefore, the KBS will produce a first design for both options. An experienced heat recovery consultant may consider this unnecessary, but it allows the users to compare the results and conclude that the heat exchanger is by far the most economical option for waste heat recovery (for a *typical* case), thereby helping the user to gain a knowledge of the merits of each type of technology. The user may also consider other drivers such as system complexity (with regards to installation and maintenance) which will be apparent in the system results (via a process flow diagram of the proposed technologies).

4.3.2. Gas-Gas Heat Exchanger Knowledge Base

The second level of equipment selection decisions is based on the operational limitations of each technology, as previously discussed in Chapter 3, and summarised in Tables 3.10-3.12.

Figures 4.5-4.7 overleaf show the gas-gas heat exchanger section of the level two selection knowledge-base. Figure 4.5 (a and b) is concerned with heat exchangers suitable for non-condensing gaseous heat sources, Figure 4.6 is concerned with heat exchangers for condensing vapour heat sources and Figure 4.7 is concerned with heat exchangers suitable for the condensation of water from a humid air heat source.

Note: in the following figures, temperature is in ^{o}C, pressure in bar and viscosity in cP.

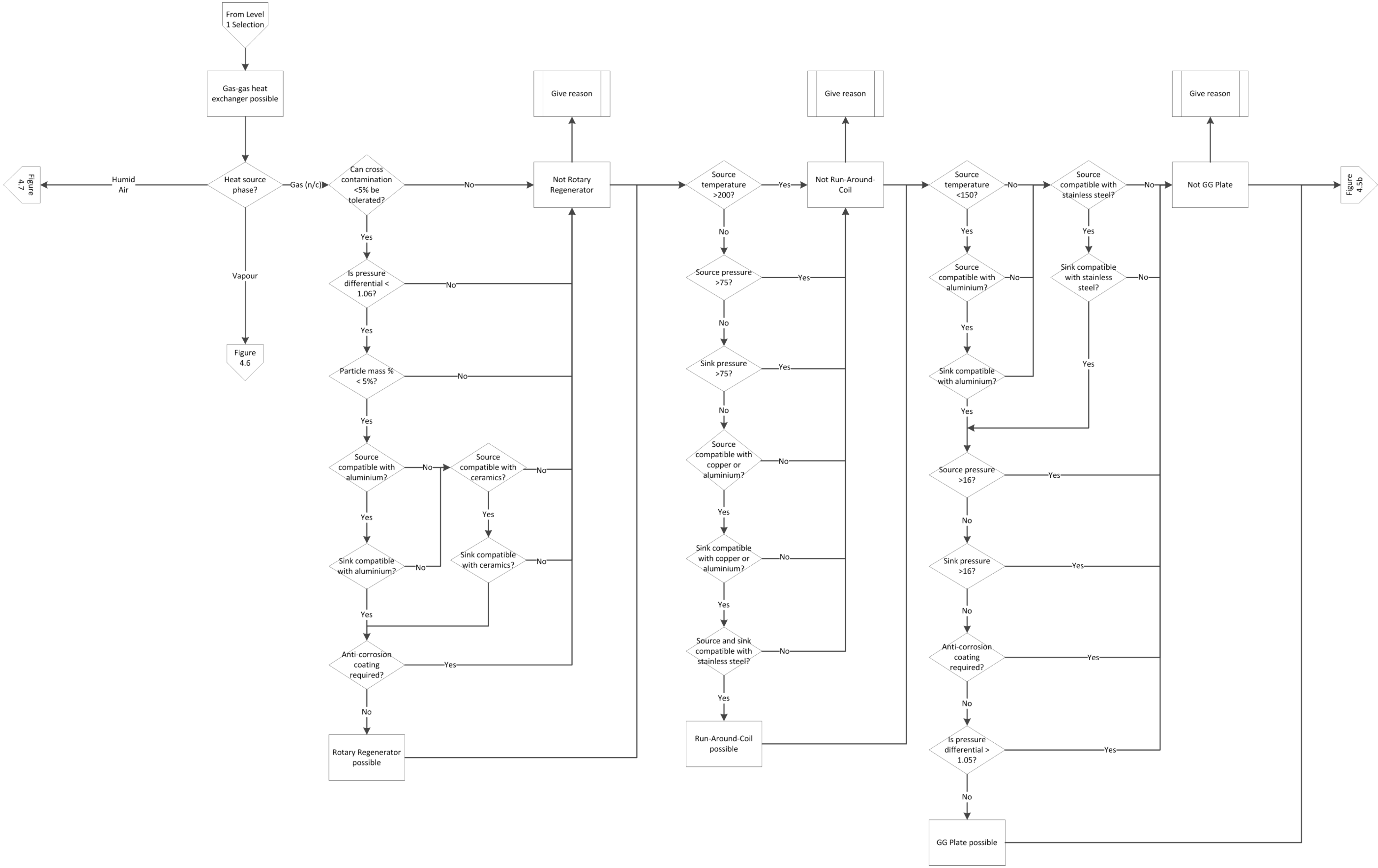


Figure 4.5. Gas-gas heat exchanger selection knowledge-base for non-condensing heat sources (part a)

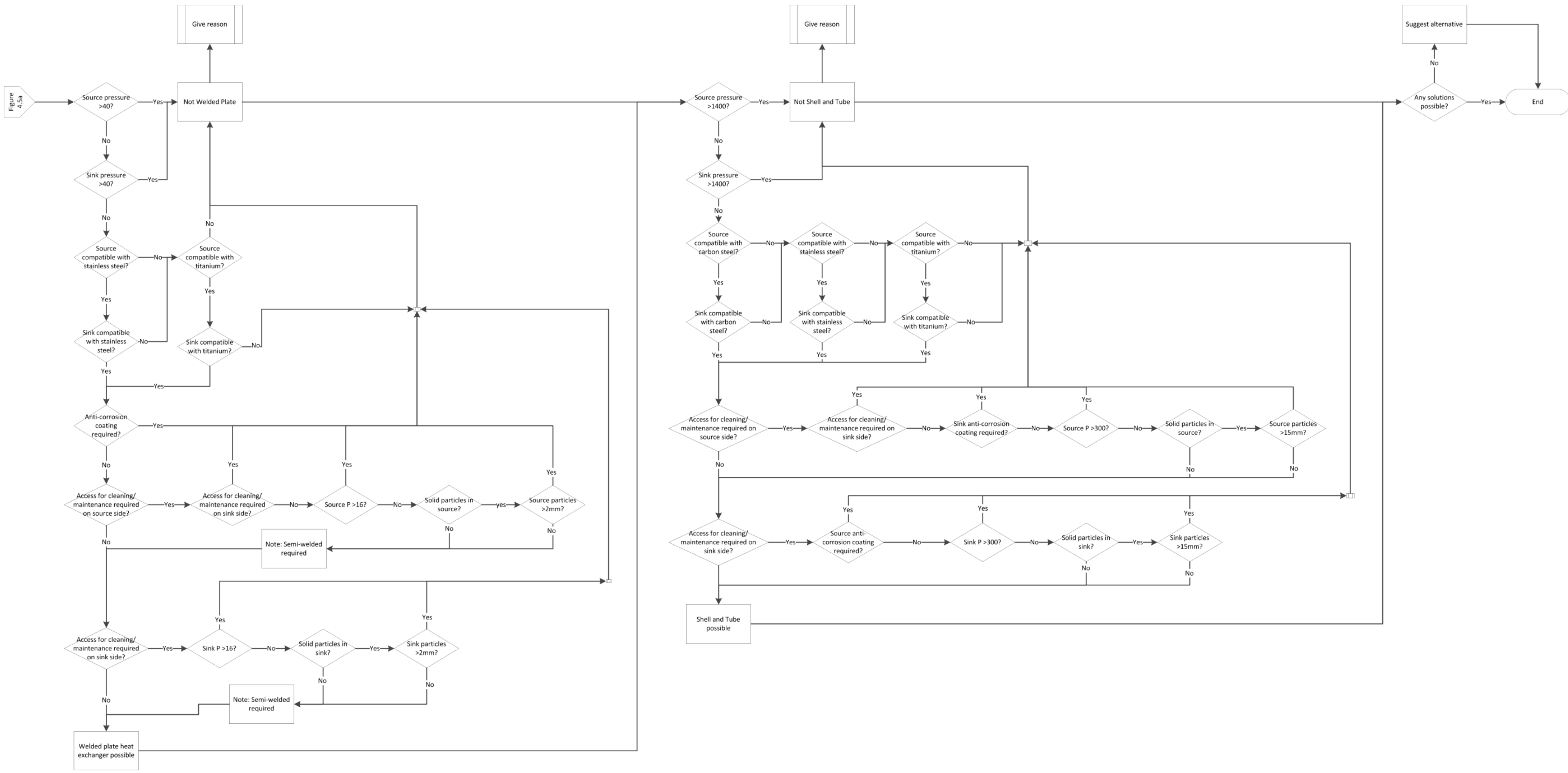


Figure 4.5. Gas-gas heat exchanger selection knowledge-base for non-condensing heat sources (part b)

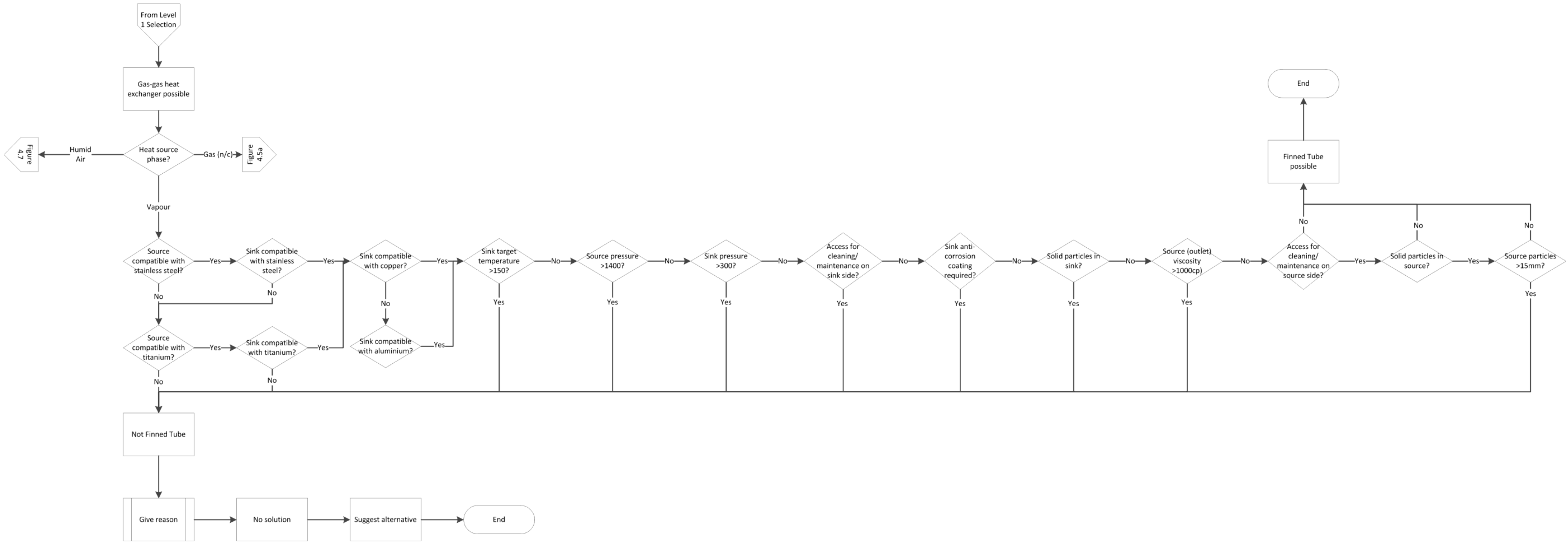


Figure 4.6. Gas-gas heat exchanger selection knowledge-base for condensing vapour heat sources

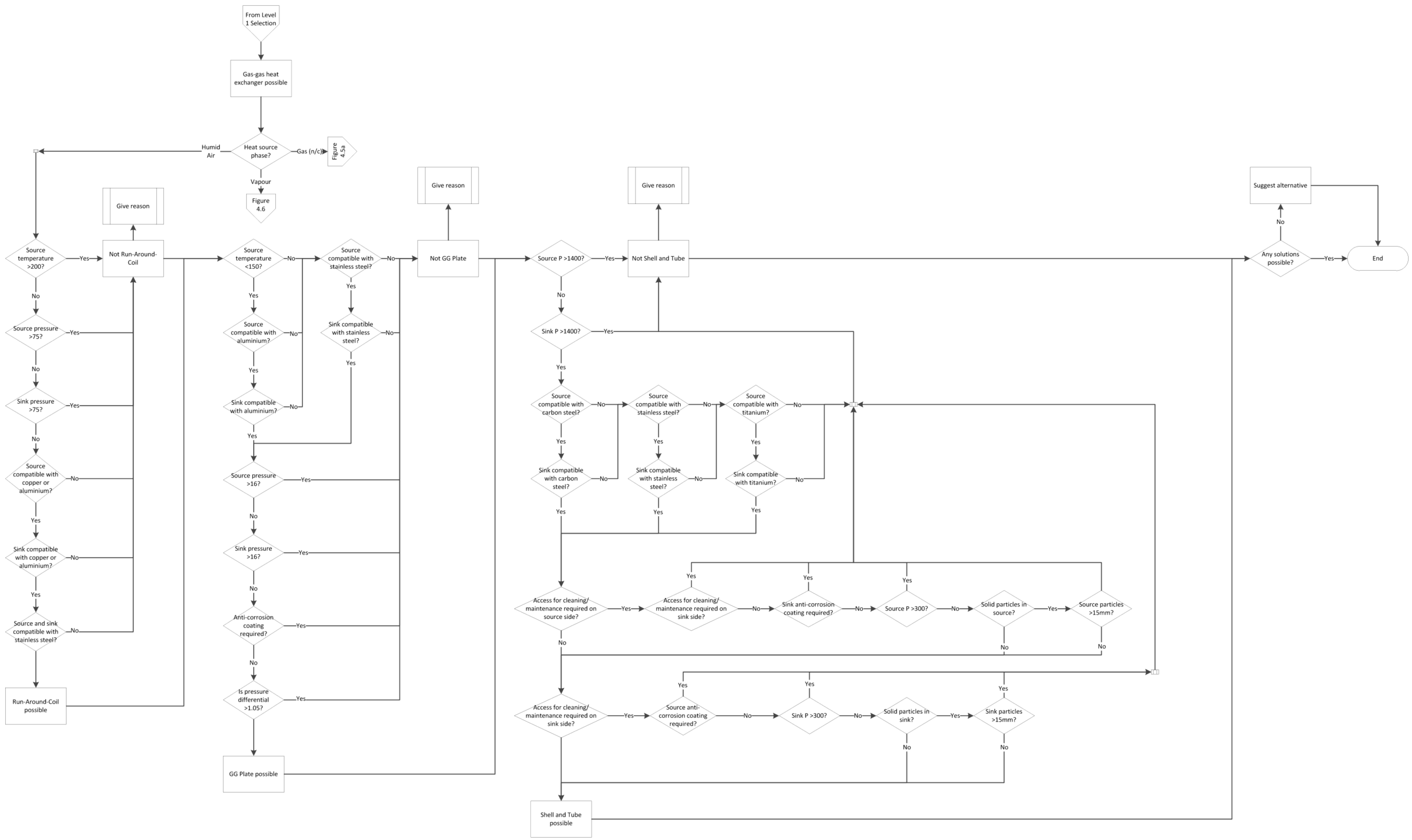


Figure 4.7. Gas-gas heat exchanger selection knowledge-base for humid air heat sources

4.3.3. Liquid-Liquid Heat Exchanger Knowledge Base

The second level of equipment selection decisions is based on the operational limitations of each technology as previously discussed in Chapter 3, and summarised in Tables 3.10-3.12.

Figure 4.8 (a and b) overleaf shows the liquid-liquid heat exchanger section of the level two selection knowledge-base.

Note: in the following figures, temperature is in ^{o}C, pressure in bar and viscosity in cp.

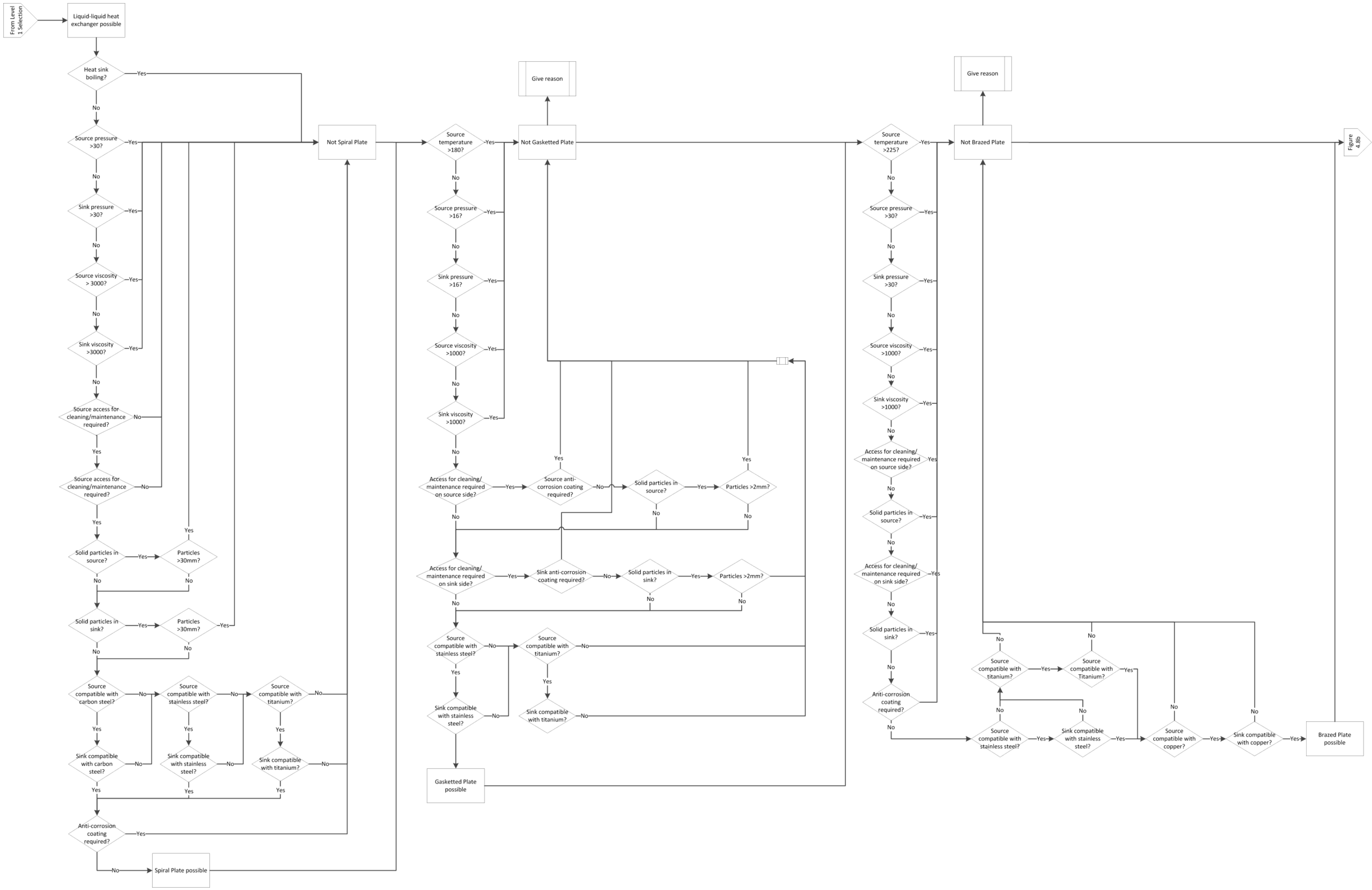


Figure 4.8. Liquid-liquid heat exchanger selection knowledge-base (part a)

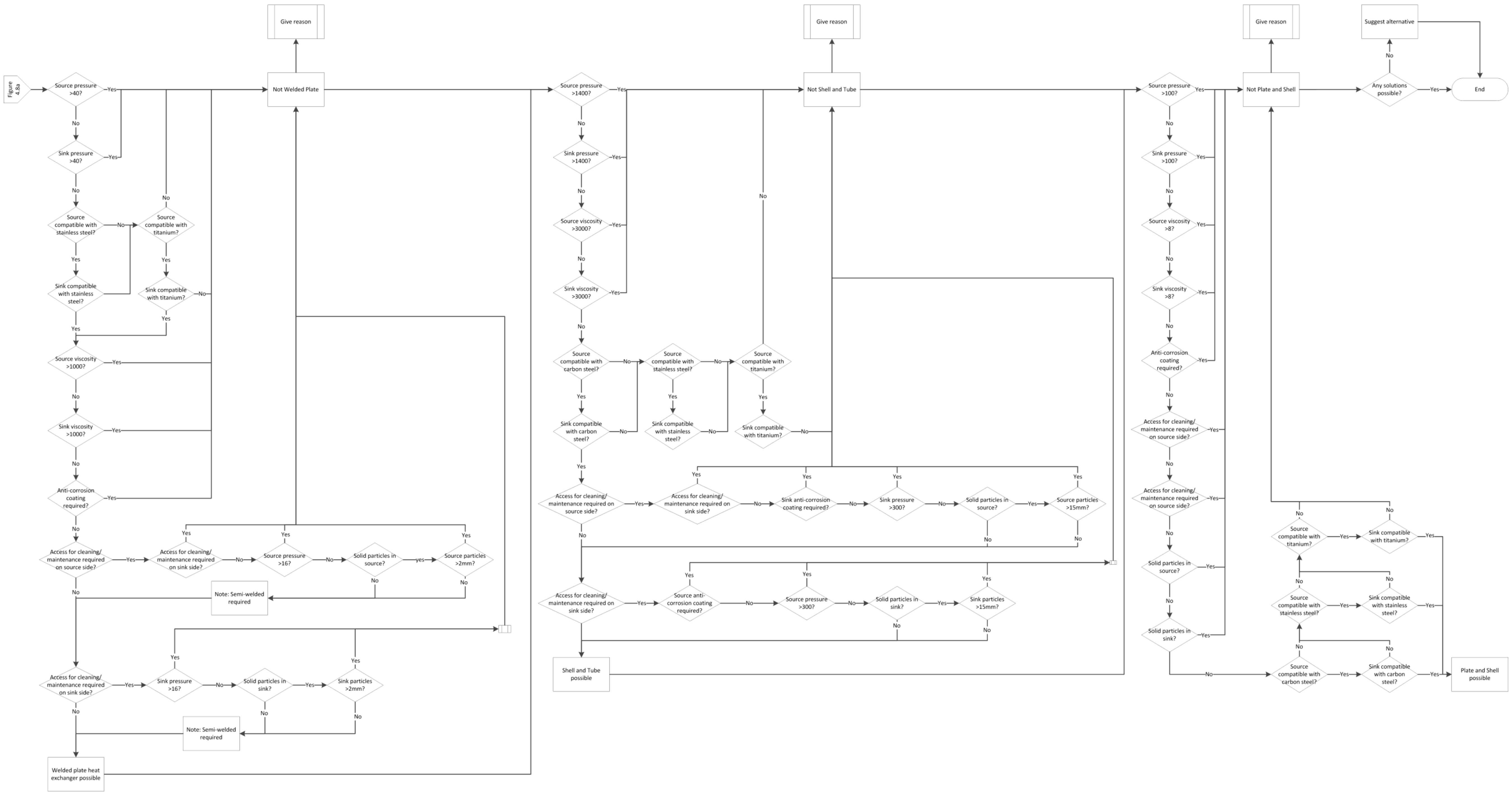


Figure 4.8. Liquid-liquid heat exchanger selection knowledge-base (part b)

4.3.4. Gas-Liquid Heat Exchanger Knowledge Base

The second level of equipment selection decisions is based on the operational limitations of each technology as previously discussed in Chapter 3, and summarised in Tables 3.10-3.12.

Figures 4.9 to 4.11 overleaf show the gas-liquid heat exchanger section of the level two selection knowledge-base. Figure 4.8 features heat exchangers suitable when the heat source is a liquid and the heat sink is gaseous. Figure 4.9 features heat exchangers suitable for a gaseous heat source (both non-condensing and humid air) and the heat sink is a liquid (boiling or constant phase). Figure 4.10 (a and b) features heat exchangers which are suitable when the heat source is a condensing vapour and the heat sink is a liquid (boiling or constant phase).

Note: in the following figures, temperature is in ^{o}C, pressure in bar and viscosity in cp.

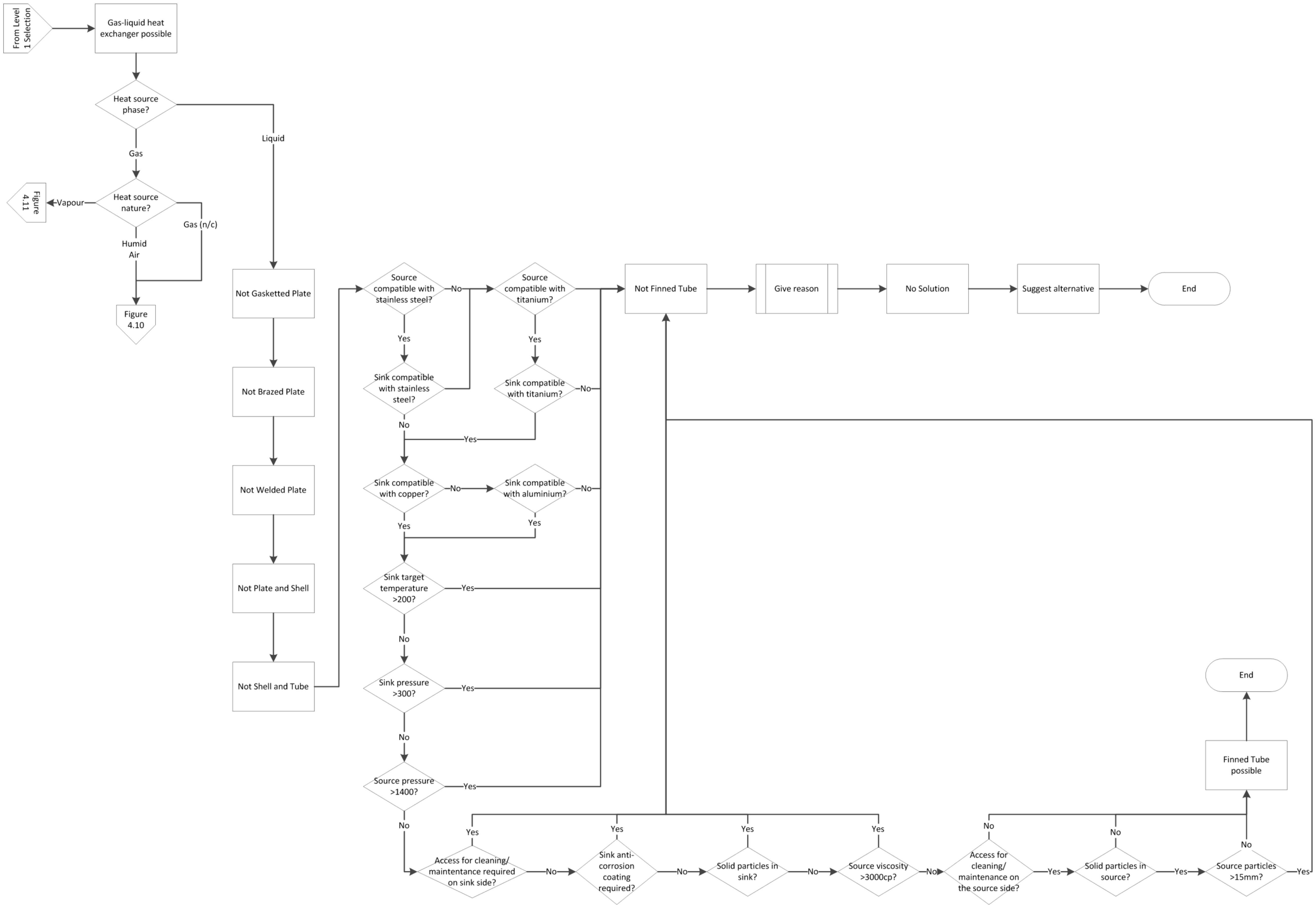


Figure 4.9. Gas-liquid heat exchanger selection knowledge-base for liquid heat sources

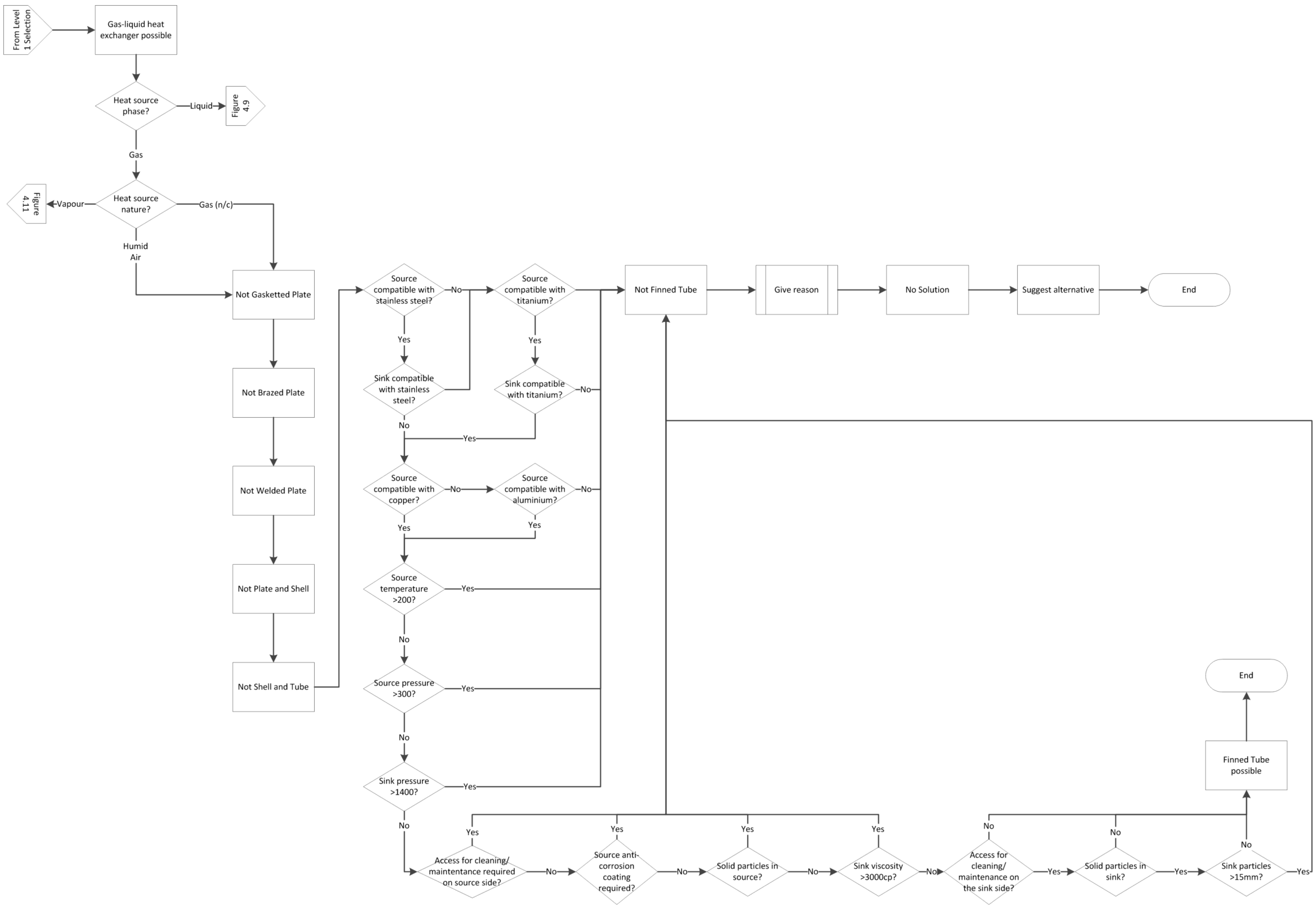


Figure 4.10. Gas-liquid heat exchanger selection knowledge-base for n/c gas and humid air heat sources

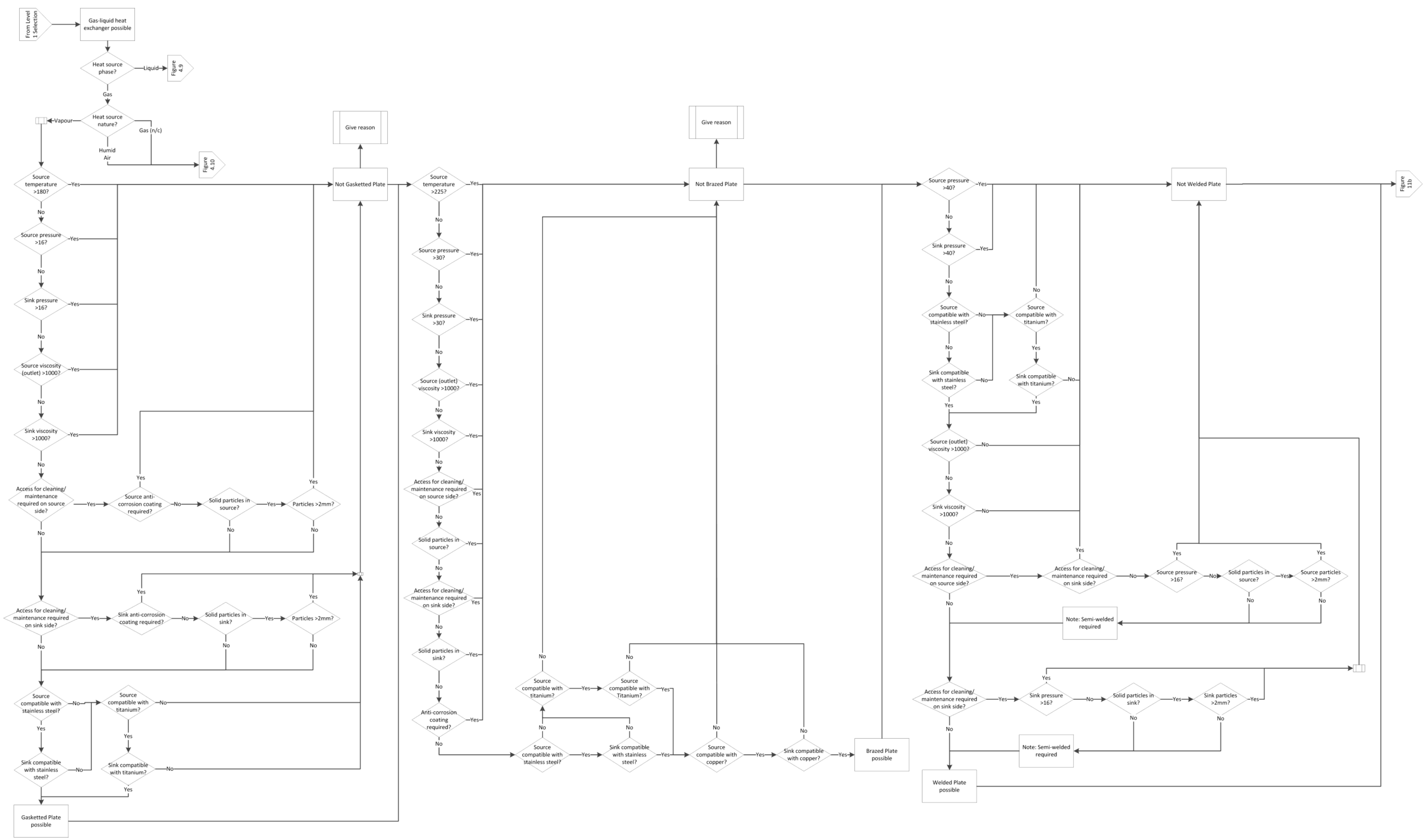


Figure 4.11. Gas-liquid heat exchanger selection knowledge-base for condensing vapour heat sources (part a)

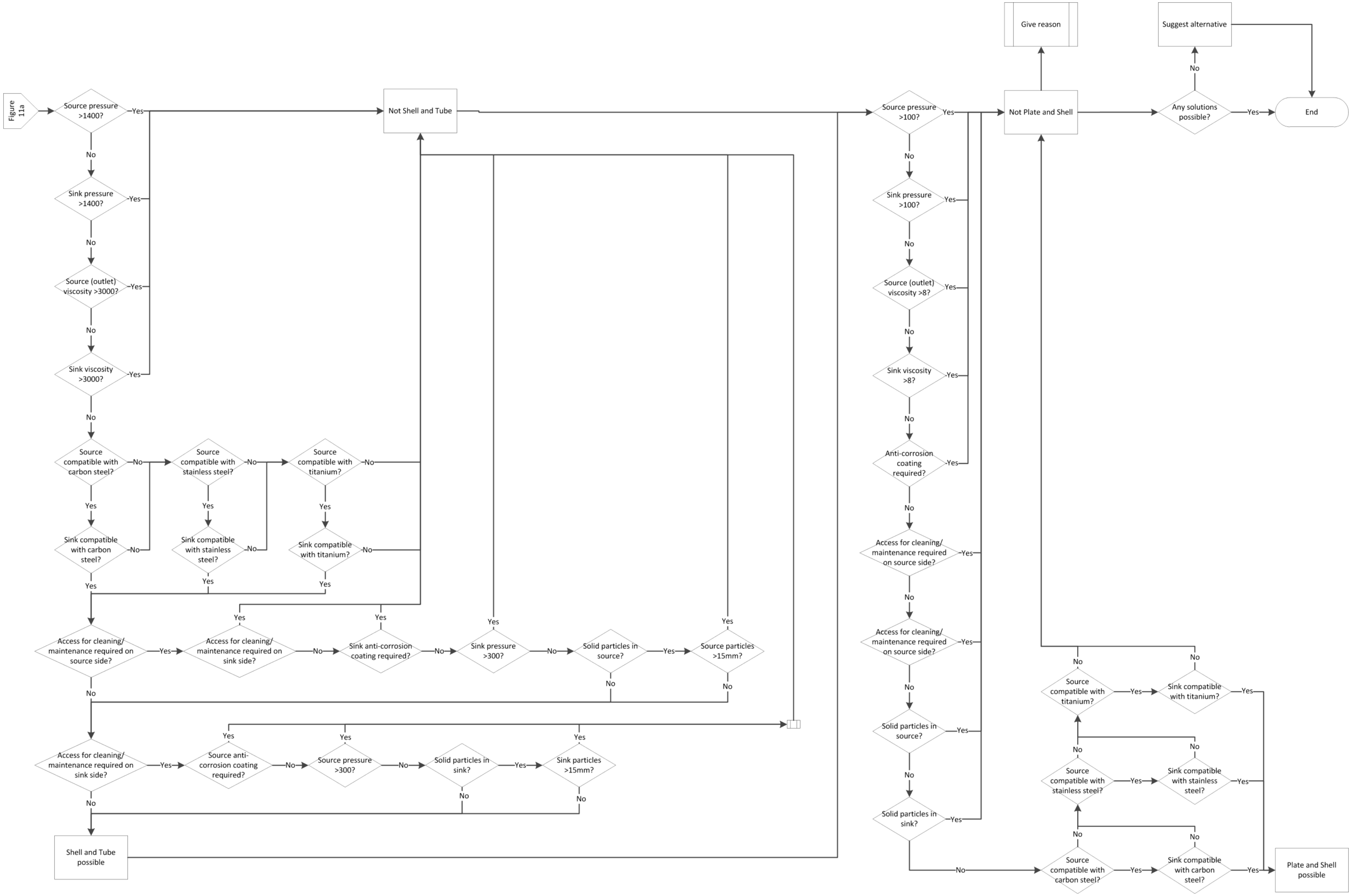


Figure 4.11. Gas-liquid heat exchanger selection knowledge-base for condensing vapour heat sources (part b)

4.3.5. Heat Pump Knowledge Base

The second level of equipment selection decisions is based on the operational limitations of each technology as previously discussed in Chapter 3, and summarised in Tables 3.10-3.12.

In the field of heat pump design, the decision is focussed on the selection of the most appropriate working fluid based on the user-input data. In general heat pump selection/design, the working fluid selection is based on the following criteria:

- Legality: only working fluids which are not ozone-depleting should be considered, in agreement with the Montreal Protocol, 1989.
- Industry Standards: only working fluids which are *"industry standard"* will be considered by this system as the results must be valid on an industrial scale (see system scope, point 8, Section 1.3.1). The definition of *"industry standard"* used here is that any working fluid must have been given an official safety rating by the American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE), as listed in the ASHRAE Fundamentals Handbook (ASHRAE, 2009).
- Health, safety and environmental properties: only working fluids which are tolerable at the site in question must be used. *i.e*. final selection must be on an individual case study basis dependant on the plant tolerance to detrimental working fluid properties, such as flammability and toxicity. Hence, working fluids must be included in the system to cover all eventualities.
- Range of operation: working fluids operate in an optimum temperature/pressure range. Hence, working fluid selection should be dependent on the heat source temperature and sink target temperature. Therefore, a wide-range of working fluids should be included to cover the entire expected temperature range.

The expected COP of each working fluid should be considered. However, the difference in COP between two suitable fluids across the same temperature range is minimal and hence this does not influence the selection to the same degree as the criteria specified above. The volumetric capacity of the working fluid (amount of energy stored per unit volume per degree centigrade) is also important as this is directly linked to the size of the equipment required and hence the capital cost. However, as the criteria above states that only *standard* working fluids should be considered, it is assumed they are all of suitable volumetric capacity. An example of this is that water would not be considered for the system, as it is not a *standard* heat pump working fluid: water shows improved COP performance compared to standard working fluids over the same operating range, but is currently not considered due to the low volumetric capacity (and other issues such as superheat during compression).

According to the criteria discussed, four heat pump working fluids were selected for inclusion in the system knowledge-base. The second level of selection for heat pumps concerns selecting which of the four is appropriate on an individual case study basis. The four working fluids are as follows (Table 4.4)

Table 4.4. Heat pump (closed cycle) working fluids selected for system knowledge-base

Name	Classification	Critical Temperature	Critical Pressure	Maximum Condenser Temperature	Global Warming Potential	ASHRAE Safety Classification
		(°C)	bar	(°C)		
R134a	HFC	101.1	40.6	85	1430	A1
R245fa	HFC	154.0	36.5	140	1030	B1
R600	HC	152.0	38.0	140	20	A3
R717	Natural	132.3	113.3	120	<1	B2

Note: HFC stands for Hydrofluorocarbon; HC stands for hydrocarbon; ASHRAE safety rating A denotes no toxicity at concentrations below 400ppm; ASHRAE safety rating B denotes toxicity at concentrations below 400ppm; ASHRAE safety rating 1 denotes non-flammable; ASHRAE safety rating 2 denotes moderate flammability; ASHRAE safety rating 3 denotes high flammability.

The fluids shown in Table 4.4 are sufficient to provide a heat pump solution for most eventualities. For example, if high flammability cannot be tolerated, three fluids are available for use (R245fa, R134a and R717) and if moderate flammability cannot be tolerated two fluids are available (R245fa and R134a). Also, if toxicity cannot be tolerated, two fluids are available for use (R600 and R134a). Finally, if both toxicity and flammability cannot be tolerated then R134a is available for use with the drawback that the maximum output (condenser) temperature of this fluid is only 85^{o}C. However, this drawback is unavoidable as no *standard* working fluids exist with both an ASHRAE safety rating of A1 and critical temperature greater than 101.1 ^{o}C.

The second level of selection for heat pumps, therefore, is the selection of one of these four working fluids based on the plant tolerance to flammability, toxicity and the outlet temperature required for the heat sink. This is depicted overleaf in Figure 4.12.

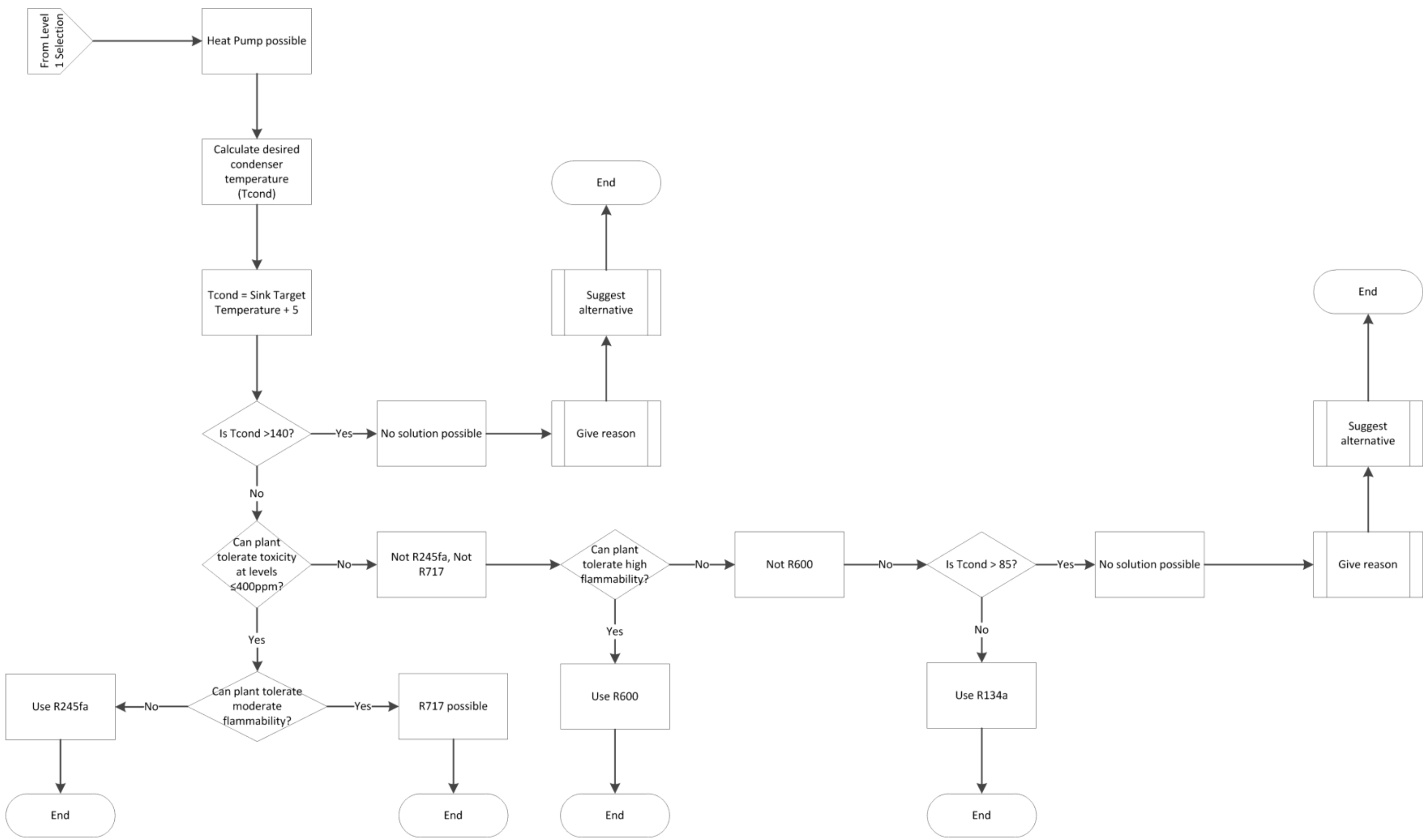


Figure 4.12. Heat pump working fluid selection knowledge-base

4.3.6. Mechanical Vapour Recompression Knowledge Base

In mechanical vapour recompression there is no secondary level of equipment selection. If MVR is deemed suitable following the first level of equipment selection, then the system proceeds straight to the design phase. The "final" decision will then be made by the user following the results of the design phase.

4.3.7. Organic Rankine Cycle Knowledge Base

The second level of equipment selection decisions is based on the operational limitations of each technology as previously discussed in Chapter 3, and summarised in Tables 3.10-3.12.

As with heat pumps, in organic Rankine cycle selection the decisions are focussed on the selection of the most appropriate working fluid based on the user-input data. Organic Rankine cycle working fluid selection is based on the same criteria as heat pumps, as outlined in Section 4.3.5. However, in this case it is also important to consider the nature of the fluid. As previously, explained in Section 3.3.2, in organic Rankine cycle design only "dry" working fluids should be considered in order to ensure a dry turbine outlet without the need for a large superheat at the inlet (which would be detrimental to the overall thermal efficiency when using a finite heat source such as industrial low-grade waste heat). Therefore, R717 (ammonia) is excluded from the list of selected working fluids for the organic Rankine cycle knowledge-base, and the considered fluids are as follows (Table 4.5).

Table 4.5. Organic Rankine cycle working fluids included in the system knowledge-base

Name	Classification	Critical Temperature	Critical Pressure	Maximum Evaporator Temperature	Global Warming Potential	ASHRAE Safety Classification
		(^{o}C)	bar	(^{o}C)		
R134a	HFC	101.1	40.6	85	1430	A1
R245fa	HFC	154.0	36.5	140	1030	B1

R600	HC	152.0	38.0	140	20	A3

Note: HFC stands for Hydrofluorocarbon; HC stands for hydrocarbon; ASHRAE safety rating A denotes no toxicity at concentrations below 400ppm; ASHRAE safety rating B denotes toxicity at concentrations below 400ppm; ASHRAE safety rating 1 denotes non-flammable; ASHRAE safety rating 2 denotes moderate flammability; ASHRAE safety rating 3 denotes high flammability.

The fluids shown in Table 4.5 are sufficient to provide an ORC solution for most eventualities. For example, if flammability cannot be tolerated, two fluids are available for use (R245fa, and R134a). Also, if toxicity cannot be tolerated, two fluids are available for use (R600 and R134a). Finally, if both toxicity and flammability cannot be tolerated then R134a is available for use with the drawback that the maximum evaporator temperature of this fluid is only 85^{o}C which will be detrimental to the overall thermal efficiency of the cycle. However, this drawback is unavoidable as no *standard* working fluids exist with both an ASHRAE safety rating of A1 and critical temperature greater than 101.1 ^{o}C.

The second level of selection for organic Rankine cycles, therefore, is the selection of one of these three working fluids based on the plant tolerance to flammability, toxicity and the available heat source temperature. Also considered at this stage is the minimum source temperature required to use an organic Rankine cycle which is defined as 73^{o}C (Brasz, 2011) (see Table 3.12). If the heat source temperature is lower than this constraint then one cannot use an organic Rankine cycle.

The second level of selection for organic Rankine cycles is depicted overleaf in Figure 4.13.

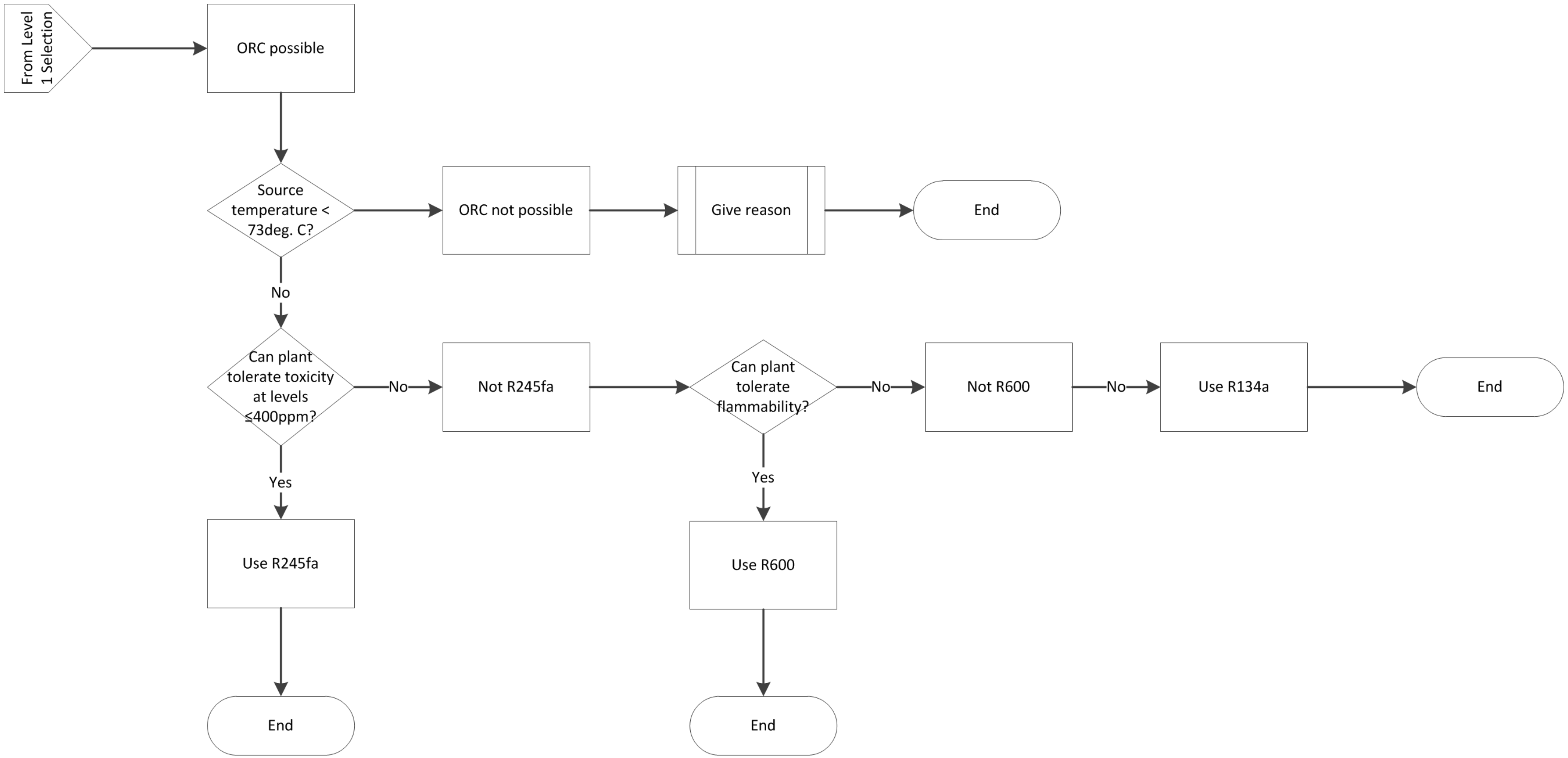


Figure 4.13. Organic Rankine cycle working fluid selection knowledge-base

4.4. Technology Design Knowledge-Base

The final stage of the KBS is to provide a *first design* of the chosen technologies in order to provide the user with data as to which is the best solution for the plant in question. Therefore, the final stage of developing the knowledge-base system is split into two parts. First of all, the results the system must generate must be specified (see Section 4.4.1). Secondly, design methodologies must be devised for each of the technologies (see Sections 4.4.2 – 4.4.4).

4.4.1. Knowledge-Based System results

The KBS must provide the user with enough data to be able to make an informed decision regarding whether waste heat recovery is suitable at the plant in question, hence, the calculated results are key to the success of the system. The following results are necessary to the "*final*" decision making process for the user and are therefore included in the system:

- Size (kW) of the units: this is crucial for all types of system. For heat exchangers this will be one value, for heat pumps this will be the size of each heat exchanger and the compressor/drive required, for organic Rankine cycles this will be the size of each heat exchanger, the turbine and the pump, and for mechanical vapour recompression this will be the heat exchanger and compressor/drive size.
- Physical dimensions: this will be provided for heat exchangers using data from manufacturers when available. This will aid the transition from KBS results to project realisation. This is more difficult for other types of system as full mechanical design of any turbines/compressors would have to be completed by a professional contractor before installation. Hence, this data is only included for heat exchangers.
- Capital cost: This will be provided for all types of system (where available). Suitable estimation techniques must be used which will vary in accuracy depending on the data available. It must be acknowledged that not all

manufacturers are willing to provide capital cost data for such projects and hence some data may be missing.

- Utility savings: To be provided for all types of system. This can be calculated based on the design data (heat exchanger duties, net power output of ORC *etc*) and the plant hours of operation per year.
- Potential cost savings: To be provided for all types of system. This can be calculated based on the utility savings and the utility costs.
- Potential greenhouse gas emission savings: To be provided for all types of system. This can be calculated based on the utility savings and the associated emissions of utilities.
- Effectiveness: This is an indicator of how efficient a heat exchanger design is and therefore must be included.
- Thermal efficiency: This is an indicator of how successful an organic Rankine cycle design is and therefore must be included.
- Coefficient of performance (COP): This is an indicator of how successful a heat pump (closed cycle and mechanical vapour recompression) design is and therefore must be included.
- Inlet/outlet temperature and pressure (pressure drop in heat exchangers): This data is included for all types of system in order to aid a physical design of the suggested waste heat recovery equipment should the user decide the results are suitable. For heat pumps, organic Rankine cycle and mechanical vapour recompression this must include a number of temperatures and pressures at each stage of the cycle in a flow diagram.

4.4.2. Heat Exchanger Design

Methods for heat exchanger design are well established and numerous shortcut methods are available for sizing the equipment which are to aid the building of the design knowledge base. Where available, manufacturer data will also be used to create accurate data for key results such as physical size, pressure drop, effectiveness and capital cost. Other data such as "typical" overall heat transfer coefficients and “typical” costs will be used where more accurate manufacturer data is not available.

The initial stage in the design of any heat exchanger is the completion of the heat balance, which is included in the system knowledge-base according to Figures 4.14-17 overleaf.

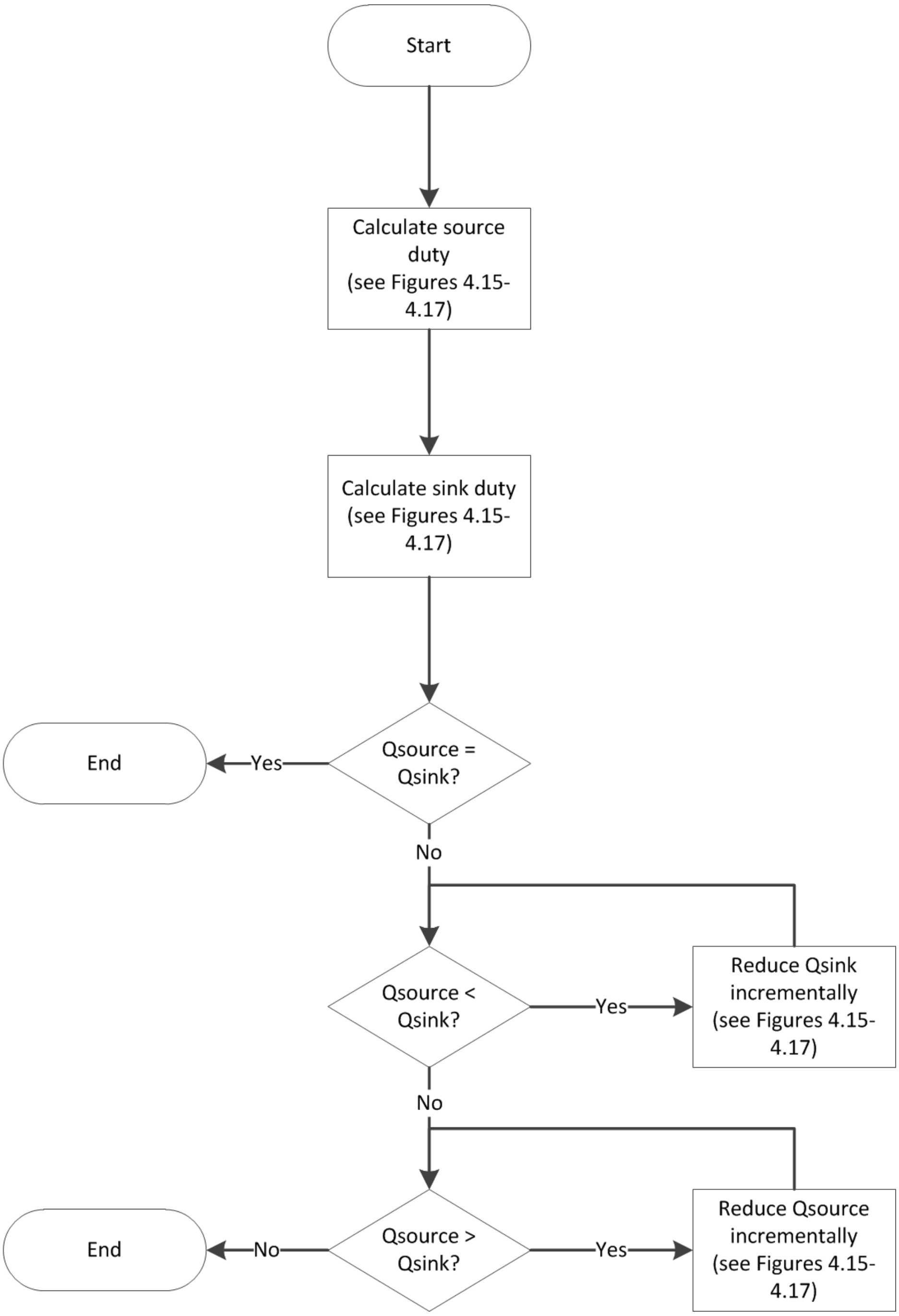


Figure 4.14. Routine to calculate the heat balance in heat exchanger design

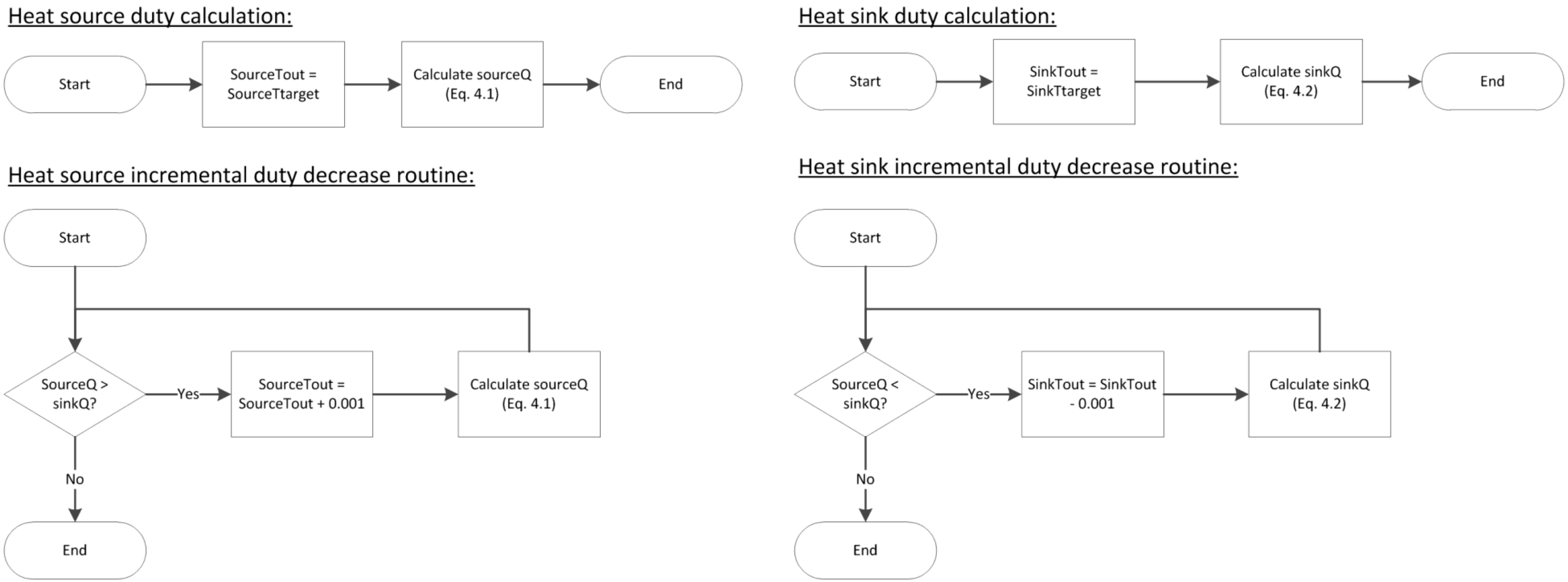


Figure 4.15. Routine to calculate heating duties and reduce duty incrementally when no phase change occurs

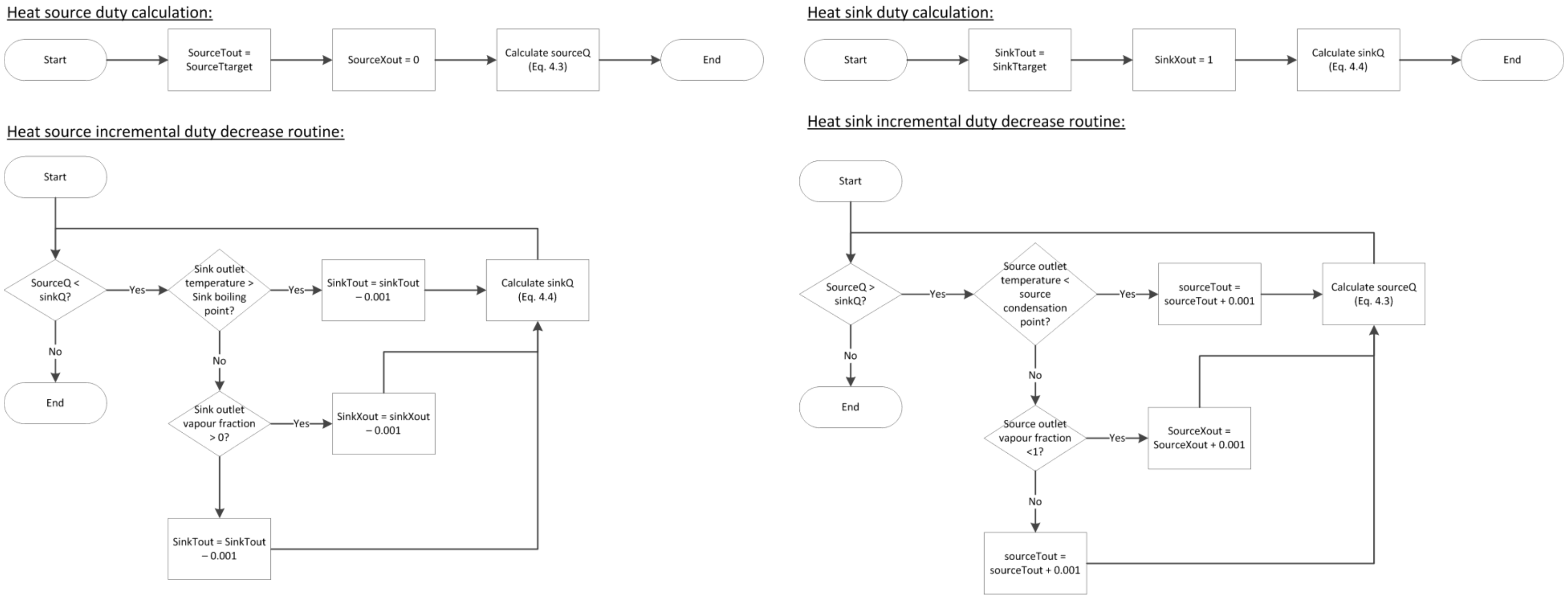


Figure 4.16. Routine to calculate heating duties and reduce duty incrementally when boiling or condensation occurs

Heat source duty calculation:

Heat source incremental duty decrease routine:

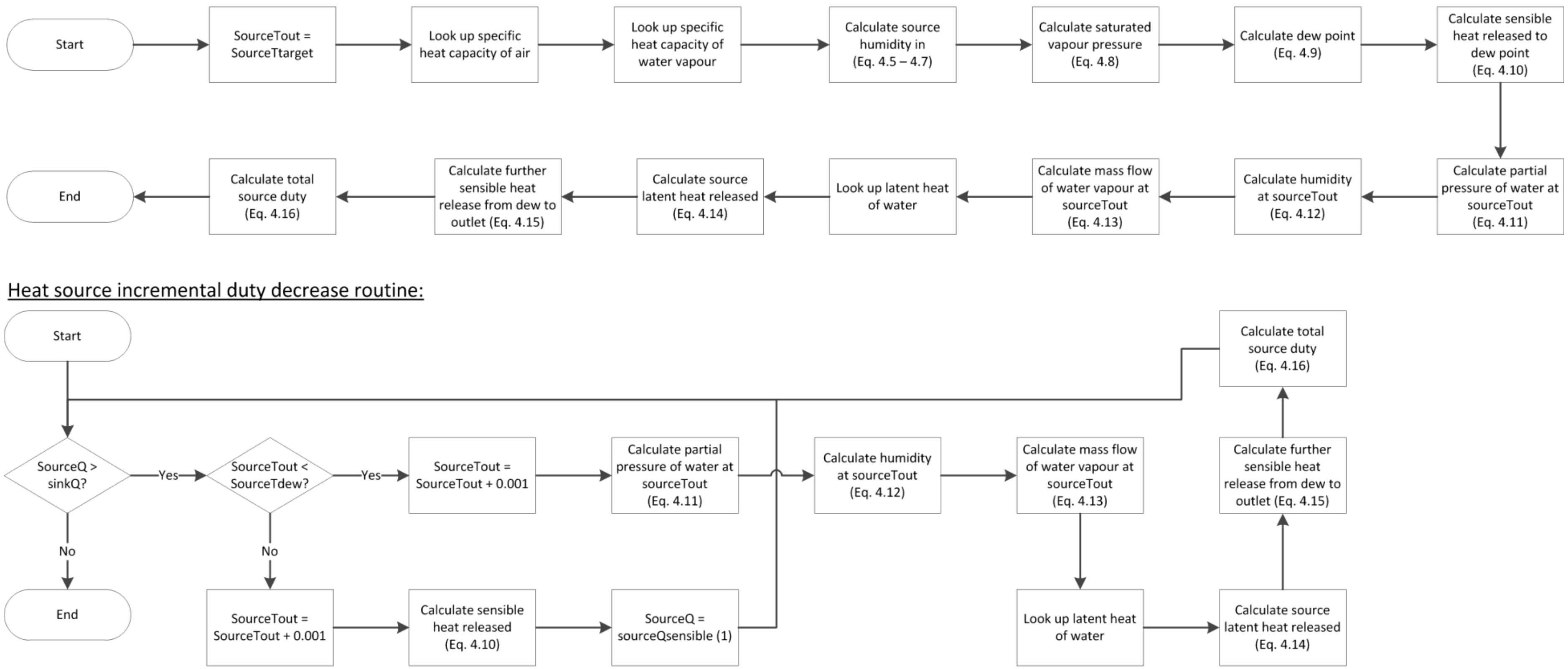


Figure 4.17. Routine to calculate heating duty and reduce duty incrementally for a humid air heat source

The list of equations relevant to Figures 4.14 - 4.17 is as follows:

$$Q_{source} = \dot{m}_{source} Cp_{source} \Delta T_{source} \quad \text{kW} \quad \text{Eq. 4.1}$$

$$Q_{sink} = \dot{m}_{sink} Cp_{sink} \Delta T_{sink} \quad \text{kW} \quad \text{Eq. 4.2}$$

$$Q_{source} = \left(\dot{m}_{source} Cp_{source,vap} (T_{source,in} - T_{source,cond})\right) + (\dot{m}_{source} \Delta H_{evap}) + \left(\dot{m}_{source} Cp_{source,liq} (T_{source,dew} - T_{source,out})\right) \quad \text{kW} \quad \text{Eq. 4.3}$$

$$Q_{sink} = \left(\dot{m}_{sink} Cp_{sink,liq} (T_{sink,boil} - T_{sink,in})\right) + (\dot{m}_{sink} \Delta H_{evap}) + \left(\dot{m}_{sink} Cp_{sink,vap} (T_{sink,out} - T_{sink,boil})\right) \quad \text{kW} \quad \text{Eq. 4.4}$$

$$\dot{m}_{water,in} = \frac{\%_{water,in}}{100} \dot{m}_{source} \quad \text{kg/s} \quad \text{Eq. 4.5}$$

$$\dot{m}_{air} = \dot{m}_{source} - \dot{m}_{water,in} \quad \text{kg/s} \quad \text{Eq. 4.6}$$

$$H_{in} = \frac{\dot{m}_{water,in}}{\dot{m}_{air}} \quad \text{kg/kg} \quad \text{Eq. 4.7}$$

$$Pwo_{sat} = \frac{H_{in} \left(\frac{M_A}{M_W}\right) . P_{source}}{1 + H_{in} (\frac{M_A}{M_W})} \quad \text{Pa} \quad \text{Eq. 4.8}$$

$$T_{source,dew} = \left[\frac{-5132}{\ln Pwo - 20.386}\right] \text{-273} \quad {}^{o}\text{C} \quad \text{Eq. 4.9}$$

$$Q_{source,sensible} = \dot{m}_{source} Cp_{source} (T_{source,in} - T_{source,dew}) \quad \text{kW} \quad \text{Eq. 4.10}$$

$$Pwo_{out} = \exp\left(20.386 - \frac{5132}{T_{source,out}}\right) * \frac{1}{0.0075} \quad \text{Pa} \quad \text{Eq. 4.11}$$

$$H_{out} = \frac{Pwo_{out}}{P_{source} - Pwo_{out}} \frac{M_W}{M_A} \quad \text{kg/kg} \quad \text{Eq. 4.12}$$

$$\dot{m}_{water,out} = H_{out} \dot{m}_{air} \quad \text{kg/s} \quad \text{Eq. 4.13}$$

$$Q_{source,latent} = \Delta\dot{m}_{water}\Delta H_{evap} \qquad \text{kW} \qquad \text{Eq. 4.14}$$

$$Q_{source,sensible2} = \dot{m}_{air}Cp_{air}(T_{source,dew} - T_{source,out}) \qquad \text{kW} \qquad \text{Eq. 4.15}$$

$$Q_{source,total} = Q_{source,sensible} + Q_{source,sensible2} + Q_{source,latent} \qquad \text{kW} \qquad \text{Eq. 4.16}$$

Note: (1) the saturated water vapour pressure found in Eq. 4.8 must be converted to units of mmHg for use in Eq. 4.9 using the conversion mmHg = 0.0075Pa. (2) The overall system pressure in Eq. 4.9 is provided by the user in units of bar which must be converted to Pa using the conversion bar = 100000Pa. (3) in Eq. 4.9, the “-273” term is introduced to convert the final temperature value from Kelvin scale to degrees centigrade. (4) In Eq. 4.11 the “(1/0.0075)” term is included to convert the final pressure from mmHg to Pa. (5). Eq. 4.11 is the vapour pressure equation for water and the constants are specific to this fluid only.

Following the procedures in Figures 4.14-4.17, a correct heat balance is in place and the source/sink duty is defined along with the outlet temperatures. At this stage, the software must make use of equations/data specific to the type of heat exchanger in question in order to produce a “*first design*” of the unit. The design data available for each type of heat exchanger varies from unit-to-unit, as listed in Table 4.6 overleaf along with brief details of the design methodology.

Table 4.6. Summary of heat exchanger design methodologies for use in KBS

Heat Exchanger Type	Data Available /Assumptions made	Design Methodology
Run-around-coil	Data from numerous manufacturers stating maximum effectiveness, maximum volumetric flow rate for varying size of units, typical pressure drops (for each size of unit). Source: Reay (1979). Relevant data also found in common texts such as Coulson and Richardson, as this unit is essentially a finned-tube-in-shell heat transfer between water and a gas (hence heat transfer coefficients are relatively easy to predict). Capital cost data may be estimated using data for finned tube heat exchangers according to Best Practice Programme (2000) data	Combination of various sources: • Use estimate heat transfer coefficients to size the coils • Use manufacturer recommendations for effectiveness to set the LMTD • Use manufacturer data for "typical" pressure drop • Size pump for carrier fluid using tubular pressure drop equations • Estimate capital cost using data for finned tube unit • See Figure 4.18
GG-Plate	Data from numerous manufacturers stating maximum effectiveness, maximum volumetric flow rate for varying size of units, typical pressure drops (for each size of unit). Source: Reay (1979)	Use manufacturer data: • Select unit size based on volumetric flow of source/sink • Set outlet temperatures based on maximum effectiveness • Calculate pressure drop according to "typical" data See Figure 4.18 (note: no accurate capital cost data available for this unit)
Rotary Regenerator	Data from numerous manufacturers stating maximum effectiveness, maximum volumetric flow rate for varying size of units, typical pressure drops (for each size of unit). Source: Reay (1979)	Use manufacturer data: • Select unit size based on volumetric flow of source/sink • Set outlet temperatures based on maximum effectiveness • Calculate pressure drop according to "typical" data See Figure 4.18 (note: no accurate capital cost data available for this unit)
Welded Plate	Data from Alfa Laval (2013) stating typical plate/overall unit sizes and maximum flow rates. Data from Best Practice Programme (2000) stating typical heat transfer coefficients for common fluid types, pressure drop equations and estimate cost functions.	Combination of both data sources: • Use Best Practice Programme data to estimate overall heat transfer coefficients • Use log mean temperature difference method (LMTD) to size unit • Use Alfa Laval data to select best plate size and calculate physical dimensions of unit • Use Best Practice Programme data to estimate pressure drops See Figure 4.17

Shell and Tube	Numerous data from a variety of sources stating design methodologies, typical shell/tube sizes, typical number of passes, pressure drop equations and overall heat transfer coefficients for common fluid types. Main sources: Coulson and Richardson (2005), TEMA (2013) and Best Practice Programme (2000).	Combination of all data sources: • Use LMTD method and estimate heat transfer coefficients to size the unit • Select from "typical" shell and tube dimensions to calculate physical dimensions of unit • Use common equations to estimate pressure drops See Figure 4.17
Gasketted Plate	Data from Alfa Laval (2013) stating typical plate/overall unit sizes and maximum flow rates. Data from Best Practice Programme (2000) stating typical heat transfer coefficients for common fluid types and pressure drop equations and estimate cost functions.	Combination of both data sources: • Use Best Practice Programme data to estimate overall heat transfer coefficients • Use log mean temperature difference method (LMTD) to size unit • Use Alfa Laval data to select best plate size and calculate physical dimensions of unit • Use Best Practice Programme data to estimate pressure drops See Figure 4.17
Brazed Plate	Data from Alfa Laval (2013) stating typical plate/overall unit sizes and maximum flow rates. Data from Best Practice Programme (2000) stating typical heat transfer coefficients for common fluid types and pressure drop equations and estimate cost functions.	Combination of both data sources: • Use best practice programme data to estimate overall heat transfer coefficients • Use log mean temperature difference method (LMTD) to size unit • Use Alfa Laval data to select best plate size and calculate physical dimensions of unit • Use best practice programme data to estimate pressure drops See Figure 4.17
Plate and Shell	Data from Alfa Laval (2013) stating typical plate/overall unit sizes and maximum flow rates. Data from Best Practice Programme (2000) stating typical heat transfer coefficients for common fluid types for other plate units is assumed to be valid for this unit due to similar flow regimes.	Combination of both data sources: • Use best practice programme data to estimate overall heat transfer coefficients • Use log mean temperature difference method (LMTD) to size unit • Use Alfa Laval data to select best plate size and calculate physical dimensions of unit See Figure 4.17 (note: no accurate cost or pressure drop data available for this unit)
Spiral Plate	Very limited data for this unit. Heat transfer coefficient estimates are published by Best Practice Programme (2000). Data from Alfa Laval state typical unit sizes and throughputs. Correlations for capital cost and pressure drop factors do not exist in the public domain	Combination of both data sources: • Use best practice programme data to estimate overall heat transfer coefficients • Use log mean temperature difference method (LMTD) to size unit • Use Alfa Laval data to select best plate size and calculate physical dimensions of unit

		See Figure 4.17 (note: no accurate cost or pressure drop data available for this unit)
Finned-Tube	Numerous data from a variety of sources stating design methodologies, typical shell/tube sizes, typical number of passes, pressure drop equations and overall heat transfer coefficients for common fluid types. Sources include Coulson and Richardson (2005), TEMA (2013) and Best Practice Programme (2000).	Combination of all data sources: • Use LMTD method and estimate heat transfer coefficients to size the unit. Here fin pitch and type must be selected based on the type of liquid (and subsequent difference in heat transfer coefficients on each side). • Select from "typical" finned-tube and shell dimensions to calculate physical dimensions of unit • Use common equations to estimate pressure drops See Figure 4.17

Note: Appendix II shows examples of the data acquired to aid the design of each type of unit

Decision trees detailing how the system then designs each type of unit (according to the brief description given in Table 4.6) are shown in Figures 4.18-4.20 overleaf. Figure 4.18 shows the design methodology for plate and tubular-type heat exchangers using a log meant temperature difference (LMTD) type method, Figure 4.19 shows the design methodology for air-handling-type heat exchangers based on *typical* effectiveness according to manufacturer data and Figure 4.20 shows the methodology for calculating utility savings due to waste heat recovery and the subsequent economic/environmental benefits.

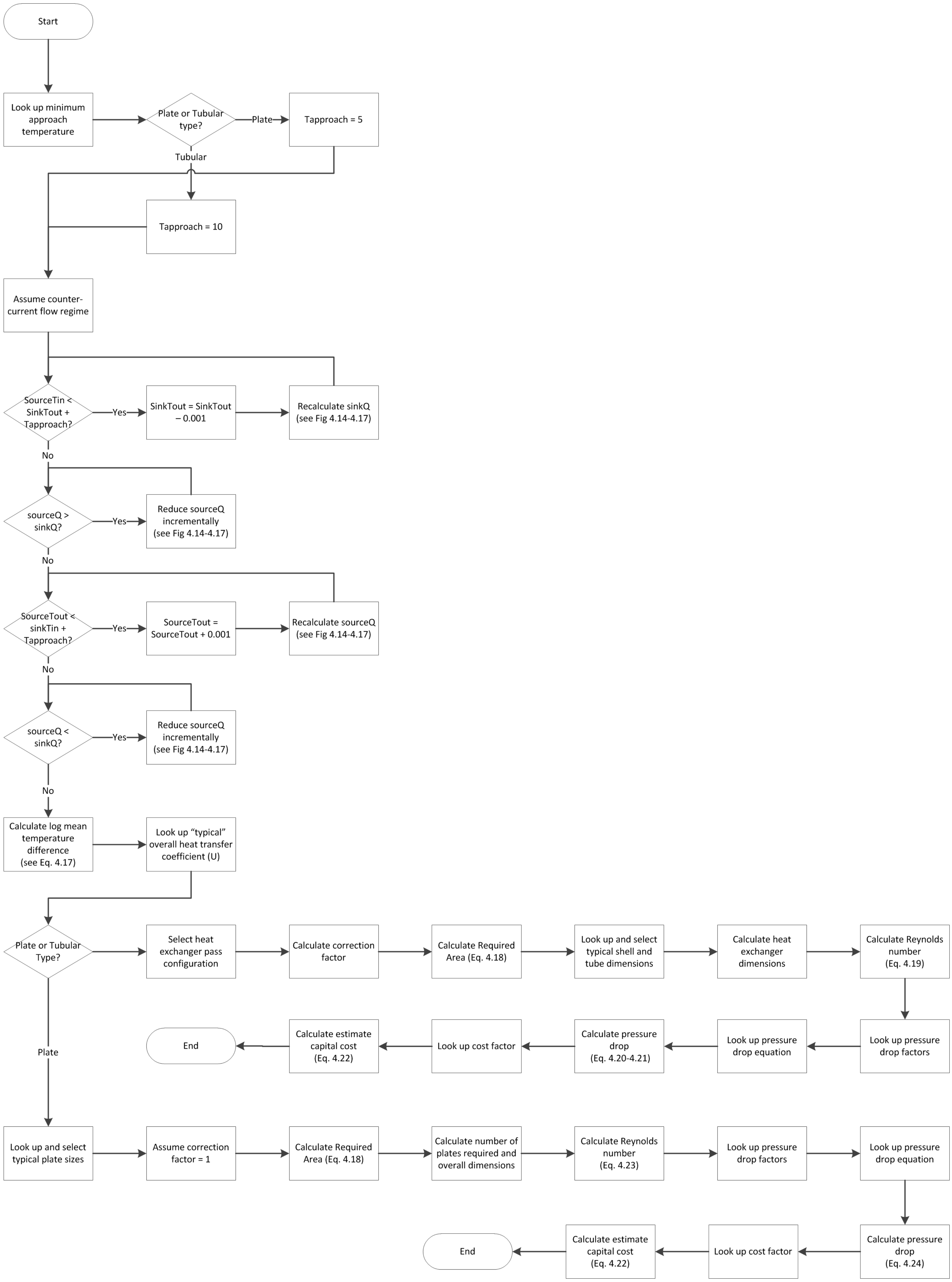


Figure 4.18. Design routine for tubular and plate-type heat exchangers

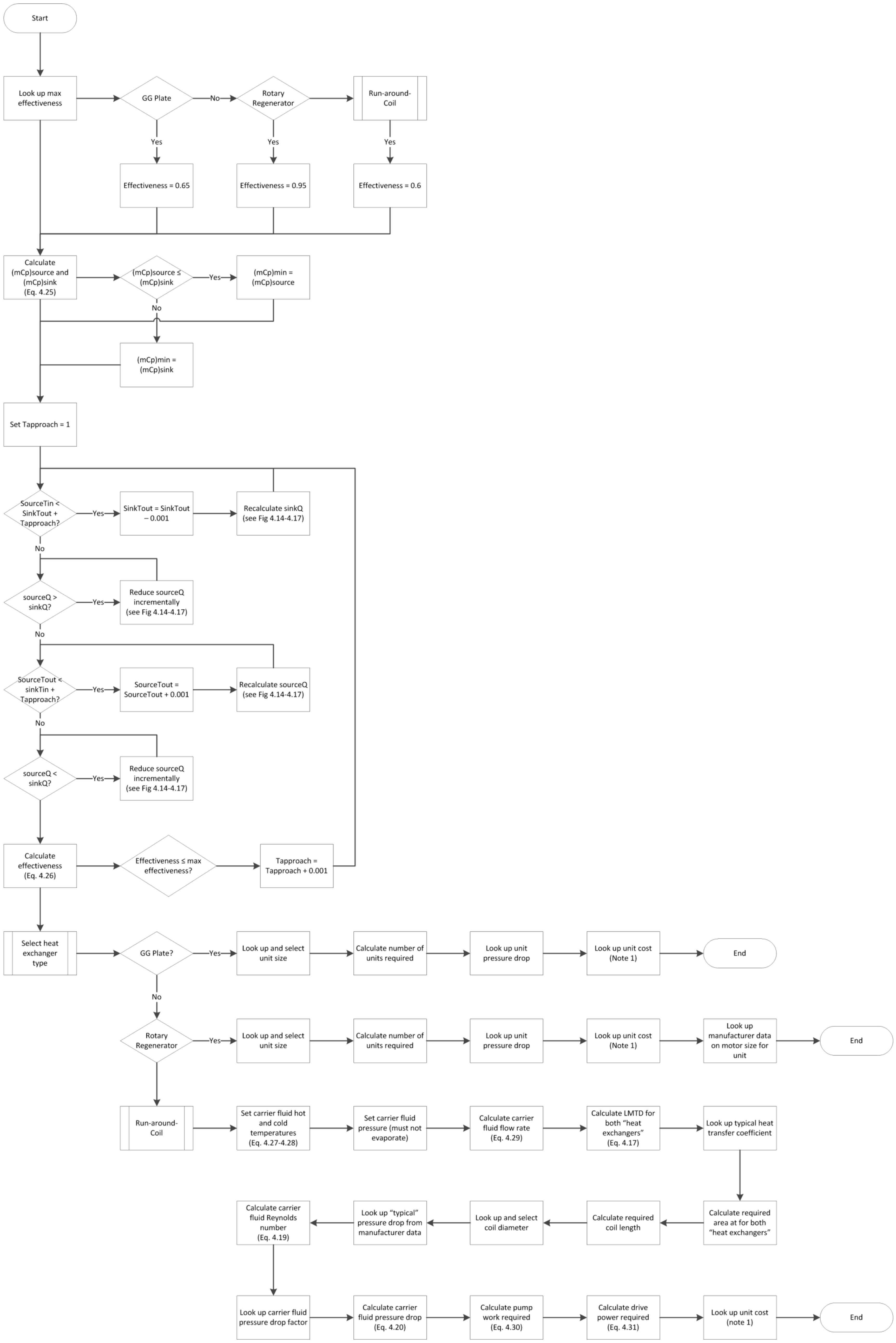


Figure 4.19. Design routine for air-handling-type heat exchangers

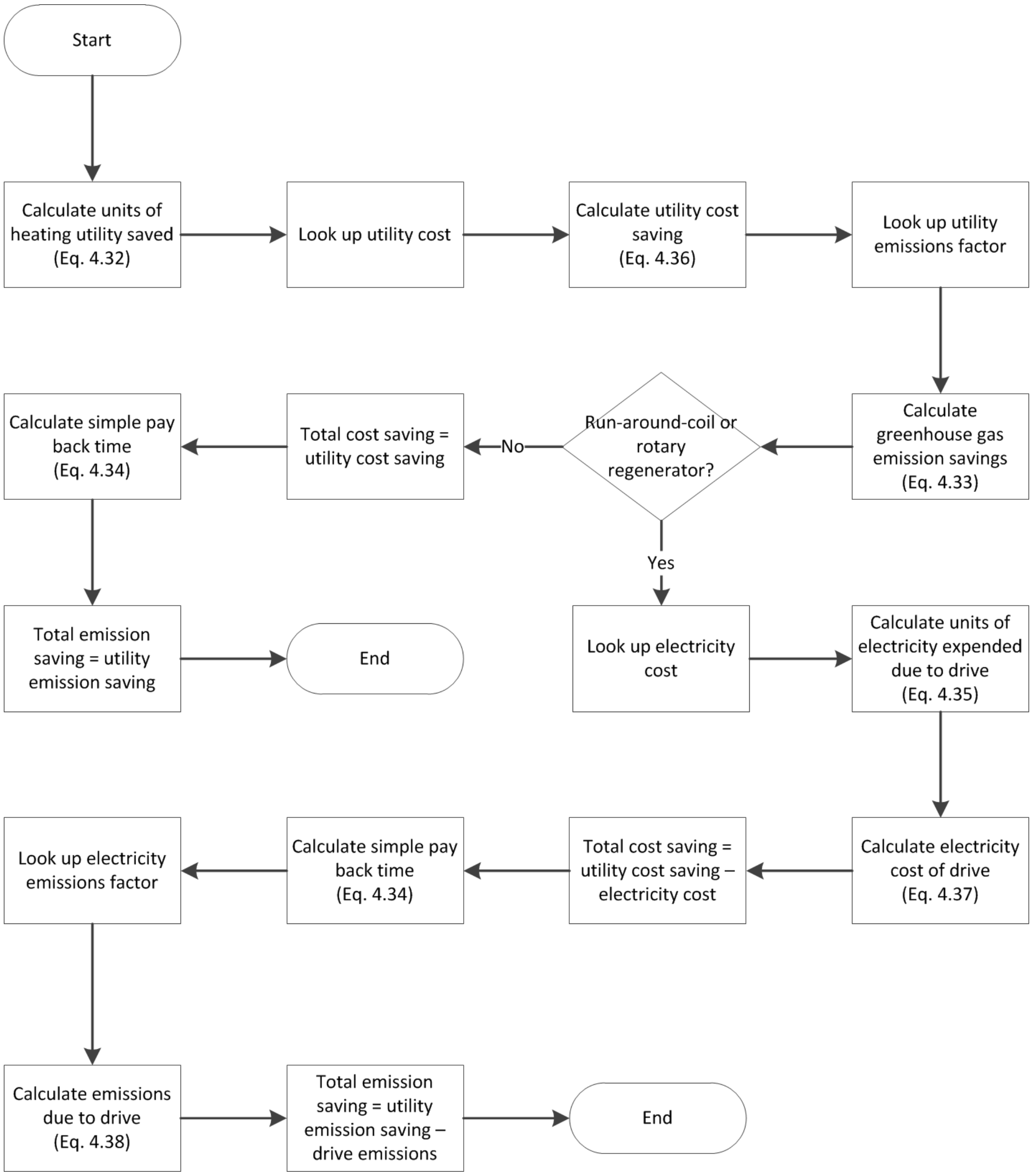


Figure 4.20. Routine for calculating utility savings and associated benefits in heat exchanger design

The list of equations relevant to Figures 4.18 - 4.20 is as follows:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$ °C Eq. 4.17

$$A = \frac{Q}{UF_T \Delta T_{LM}}$$ m^2 Eq. 4.18

$$Re = \frac{\rho v d}{\mu}$$ Eq. 4.19

$$\Delta P = N_P \left[4j_f \left(\frac{l}{d_i}\right) + 1.25\right] (\rho u^2)$$ Pa Eq. 4.20

$$\Delta P = 4j_f \frac{d_s}{d_e} \frac{l}{l_b} \rho u^2$$ Pa Eq. 4.21

$$C_{capital} = C_f * \frac{Q}{(F_T \Delta T_{LM})}$$ £ Eq. 4.22

$$Re = \frac{C_{Re} \dot{m}}{\mu}$$ Eq. 4.23

$$\Delta P_{plate} = f_{plate} \left(\frac{\dot{m}^2}{\rho}\right)$$ Pa Eq. 4.24

$$(mCp)_{source} = \dot{m}_{source} * Cp_{source}$$ kW/Ks Eq. 4.25

$$\eta_{eff} = \frac{Q}{Q_{max}} = \frac{Q}{(\dot{m}Cp)_{min}(T_{source,in} - T_{sink,in})}$$ Eq. 4.26

$$T_{carrier,hot} = \frac{T_{source,in} - T_{sink,out}}{2}$$ °C Eq. 4.27

$$T_{carrier,cold} = \frac{T_{source,out} - T_{sink,in}}{2}$$ °C Eq. 4.28

$$\dot{m}_{carrier} = \frac{Q}{Cp_{carrier}(T_{carrier,hot} - T_{carrier,cold})}$$ kg/s Eq. 4.29

$$W_{pump} = \frac{\dot{m}_{carrier}}{\rho_{carrier}} \frac{\Delta P_{carrier}}{\eta_{pump}} (0.001)$$ kW Eq. 4.30

$$W_{drive} = \frac{W_{pump}}{\eta_{drive}}$$ kW Eq. 4.31

$$N_{units,saved} = Q_{sink} h_{plant} \quad \text{kWh/year} \quad \text{Eq. 4.32}$$

$$E_{saved} = N_{units,saved} * E_{utility} \quad tCO_2\text{eq/year} \quad \text{Eq. 4.33}$$

$$PB = \frac{C_{capital}}{C_{saved}} \quad \text{Year} \quad \text{Eq. 4.34}$$

$$N_{units,drive} = W_{drive} h_{plant} \quad \text{kWh/year} \quad \text{Eq. 4.35}$$

$$C_{saved} = N_{units,saved} C_{utility} \quad \text{£/year} \quad \text{Eq. 4.36}$$

$$C_{drive} = N_{units,drive} C_{electricity} \quad \text{£/year} \quad \text{Eq. 4.37}$$

$$E_{drive} = N_{units,drive} E_{electricity} \quad tCO_2\text{eq/year} \quad \text{Eq. 4.38}$$

Note: (1) Eq. 4.23 and 4.24 represent the Reynolds number and pressure drop equations specifically for Alfa Laval plate heat exchangers. As Alfa Laval dimensions are used in the heat exchanger design, it is assumed this method is more accurate than “standard” pressure drop/Reynolds number equations; (2) In Eq. 4.25, for the case of the heat sink, replace source data with sink data; (3) in Eq. 4.30 the pump efficiency is taken as 75%; (4) in Eq. 4.30 the “0.001” term is included to convert the units to kW from W; (5) in Eq. 4.31 the drive efficiency is taken as 95%

4.4.3. Heat Pump and MVR Design

Design routines for heat pumps and MVR are not as common as for heat exchangers (as discussed in Section 4.4.2). Therefore, methods must be created which encapsulate the following aspects:

- Limitations of working fluids (temperature, pressure limits *etc*) (closed cycle vapour compression heat pumps only)
- The minimum system COP required for the system to be profitable in terms of both economics and greenhouse gas emissions (as discussed in Section 3.2.1)
- An attempt to achieve user-requested target temperatures for the heat source/sink
- The ability of the heat pump to provide simultaneous useful heating and cooling (closed cycle vapour compression heat pumps only)

- The ability of an MVR heat pump to accommodate existing heat exchangers thereby reducing the capital cost required of the system (where possible, this will be done and both results will be shown to the user)

Only the standard cycle configurations are considered, as shown below in Figures 4.21-4.23. This is deemed sufficient for cycle "*first design*". It is assumed that any further optimization or inclusion of further heat exchangers would occur during full physical design.

Note that for some working fluids, superheat is required at the compressor inlet to ensure a dry outlet feed. In such cases, the superheated cycle is used. For MVR design, two batch-type configurations were considered as shown in Figure 4.22 and 4.23. Figure 4.22 shows the use of an internal heating coil to transfer heat to the boiling vessel, whereas Figure 4.23 shows the use of an external heat exchanger to transfer heat to the vessel. In MVR systems using an external heat exchanger (Figure 4.23), the recirculating fluid (from the evaporator) is kept at a certain pressure to ensure the heat exchanger outlet temperature is below the boiling point. Boiling then occurs upon re-entry to the vessel.

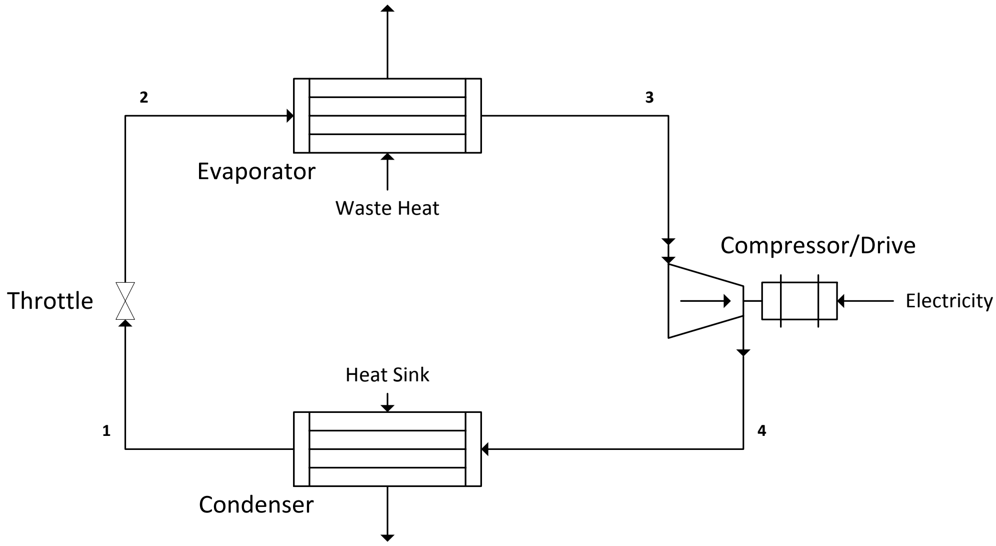


Figure 4.21. Standard vapour compression heat pump configuration

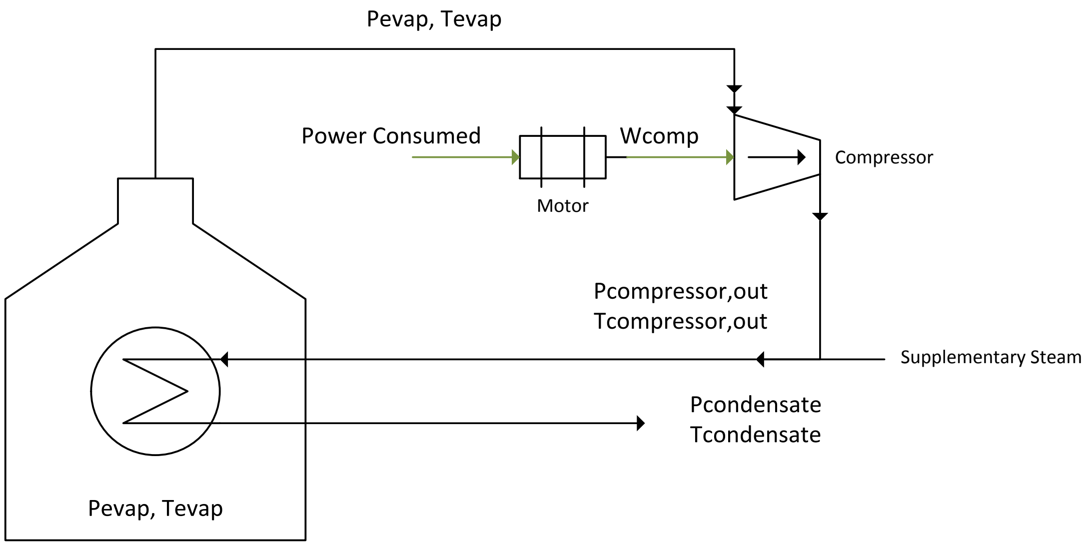


Figure 4.22. Standard MVR configuration with internal heating coils

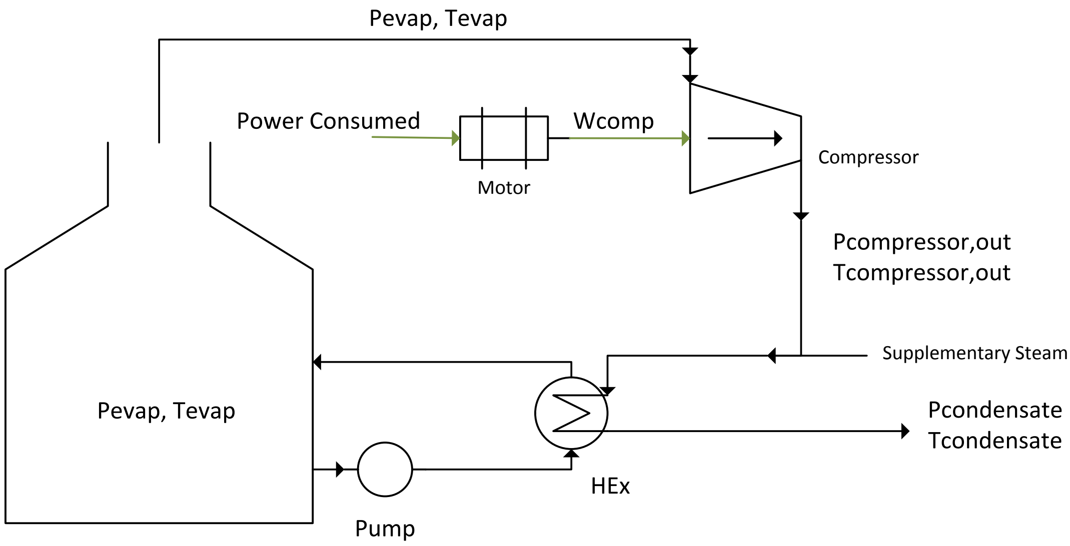


Figure 4.23. Standard MVR configuration with an external recirculation heat exchanger

The methods created are displayed in the form of decision trees in Figures 4.24-4.26 shown overleaf. Figure 4.24 shows the design method for closed cycle vapour compression heat pumps, Figure 4.25 shows the initial design stage for MVR to determine whether or not the existing heat exchanger may be used for MVR and Figure 4.26 shows the subsequent design procedure.

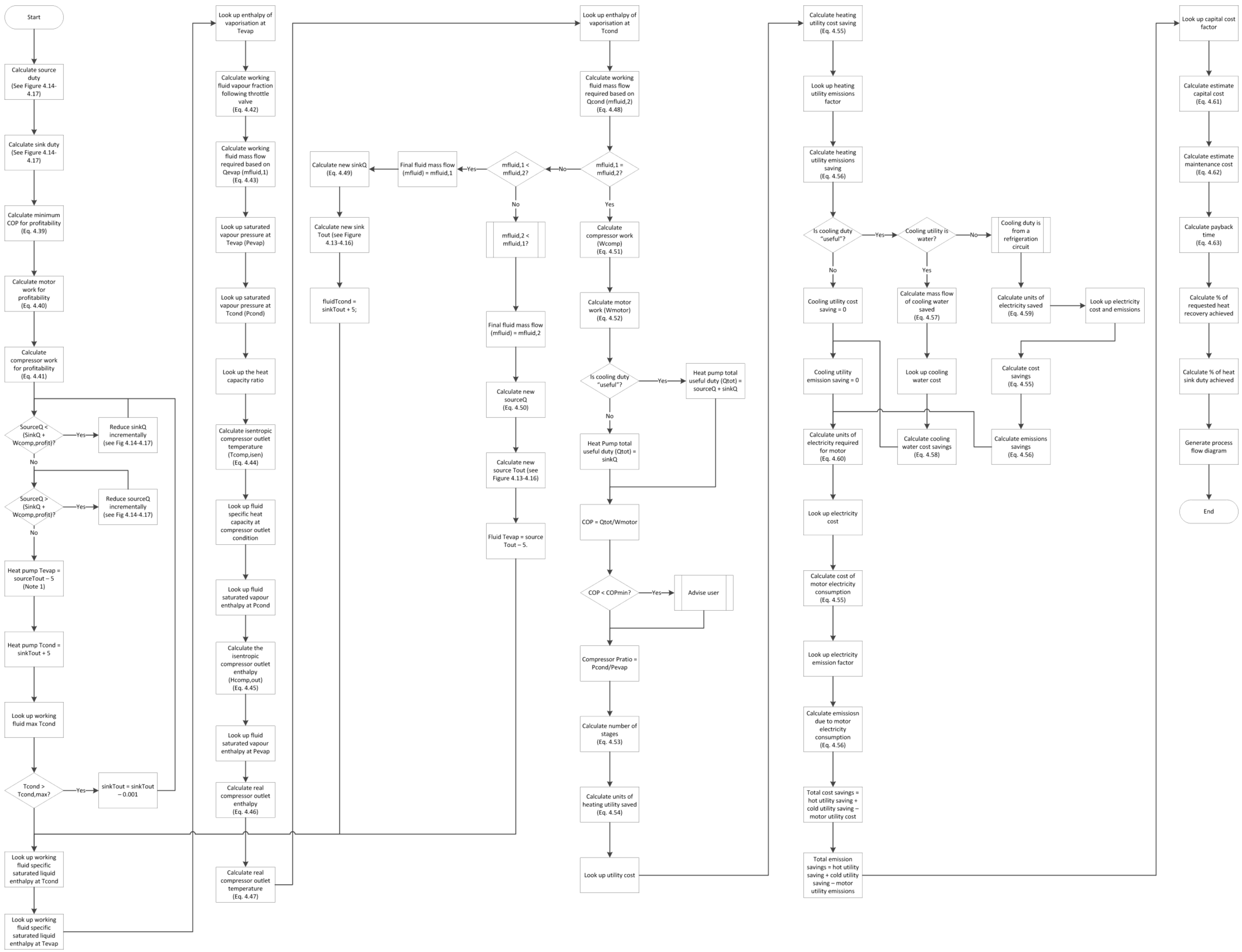


Figure 4.24. Design routine for close-cycle vapour compression heat pumps

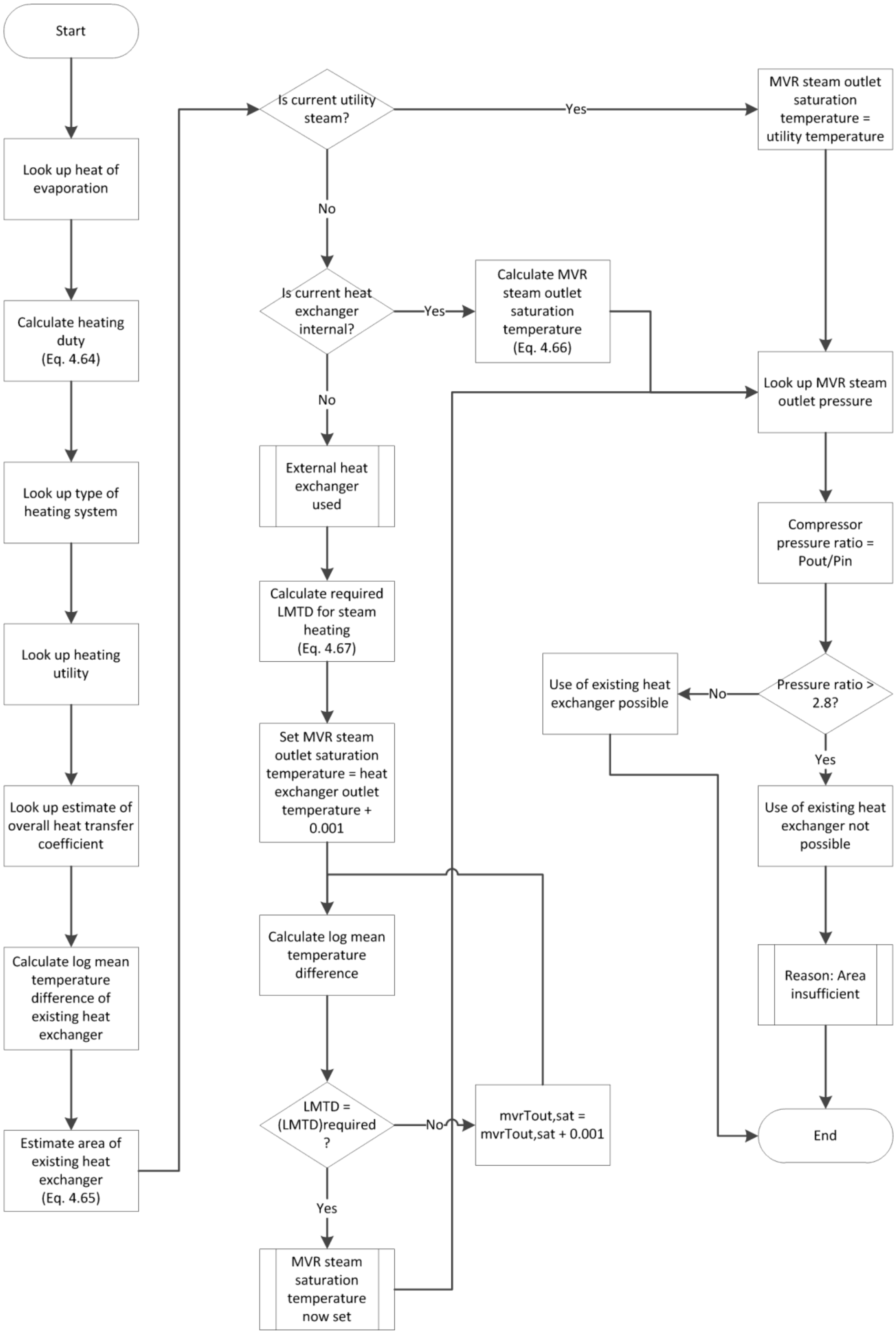


Figure 4.25. Routine to assess current heat exchanger suitability for MVR system

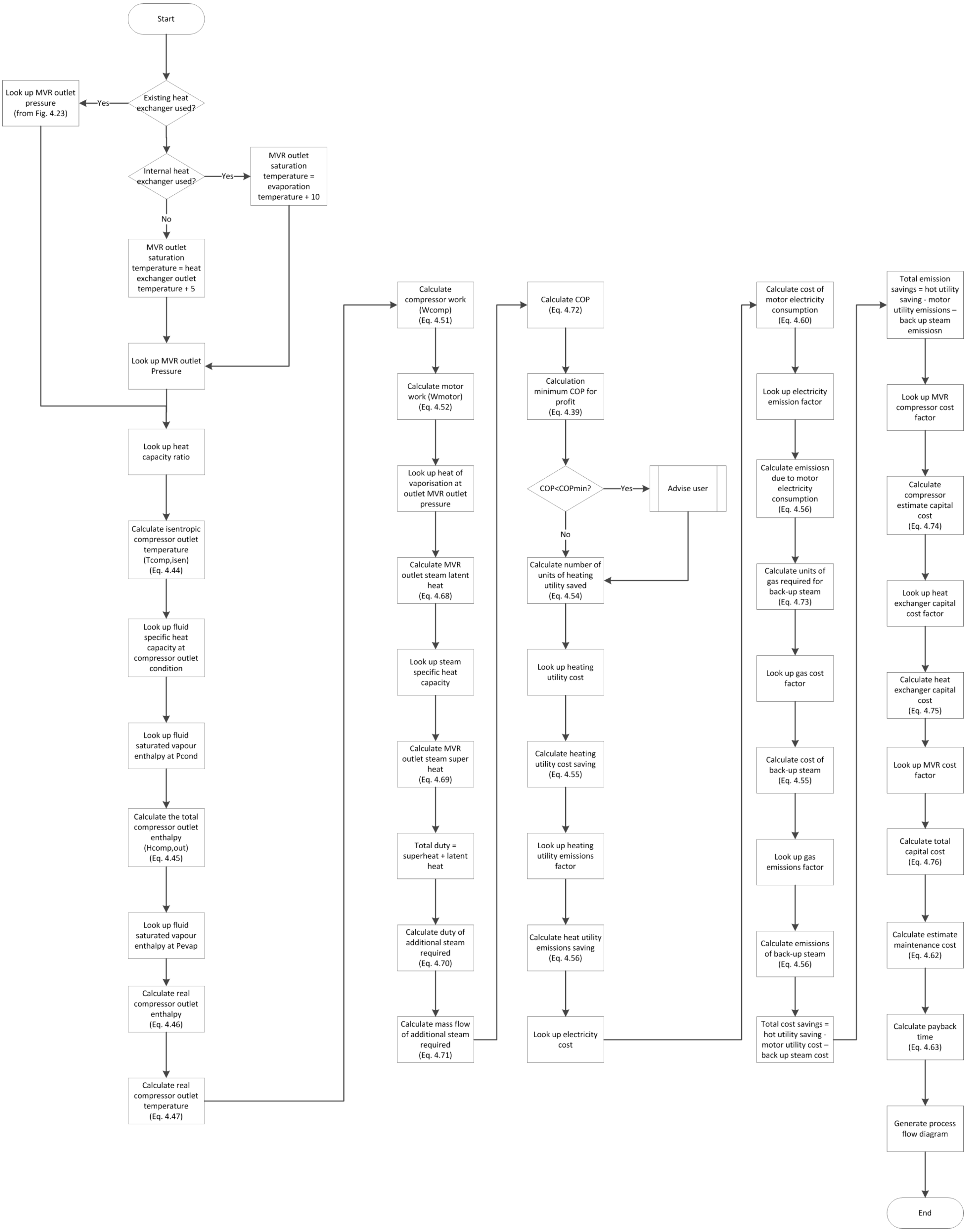


Figure 4.26. Design routine for MVR

The following data/assumptions are used during the closed cycle vapour compression heat pump and MVR design procedures shown in Figures 4.24-4.26:

- Compressor isentropic efficiency of 80%.
- Motor efficiency of 95%.
- Back-up steam in MVR is provided by a gas boiler of 75% efficiency
- MVR heat exchangers are assumed to be copper coil for internal heating systems (with a 10^{o}C typical approach temperature) and a gasketted plate heat exchanger (with a 5^{o}C typical approach temperature) for external heating systems.
- Working fluid data for close-cycle heat pumps and data for water/steam in MVR is provided by ASHRAE, 2009. See Appendix III for an example of the thermophysical data tables used. This data is programmed into the KBS database and the software can "*look-up*" the data as required.
- Heat transfer coefficients required for calculations in MVR design (particularly when assessing whether the existing heat exchanger is suitable for the new MVR duty) is taken from Coulson and Richardson, 2005.
- Capital cost factors for close-cycle heat pumps are taken from the US Department of Energy, 2008. Historical currency conversion data and the engineering price indices (The Engineer, 2013; Coulson and Richardson, 2005) are used to create cost factors valid in the UK at the present date.
- Compressor cost factors for MVR are taken from Coulson and Richardson, 2005. Again, engineering price indices are used to create cost factors valid for the present date.
- An overall system cost factor for MVR is used to account for key ancillary equipment such as de-misters, piping and civil engineering. This is taken from Allen *et al*, 1983, as 2 times the compressor cost.
- Maintenance cost factors for MVR and vapour compression heat pumps are taken from Coulson and Richardson, 2005.
- The compressor type is assumed to be centrifugal with a max. single stage pressure ratio of 2.8. The maximum number of stages for MVR systems is 1 as steam compressors command a higher capital cost than closed cycle

vapour compression heat pumps due to the low volumetric capacity of steam compared to *typical* working fluids (HFC's *etc*). Brotherton (2012) states that MVR project profitability is severely reduced if the number of MVR compression stages is greater than 1, hence, here the maximum is limited to 1 stage.

- Heating coil cost factors (for MVR internal heating systems) are taken from Coulson and Richardson, 2005. Gasketted plate heat exchanger cost factors are taken from the Best Practice Programme (2000), as described in Section 4.4.2.

The list of equations relevant to Figure 4.24-4.26 is as follows:

$$COP_{min,profit} = \frac{C_{elec}}{(C_{current}/\eta_{current})} \qquad \text{Eq. 4.39}$$

$$W_{motor} = \frac{Q_{sink}}{COP_{min,profit}} \qquad \text{kW} \qquad \text{Eq. 4.40}$$

$$W_{compressor} = W_{motor} * \eta_{motor} \qquad \text{kW} \qquad \text{Eq. 4.41}$$

$$x = \frac{H_{sat\ liquid,Pcond} - H_{sat\ liquid,Pevap}}{\Delta H_{evap,Pevap}} \qquad \text{Eq. 4.42}$$

$$m_{fluid,1} = \frac{Q_{source}}{(1-x)\Delta H_{evap,Pevap}} \qquad \text{kg/s} \qquad \text{Eq. 4.43}$$

$$T_{comp,out,isen} = T_{comp,in} * \frac{P_{cond}}{P_{evap}}^{1-\frac{1}{\gamma}} \qquad \text{K} \qquad \text{Eq. 4.44}$$

$$H_{comp,out,isen} = (Cp_{vap} * (T_{comp,out,isen} - T_{comp,out,sat})) + H_{comp,out,sat-vap} \qquad \text{kJ/kg} \qquad \text{Eq. 4.45}$$

$$H_{comp,out,real} = \frac{(H_{comp,out,isen} - H_{comp,in})}{\eta_{isen}} + H_{comp,in} \qquad \text{kJ/kg} \qquad \text{Eq. 4.46}$$

$$T_{comp,out,real} = T_{cond} + \frac{H_{comp,out,real} - H_{comp,out,sat-vap}}{Cp_{vap}} \qquad {}^{o}\text{C} \qquad \text{Eq. 4.47}$$

$m_{fluid,2} = \frac{Q_{sink}}{(Cp_{fluid,vap}(T_{comp,out,real} - Tcond) + \Delta H_{evap,Pcond})}$	kg/s	Eq. 4.48
$Q_{sink} = m_{fluid}(Cp_{fluid,vap}(T_{comp,out,real} - Tcond) + \Delta H_{evap,Pcond})$	kW	Eq. 4.49
$Q_{source} = m_{fluid}(1 - x_{fluid})\Delta H_{evap,Pevap}$	kW	Eq. 4.50
$W_{comp} = m_{fluid}(H_{comp,out,real} - H_{in})$	kW	Eq. 4.51
$W_{motor} = \frac{W_{comp}}{\eta_{drive}}$	kW	Eq. 4.52
$n_{stages} = \frac{\ln P_{ratio}}{\ln 2.8}$		Eq. 4.53
$N_{units,saved} = Q_{sink/source} h_{plant}$	kWh/year	Eq. 4.54
$C_{saved/spent} = N_{units,saved/spent} C_{utility}$	£/year	Eq. 4.55
$E_{saved/spent} = N_{units,saved/spent} * E_{utility}$	tCO_2eq/year	Eq. 4.56
$m_{water} = \frac{Q_{source}}{Cp_{water} * \Delta T} = \frac{Q_{source}}{42}$	kg/s	Eq. 4.57
$C_{water} = (m_{water} C_{water,factor}) * 3600 * h_{plant}$	£/year	Eq. 4.58
$N_{units,cooling,circuit} = \frac{Q_{source}}{COP_{circuit}} * h_{plant}$	kWh/year	Eq. 4.59
$N_{units,motor} = W_{motor} * h_{plant}$	kWh/year	Eq. 4.60
$C_{capital} = C_{factor,heatpump} * Q_{sink}$	£	Eq. 4.61
$C_{maintenace} = C_{factor,maintenance} * C_{capital}$	£/year	Eq. 4.62
$PB = \frac{C_{capital}}{C_{saved} - C_{maintenace}}$	Year	Eq. 4.63
$Q = m_{evap}\Delta H_{evap}$	kW	Eq. 4.64
$A = \frac{Q}{\Delta T_{LM}.U_{old}}$	m^2	Eq. 4.65
$T_{MVR,out,sat} = \frac{Q}{U_{MVR} + A} + T_{evap}$	^oC	Eq. 4.66

$$LMTD_{required} = \frac{Q}{U_{MVR} + A}$$ °C Eq. 4.67

$$Q_{latent} = m_{evap}\Delta H_{evap}$$ kW Eq. 4.68

$$Q_{latent} = m_{evap}Cp(T_{comp,out,real} - T_{comp,out,sat})$$ kW Eq. 4.69

$$Q_{additional} = Q_{evap} - Q_{cond}$$ kW Eq. 4.70

$$m_{additional} = \frac{Q_{additional}}{\Delta H_{evap}}$$ kg/s Eq. 4.71

$$COP = \frac{Q_{cond}}{W_{motor}}$$ Eq. 4.72

$$N_{gas,back-up\ steam} = \frac{Q_{additional}}{0.75} * h_{plant}$$ kWh/year Eq. 4.73

$$C_{capital,compressor} = C_{Factor,compressor} * W_{motor}$$ £ Eq. 4.74

$$C_{capital,HEx} = C_{factor,HEx} * Q_{cond}$$ £ Eq. 4.75

$$C_{MVR} = \left(C_{capital,compressor} * C_{factor,MVR}\right) + C_{capital,HEx}$$ £ Eq. 4.76

Note: (1) The KBS does not consider renewable heat levies, hence Eq. 4.39 only considers utility costs. (2) In Eq. 4.44 the temperature is in Kelvin but is displayed in °C throughout other areas. A sub-routine is programmed into the KBS to alter units where necessary. (3) The number of compression stages (Eq. 4.53) is rounded up to a whole number. (4) In Eq. 4.57, plant cooling water is assumed to allow a temperature change of 10°C. (5) In Eq. 4.71 if the additional duty required is less than 0, this is set to 0. (6)In Eq. 4.53 "2.8" is the max. pressure ratio of a centrifugal type compressor. This constant would vary if using a different compressor.

4.4.4. Organic Rankine Cycle Design

Design routines for organic Rankine cycles are not as common as for heat exchangers (as discussed in Section 4.4.2). Therefore, a method must be created which encapsulates the following aspects:

- Limitations of working fluids (temperature, pressure limits *etc*)
- Maximising the net power output. This is chosen rather than the thermal efficiency as waste heat is taken as a *free* resource (in terms of both

emissions and cost). Therefore, project economics and greenhouse gas reductions are proportional only to the net power output, rather than the thermal efficiency

- An attempt to achieve the user-defined heat source target temperature

Only the standard organic Rankine cycle configuration is considered, as shown below in Figure 4.27. This is deemed sufficient for a "*first design*" of the cycle. It is assumed that any further optimization or inclusion of further heat exchangers (such as a recuperator) would occur during full physical design. Note: for some working fluids (R245fa, R600) superheat is required at the turbine inlet to ensure a dry outlet feed. In such cases, the superheated cycle is used. Further discussion of organic Rankine cycle configurations is provided in Sections 2.2.3 and 3.3.2.

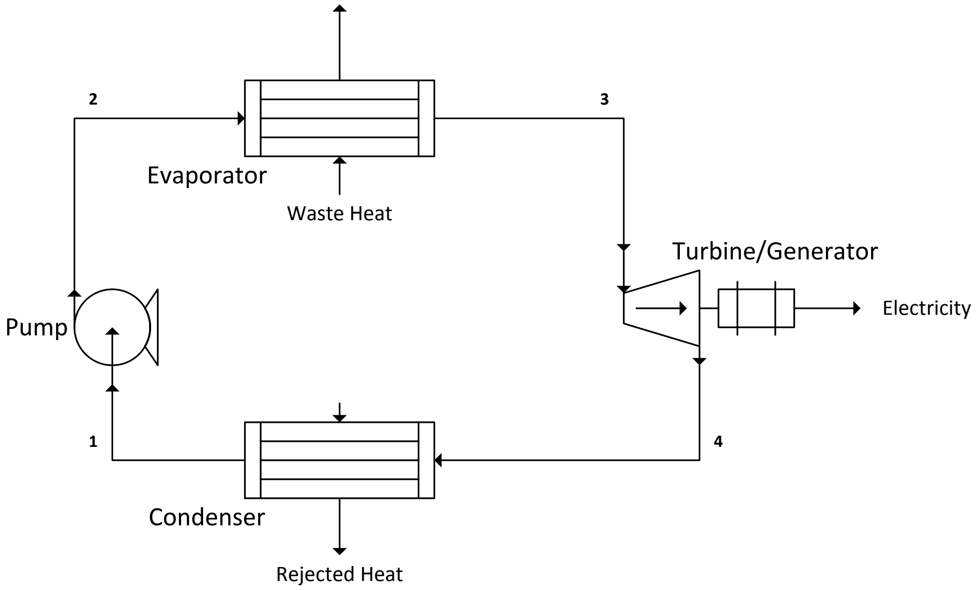


Figure 4.27. Standard ORC configuration

The methods created for automated ORC design are shown overleaf in Figure 4.28-4.29. Figure 4.28 is for heat sources with either sensible heat only (liquid or non-condensable gas) or humid air heat sources while Figure 4.29 is for (isothermal) condensing vapour heat sources.

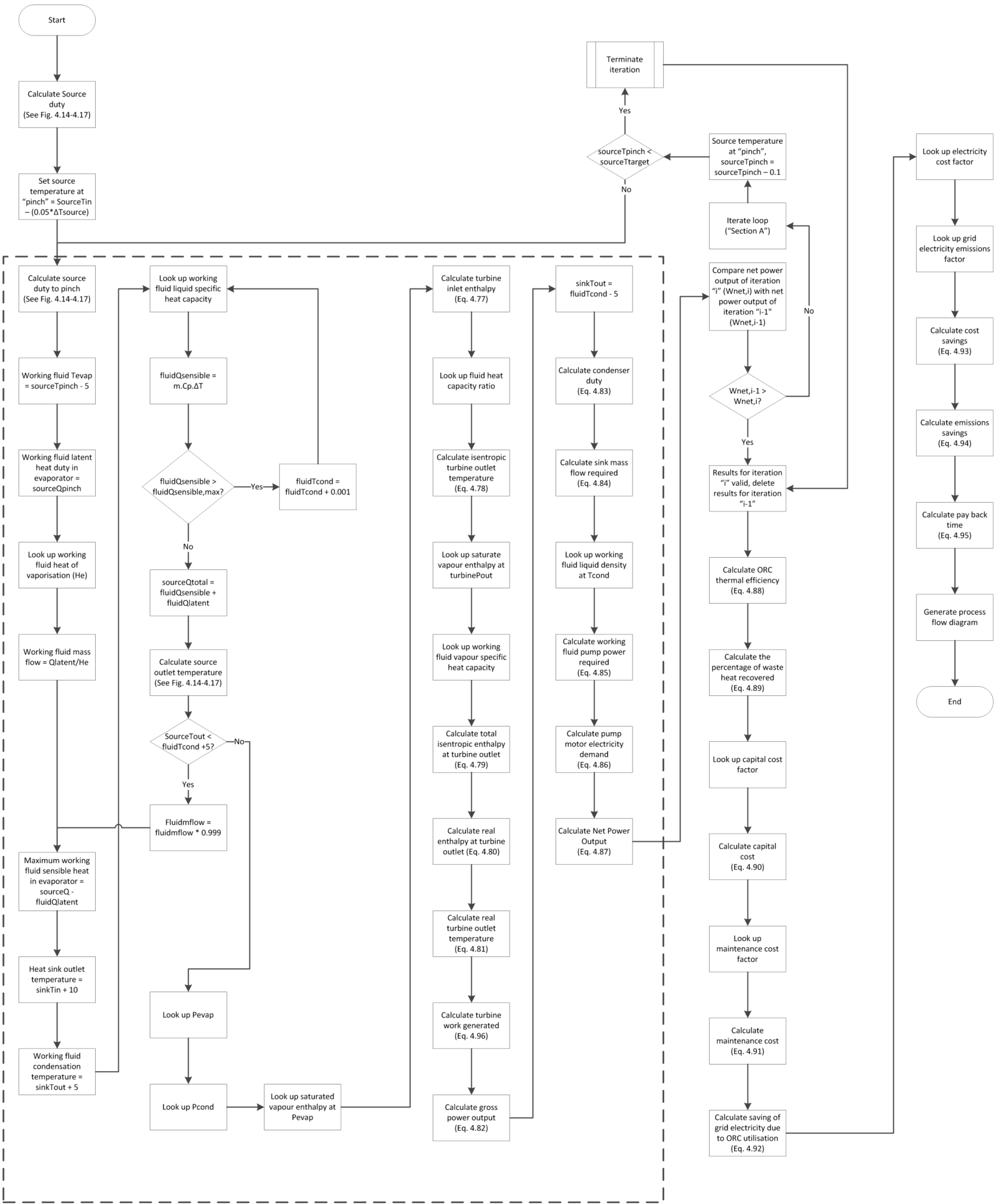


Figure 4.28. Design routine for ORC with heat sources containing sensible heat only or humid air heat sources

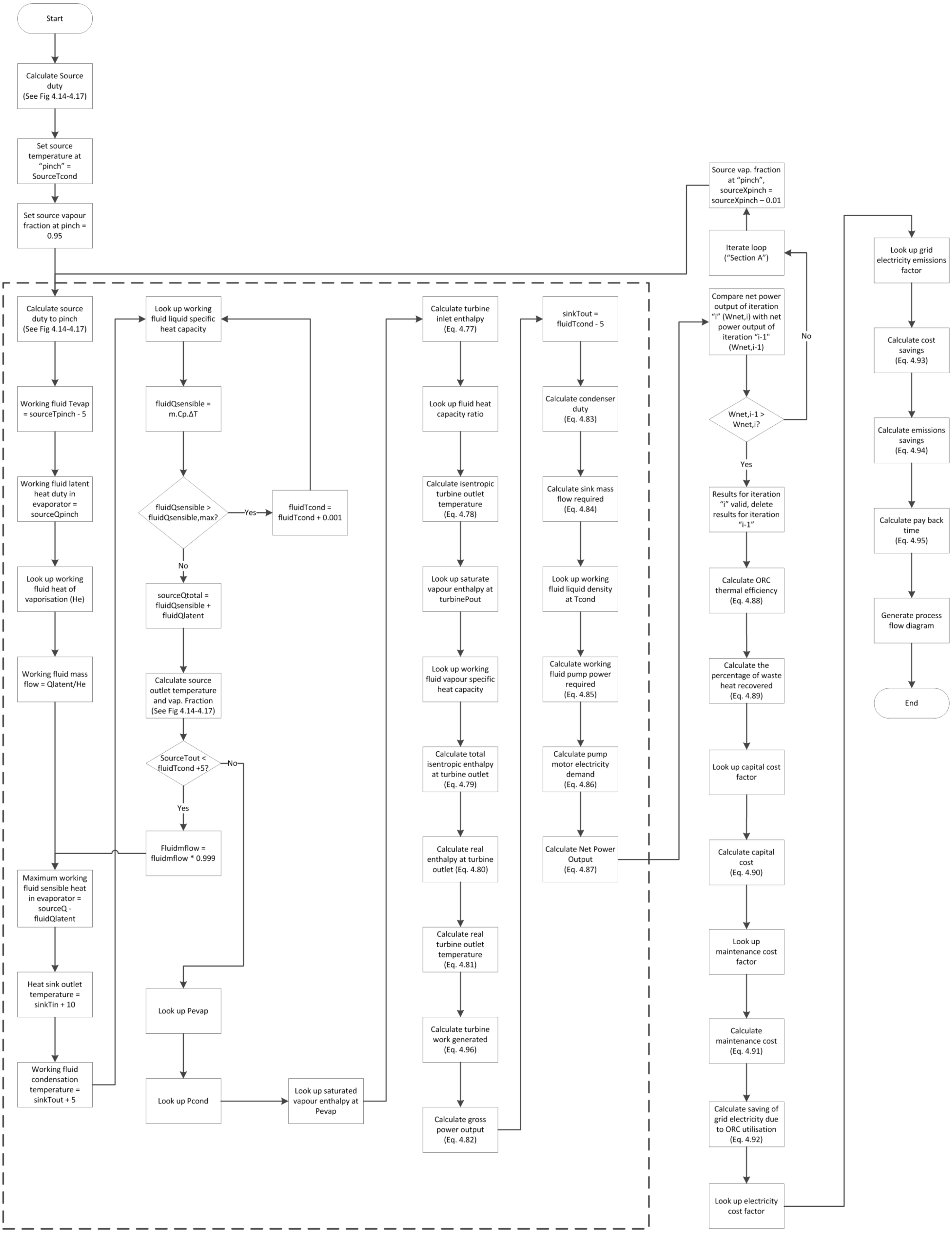


Figure 4.29. Design routine for ORC with condensing vapour heat sources

The following data/assumptions are used during the ORC design procedures shown in Figures 4.28-4.29:

- Turbine isentropic efficiency of 85%.
- Generator efficiency of 95%.
- Pump efficiency of 75%. Pump motor efficiency of 95%.
- 5^{o}C minimum approach (or "pinch") temperature in heat exchangers.
- Working fluid data is provided by ASHRAE, 2009. See Appendix III for an example of the thermophysical data tables used. This data is programmed into the KBS database and the software can "look-up" the data as required.
- Capital cost factors are taken from the US Department of Energy, 2008. Historical currency conversion data and the engineering price indices (The Engineer, 2013; Coulson and Richardson, 2005) are used to create cost factors valid in the UK at the present date.

The list of equations relevant to Figure 4.28-4.29 is as follows:

$$H_{turb,in} = H_{sat,vap,Pevap} + (\Delta T_{super}.Cp) \qquad \text{kJ/kg} \qquad \text{Eq. 4.77}$$

$$T_{turb,out,isen} = T_{comp,in} * \frac{P_{cond}}{P_{evap}}^{1-\frac{1}{\gamma}} \qquad \text{K} \qquad \text{Eq. 4.78}$$

$$H_{turb,out,isen} = (Cp_{vap} * (T_{turb,out,isen} - T_{turb,out,sat})) + H_{turb,out,sat,vap} \qquad \text{kJ/kg} \qquad \text{Eq. 4.79}$$

$$H_{turb,out,real} = H_{turb,in} - \eta_{isen}(H_{turb,in} - H_{turb,out,isen}) \qquad \text{kJ/kg} \qquad \text{Eq. 4.80}$$

$$T_{tub,out,real} = T_{cond} + \frac{H_{turb,out,real} - H_{turb,out,sat,vap}}{Cp_{vap}} \qquad {}^{o}\text{C} \qquad \text{Eq. 4.81}$$

$$W_{gen} = \eta_{gen} W_{turb} \qquad \text{kW} \qquad \text{Eq. 4.82}$$

$$Q_{cond} = \left(m_{fluid} Cp_{vap}\left(T_{turb.out,real} - T_{cond}\right)\right) + m_{fluid}\Delta H_{vap} \qquad \text{kW} \qquad \text{Eq. 4.83}$$

$$m_{sink} = \frac{Q_{cond}}{Cp_{sink}\Delta T_{sink}}$$ kg/s Eq. 4.84

$$W_{pump} = \frac{m_{fluid}}{\rho_{fluid}} \frac{P_{evap} - P_{cond}}{\eta_{pump}}$$ kW Eq. 4.85

$$W_{pump,motor} = \eta_{motor} W_{pump}$$ kW Eq. 4.86

$$W_{net} = W_{gen} - W_{pump,motor}$$ kW Eq. 4.87

$$\eta_{thermal} = \frac{W_{net}}{Q_{evap}}$$ Eq. 4.88

$$\%_{WHR} = \frac{Q_{evap}}{Q_{source,desired}}$$ kW Eq. 4.89

$$C_{capital} = C_{factor,ORC} * W_{gen}$$ £ Eq. 4.90

$$C_{maintenace} = C_{factor,maintenance} * C_{capital}$$ £/year Eq. 4.91

$$N_{units,electricity,saved} = W_{net} h_{plant}$$ kWh Eq. 4.92

$$C_{electricty,saved} = N_{units,electricity,saved} C_{utility}$$ £/year Eq. 4.93

$$E_{electricity,spent} = N_{units,electricity,saved} * E_{utility}$$ tCO_2eq/year Eq. 4.94

$$PB = \frac{C_{capital}}{C_{electricity,saved} - C_{maintenace}}$$ Year Eq. 4.95

$$W_{turb} = m_{workingfluid} * (H_{turb.in} - H_{turb,out,real})$$ kW Eq. 4.96

Note: (1) In Eq. 4.85, pressure is in kPa rather than bar. The KBS has a routine to change units where required.

4.5. Chapter Conclusions

This chapter has presented methods for the selection and design of equipment for the recovery of low-grade industrial waste heat and thereby forms the equation, logic and methods of the knowledge-based system presented in this thesis.

The programming of this data into the knowledge based system is described in the next chapter, Chapter 5: System Programming.

Chapter 5

This chapter covers the programming and compilation of the knowledge-based system using a suitable computer language. The choice of programming language is discussed, along with screenshots of the graphical user interface produced.

5. System Programming

The programming of the system is a key step in the overall production of the KBS and must lead to a programme that adheres to the rules set out in the scope of the system in Section 1.3.1.

As the system equations, logic and methods have already been devised (Chapter 4), the key step here is the selection of a suitable programming language. The selected language must allow the production of a clear and functional user interface, whilst allowing wide, preferably free, dissemination into the public domain. This is covered in Section 5.1, while Section 5.2 briefly discusses some example code (found in Appendix IV) and Section 5.3 shows screen shots of the system GUI.

5.1. Selection of Suitable Programming Language

A number of programming languages and techniques were considered for use in writing the software including Java, C++, Matlab and Visual Basic. Java was chosen according to design constraint number 4 (see Section 1.3.1): for ease of dissemination into the public domain. Java allows the developer to write and compile code which may be ran anywhere and everywhere - *write once, run anywhere/everywhere* (WORA/WORE) (Lewis and Loftus, 2005). Therefore, developed java applications will run on any operating system which supports the java runtime environment without further compilation. This includes Linux, Solaris, Windows and Mac OS (Lewis and Loftus, 2005). Furthermore, the java runtime environment is available as a free download, meaning that there would be no financial constraints on the dissemination of the software.

This gave java a distinct advantage over the other options considered. For example, C++ programming operates according to the *write once, compile anywhere* (WOCA) principle. Therefore, a different version would have to be created for each different type of operating system, or the end user would be required to compile the code prior to running the software. Visual Basic programs are only available for use on the Windows operating system, therefore multi-platform dissemination using this language would not be possible. Finally, any code written in Matlab would require the user to have a Matlab license in order to run the software. This would create a financial constraint which would be detrimental to the use of the system.

Other advantages of using Java to write this system are summarised in Table 5.1 below. [Note: information taken from a combination of sources (Lewis and Loftus, 2005; Oracle, 2013) and the authors own experience].

Table 5.1. .Summary of advantages of the java programming language for the KBS (Adapted from Lewis and Loftus, 2005; Oracle, 2013; authors knowledge)

Ease of use	Java is relatively simple to learn, write, compile and debug. This allows more time to be focussed on the technical (engineering) content of the program.
Object-Oriented Language	This allows the creation of modular and reusable code. This is useful for this system as a lot of design code will be shared between the different types of heat exchangers, for example.
Platform Independence	As discussed above.
Security	Java was developed with safety in mind. The result of this is that online downloads of Java programs may be

	trusted - crucial for ease of dissemination via the web.
Robustness	The Java compiler places emphasis on early detection of errors. This allows more time to be focussed on the technical (engineering) content of the program.
Graphical user interface	Java allows the use of commonly known HTML tags to build up the graphical user interface. This allows a simple, functional interface to be created with relative ease

5.1. Example Code

An example of the java code written for this system is shown in Appendix IV. Note: this code corresponds to the section of the system knowledge-base depicted in Figures 4.14-4.16 in Section 4.4.2.

5.2. System User Interface

The system user interface is designed to be both simple and functional. *i.e*, it must be simple to use and navigate, and it must display all of the required data fields (for example, all of the required data input to the system and also all of the required results data).

Figure 5.1 below shows the system home screen, Figure 5.2 shows the initial user questions screen designed to guide the user towards the correct type of waste heat recovery technology (as depicted by the part of the system knowledge-base shown in Figure 4.2-4.3, Section 4.3.1), Figure 5.3 shows an example of the data input screen (for an organic Rankine cycle system), Figure 5.4 shows an example of the general results screen (again, for an organic Rankine cycle system) and Figure 5.5 shows an example of detailed process flow diagram results (again for an organic

Rankine cycle system). Note: Figures 5.1-5.4 are representative of the steps taken in Case Study 1 shown in Section 6.1.

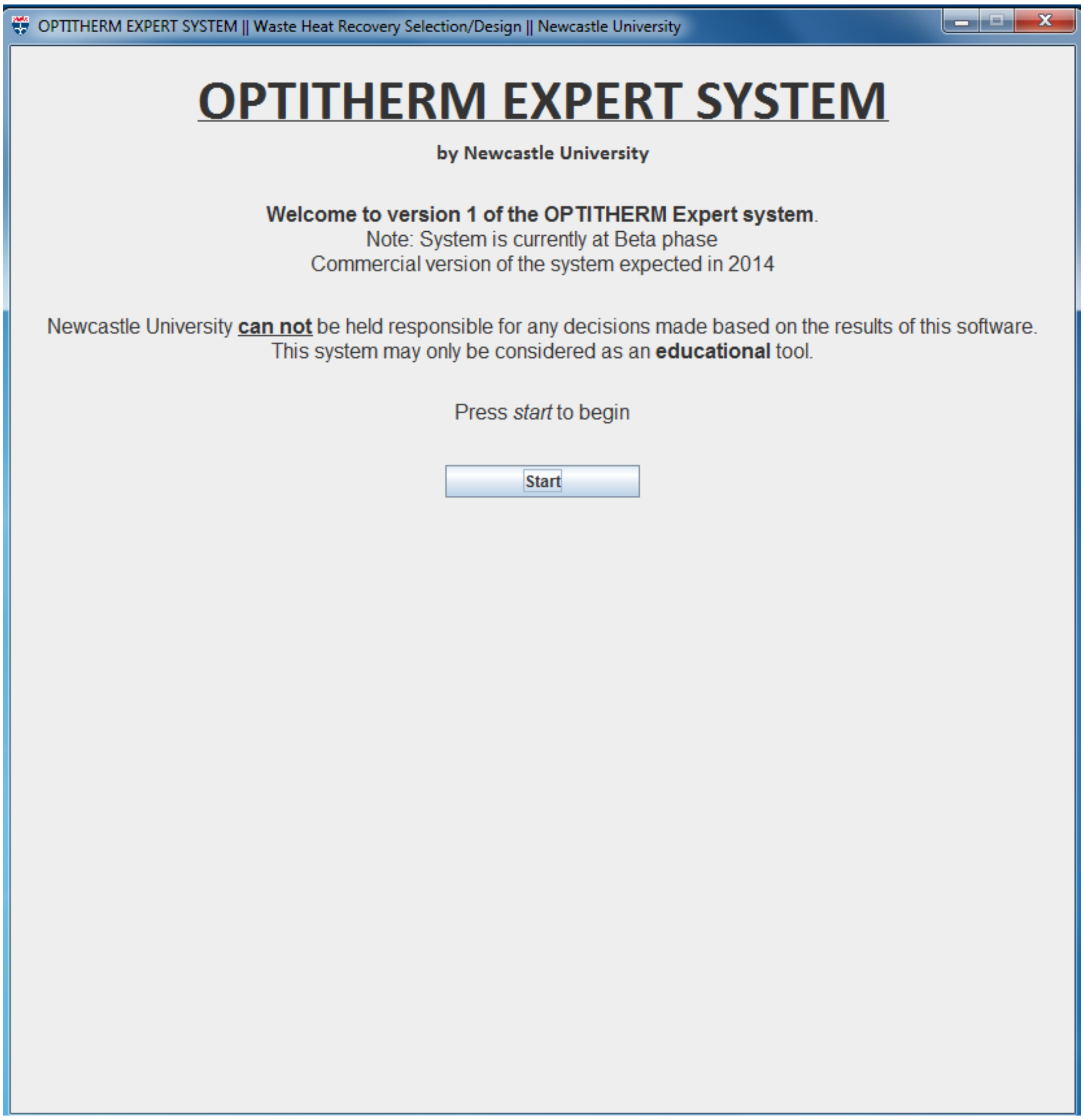


Figure 5.1. KBS home screen

Figure 5.1 shows the KBS home screen. The interface is designed to be basic and functional, displaying key data and points in an ordered manner. This screen briefly introduces the software and features a “start” button which launches the main application.

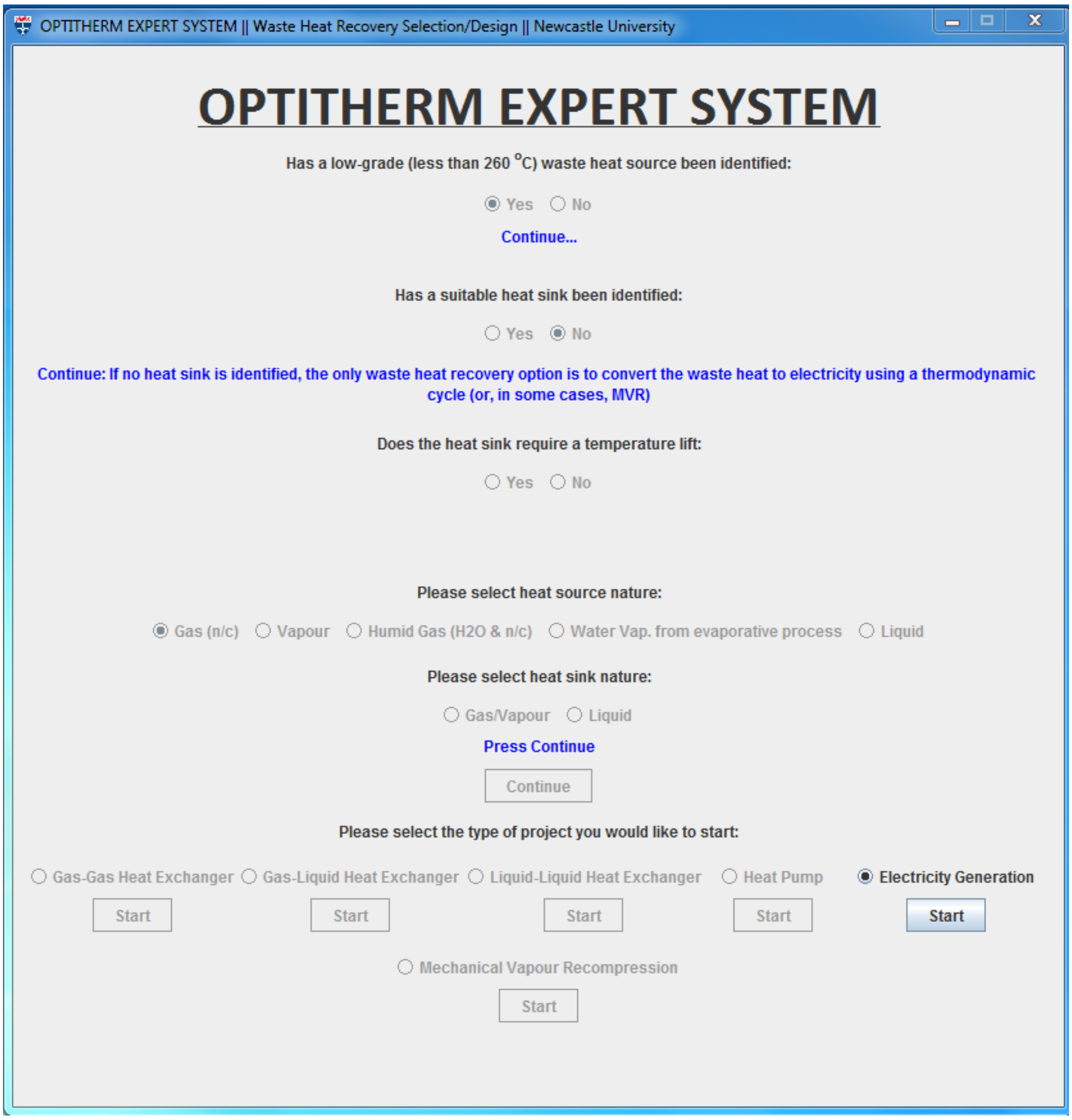


Figure 5.2. KBS initial user questions

Figure 5.2 shows the initial questions the user is asked in order to select which categories of waste heat recovery technology it is possible to use in the case study (as described in Figures 4.2-4.3 in Section 4.3.1). In this case, the user has selected that no heat sinks have been identified, in which case only “Electricity Generation” is possible. Hence, the “Start” button underneath “Electricity Generation” is activated. The GUI is, again, simple and functional with clear questions given in the black font and explanation of the decisions displayed in the blue font.

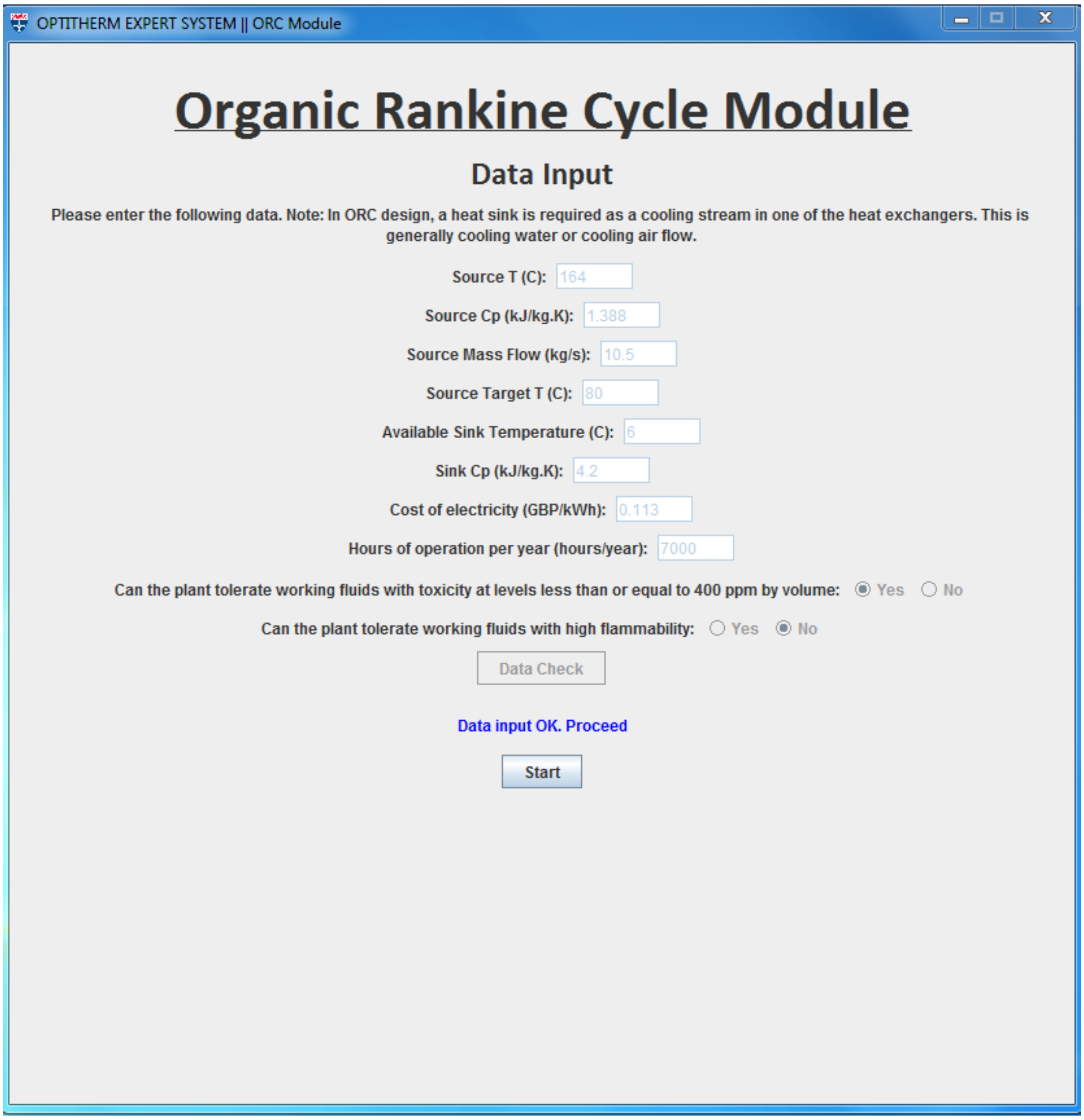
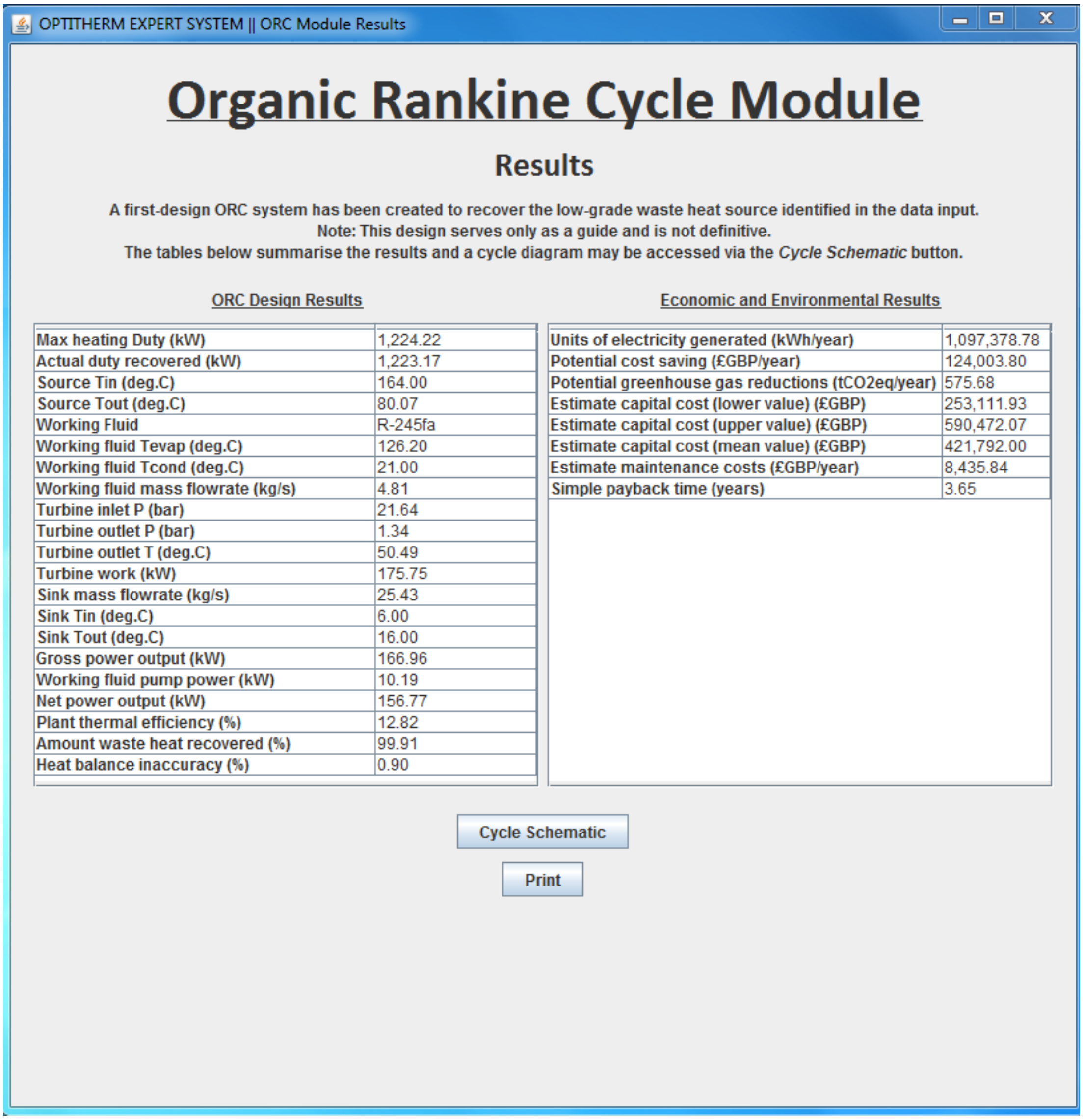


Figure 5.3. KBS data input (ORC)

Figure 5.3 shows the source data input required for the system to design an ORC. Note that this differs for each type of technology, and further screenshots are shown in Appendix V. Here, the user has inputted the required data, ran the data check process and hit the start button. The data did not contain any errors, hence "Data Input OK, Proceed" is displayed on the screen and the user was able to press the "Start" button. The "Results" button is now activated as the KBS has generated results for the case study.



OPTITHERM EXPERT SYSTEM || ORC Module Results

Organic Rankine Cycle Module

Results

A first-design ORC system has been created to recover the low-grade waste heat source identified in the data input.
Note: This design serves only as a guide and is not definitive.
The tables below summarise the results and a cycle diagram may be accessed via the *Cycle Schematic* button.

ORC Design Results

Max heating Duty (kW)	1,224.22
Actual duty recovered (kW)	1,223.17
Source Tin (deg.C)	164.00
Source Tout (deg.C)	80.07
Working Fluid	R-245fa
Working fluid Tevap (deg.C)	126.20
Working fluid Tcond (deg.C)	21.00
Working fluid mass flowrate (kg/s)	4.81
Turbine inlet P (bar)	21.64
Turbine outlet P (bar)	1.34
Turbine outlet T (deg.C)	50.49
Turbine work (kW)	175.75
Sink mass flowrate (kg/s)	25.43
Sink Tin (deg.C)	6.00
Sink Tout (deg.C)	16.00
Gross power output (kW)	166.96
Working fluid pump power (kW)	10.19
Net power output (kW)	156.77
Plant thermal efficiency (%)	12.82
Amount waste heat recovered (%)	99.91
Heat balance inaccuracy (%)	0.90

Economic and Environmental Results

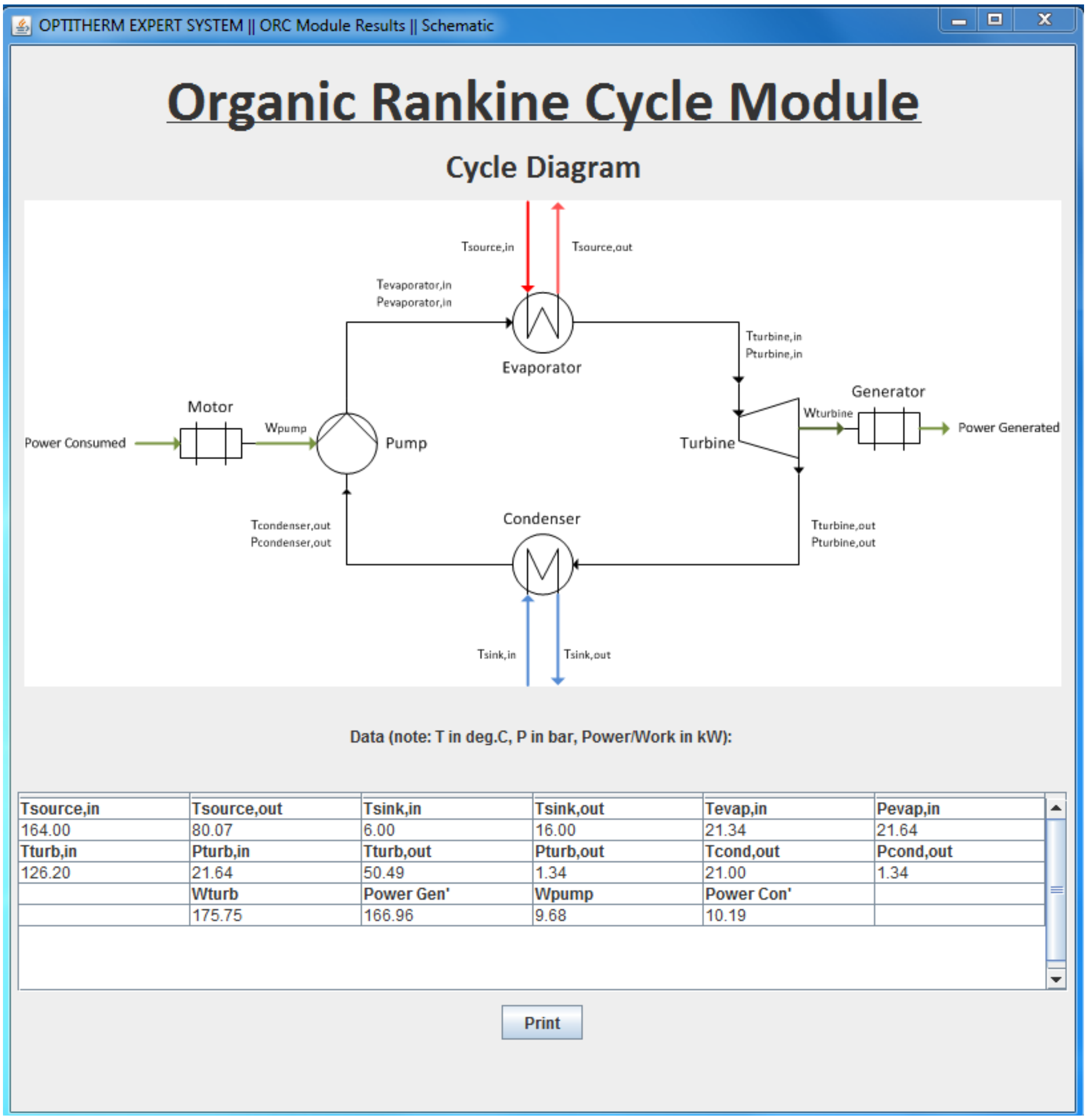
Units of electricity generated (kWh/year)	1,097,378.78
Potential cost saving (£GBP/year)	124,003.80
Potential greenhouse gas reductions (tCO2eq/year)	575.68
Estimate capital cost (lower value) (£GBP)	253,111.93
Estimate capital cost (upper value) (£GBP)	590,472.07
Estimate capital cost (mean value) (£GBP)	421,792.00
Estimate maintenance costs (£GBP/year)	8,435.84
Simple payback time (years)	3.65

Cycle Schematic

Print

Figure 5.4. KBS results (ORC)

Figure 5.4 shows the results generated during ORC design.. Note that the results differ for each type of technology, and further screenshots are shown in Appendix V. The user has the option to “Print” the results screen, and also view a cycle schematic (as shown in Figure 5.5).



Tsource,in	Tsource,out	Tsink,in	Tsink,out	Tevap,in	Pevap,in
164.00	80.07	6.00	16.00	21.34	21.64
Tturb,in	Pturb,in	Tturb,out	Pturb,out	Tcond,out	Pcond,out
126.20	21.64	50.49	1.34	21.00	1.34
	Wturb	Power Gen'	Wpump	Power Con'	
	175.75	166.96	9.68	10.19	

Figure 5.5. KBS process flow diagram of results (ORC)

Figure 5.5 shows a process flow diagram of the cycle designed by the KBS. The screen is, again, simple but functional and displays all of the required data. Note that the process flow diagram screen differs for each type of technology, and further screenshots are shown in Appendix V. The user again has an option to "Print" these results.

Figures 5.1-5.5 show that the GUI is simple but functional, therefore making the system easy to use. The GUI also features all of the necessary data entry fields and results as set out in Section 4.4.1.

Further screenshots from various case studies are shown in Appendix V. Further details about the corresponding case studies can be found in Chapter 6.

5.3. Chapter Conclusions

A number of programming languages were considered for use including Java, C++, and Visual Basic. Java was selected for use because of a number of key factors including ease of dissemination (due to the "write once, run anywhere" nature of the language), ease of use and the robustness of the language.

A simple but functional GUI has been created which clearly defines the user data input requirements, provides explanation of the system decisions and displays all of the required system results.

Chapter 6

This chapter covers testing of the knowledge-based system via case studies. The case studies were based on both published literature and original data from industrial partners.

6. Testing of Knowledge-Based System

Testing the knowledge-based system (KBS) was necessary in order to assess the success of the system in terms of making a positive contribution to the methods and tools available to aid industrial waste heat recovery, and comparing the final system to the criteria set out in the scope of the system (Section 1.3.1).

Testing was completed via 5 case studies (summarised in Table 6.1) which were chosen to cover a broad range of the process industry subsectors and scenarios. The case study results also covered the 4 main recovery technologies in the system knowledge-base: heat exchangers, heat pumps (closed-cycle vapour compression), mechanical vapour recompression and organic Rankine cycles. Hence, results for each of the included technology groups are analysed. The case studies are a combination of existing published case-studies and *new* case-studies provided by collaborative partners.

Table 6.1. Summary of case studies

Case Study	**Reference**	**Process Industry Sector**	**Primary technology category**
1. Potato Crisp Production	Aneke *et al* (2012) Aneke (2012)	Food and beverage	Electricity generation: ORC
2. Textile Production	Pulat *et al* (2009)	Textiles	Heat transfer to *matching* heat

			sink: heat exchanger; Electricity generation: ORC
3. Industrial Washing Process	Paarske (2011)	Metal Products	Heat upgrade: vapour compression heat pump
4. Brewery	N/A: Original Case Study	Food and beverage	Heat upgrade: mechanical vapour recompression
5.Inorganic Chemical Production	N/A: Original Case Study	Chemicals	Heat transfer to *matching* heat sink: heat exchanger; Electricity generation: ORC

6.1. Case Study 1: Potato Crisp Production

This case study was originally published by Aneke *et al* (2012) and is further investigated in the thesis by Aneke (2012) which provided additional information, particularly economic and carbon emission data. This was a theoretical case study, with modelling results generated using the IPSEpro (SimTech, 2013) modelling package. However, it was based on real process data from a food processing plant in the UK.

6.1.1. Case Study Summary

Waste heat recovery was investigated in a UK food processing plant producing potato crisps. The main area of interest was the frying process, the most energy intensive process in the plant. A process flow diagram of the fryer is shown in Figure 6.1.

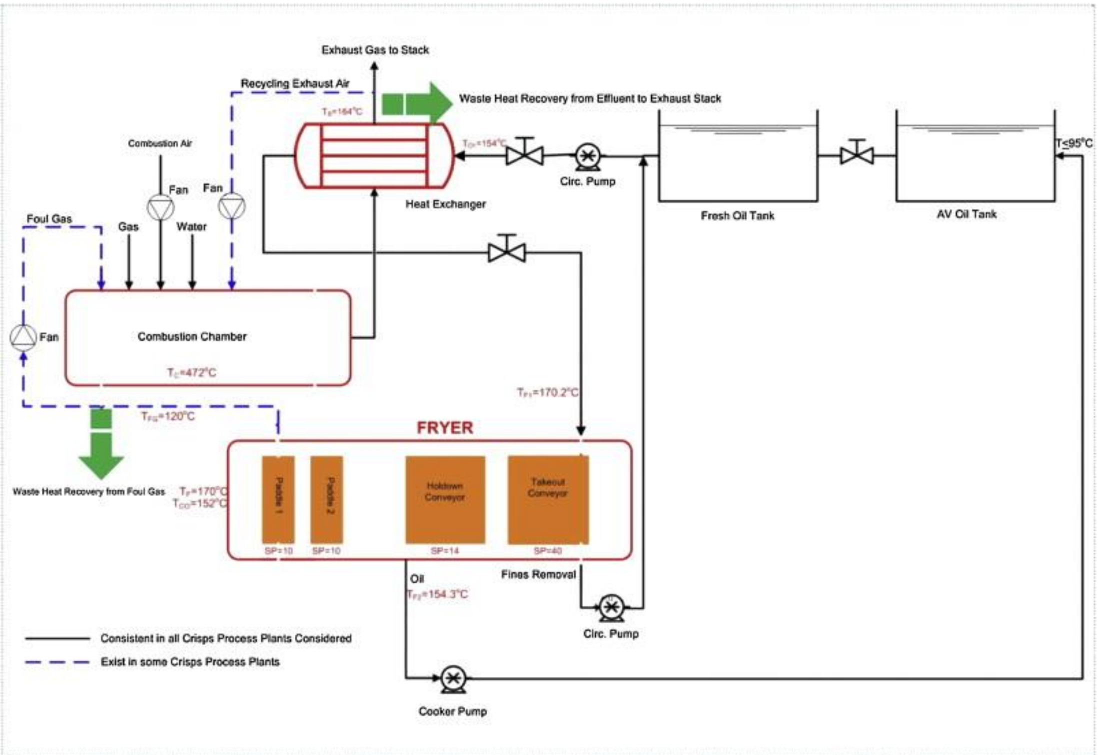


Figure 6.1. Process flow diagram of “fryer” from case study 1(Aneke *et al*, 2012)

Two main heat sources were identified and are denoted by large green arrows in Figure 6.1: the “Exhaust Gas to Stack” [leaving the “heat exchanger”] and the “Foul Gas” [leaving the “FRYER”]. No suitable heat sinks were identified to allow waste heat recovery via a heat exchanger or heat pump, hence it was decided that waste-heat driven electricity generation should be investigated.

However, it was not stated whether or not other methods of recovery were investigated. For example, most food processing plants require large quantities of hot wash water and it could be hypothesized that an opportunity for waste heat recovery via a heat exchanger may have been possible. In this case, only the data as presented can be considered, and so it must be concluded that the authors considered all options for waste heat recovery to find that electricity generation was the only valid option for waste heat recovery.

6.1.2. Source/Sink Data Provided

Tables 6.2 and 6.3 below show data for the two heat sources, and the heat sinks available for use in the condenser end of the ORC.

Table 6.2. Case study 1: source/sink data for foul gas (Aneke *et al*, 2012)

Heat Source Name	Foul Gas
Heat Source Nature	Non-condensable gas[1]
Heat Source Temperature (^{o} C)	120
Heat Source Target Temperature (^{o}C)	87.0[1]
Heat Source Specific Heat Capacity (kJ/kgK)	1.548
Heat Source Mass Flow Rate (kg/s)	3.17
Heat Sink Nature	Liquid (water)
Heat Sink Temperature (^{o} C)	6.00
Heat Sink Specific Heat Capacity (kJ/kgK)	4.20

[1]The paper stated it is “desirable” to operate above the dew point of the foul gas. This is assumed to be upon request of the host plant and is therefore taken as the source target temperature, and the source is treated as a non-condensable gas.

Table 6.3. Case study 1: source/sink data for exhaust gas (Aneke *et al*, 2012)

Heat Source Name	Exhaust Gas
Heat Source Nature	Non-condensable gas[1]
Heat Source Temperature (^{o} C)	164
Heat Source Target Temperature (^{o}C)	80.0[1]
Heat Source Specific Heat Capacity (kJ/kgK)	1.388
Heat Source Mass Flow Rate (kg/s)	10.5
Heat Sink Nature	Liquid (water)
Heat Sink Temperature (^{o} C)	6.00
Heat Sink Specific Heat Capacity (kJ/kgK)	4.20

[1]The paper stated it is "desirable" to operate above the dew point of the exhaust gas. This is assumed to be upon request of the host plant and is therefore taken as the source target temperature, and the source is treated as a non-condensable gas.

Table 6.4 shows general plant data from Aneke (2012), including plant hours of operation, utility costs and plant tolerance to harmful working fluid properties

Table 6.4. Case study 1: general plant data (Aneke, 2012)

Plant hours of operation (hours/year)	7000
Cost of electricity (£GBP/kWh)	0.113[1]
Plant tolerant to toxic working fluids (according to ASHRAE safety classification "B")	Yes[2]
Plant tolerant to flammable working fluids (according to ASHRAE safety classification "3")	No[2]

[1]The current cost of electricity at the plant is not discussed here. In the economic evaluation, the authors compared project value over a range of hypothesized electricity prices ranging from 0.01£/kWh to 0.35 £/kWh. Hence, for this case study the cost of electricity is obtained from DECC (Department for Energy and Climate Change) and is taken as £0.113 £/kWh (DECC (c), 2013). [2]Working fluid selection with regards to plant constraints is not discussed (the author simply compares R-245a to R134a, the "two industry standards" and conclude R-245fa has a more useful operating range). However, it is assumed the plant may tolerate B-class working fluids as R-245fa is used in the paper. It is also assumed that the plant would not tolerate "3-class", highly flammable fluids, as food processing sites may not have the health and safety measures in place to deal with such hazardous materials.

6.1.3. Results and Discussion

In the published results (in both the journal article and the thesis), the aim was to optimize the net power output of the system by integration of both sources into the same cycle. Five cycles were produced as follows:

1. Single source cycle using "Foul Gas" source
2. Single source cycle using "Exhaust Gas" source

3. Dual source using two sources in parallel and one turbine. “Foul Gas” source provides pre-heating and “Exhaust Gas” source drives evaporation
4. Dual source using two sources in parallel and one turbine. “Exhaust Gas” source provides pre-heating and “Foul Gas” source drives evaporation
5. Dual source using two heat exchanger-turbine combinations in a re-heat cycle configuration. “Exhaust Gas” source drives the initial pre-heat/evaporation for the “high pressure turbine” and the “foul gas” re-heats the exit vapour which then drives the low-pressure turbine.

Figure 6.2 below shows the “dual heat exchanger” cycles described by cycles 3 and 4 above, and Figure 6.3 shows the “re-heat” cycle described by cycle 5 above. Cycles 1 and 2 are standard single-source cycles described numerous times previously (for example Figure 3.20, Section 3.3.2 and Figure 4.27, Section 4.4.4).

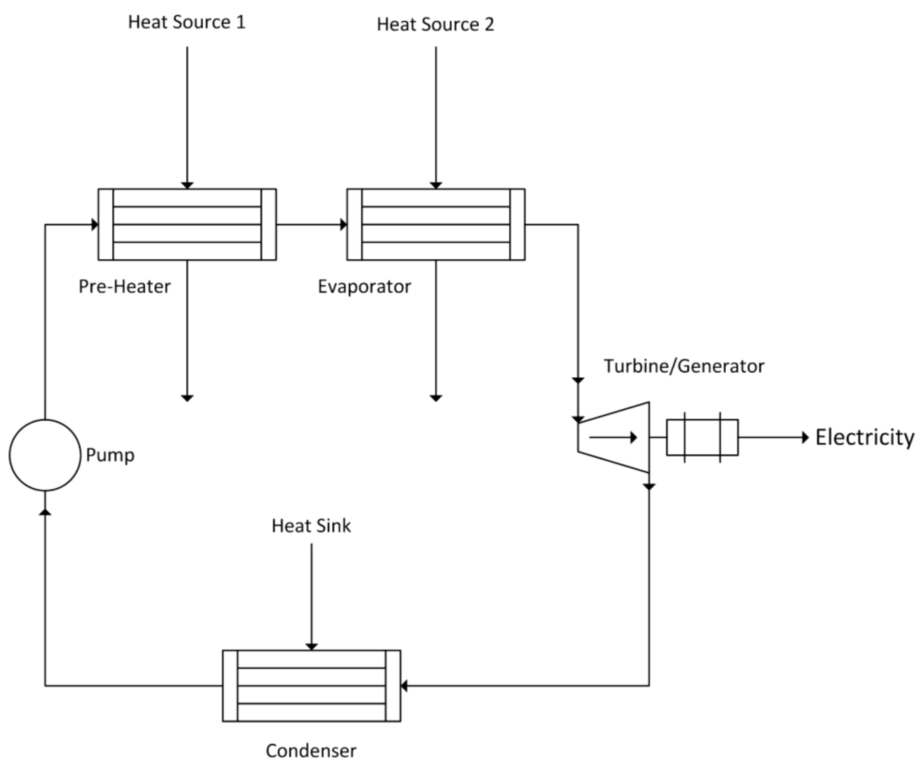


Figure 6.2. Dual source ORC using two heat exchangers in parallel (adapted from Aneke et al, 2012)

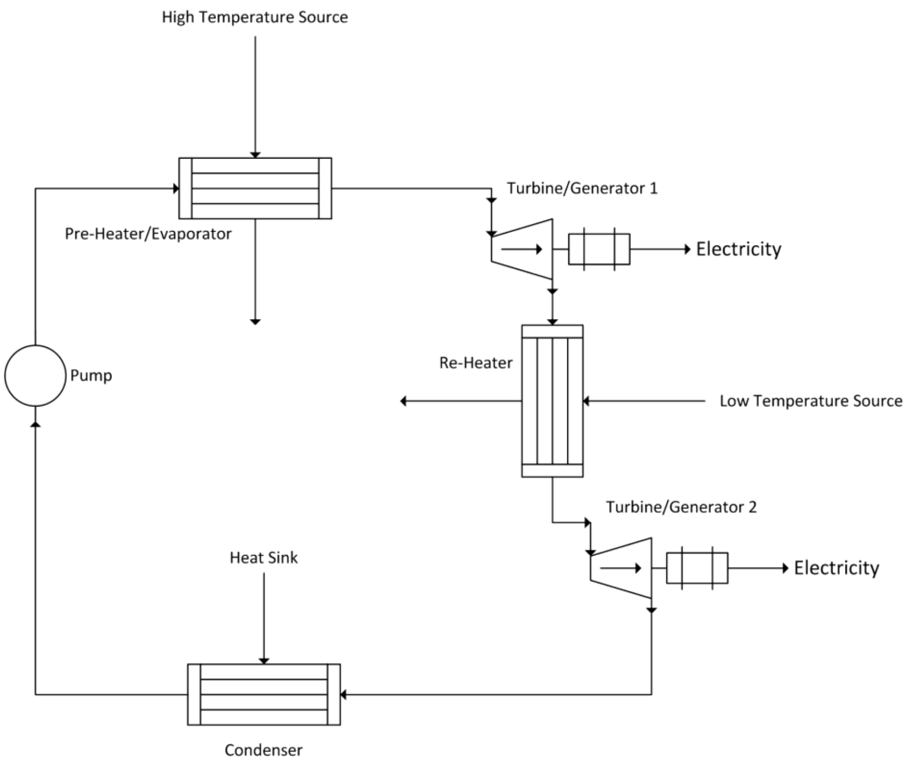


Figure 6.3. Dual source ORC using a dual turbine re-heat cycle (adapted from Aneke *et al*, 2012)

The cycles were modelled using IPSEpro (SimTech 2013). In all cases, the models were optimised for maximum power generation according to the following constraints:

- Minimum pinch point in evaporator of $5^{o}C$
- $4^{o}C$ temperature rise in cooling water
- Saturated vapour at turbine inlet
- Heat source outlet temperature greater than dew point

As the KBS only considers standard ORC cycles, only the results of cycles 1 and 2 are used for comparison. The dual source cycles are considered to be novel, optimised cycles and as such are out of the scope of the system. The relevant results by Aneke *et al* are displayed in Tables 6.5 and 6.6 below.

Table 6.5. Technical results for foul gas heat source (Aneke *et al*, 2012)

Working Fluid	R-245fa
Heat Source Inlet Temperature (^{o} C)	120
Heat Source Outlet Temperature (^{o} C)	87.0
Pre-heater + Evaporator Duty (kW)	162
Heat Sink Inlet Temperature (^{o} C)	6.00
Heat Sink Outlet Temperature (^{o} C)	10.0
Condenser Duty (kW)	138
Heat Sink Mass Flow Rate (kg/s)	8.23
Working Fluid Mass Flow Rate (kg/s)	0.618
Working Fluid Turbine Inlet Temperature (^{o} C)	99.6
Working Fluid Turbine Inlet Pressure (bar)	12.5
Working Fluid Turbine Outlet Temperature (^{o} C)	40.7
Working Fluid Turbine Outlet Pressure (bar)	1.02
Working Fluid Condensation Temperature (^{o} C)	12.6
Plant Gross Power Output (kW)	23.1
Working Fluid Pump Power (kW)	1.03
Net Power Output (kW)	22.1
Plant Thermal Efficiency (%)	14.0

Table 6.6. Technical results for exhaust gas heat source (Aneke *et al*, 2012)

Working Fluid	R-245fa
Heat Source Inlet Temperature (^{o} C)	164
Heat Source Outlet Temperature (^{o} C)	80.1
Pre-heater + Evaporator Duty (kW)	1223
Heat Sink Inlet Temperature (^{o} C)	6.00
Heat Sink Outlet Temperature (^{o} C)	10.0
Condenser Duty (kW)	1011
Heat Sink Mass Flow rate (kg/s)	60.2
Working Fluid Mass Flow Rate (kg/s)	4.40

Working Fluid Turbine Inlet Temperature (^{o} C)	133.6
Working Fluid Turbine Inlet Pressure (bar)	25.0
Working Fluid Turbine Outlet Temperature (^{o} C)	47.5
Working Fluid Turbine Outlet Pressure (bar)	1.024
Working Fluid Condensation Temperature (^{o} C)	12.6
Plant Gross Power Output (kW)	208.7
Working Fluid Pump Power (kW)	15.2
Net Power Output (kW)	193.5
Plant Thermal Efficiency (%)	16.0

The case study source, sink and general plant data displayed in Tables 6.2-6.4 was used to run the knowledge based system, and the results are as follows. Note that screen shots from this case study are shown in Figures 5.1-5.5 in Section 5.2.

Table 6.7 shows the results from the first stage of technology selection, whereby the general categories of waste heat recovery technology are selected for the case study. These results show that the KBS found that waste-heat driven electricity generation (via an organic Rankine cycle) is the only option in this case study. This is in agreement with that of the Aneke *et al* (2012) and Aneke (2012).

Table 6.7. Case study 1: KBS initial selection results

		Reason
Heat Exchanger Heat Recovery Possible	No	No *matching* heat sink
Closed-Cycle Vapour Compression Heat Pump Possible	No	No heat sink within reasonable temperature lift
Mechanical Vapour Recompression Possible	No	Not an evaporative process
Organic Rankine Cycle Possible	Yes	N/A

Tables 6.8 and 6.9 below show the technical results generated by the KBS and comments comparing the data to that in the published case study.

Table 6.8. Case study 1: KBS technical results for foul gas heat source

		Comment on comparison with published data
Working Fluid	R-245fa	Equal
Source Inlet Temperature (^{o} C)	120	N/A
Heat Source Outlet Temperature (^{o} C)	87.0	Approximately equal (<1% larger [compared to ambient])
Pre-heater + Evaporator Duty (kW)	162	Approximately equal (<1% smaller)
Heat Sink Inlet Temperature (^{o} C)	6.00	N/A
Heat Sink Outlet Temperature (^{o} C)	16.0	Higher (by 6 ^{o}C)
Condenser Duty (kW)	141	Larger (2.2%)
Heat Sink Mass Flow Rate (kg/s)	3.36	Smaller (59.2%)
Working Fluid Mass Flow Rate (kg/s)	0.66	Larger (6.8%)
Working Fluid Turbine Inlet Temperature (^{o} C)	96.2	Lower (by 3.4 ^{o}C)
Working Fluid Turbine Inlet Pressure (bar)	11.5	Lower (by 1 bar)
Working Fluid Turbine Outlet Temperature (^{o} C)	42.1	Higher (by 1.4 ^{o}C)
Working Fluid Turbine Outlet Pressure (bar)	1.34	Higher (by 0.32 bar)
Working Fluid Condensation Temperature (^{o} C)	21.0	Higher (by 8.4 ^{o}C)
Plant Gross Power Output (kW)	20.9	Lower (9.7%)
Working Fluid Pump Power (kW)	0.70	Lower (32%)
Net Power Output (kW)	20.2	Lower (8.8%)

Plant Thermal Efficiency (%)	12.5	Lower (11%)

Table 6.9. Case study 1: KBS technical results for exhaust gas heat source

		Comment on comparison with published data
Working Fluid	R-245fa	Equal
Source Inlet Temperature (^{o} C)	164.0	N/A
Heat Source Outlet Temperature (^{o} C)	80.0	Approximately equal (<1% smaller [compared to ambient]
Pre-heater + Evaporator Duty (kW)	1223	Approximately equal (<1% smaller)
Heat Sink Inlet Temperature (^{o} C)	6.00	N/A
Heat Sink Outlet Temperature (^{o} C)	16.0	Higher (by 6 ^{o}C)
Condenser Duty (kW)	1068	Larger (5.7%)
Heat Sink Mass Flow Rate (kg/s)	25.4	Smaller (57.8%)
Working Fluid Mass Flow Rate (kg/s)	4.81	Larger (9.3%)
Working Fluid Turbine Inlet Temperature (^{o} C)	126	Lower (by 7.4 ^{o}C)
Working Fluid Turbine Inlet Pressure (bar)	21.6	Higher (by 3.4 bar)
Working Fluid Turbine Outlet Temperature (^{o} C)	50.5	Higher (by 3 ^{o}C)
Working Fluid Turbine Outlet Pressure (bar)	1.34	Higher (by 0.32 bar)
Working Fluid Condensation Temperature (^{o} C)	21.0	Higher (by 8.4 ^{o}C)
Plant Gross Power Output (kW)	167	Lower (20.0%)
Working Fluid Pump Power (kW)	10.2	Lower (19.1%)
Net Power Output (kW)	156	Lower (19.0%)
Plant Thermal Efficiency (%)	12.8	Lower (19.9%)

The results show a very good match between the KBS cycle results and those generated using IPSEpro by Aneke *et al,* for both cases. The relatively simple optimisation strategy used in the KBS, which alters the evaporation temperature of the working fluid by reducing it incrementally, finds an optimum extremely close to that of the published results (KBS turbine inlet temperature within 3.5^{o}C for foul gas, and within 7.4^{o}C for the exhaust gas).

The main difference between the results lies in the turbine outlet temperature, condensation temperature, working fluid mass flow rate and heat sink results. This ultimately culminates in a relatively large difference between the net power output results for the KBS and published results (8.8% for the foul gas and 19.9% for the exhaust gas source).

This difference, however, is explained by the model assumptions rather than errors in the KBS methodology. Aneke *et al* assume only a 4^{o}C rise in the cooling water, while the KBS assumes 10^{o}C. Furthermore, the model in the KBS assumes a pinch point of 5^{o}C in the condenser, while the results by Aneke *et al* show a pinch point of around 3^{o}C. Therefore, the model by Aneke *et al* allows a significantly lower condensation temperature and pressure than the model in the KBS (12.6^{o}C/1.02bar as opposed to 21^{o}C/1.34bar). This explains the difference in heat sink flow rate (according to Equation 6.1 & 6.2 below), working fluid flow rate (according to Equation 6.3 below), gross power output (according to Equation 6.4 below), net power output (according to Equation 6.5 below) and thermal efficiency (according to Equation 6.6 below).

$$Q_{cond} = m_{ref}(\Delta H_{vap,,ref} + \left(Cp_{ref} * \left(T_{turb,out} - T_{cond}\right)\right) = m_{sink}Cp_{sink}\Delta T_{sink} \qquad \text{kW} \qquad \text{Eq. 6.1}$$

$$m_{sink} = \frac{m_{ref}(\Delta H_{vap,ref} + \left(Cp_{ref} * \left(T_{turb,out} - T_{cond}\right)\right)}{Cp_{sink}\Delta T_{sink}} \qquad \text{kg/s} \qquad \text{Eq. 6.2}$$

The heat sink mass flow rate (m_{sink}) in the results by Aneke *et al* is larger than in those by the KBS as it is inversely proportional to the temperature change in the heat sink (ΔT_{sink}). Furthermore, the temperature change between the turbine outlet ($T_{turb,out}$) and condensation point (T_{cond}) is larger, as is the latent heat of vaporisation ($\Delta H_{vap,ref}$) at lower temperatures (although this difference is minimal).

$$m_{ref} = \frac{Q_{evap}}{\Delta H_{vap,ref} + Cp_{ref}(T_{evap} - T_{cond})} \qquad \text{kg/s} \qquad \text{Eq. 6.3}$$

Note: Q_{evap} is equal to the source duty

The refrigerant mass flow rate (m_{ref}) is equal to the pre-heater/evaporator duty (Q_{evap}) (determined from the heat available in the heat source) divided by the specific sensible ($Cp_{ref} * (T_{evap} - T_{cond})$) and latent heat ($\Delta H_{vap,ref}$) required to fully evaporate the working fluid. In the results by Aneke *et al* the temperature change is significantly larger than in the results generated by the KBS, while the difference in latent heat of vaporisation is minimal. Hence, the working fluid mass flow rate is larger in the results by Aneke *et al*.

$$W_{gross} = \eta_{gen}(m_{ref}(H_{turb,in} - H_{turb,out})) \qquad \text{kW} \qquad \text{Eq. 6.4}$$

$$W_{net} = W_{gross} - W_{pump} \qquad \text{kW} \qquad \text{Eq. 6.5}$$

The gross power output (W_{gross}) in the results by Aneke *et al* is larger than in the results generated by the KBS as, firstly, the turbine inlet temperature/pressure is larger, which leads to a greater specific enthalpy at the turbine inlet ($H_{turb,in}$), and secondly, the turbine outlet pressure is smaller which leads to a smaller specific enthalpy at the turbine outlet ($H_{turb,out}$). The generator efficiency (η_{gen}) is also assumed to be higher by Aneke *et al* (96% as opposed to 95%). This also leads to a larger net power output (W_{net}) as the difference in pumping power (W_{pump}) is insignificant.

$$\eta_{thermal} = \frac{W_{net}}{Q_{evap}} \quad - \quad \text{Eq. 6.6}$$

The thermal efficiency ($\eta_{thermal}$) in the results by Aneke *et al* is also larger due to the larger net power output (W_{net}) and approximately equal pre-heater/evaporator duty (Q_{evap}), due to the source duties being approximately equal.

These results suggest that it may be beneficial to change the model assumptions in the KBS in order to produce solutions approaching the optimal, as presented by Aneke *et al*. However, in this case the conservative assumptions are beneficial to the overall aims of the software. The KBS is designed to provide only a "first" design, hence the use of conservative assumptions is advised. The results may then be later optimised. The case for this is strengthened when one considers the results for the condenser end of the cycle presented by Aneke *et al*. Here, for the exhaust gas heat source, 60.2 kg/s of cooling water is required (approximately 217 tonnes /hour). Such a large water requirement may seem excessive to some users, particular those who are new to organic Rankine cycle waste heat recovery. Furthermore, the pinch point of 3.2^{o}C used by Aneke *et al* would necessitate very large heat exchangers. Whilst it is not argued that this was not acceptable at the plant in question, it may not be suitable for every process plant.

Therefore, it is concluded that it is best to maintain conservative assumptions in the KBS while noting that further optimisation could be completed later, once the user has decided that an ORC machine would be beneficial at their site. This further optimisation may lead to smaller condenser pinch points and heat sink temperature rise according to the requirements/limitations of the host plant.

Economic and environmental results for this case study are presented by Aneke (2012) in the PhD thesis. However, this is only provided for the dual-heat source cycle (with foul gas providing pre-heating, exhaust gas providing evaporation heat) is considered. Hence, a direct comparison is not possible with the KBS results.

However, it should be noted that the dual source results are analogous to the exhaust gas results. Both cycles are gas-source driven ORC cycles, hence the only difference is that a larger quantity of waste heat is available, necessitating larger equipment. Therefore, the results are comparable as long as the difference in size is accounted for.

The economic results presented by Aneke are shown in Table 6.10 below and the results for the key cycle units are shown in Table 6.11 along with a comparison to the results generated by the KBS for exhaust gas case study.

Table 6.10. Case study 1: Comparison of economic and environmental results by the KBS and Aneke *et al* (2012)

	Aneke (2012) Dual Source	KBS Results Exhaust Gas Source
Estimate Capital Cost (£GBP)	525990	421792
Estimate Maintenance Cost (£GBP/year)	10520	8436
Estimate Cost Savings (£GBP/year)	157725	124004
Estimate Payback Period (years)	3.57	3.65
Estimate Carbon Dioxide Reductions (tCO_2eq/year)	549.9	575.7

Table 6.11. Case study 1: Comparison of key unit duty of KBS exhaust gas results and dual source results by Aneke (2012)

	Aneke (2012) Dual Source	KBS Results Exhaust Gas Source	Comment on comparison with published data
Pre-heater/Evaporator Duty (kW)	1385.4	1223.2	Smaller (11.7%)
Gross Power Output (kW)	214.6	166.97	Smaller (22.2%)
Turbine Inlet Temperature (°C)	127.0	126.2	Approximately equal

Pump Power (kW)	15.2	10.2	Smaller (32.9%)
Condenser Duty (kW)	1175.5	1068.1	Smaller (9.1%)

The capital cost of the dual source ORC system by Aneke is expectedly larger than the capital cost estimate of the single source system calculated by the KBS. If the concept of dual source is ignored in this analysis, and it is seen simply as a gas-source ORC then the difference in capital cost can be explained by the difference in size of each unit (in terms of duty).

In the KBS single-source results (exhaust gas source) each of the required duties is smaller. The pre-heater/evaporator is 11.7% smaller (in terms of duty, which would approximately correspond to area as the heat transfer coefficients and pinch points are equal between the two cases). The gross power output (or turbine work) is 22.2% smaller for approximately the same inlet temperature/pressure (hence the specific volume of the refrigerant would be equal and the physical size difference would be correlated to the mass flow through the turbine and the subsequent work generated). The pump power is 32.9 % smaller (for approximately the same pressure change, hence this can be correlated to mass flow rate and physical size). Finally, the condenser duty is 9.1% smaller (in terms of duty, which would approximately correspond to area as the heat transfer coefficients are similar between the two cases). This data leads to the conclusion that the physical size of the equipment required in the case by Aneke would be larger than the single-source case investigated by the KBS. Hence, the capital cost is larger.

The ratio of capital cost to cost savings remains approximately equal, as the project payback periods are 3.57 (Aneke) and 3.65 (KBS), a difference of only 2.2%.

The only large discrepancy between the results is the greenhouse gas reductions achieved by the ORC installation. Here, one would expect the greenhouse gas emission savings generated by the KBS to be lower due to the smaller net power output of this system. However, it is larger (575.7 as opposed to 549.9 tCO_2eq/year). This is explained by different emissions factors assumed by each method. Aneke used a method based on a mass balance of an “energy mix” power plant which resulted in an emissions factor of 0.394kg/kWh whereas the KBS assumes a value of 0.525 kg/kWh, as recommended by the Carbon Trust (2012) as the true emissions factor of grid electricity in the UK. Hence, it is deemed that the ORC model assumption remains valid in this case.

Furthermore, a hand calculation of the greenhouse gas emission savings for the KBS data using the emissions factor suggested by Aneke (2012) leads to greenhouse gas reductions of approximately 432tCO_2eq/year for the case study. This is 78% of the value of the data by Aneke (2012), which is correlated to the difference in net power outputs between the two cases (also 78%).

Overall, this case study has proven that the relatively simple methods employed by the KBS can produce results that are comparable with those achieved using a proven power cycle software package such as IPSEpro (SimTech, 2013). The optimisation loop used in the KBS knowledge-base to maximise net power output produces comparable results (in terms of turbine inlet temperature, and net power output) to the published methods. Also, the methods for producing economic and environmental data are comparable.

Furthermore, it has shown that the KBS can suggest a viable waste heat recovery solution for a process plant where, superficially, no obvious solutions were present.

Finally, drawbacks exist such as the inability to combine multiple heat sources into one cycle and the inability to optimise via addition of heat exchangers. However, the results have shown that the system produces a good initial analysis which is enough to base an initial decision on. Further optimisation may then be completed later during the final design phase.

6.2. Case Study 2: Textile Industry

This case study is based on the data of Pulat *et al* (2009). It presents the potential for waste heat recovery from textile production in the Turkish city of Bursa. This is primarily a theoretical study using in-house models generated by the authors, although it is based on real plant data.

6.2.1. Case Study Summary

Waste heat recovery was investigated for an "average" textile plant in Bursa, Turkey. Data was collated for over 200 active dyeing plants in the city, and average values produced for various heat sources as shown in Table 6.12.

Table 6.12. Summary of waste heat availability in Turkish textile industry (Pulat *et al*, 2009)

Process	Effluent Temperature (^oC)	Volume per shift (L/shift)
Bleaching	96	5000
Washing	96	5000
Acidification	50	5000
Dyeing	96	5000
Washing	90	5000
Hot Rinse	70	5000

The authors state that an accepted approach to waste heat recovery at these sites was to store the effluents in a well-insulated buffer tank, thereby creating a steady-state supply of hot waste water at an approximately constant temperature. It was stated that filters were used to ensure no insoluble solids are present in the heat source stream. This resulted in an approximately steady-state heat source which is further described in Section 6.2.2.

The heat sink is defined as the feed to the low-temperature water store to be used in the *"finishing"* process. The required water temperature is stated as 60^oC. Further data for this heat sink is given in Section 6.2.2.

The authors only considered waste heat recovery by shell-and-tube heat exchanger. The KBS will consider further options to find the most recommended approach to waste heat recovery at this plant. Parametric exergy analysis was also carried out by the authors but this is outside the scope of the KBS, and this analysis is based on the modal values stated by the authors.

6.2.2. Data Provided

The heat source (as described above) is a combined waste water source and the heat sink is the water feed to the low-temperature store. This data, along with general plant data, is shown in Table 6.13.

Table 6.13. Case study 2: heat source data (Pulat *et al,* 2009*)*

Source Nature	Liquid (waste water)
Source Temperature (^oC)	83.0
Source Target Temperature (^oC)	20.0[1]
Source Mass Flowrate (kg/s)	8.33
Source Pressure (bar)	1.01
Insoluble solids in source	No
Source Viscosity (cp)	1.00[2]
Source Density (kg/m^3)	1000[2]
Source Material Compatibility	Only Stainless Steel
Source Access for Cleaning Required	Yes
Sink Nature	Liquid (water)
Sink Temperature	20.0
Sink Target Temperature	60.0
Sink Mass Flowrate (kg/s)	12.1
Sink Pressure (bar)	4.00
Insoluble Solids in Sink	No
Sink Viscosity (cp)	1.00
Sink Density (kg/m^3)	1000
Sink Material Compatibility	No constraints listed (source

	limiting)
Sink Access for Cleaning Required	Yes (scaling possible)
Current Heating Utility	Gas
Efficiency of Current Heating Method	Assumed in paper to be 100%
Cost of Gas (£/kWh)	0.022[3]
Cost of Electricity (£/kWh)	0.079[3]
Operating hours/year	7200
Plant tolerant to working fluids with toxicity levels of less than or equal to 400 ppm by volume	Yes[4]
Plant tolerant to working fluids with high or moderate flammability?	Yes[4]

Note: [1]This is not explicitly given but is taken as the ambient condition stated by the authors. [2]Values of viscosity and density are taken as those for water under standard conditions. [3]The cost of electricity is not given. Therefore, this was acquired using IEA data for Turkey via DECC (d) (2013). The gas cost data is therefore taken from the same source for consistency purposes. [4]This data was not given explicitly as heat pumps/ORCs are not considered by the authors. However, the KBS results showed that an ORC was viable (see Section 6.2.3), therefore this data is inferred from the fact that various hazardous chemicals are used during textile manufacture (during bleaching, for example). Hence, it is assumed toxic and flammable chemicals can be tolerated.

6.2.3. Results and Discussion

The published results show the design of a shell and tube heat exchanger with four tube passes and one shell pass. The technical results are as follows:

Table 6.14. Technical design results by Pulat *et al*, 2009

Heat Source Inlet Temperature (°C)	83.0
Heat Source Outlet Temperature (°C)	25.0
Heat Source Pressure Drop (bar)	0.00 (assumed by author, not calculated)
Heat Sink Inlet Temperature (°C)	20.0
Heat Sink Outlet Temperature (°C)	60.0
Heat Sink Pressure Drop (bar)	0.00 (assumed by author, not calculated)
Heat Exchanger Duty (kW)	2030
Heat Exchanger Area (m^2)	228.4
Number Tubes	185
Number Tube Passes	4
Tube Outer Diameter (m)	0.025
Number Shell Passes	1
Heat Exchanger Effectiveness (%)	92.1

Table 6.15, below, shows the results of the initial analysis to decide which waste heat recovery technology categories are possible for the case study. Screenshots from this case study are shown in Appendix V.

Table 6.15. Case study 2: KBS initial selection results

		Reason
Heat Exchanger Heat Recovery Possible	Yes	N/A
Closed-Cycle Vapour Compression Heat Pump Possible	No	Heat sink does not require a temperature lift in source

Mechanical Vapour Recompression Possible	No	Not an evaporative process
Organic Rankine Cycle Possible	Yes	N/A

Table 6.15 shows that the results are in agreement with Pulat *et al* in that heat exchanger waste heat recovery is possible. However, it also shows that organic Rankine cycle waste heat recovery is also a possibility. Hence, results are generated for both options as shown in Tables 6.16-6.18 below.

Table 6.16 shows the selection of particular liquid-liquid heat exchangers available for the duty.

Table 6.16. Case study 2: KBS heat exchanger selection results

Heat Exchanger Type	**Selection**	**Reason**
Plate and Frame	Yes	N/A
Brazed Plate	No	Access for cleaning/maintenance not possible
Welded Plate	No	Access for cleaning/maintenance not possible
Plate and Shell	No	Access for cleaning/maintenance not possible
Shell and Tube	Yes	N/A
Spiral	No	Only considered when at least one fluid is a slurry

Table 6.16 shows that the results are in agreement with Pulat *et al* regarding the suitability of a shell and tube heat exchanger for this duty. However the plate and frame (or gasketted plate) heat exchanger is also suitable. The technical results for each are shown in Tables 6.17-18 below. Note: the results for shell and tube heat exchanger also include comments comparing the results to those by Pulat *et al.*

Table 6.17. Case study 2: KBS shell-and-tube technical design results

		Comment
Heat Source Inlet Temperature (^oC)	83.0	N/A
Heat Source Outlet Temperature (^oC)	30.0	Higher (5^oC)
Heat Source Pressure Drop (bar)	0.0001	Larger
Heat Sink Inlet Temperature (^oC)	20.0	N/A
Heat Sink Outlet Temperature (^oC)	56.5	Lower (3.5^oC)
Heat Sink Pressure Drop (bar)	0.0001	Larger
Heat Exchanger Duty (kW)	1854	Smaller (8.7%)
Heat Exchanger Area (m^2)	175.1	Smaller (23.3%)
Number Tubes	457	Larger (factor of around 2.5)
Number Tube Passes	1	Lower (factor of 4)
Tube Outer Diameter (m)	0.05	Larger (factor of 2)
Number Shell Passes	1	Equal
Unit Length (m)	2.44	Not stated by Pulat *et al*
Unit Diameter (m)	1.66	Not stated by Pulat *et al*
Heat Exchanger Effectiveness (%)	84.1	Lower (8.6%)

The results show some discrepancies between the KBS and methods of Pulat *et al*. Firstly, the source outlet temperature is higher by 5^oC which results in a lower sink outlet temperature (due to the heat balance). This is due to the KBS's assumption that the minimum approach temperature in a shell-and-tube heat exchanger should be 10^oC, whereas it appears this has been set as 5^oC by Pulat *et al*. An approach temperature of 10^oC is standard for the "first design" of shell and tube heat exchangers, as suggested by Coulson and Richardson (2005), Kern (1950) and Perry and Green (2008), amongst others. Therefore, this assumption should not be changed.

Secondly, the KBS results show a lower required area ($175.1m^2$ as opposed to $228.4m^2$, a difference of 23.3%). This is again due to variation in the inlet/outlet temperatures of the source/sink. The KBS results have a logarithmic mean temperature difference (LMTD) of 16.9°C as opposed to 11.8°C in the results by Pulat *et al*. This difference in LMTD explains the smaller area required in the KBS results and show that the overall heat transfer coefficient estimates used in the knowledge-base are approximately equal to those used in the methods by Pulat *et al*. This suggests that the estimate heat transfer coefficients are sufficient to create a "first design" of the heat exchangers, as required in the scope of the system.

Finally, the configuration of heat exchanger is very different. The KBS results show a 1-shell and 1-tube pass heat exchanger, whereas Pulat *et al* suggest the use of a 1-shell and 4-tube pass unit. Initially it appeared that this was a possible error in the KBS code, as generally one would employ a multi-pass shell-and-tube heat exchanger where possible. However, the KBS results were proven accurate, for the following reason:

The KBS considers the three most common pass arrangements for shell-and-tube heat exchangers as follows (in order of preference): (1) 2-shell-pass, 4-tube-pass; (2) 1-shell-pass, 2-tube-pass; (3) 1-shell-pass, 1-tube-pass. The system uses the correction factor equations (Perry and Green, 2008) to analyse whether each pass configuration is possible as shown below in Equations 6.7-6.12. The KBS calculates the correction factor, and if the correction factor is found to be less than 0.8, or the configuration is not possible due to temperature crosses, then the configuration in question is not possible. If both multi-pass configurations are not possible then the design reverts to a single pass counter-flow design.

$$F_{1-2} = \frac{\left[\frac{\sqrt{R^2}}{R-1}\right] \ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{A+\sqrt{R^2+1}}{A-\sqrt{R^2+1}}\right]} \qquad \text{Eq. 6.7}$$

$$F_{2-4} = \frac{\left[\frac{\sqrt{R^2}}{2(R-1)}\right] \ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{A+B+\sqrt{R^2+1}}{A+B-\sqrt{R^2+1}}\right]} \qquad \text{Eq. 6.8}$$

Where F_{1-2} is the correction factor for a 1 shell pass, 2 tube pass configuration, F_{2-4} is the correction factor for a 2 shell pass, 4 tube pass configuration, and R, P, A and B are equal to the following:

$$R = \frac{T_1 - T_2}{t_2 - t_1} \quad \text{Eq. 6.9}$$

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad \text{Eq. 6.10}$$

$$A = \frac{2}{P} - 1 - R \quad \text{Eq. 6.11}$$

$$B = \frac{2}{P}\sqrt{(1-P)(1-PR)} \quad \text{Eq. 6.12}$$

Where upper case “T” denotes the source temperature, lower case “t” represents the sink temperature, “1” denotes the inlet, “2” denotes the outlet.

For both configurations, upon input of the source/sink temperature data, the denominator of the correction factor equation is the natural logarithm of a negative number which is not a meaningful value. This indicates that the use of each configuration would result in a temperature cross and hence they are not possible. Therefore, the KBS suggested a 1-shell and 1-tube pass configuration. It is noted that the results by Pulat *et al* show a 1-shell and 4-tube pass configuration. However, the KBS could not be expected to produce such a result as this is a non-standard configuration, which is not listed in the common texts of Coulson and Richardson (2005) and Kern (1950). The KBS, however, has produced a valid design, comparable to that produced by Pulat *et al*.

The results for plate and frame heat exchanger are as follows:

Table 6.18. Case study 2: KBS plate-and-frame technical design results

Heat Source Inlet Temperature (^oC)	83.0
Heat Source Outlet Temperature (^oC)	25.0
Heat Source Pressure Drop (bar)	0.0021
Heat Sink Inlet Temperature (^oC)	20.0
Heat Sink Outlet Temperature (^oC)	59.9
Heat Sink Pressure Drop (bar)	0.0021
Heat Exchanger Duty (kW)	2029

Heat Exchanger Area (m^2)	46.8
Plate Height (m)	0.72
Plate Width (m)	0.23
Unit Depth (m)	1.03
No. Plates	282
Heat Exchanger Effectiveness (%)	92.1

Table 6.18 shows that the use of a plate and frame heat exchanger in this case would be preferred to a shell-and-tube from a technical point of view. The plate and frame achieves a higher effectiveness, allowing it to achieve the heat sink target temperature of 60^{o}C as requested by the user. The results also show a much more compact unit, with unit dimensions of 0.72 x 0.23 x 1.03m (height x width x depth) as opposed to 1.66 x 1.66 x 2.44m for the shell and tube. This results in a volume of 0.171m^2 for the plate and frame, and 5.28m^3 for the shell and tube, a 31-fold reduction. This is often preferred in retro-fit waste heat recovery systems due to space limitations.

There is also the disadvantage of increased pressure drop with 0.002bar for plate and frame and 0.0001 bar for the shell and tube (for both source and sink). However, this remains less than 1% of the inlet pressure for both sides, hence the pressure drop is insignificant and would not affect the final choice of heat exchanger.

The heat exchanger design results for this case study highlight the advantage of using the KBS to investigate waste heat recovery rather than considering only one technology, in this case conventional shell and tube heat exchangers.

The final option for waste heat recovery in this case study is for power generation via an ORC. The data generated for this option is shown in Table 6.19 below.

Table 6.19. Case study 2: KBS ORC technical design results

Working Fluid	R-245fa
Source Inlet Temperature (° C)	83.0
Heat Source Outlet Temperature (° C)	58.2
Pre-heater + Evaporator Duty (kW)	868
Heat Sink Inlet Temperature (° C)	20.0
Heat Sink Outlet Temperature (° C)	30.0
Condenser Duty (kW)	824.5
Heat Sink Mass Flow Rate (kg/s)	19.6
Working Fluid Mass Flow Rate (kg/s)	4.36
Working Fluid Turbine Inlet Temperature (° C)	56.9
Working Fluid Turbine Inlet Pressure (bar)	4.19
Working Fluid Turbine Outlet Temperature (° C)	41.3
Working Fluid Turbine Outlet Pressure (bar)	2.18
Working Fluid Condensation Temperature (° C)	35.0
Plant Gross Power Output (kW)	42.9
Working Fluid Pump Power (kW)	0.94
Net Power Output (kW)	41.9
Plant Thermal Efficiency (%)	4.82

These results imply that an ORC would not be a suitable option for waste heat recovery at this site. The thermal efficiency of the cycle is very low at 4.82% and the net power output of 41.9kW would not generate a large revenue in terms of utility savings: typically, the thermal efficiency would be greater than 10% to be considered viable (Brasz, 2011). The increased complexity of the cycle (as highlighted by the cycle diagram generated by the KBS, shown in the case study screen shots in Appendix V) also implies that the ORC would not be suitable for the case study compared to the heat exchanger options that are available.

However, as stated in the scope of the system (Section 1.3.1), it is important for the KBS to consider and display all available options as this enhances the educational value of the system.

The final decision on which technology is most suitable for this case study should also consider the economic and environmental results, as shown in Table 6.20. Here, a comparison of the results by Pulat *et al*, the shell and tube results by the KBS, the plate and frame results by the KBS and the ORC results by the KBS is presented. (note: the data by Pulat *et al* is converted from $USD to £GBP using the 2011 exchange rate of 1.61 $USD to £1GBP [Oanda, 201]).

Table 6.20. A comparison of the economic and environmental results generated by the KBS and Pulat *et al*, 2009

	Pulat *et al*: Shell and Tube Heat Exchanger	KBS: Shell and Tube Heat Exchanger	KBS: Plate Heat Exchanger	KBS: ORC
Estimate Capital Cost (£GBP)	68803 (34400 for heat exchanger alone)	12204	5308	108260
Estimate Maintenance Cost (£GBP/year)	2981	Not given	Not given	2165
Estimate Cost Savings (GBP/year)	263253	293708	321416	23839
Estimate Payback Period (years)	0.29	0.04	0.02	4.99

Estimate Greenhouse Gas Reductions (tCO_2eq/year)	Not given	2451	2682	158

Comparison of Shell and Tube data by Pulat *et al* and the KBS results

The economic results produced by both Pulat *et* al and the KBS for the shell and tube heat exchanger show a large difference in the capital cost and maintenance costs. Pulat *et al* estimate the capital cost to be 5.6 times greater than the KBS: £68803 as opposed to £12204. The method by Pulat *et al* included the cost of the heat exchanger, installation, pumps, valves, connections, freight, local taxes, retrofit of current system and testing. It was stated that the additional costs (non-heat exchanger capital costs) are approximately double the cost of the heat exchanger itself. Hence, the heat exchanger itself would cost an estimate £34400. This remains almost three times larger than the estimate by the KBS. It is devised using a similar cost factor method to that used in the KBS, albeit with larger cost factors.

Both methods use cost factors originating from studies of manufacturer data. Hence, it is difficult to determine which is more accurate or define the accuracy without obtaining a cost estimate from a potential manufacturer. Hence, it is difficult to judge whether the heat exchanger capital costs estimation methods employed by the KBS are accurate based on this case study. This highlights a flaw in the KBS methodology as the accuracy of cost factor methods is highly subjective. However, this could only be rectified by obtaining a significant amount of economic data (for all heat exchangers) from a wide range of industrial partners.

The KBS also does not consider the need for maintenance costs to be included in heat exchanger economic analysis, whereas Pulat *et al* include a maintenance charge of £2981 per year. It is not stated where this figure was calculated from, hence it is difficult to analyse whether this is accurate.

The cost savings estimated by the KBS and Pulat *et al* differ by 11.5%. This is due to a varying estimate of gas prices. Pulat *et al* use a Turkish source which was not accessible rather than stating the gas cost at the plant in question. During KBS operation, a value was taken from IEA (via DECC (d), 2013) as this can be considered accurate. Therefore, a direct comparison is not possible. However, the error is not due to the methods of the KBS, but instead due to the differences in the input data to the two models.

The difference in both the capital cost and cost savings estimates leads to a rather large difference in payback time estimate, with Pulat *et al* estimating 0.29 years and the KBS estimating 0.04 years. Again, this is down to different cost factors being used in the two methods.

<u>Discussion of KBS results</u>

First of all, the economic and environmental results for the ORC, as generated by the KBS, confirm that this technology is not the most suitable for this case study. The capital cost estimate is significantly higher than the other options, and the estimate cost savings are significantly lower, as are the potential greenhouse gas reductions. When this is coupled with the increased complexity of this solution, this technology can be ruled out for use in this case study. However, this result is important from an educational point of view, as it highlights the advantages of heat exchanger waste heat recovery (compared to ORC), and shows that one would not use such a complex solution if a simpler solution was available, for cost reasons alone.

From this point of view, the results are in agreement with Pulat *et al* in that a heat exchanger is the recommended technology for waste heat recovery in this case study.

However, the KBS results show that the plate and frame heat exchanger would be preferred to the shell-and-tube heat exchanger. This heat exchanger is around half the capital cost of the shell-and-tube, and achieves higher cost savings and greenhouse gas reductions (due to the high effectiveness of this unit, as discussed previously). Therefore, the economic and environmental results are in agreement with the technical results (as discussed following Table 6.18) and the plate and frame heat exchanger would be the preferred heat exchanger for this case study.

In conclusion, this case study has shown that the KBS has selected heat exchanger waste heat recovery as the most suitable technology for waste heat recovery, in agreement with the work of Pulat *et al*. However, the KBS recommends the use of the plate-and-frame heat exchanger which has numerous advantages such as compact size, higher effectiveness and greater project economic value. There is a large discrepancy between the capital cost estimates employed by Pulat *et al* and those programmed into the KBS. The accuracy of the methods is highly subjective and cannot be judged without obtaining a genuine cost estimate from a supplier.

6.3. Case Study 3: Industrial Washing Process

This case study was originally published by Paarske (2011). It presents both theoretical and experimental (via a technology trial) data.

6.3.1. Case Study Summary

Waste heat recovery was investigated in a metal-parts processing plants in Austria by a group of energy consultants from the Danish Technological Institute. The washing section of the process was the main focus as it is the only significant consumer of low-grade thermal energy. In this section of the plant, a number of small continuous drum-type washers are used to clean metal parts. Each requires an average of 25.0kW of thermal energy in the final wash/rinsing stage, to provide hot demineralised water at 62^{o}C. Currently, this heat sink is heated using electrical heating elements. The case study was concerned with only one of the washing units, although comments are made regarding the impact of applying waste heat recovery to every unit in the plant. A sketch of the process is shown in Figure 6.4.

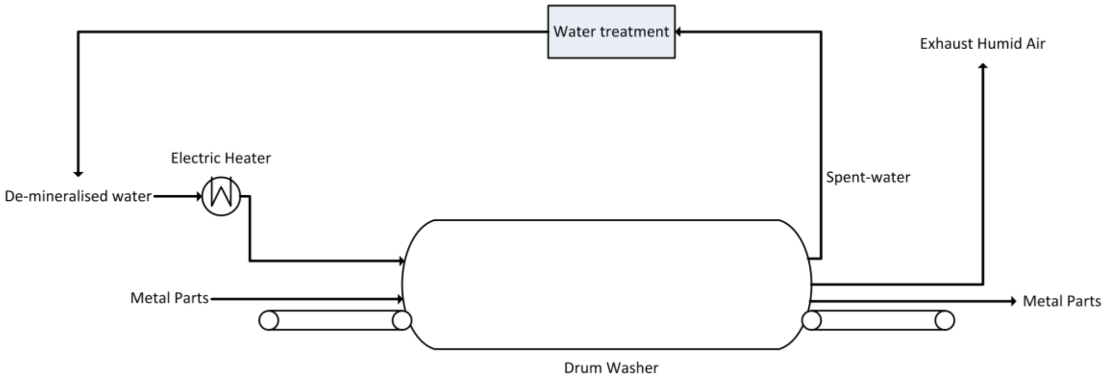


Figure 6.4. Sketch of industrial washing process

6.3.2. Data Provided

The heat source identified was the exhaust gas leaving the washers which was described as a humid air stream. The case study data (including source data, sink data and general plant data) required to run the KBS is shown in Table 6.21.

Table 6.21. Case study 3: Case study data (Paarske, 2011)

Source Nature	Humid Air
Source Temperature (^{o}C)	53.0
Source Target Temperature (^{o}C)	20.0[1]
Source Mass Flowrate (kg/s)	0.122
Source Pressure (bar)	1.01
Water Vapour Fraction (mass %)	8.3[2]
Sink Nature	Liquid (Demineralised water)
Sink Temperature (^{o}C)	58.0
Sink Target Temperature (^{o}C)	62.0
Sink Mass Flowrate (kg/s)	1.49
Sink Specific Heat Capacity (kJ/kgK)	4.20

Current Method of Heating Sink	Electricity
Efficiency of Current Heating Method (%)	99.0[3]
Cost of Electricity (£GBP/kWh)	0.09
Operating hours/year	8000
Plant tolerance to working fluids with toxicity levels of less than or equal to 400 ppm by volume?	No
Plant tolerance to working fluids with high or moderate flammability?	No
Is heat sink duty "useful"?	No

Note: [1]This is not given and is therefore taken as a *typical* ambient temperature. [2]This is not given and is therefore calculated from the data stated which is a relative humidity of 95%. [3]This is not given but assumed to be this relatively high value as electric heaters are generally highly efficient.

6.3.3. Results and Discussion

The published results cover the design of a standard heat pump cycle with a scroll compressor. The technical results are summarised in Table 6.22.

Table 6.22. Technical heat pump design results (Paarske, 2011)

Working Fluid	R-134a
Heat Source Inlet Temperature (^{o} C)	53.0
Heat Source Outlet Temperature (^{o} C)	38.0
Evaporator Duty (kW)	18.0
Heat Sink Inlet Temperature (^{o} C)	58.0
Heat Sink Outlet Temperature (^{o} C)	62.0
Condenser Duty (kW)	25.0
Working Fluid Mass Flow Rate (kg/s)	0.189 [1]
Working Fluid Evaporation Temperature (^{o}C)	15.0
Working Fluid Evaporation Pressure (bar)	4.89[2]

Working Fluid Condensation Temperature (^oC)	65.0
Working Fluid Condensation Pressure (bar)	18.9 [2]
Compressor Work (kW)	Not Specified
Motor Work (kW)	7.00
Compressor Pressure Ratio (P_{cond}/P_{evap})	3.87
No. Stages	1.00
COP	3.57

[1]Working fluid mass flow rate is not explicitly given. Therefore, this is calculated using the condenser heating duty and heat of vaporisation data from the ASHRAE Fundamentals Handbook (ASHRAE, 2009). [2]Working fluid pressures are not explicitly given. Therefore, they are found using vapour pressure data from the ASHRAE fundamentals handbook (ASHRAE, 2009).

Table 6.23 shows the results of the initial analysis to decide which waste heat recovery technology categories are possible for the case study. NB: Screenshots from this case study are shown in Appendix V.

Table 6.23. Case study 3: KBS initial selection results

		Reason
Heat Exchanger Heat Recovery Possible	No	No *matching* heat sink
Closed-Cycle Vapour Compression Heat Pump Possible	Yes	N/A
Mechanical Vapour Recompression Possible	No	Not an evaporative process
Organic Rankine Cycle Possible	No	Source temperature too low (less than 73^oC)

The results above are in agreement with that of Paarske, and show that a vapour compression heat pump waste heat recovery is the only option in this case study.

The heat pump design results generated by the KBS are shown in Table 6.24 below along with comments comparing the data to that published by Paarske.

Table 6.24. Case study 3: KBS technical heat pump design results

		Comment
Working Fluid	R-134a	Equal
Heat Source Inlet Temperature (^{o} C)	53.0	N/A
Heat Source Outlet Temperature (^{o} C)	20.0	Lower (18^{o}C)
Evaporator Duty (kW)	19.4	Larger (1.4)
Heat Sink Inlet Temperature (^{o} C)	58.0	N/a
Heat Sink Outlet Temperature (^{o} C)	62.0	Equal
Condenser Duty (kW)	25.0	Equal
Working Fluid Mass Flow Rate (kg/s)	0.18	Smaller (4.76%)
Working Fluid Evaporation Temperature (^{o}C)	15.0	Equal
Working Fluid Evaporation Pressure (bar)	4.89	Equal
Working Fluid Condensation Temperature (^{o}C)	67.0	Higher (2^{o}C)
Working Fluid Condensation Pressure (bar)	19.7	Higher (1.8 bar)
Compressor Work (kW)	5.62	N/A
Motor Work (kW)	5.91	Smaller (15.6%)
Compressor Pressure Ratio (P_{cond}/P_{evap})	4.03	Higher (4.1%)
No. Stages	2	Higher (100%)
COP	4.22	Higher (9.0%)

Table 6.24 shows a good agreement between the data presented by Paarske and the results of the KBS. The main difference is in the heat source outlet temperature which is 18^{o}C lower in the KBS results. This suggests a difference in the heat balance calculations between the two methods.

The total heating duty of the source is made up of sensible heat by air cooling and the latent heat of water condensation as shown in Equations 6.12-6.17 below. The sensible heat of liquid water cooling is ignored in the KBS method, as it may "drip" from the heat transfer surface before cooling. Hence, it cannot be guaranteed that this heat will be transferred to the sink.

$$Q_{source} = Q_{latent} + Q_{sensible} \qquad \text{kW} \qquad \text{Eq. 6.12}$$

$$Q_{latent} = \left(m_{water,vap,in} - m_{water,vap,out}\right) * \Delta H_{evap} \qquad \text{kW} \qquad \text{Eq. 6.13}$$

$$Q_{sensible} = m_{air} Cp_{air} (T_{in} - T_{out}) \qquad \text{kW} \qquad \text{Eq. 6.14}$$

The mass of water vapour out for the results by Paarske can be found by firstly calculating the water vapour pressure at the outlet temperature (Eq. 6.15), then calculating the stream humidity (Eq. 6.16) before finally converting this into a mass flow, as follows:

$$P_{w,out} = \exp[20.838 - \frac{5132}{T}] \qquad \text{mmHg} \qquad \text{Eq. 6.15}$$

Note: T here is in Kelvin and is therefore 311K for the Paarske results.

The partial pressure of water vapour at the outlet ($P_{w,out}$)is found to be 76.4mmHg.

$$H_{w,out} = \frac{\mathrm{P_{w,out}}}{\mathrm{P_{system}} - \mathrm{P_{w,out}}} \frac{\mathrm{M_W}}{\mathrm{M_A}} \qquad \text{kg/kg} \qquad \text{Eq. 6.16}$$

Note: The system pressure is atmospheric pressure (as stated in the case study data) and is input as 760mmHg for unit consistency. M_w and M_A are the molar mass of water (18.0) and air (29.0) respectively.

The air humidity at the outlet ($H_{w,out}$) is found to be 0.0695kg/kg.

$$m_{water,vap,out} = H_{w,out} * m_{air} \qquad \text{kg/s} \qquad \text{Eq. 6.17}$$

The water vapour mass flow rate at the outlet was found to be 0.00778 kg/s. The water vapour mass flow rate at the inlet is 0.0101 kg/s. Hence, the source duty in the results presented by Paarske can be calculated using this data and the inlet/outlet temperatures using Equation 6.12-14, and was found to be 7.12 kW, which is significantly lower than the 18.0 kW reported. Hence, the results of the KBS are correct for this case study and an error in the heat balance is evident in the results of Paarske.

One other result showing a discrepancy is the motor work of the compressor. Here, the KBS results indicate that the motor work required for compression is 5.91kW whereas Paarske's results show a motor size of 7.00 kW. This is probably due to the discrepancy between the assumed efficiency of the compressor and motor in the KBS method (75% and 95% respectively). However, no comparison is available as this data is not specified in the results by Paarske.

The final difference between the results is in the selection of compressor. The KBS results show the use of a 2-stage centrifugal compressor, while Paarske has designed a system using a single-stage scroll compressor. This highlights a drawback of the KBS whereby only one compressor type is considered. In this case study, it is clearly beneficial to use the scroll compressor suggested by Paarske, as a single stage compression system is less complex than a 2-stage system.

Overall, however, there is a good agreement between the KBS models and the results by Paarske. The key temperatures and pressures throughout the cycle are approximately equal, and only small discrepancies exist between other key results such as motor work and evaporator duty. Hence, it can be concluded that the technical results of the KBS are suitable for a "first design" of the system, as was specified in the system scope. Further improvements (such as optimal compressor selection, for example) would then be investigated later once this method of waste heat recovery was chosen.

The economic and environmental results presented by Paarske are shown in Table 6.25 below, along with those generated by the KBS. As the case study is from Austria, the economic results are presented in the Euro currency. Therefore, this has been converted using the 2011 exchange rate of 1.11 (Euro per GBP) (Oanda, 2011).

Table 6.25. A comparison of the economic and environmental results generated by the KBS and Paarske, 2011

	Results, Paarske (Cost in £GBP)	**Results, KBS (Cost in £GBP)**
Estimate Capital Cost (Unit Currency)	27027	13918
Estimate Maintenance Cost (Unit Currency/year)	Not Given	278
Estimate Cost Savings (Unit Currency/year)	10909	11595
Estimate Payback Period (years)	2.50	1.23
Estimate Carbon Dioxide Reductions (tCO_2eq/year)	50	81.1

Table 6.25 shows that the capital cost provided by Paarske is almost twice that estimated by the KBS. This is due to the fact that the data presented by Paarske was for the full installed system, including retrofit costs, freight, testing and monitoring, control systems and local taxes. The cost estimates provided by the KBS are for the heat pump system *only*. When all of the additional costs are considered, it can be concluded that the KBS estimate capital cost is accurate enough for a “first design” although this is difficult to quantify due to the inherent subjective nature of the estimates, and geographic and temporal variation of such costs.

There is a good match between the cost savings. The small difference between the two values is due to increased COP and decreased motor work in the KBS results.

The heat pump's electricity demand is therefore lower, which increases the overall cost savings.

The KBS results show larger associated greenhouse gas emission reductions than the results by Paarske. This can be explained as the KBS assumes a UK value for grid electricity associated emissions of 0.525 kg/kWh. As the case study is based in Austria, this assumption is not valid. The grid electricity associated emissions in Austria ars 0.310 kg/kWh (IEA, via DECC (e), 2013). Hence the associated emissions in the UK are 69.4% larger while the KBS result for reduction in greenhouse gas emissions is 62.2% larger. Hence, it can be concluded that this discrepancy in results is due to the varying associated emissions between the two countries. This highlights a key drawback of the KBS in that it has been programmed primarily for use in the UK, therefore any assumptions made are generally based on UK data which may not be valid elsewhere.

Overall, this case study has validated the KBS. The technical and economic/environmental results generated by the KBS are of reasonable accuracy compared to those of Paarske. The KBS results are therefore comparable with those of an expert waste heat recovery contractor and the KBS may be used in the initial stages of investigation as a viable alternative to such services.

6.4. Case Study 4: Brewery

This is an original case study using data provided by an industrial partner: a brewery located in southern Scotland, UK.

6.4.1. Case Study Summary

The brewery is best described as a "*medium*" sized brewery, producing approximately 2,500,000 litres of beer per annum. A variety of beers are produced, and this analysis is based on the most commonly produced beer which accounts for around 90% of the total production.

The plant has an active energy manager who has implemented a number of measures to increase the energy efficiency of the process, including energy audits, installation of energy efficient/variable speed drives and heat integration.

However, the “Wort Boiling” process remains extremely inefficient. Here, water is boiled from the “wort” (a brewing liquor made up of organic sugars from the “malting” grains dissolved into water) in order to concentrate the solution before fermentation. This is currently achieved using a calandria gas heater, in a run-around configuration, as depicted in Figure 6.5.

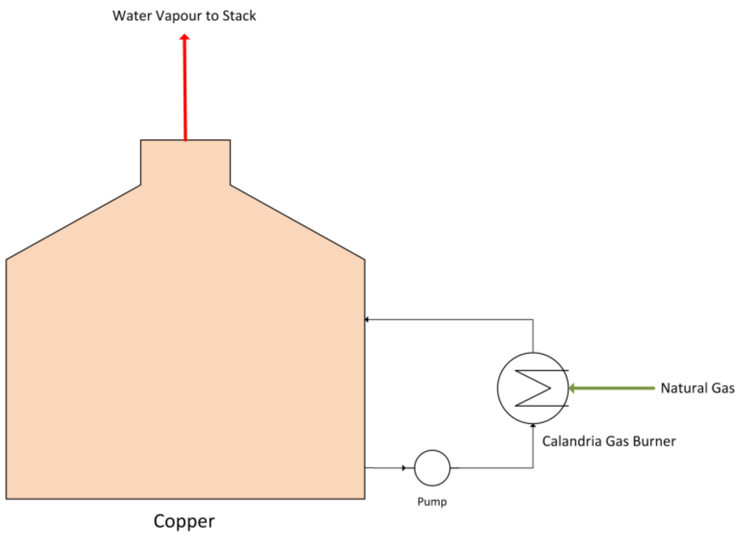


Figure 6.5. "Wort boiling" process from case study 4

The evaporation vessel (or “Copper” as it is referred to in brewing) is operated as a batch process. Initially, pre-heating is required to heat the wort from the inlet temperature of 65^{o}C to the boiling point of approximately 100^{o}C. This takes around 2 hours. Following this, 4% of the total mass of water is evaporated in a boiling process which is effectively at steady-state for 70 minutes. 784kg of water representing approximately 1770 MJ of latent heat is boiled from the wort and vented to the atmosphere. The water contains a negligible amount of volatile organic compounds and can be treated as pure water for the sake of thermodynamic analysis (as described for the case of both brewing and whiskey production by Brotherton (2012)).

No *suitable* heat sinks were identified for the transfer of the waste heat from the source via either a heat exchanger or vapour compression heat pump. It was suggested that the vapour could be used to heat water stored in the “hot well” of the process (used in production of the wort). However, this was rejected by the plant manager for the following reasons:

- The plant boiler is designed and optimised for current load. Reduction in load would be detrimental to boiler performance and service plan.
- The intermittent nature of the heat source (due to batch processing) would provide intermittent heat to the hot well feed. Hence, a new and relatively complex fuzzy controller would be required to utilise the waste heat as and when it was available, and to use conventional process steam when it was not available. The plant manager was not willing to employ such a system at the plant.

Therefore, the problem was reduced to one heat source (vapour from wort boiling) and no (*“standard”*)heat sinks.

6.4.2. Data Provided

As previously stated, the “copper” boils at approximately steady-state for 70 minutes per batch. There are four batches per day (Monday-Friday) and two batches at weekends. The plant has two weeks of down-time per year. Hence, 1200 batches are produced per year, giving a total steady-state operation of 1400 hours per year.

The source/process and general plant data is summarised in Table 6.26 below:

Table 6.26.Case Study 4: data

Evaporation Temperature (^oC)	100
Evaporation Pressure (bar)	1.01
Evaporative Rate (kg/s)	0.187

Circulation Mass Flow Rate (kg/s)	5.28
Circulation Pressure (bar)	2.00
Wort Temperature at Heat Exchanger Outlet (°C)	119
Current Heating Utility	Gas
Heat Exchanger Type	Tubular
Efficiency of Current Heating Method (%)	60.0
Hours of "steady state" evaporation per cycle	1.17
Cycles per day	3.43[1]
Operating days per year	351
Total hours of "steady state" evaporation per year	1400
Cost of Gas (£GBP/kWh)	0.024
Cost of Electricity (£GBP/kWh)	0.105

Note: [1]This is the average of 24 cycles per week divided by 7 days

6.4.3. Results and Discussion

The data provided was used by the KBS to generate the results shown below in Tables 6.27-6.28. As there is no published data to compare the results to in this case study, the technical results are compared to the results of using Aspen Plus (Aspen Tech, 2012) to design the system. Unfortunately there is therefore no valid economic comparison for this case study. This is because methods from literature were reviewed, analysed and used in building the section of the system knowledge-base which produces the estimate capital costs, cost savings and other related data. Hence, if the same methods were then used manually for comparison, the same result would be found so, this comparison would be invalid.

The initial stage of results generated by the KBS is as follows and screenshots from this case study are shown in Appendix V. Table 6.27 shows the results of the initial analysis to decide which waste heat recovery technology categories are possible for the case study. Here, as expected based on the source/sink data provided, the only option for waste heat recovery is to use Mechanical Vapour Recompression (MVR).

Table 6.27. Case study 4: KBS initial selection results

		Reason
Heat Exchanger Heat Recovery Possible	No	No *matching* heat sink
Closed-Cycle Vapour Compression Heat Pump Possible	No	No *standard* heat sink within a reasonable temperature lift
Mechanical Vapour Recompression Possible	Yes	N/A
Organic Rankine Cycle Possible	No	Only continuous processes considered

Another result generated by the KBS is the determination of whether existing heat exchangers may be used in the new MVR system. In this case, the result is "No". This is because it is not possible to modify a gas burner in such a way.

Table 6.28 below shows the technical results generated by the KBS for MVR design.

Table 6.28. Case study 4: KBS technical MVR design results

Evaporative Duty (kW)	421.9
Compressor Inlet Temperature (^oC)	100
Compressor Outlet Saturation Temperature (^oC)	124
Maximum Possible Condenser Duty (kW)	438
Compressor Inlet Pressure (bar)	1.01
Compressor Outlet Pressure (bar)	2.29
Compressor Work (kW)	35.1
Drive Work (kW)	36.9
Compressor Pressure Ratio (P_{cond}/P_{evap})	2.29
COP	11.9

The system was simulated in Aspen Plus (AspenTech, 2012), according to the data shown in Table 6.26 (source data) and the model flow diagram shown in Figure 6.6, in order to compare the technical results of the KBS with a standard industrial modelling tool. The steam tables fluid property package was used as this was the most accurate for water/steam calculations. The Aspen model uses the same assumptions as the KBS (discussed in Section 4.4.3) *i.e.* the compressor isentropic efficiency is assumed to be 75%, the motor efficiency of the compressor is assumed to be 95% and the heat exchanger pinch point is assumed to be 5°C.

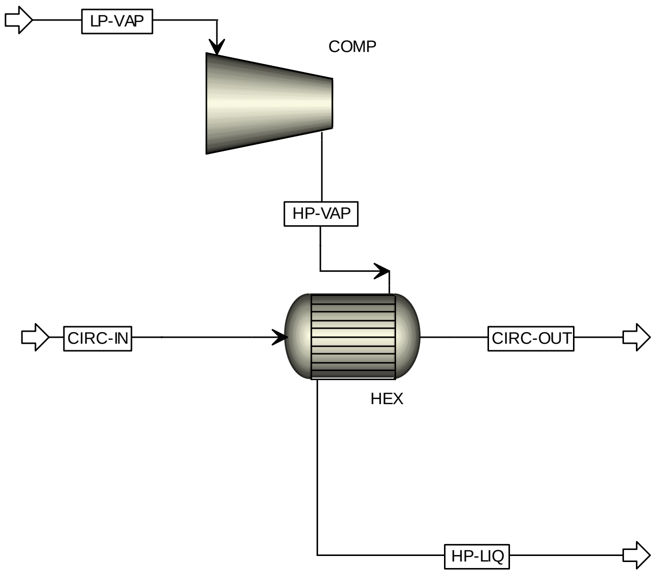


Figure 6.6. Simplified MVR model in Aspen Plus

The model results are shown in Table 6.29 along with comments comparing the results to those generated by the KBS in Table 6.28.

Table 6.29. Case study 4: Technical MVR design results generated using Aspen Plus

		Comment
Evaporative Duty (kW)	423.5	Equal

Compressor Inlet Temperature (^oC)	100	Equal
Compressor Outlet Saturation Temperature (^oC)	124	Approximately equal (less than 1% difference compared to ambient temperature)
Maximum Possible Condenser Duty (kW)	437	Approximately equal (less than 1 % difference)
Compressor Inlet Pressure (bar)	1.01	Equal
Compressor Outlet Pressure (bar)	2.3	Equal
Compressor Work (kW)	34.8	Approximately equal (less than 1% difference)
Drive Work (kW)	36.6	Approximately equal (less than 1% difference)
Compressor Pressure Ratio (P_{cond}/P_{evap})	2.29	Equal
COP	12.0	Approximately equal (<1% difference)

The results for the Aspen model are almost identical to those calculated by the KBS model. This shows that the KBS models function correctly and can produce results comparable with conventional modelling software. Hence, the KBS is capable of replacing an expert contractor with access to such software in the initial stages of waste heat recovery system design, as specified in the scope of the system (Section 1.3.1)

The economic and environmental results generated by the KBS are shown in Table 6.30. As previously stated, there is no basis for comparison for this case study as it is from original data rather than a publication or other source.

Table 6.30. Case study 4: KBS economic and environmental results

Units of gas saved per year (kWh/year)	1028459
Units of electricity required per year (kWh/year)	51987
Cost saving (£GBP/year)	19224
Potential GHG saving (tCO_2eq/year)	162
Capital cost estimate (£GBP)	55826
Estimate Maintenance Costs (£GBP/year)	1117
Simple payback time (years)	3.08

The data in Table 6.30 show that the MVR is suitable for this case study. The system has the potential to reduce utility bills by around £19000 per year due to a significant reduction in the gas demand, whilst expending a relatively small amount on electricity to drive the MVR compressor. The cost estimate generated is around £56000, giving a simple payback time of 3.08 years. The system also shows potential to save 162 tCO_2eq/year, which would be a direct emission saving at the plant due to the removal of a gas burner. The data in Table 6.30 would be sufficient to make a decision on whether to proceed with waste heat recovery at this plant.

Although a direct economic comparison was not available for this case study, it is noted that the results for payback time and system COP are similar to those reported in other works. For example, Staveley Chemicals Ltd (published in the Energy Efficiency Demonstration Scheme, Expanded Project Profile 259, 1989) installed an MVR system to provide the heating duty in a by-product evaporator. Here it was stated that typical pay back times for MVR systems would be in the range of 2-4 years. Similarly, Brotherton (2012) (an expert contractor in the field of MVR) states that typical MVR payback times in the whiskey industry (with highly similar production methods to brewing) are in the region of 3-5 years. (Note: a review of MVR literature is provided in Section 3.2.2). The payback time for this case study falls into this range, thereby suggesting that the methods employed are correct.

In conclusion, this case study has shown that the KBS has correctly identified that MVR is the only course of action for waste heat recovery at the plant in question. The KBS technical results were approximately equal to those generated using the conventional modelling tool Aspen Plus (Aspen Tech, 2013), thereby proving that the KBS may be used to replace an expert contractor in the initial stages of waste heat recovery design. Finally, the KBS economic and environmental results are detailed enough to allow a decision on whether or not this technology is feasible at the plant, and despite the lack of a direct comparison, data from previously published case studies suggest that the economic data generated by the KBS is reasonably accurate.

6.5. Case Study 5: Inorganic Chemicals Drying

This is an original case study using data provided by an industrial collaborator: an inorganic chemical production plant based in northern England. In particular, the case study concerns a large spray dryer unit in the downstream processing section of the plant. This is one of the largest consumers of low-grade thermal energy in the plant, consuming around 17.9 GWh of natural gas per year.

6.5.1. Case Study Summary

The process under investigation is a large spray drying tower. Here, fresh air is input heated to (on average) 300^{o}C before entering the dryer via a gas burner. The exhaust feed from the dryer is typically at 95^{o}C with significant moisture content. A sketch of the process is shown below in Figure 6.7.

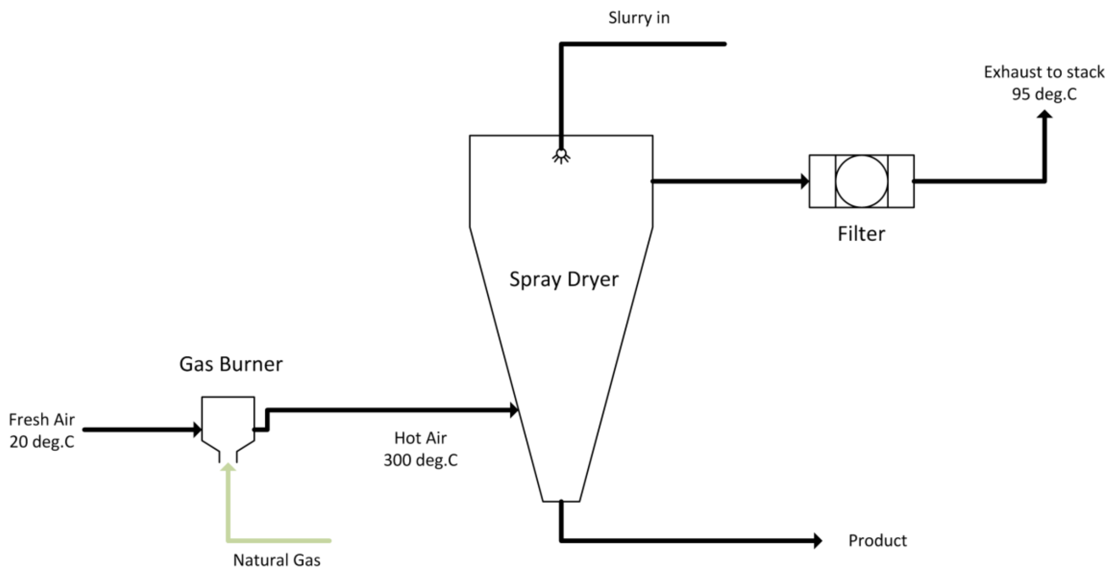


Figure 6.7. Spray dryer process from case study 5

The heat source is the spray dryer exhaust which is currently sent to the stack. This is a humid air heat source, containing solid particles. The mass fraction of the solid particles is known to be less than 1%, and the particle size is less than 20µm, and it is known that any heat exchanger installed would require cleaning/maintenance due to fouling. The only heat sink of interest to the host plant was the fresh air inlet to the spray dryer. This is because they would be unwilling to consider heat sinks further from the source due to civil engineering complexity and cost. The process is continuous, operating for 24 hours per day, 7 days per week for 48 weeks per year.

6.5.2. Data Provided

The heat source, as discussed above, is a humid air heat source from a spray dryer, and the heat sink is the fresh air feed to the process. The data provided by the host plant is summarized below in Table 6. 31.

Table 6.31. Case study 5: data

Source Nature	Humid Air

Source Temperature (^{o}C)	95.0
Source Target Temperature (^{o}C)	20.0
Source Mass Flowrate (kg/s)	6.21
Source Pressure (bar)	1.01
Water Vapour Fraction (mass %)	10.4
Solids in source	Yes.
Solid Size (mm)	0.02
Solid Fraction (mass %)	<1
Source Viscosity (cp)	0.0198[1]
Source Density (kg/m^3)	1.00[1]
Source Material Compatibility	No constraints
Source Access for Cleaning Required	Yes
Sink Nature	Gas (Air)
Sink Temperature	20.0
Sink Target Temperature	50.0
Sink Mass Flowrate (kg/s)	5.56
Sink Specific Heat Capacity (kW/kgK)	1.00
Sink Pressure (bar)	1.01
Insoluble Solids in Sink	No
Sink Viscosity (cp)	0.0198[1]
Sink Density (kg/m^3)	1.00[1]
Sink Material Compatibility	No constraints
Source Access for Cleaning Required	No
Current Heating Utility	Gas
Efficiency of Current Heating Method (%)	70
Cost of Gas (£/kWh)	0.0266
Cost of Electricity (£/kWh)	0.0867
Operating hours/year	8060

Plant tolerance to working fluids with toxicity levels of less than or equal to 400 ppm by volume	Yes[2]
Plant tolerance to working fluids with moderate flammability?	Yes[2]
Plant tolerance to working fluids with high flammability?	No[2]
ORC Heat Sink Availability?	Yes - Cooling water
Heat Sink Temperature (^{o}C)	10

Note: [1]Density and viscosity for both source and sink taken as data for air under standard conditions. [2]Plant stated that they would not be willing to employ extra health and safety measures required when using "flammable working fluids such as hydrocarbons" but toxicity/flammability of "ammonia levels" is acceptable (an ammonia chiller circuit is employed elsewhere on site).

6.5.3. Results and Discussion

The data provided was used by the KBS to generate the results shown below in Tables 6.32-6.36 and screen shots for this case study are shown in Appendix V. Firstly, the general categories of waste heat recovery technologies are devised as shown in Table 6.31 below.

Table 6.32. Case study 5: KBS initial selection results

		Reason
Heat Exchanger Heat Recovery Possible	Yes	N/A
Closed-Cycle Vapour Compression Heat Pump Possible	No	No heat sink within a reasonable temperature lift
Mechanical Vapour	No	Not an evaporative

Recompression Possible		process
Organic Rankine Cycle Possible	Yes	N/A

The KBS results in Table 6.32 show that two options are possible for waste heat recovery in this case study: heat transfer from source to sink via a heat exchanger and electricity generation via an ORC.

The results for which type of gas-gas heat exchangers may be suitable for this application are then generated, as shown in Table 6.33.

Table 6.33. Case study 5: KBS heat exchanger selection results

Heat Exchanger Type	Selection	Reason
Run-around-Coil	Yes	N/A
Gas-Gas Plate	Yes	N/A
Shell-and-Tube	Yes	N/A

Table 6.33 shows that all three of the heat exchangers suitable for a humid gas duty in the knowledge-base are suitable for use in this case study. Hence, design results are generated for each of the three heat exchangers (see Table 6.34 and also the other option shown in Table 6.32, organic Rankine cycle (see Table 6.35)

Table 6.34.Case study 5: KBS technical heat exchanger design results

	Run-around-Coil	Gas-gas Plate	Shell-and-Tube
Heat Source Inlet Temperature (^oC)	95.0	95.0	95.0
Heat Source Outlet Temperature (^oC)	57.4	57.4	57.4

Heat Source Pressure Drop (bar)	0.0009	0.0047	0.220
Heat Sink Inlet Temperature (^oC)	20.0	20.0	20.0
Heat Sink Outlet Temperature (^oC)	50.0	50.0	50.0
Heat Sink Pressure Drop (bar)	0.0008	0.0042	0.0144
Heat Exchanger Duty (kW)	166.7	166.7	166.7
Heat Exchanger Area (m^2)	113.7	Not given	23.2
Number Tubes	N/A	N/A	41
Number Tube Passes	N/A	N/A	2
Tube Outer Diameter (m)	N/A	N/A	0.05
Number Shell Passes	N/A	N/A	1
Unit Length (m)	N/A	1.27	1.83
Unit Height (m)	N/A	1.02	1.13
Unit Depth (m)	N/A	1.65	1.13
No Units (m)	1	2	1
Heat Exchanger Effectiveness (%)	40.0	40.0	40.0

The data in Table 6.34 shows equal performance by the three heat exchanger options in terms of effectiveness. This is due to a conservative heat sink target temperature. Hence, the *required* effectiveness of the unit is 40% and this is not a constraint of the units (run-around-coils can achieve a maximum effectiveness of 60%, gas-gas plate 65%, shell-and-tube 90% as discussed in Section 3.1.1).

Hence, the analysis of the three options and decision is based on other factors. The shell and tube results show a pressure drop of 0.22 bar on the source side, which is an order of magnitude greater than the other two options. This shows why *typically* shell-and-tube heat exchangers would not be used for gas-gas waste heat recovery duties when it is possible to use specific air handling units (such as the gas-gas plate, and run-around-coil for example). The heat sink pressure drop is also larger than the other two options: around 3.5 times greater than the gas-gas plate and 18 times greater than the run-around-coil.

However, the shell-and-tube heat exchanger performance is greater than both of the other two options. This is due to higher heat transfer coefficients in the shell and tube unit due to greater levels of turbulence. Hence, the shell-and-tube heat transfer area is around 20% of that required by the run-around-coil, and the overall shell-and-tube size is around half that of the two gas-gas plate units required. Furthermore, the run-around-coil requires a water pump which increases the complexity of maintenance procedures (this is made evident to the user by the flow diagram of the heat exchangers, shown in the case study screenshots in Appendix V). Clearly, each of the three heat exchangers has advantages and disadvantages, and the *final* decision would not be solely based on the technical design results.

The results in Table 6.34 also highlight a drawback of the system, which is lack of design data for certain heat exchangers. "*Off-the-shelf*" type heat exchangers such as run-around-coils and gas-gas plates do not have full design algorithms for use in the KBS. Hence, some data is missing, such as heat exchanger dimensions for the run-around-coil, and overall heat transfer surface area for the gas-gas plate. Further information of the design of such units can be found in Section 4.4.2.

As shown in Table 6.32, organic Rankine cycles are also viable for this case study. Table 6.35 shows the technical design results for an ORC for comparison with the use of a heat exchanger.

Table 6.35. Case study 5: KBS technical ORC design results

Working Fluid	R-245fa
Source Inlet Temperature (^{o} C)	95.0
Heat Source Outlet Temperature (^{o} C)	55.9
Pre-heater + Evaporator Duty (kW)	1104
Heat Sink Inlet Temperature (^{o} C)	10.0
Heat Sink Outlet Temperature (^{o} C)	20.0
Condenser Duty (kW)	1050
Heat Sink Mass Flow Rate (kg/s)	25.0
Working Fluid Mass Flow Rate (kg/s)	5.41
Working Fluid Turbine Inlet Temperature (^{o} C)	45.5
Working Fluid Turbine Inlet Pressure (bar)	3.01
Working Fluid Turbine Outlet Temperature (^{o} C)	30.4
Working Fluid Turbine Outlet Pressure (bar)	1.56
Working Fluid Condensation Temperature (^{o} C)	25.0
Plant Gross Power Output (kW)	51.7
Working Fluid Pump Power (kW)	0.83
Net Power Output (kW)	50.9
Plant Thermal Efficiency (%)	4.61

The results in Table 6.35 show that the organic Rankine cycle is capable of recovering significantly more of the available waste heat than the heat exchanger options, 1104kW as opposed to 167kW. This is due to the fact that the heat source is cooled to a lower temperature in the organic Rankine cycle pre-heater/evaporator, as it is not limited by a specific heat sink target temperature/duty as in the heat exchanger design.

The thermal efficiency of the ORC is 4.61% and only 50.9 kW of useful power (electricity) is generated from the cycle, whereas all of the heat recovered by the heat exchanger options is utilized in heating the sink. However, as electricity is of higher energetic and economical value than low-grade thermal energy, the economic and environmental results must be studied in order to make a final decision on which technology is most suitable for waste heat recovery in this case study.

The economic and environmental results for all options (three heat exchangers and ORC) are shown in Table 6.36.

Table 6.36. Case study 5: KBS economic and environmental results

	Run-around-coil	**Gas-Gas Plate**	**Shell-and-Tube**	**ORC**
Estimate Capital Cost (£GBP)	Not available	Not available	7614	158055
Estimate Maintenance Cost (£GBP/year)	N/A	N/A	N/A	3161
Estimate Cost Savings (GBP/year)	51073	51088	51088	35554
Estimate Payback Period (years)	Not available	Not available	0.15	4.88
Estimate Greenhouse Gas Reductions (tCO_2eq/year)	352.5	352.6	352.6	215

Table 6.36 highlights another limitation of the KBS in that economic data is not available for some types of heat exchanger, particularly the gas-gas plate and run-around-coil in this case study. This is an unavoidable limitation due to lack of published data in this particular area, and the lack of suitable industrial partners on the project. However, the economic data for the shell-and-tube heat exchanger and the organic Rankine cycle can be compared, and greenhouse gas reductions for all of the options.

The associated greenhouse gas reductions for each of the heat exchanger options are approximately equal. Only the run-around-coil option is slightly lower due to the need of a small pump to circulate the water in the coil. Each of the heat exchangers reduce associated emissions by around 352tCO_2eq/year, which is approximately 60% larger than the ORC (215tCO_2eq/year). Hence, from this viewpoint the heat exchanger options would be preferred.

The capital cost, cost savings and payback time of the shell-and-tube option is also greatly preferred to the ORC. The capital cost is approximately 5% of that for the ORC (£7614 as opposed to £158055) while the cost savings are 44% larger (£51088/year as opposed to £35554). Hence, from the economic viewpoint this option is preferred.

Overall, for this case study the heat exchanger waste heat recovery option would be preferred as it shows the best results from both a technical and economic/environmental viewpoint. Firstly, it provides a useful heating duty of 167 kW as opposed to 50.9kW of electricity generated by the ORC. The organic Rankine cycle is also a significantly more complex solution, as shown in the process flow diagram of each technology in the case study screenshots shown in Appendix V.

Secondly, the economic and environmental analyses show that the increased value of electricity compared to low-grade thermal energy does not transfer into greater economic/environmental performance, and the shell-tube-heat exchanger outperforms the ORC in terms of lower capital cost, greater cost savings, lower payback period and greater reduction in associated greenhouse gas emissions.

The case study also highlights some flaws in the KBS performance. Some design and capital cost data is missing for some heat exchanger options (particularly the gas-gas heat exchanger and run-around-coil in this case). Hence, a full comparison of the different heat exchanger options was not possible based on the results of the KBS alone.

6.6. Chapter Conclusions

Testing the KBS has shown that the system has been successful in achieving the aims set out in the scope of the system (Section 1.3.1). In particular, points 5-7 are of interest here, as shown below:

5. Must allow a comparison of various technologies: this will educate users as to the benefits of each type of technology (when appropriate)
6. Must give accurate results: results from this software must be comparable with other modelling tools (to be validated by case studies)
7. Must include a variety of waste heat recovery techniques: this will allow a wide range of possible process conditions to be accommodated

Case studies 1 and 5 in particular have shown that the KBS is capable of allowing a comparison of various technologies thereby educating the user about the benefits of each type of technology (as stated in point 5 above). In both these cases, heat exchangers and organic Rankine cycles were selected as suitable for use. The benefits and drawbacks of each technology were displayed to the user via the technical and economic/environmental results of the KBS. In both cases, heat exchangers were found to be the most appropriate solution, as would be expected by an expert heat recovery consultant.

The results for each category of waste heat recovery technology (heat exchangers, vapour compression heat pumps, mechanical vapour recompression and organic Rankine cycles) have been proven to be accurate and comparable to the use of more conventional methods. Organic Rankine cycle results are shown to be comparable to the use of IPSE Pro (SimTech 2013) in case study 1, heat exchanger results are shown to be comparable to common computational design methods in case study 2, vapour compression heat pump results are shown to be comparable to both the results of an expert contractor (using various modelling techniques) and test data in case study 3, and mechanical vapour recompression results are shown to be comparable to those of Aspen Plus (AspenTech 2013) in case study 4. Hence, the aims of point 6 of the system scope have been achieved.

Point 7 of the system scope has also been achieved. The results show that results were generated for a variety of scenarios across a range of process industry sub-sectors. Furthermore, systems involving various different types of heat source (and sink, where available), such as liquid waste water, dry and humid exhaust gases, and low pressure water vapour have been investigated.

The testing process has also highlighted some of the inherent limitations of the system: full design and economic data is not available for every technology in the knowledge-base, such as the gas-gas plate heat exchanger and the run-around-coil. This flaw is unavoidable as such data is not published. This may be rectified at a later date by collaboration further industrial partners. Furthermore, the accuracy of all of the capital cost estimate methods used by the KBS are difficult to design due to the inherently qualitative nature. This, however, is unavoidable without the use of further industrial/manufacturing collaborators in the project to provide accurate and realistic data.

6.7. Case Study References

The full references of the published case studies referred to in this chapter (case studies 1-3) are as follows:

Table 6.37. Full references of cited case studies

Case Study 1	Aneke, M., Agnew, B., Underwood, C., Wu, H. & Masheiti, S. (2012) Power generation from waste heat in a food processing application. *Applied Thermal Engineering,* 36**,** 171-180. Aneke, M. C. (2012) Optimising Thermal Energy Recovery, Utilisation and Management in the Process Industries, PhD Thesis, Northumbria University, Newcastle, UK
Case Study 2	Pulat, E., Etemoglu, A. B. & Can, M. (2009) Waste-heat recovery potential in Turkish textile industry: Case study for city of Bursa. *Renewable and Sustainable Energy Reviews*, 13(3), 663-672
Case Study 3	Paarske, B. (2011) Heat pumps in industrial washing applications. European Heat Pump Summit 2011. 28-29/09/2011. Nuremburg. Germany

Chapter 7

In this chapter, conclusions are drawn based on the findings of the work and recommendations for future work are presented.

7. Conclusions and Further Work

7.1. Conclusions

This thesis reports the development of a novel knowledge-based system (KBS) for low-grade waste heat recovery equipment selection and (preliminary) design. The aim of the system was to provide a tool capable of encouraging the recovery of low-grade waste heat in the process industries by addressing the following two barriers:

i. Cost of consultancy: KBS provides a free (or very low cost) alternative to outside consultancy during the initial stages of low-grade waste-heat recovery projects.
ii. Awareness of best-available/novel technologies: they are highlighted when suitable (in an educational format).

A literature review of methods for heat recovery selection and design was conducted and a gap in this particular area was identified. For example, pinch methods for heat integration have been highly successful, but do not provide a selection/design of the most suitable waste heat recovery equipment. Furthermore, many modelling tools are available for thermodynamic cycle analysis but they are highly expensive, do not offer advice on when to use each type of cycle, time consuming and (often) require specialist training. Hence, the KBS will make a contribution to knowledge in this area as it is a system that simultaneously selects and designs waste heat recovery systems from a range of options. This is complementary to pinch methods which can highlight potential heat sources and sinks, and it expands on other previously developed methods for heat exchanger selection by including more advanced technologies (such as organic Rankine cycles and heat pumps).

Various types of waste heat recovery technology were analysed for inclusion in the knowledge-base and heat exchangers (various types), vapour compression heat pumps, mechanical vapour compression and organic Rankine cycles were considered the most appropriate according to the scope of the system and hence included. These technologies cover a wide range of scenarios including heat sources of various temperature and phase, availability of *"matching"* heat sinks, availability of a heat sink requiring a temperature lift and lack of heat sink to transfer waste heat.

The system knowledge-base was built according to a "decision tree" type methodology which has the advantage of mapping the knowledge-base with inherent explanation as the decisions are visualised in a flow diagram, which is relevant to the educational aims of the software. Furthermore, this methodology has proven to be successful in the building of expert systems for *similar* engineering problems such as in selection of food drying equipment and selection of shell-and-tube bundle types in TEMA design of shell-and-tube heat exchangers.

The system was programmed using the Java language which has a number of advantages for this application, the primary being that it allows ease of dissemination into the industrial domain due to the "*write once, run anywhere/everywhere"* (WORA/WORE) principle of the language. This allows the software to run on any operating system upon download of the free Java runtime environment.

Testing of the system was achieved via case-studies derived from both published literature and new data from industrial partners. Overall, testing was a success and the results closely matched those produced by published results, common process modelling software and the work of expert consultants. In some cases, the KBS suggested technologies for waste heat recovery that were more appropriate than those considered in the published investigation.

However, testing also highlighted some of the inherent problems in the KBS methodology particularly that the accuracy of cost factor estimates is subjective and hence difficult to quantify. Also, cost factors are missing from some types of heat exchanger as they were unobtainable.

In order to judge the overall success of the system, it is important to revisit its scope and compare the results to what was initially intended. Hence, the scope of the system is listed as follows (originally displayed in Section 1.3.1).

1. Must be simple and intuitive to use: To aid users with no previous experience of process waste heat recovery.

This has been achieved, as is evident in the screen shots of the KBS operation. The system has a simple but functional GUI that requires only text-box and radio-button data entry.

2. Must make use of easy-to-access data: this will aid users with limited time in the collection of data for use in the software.

This has been achieved. Only data that was judged to be easily accessible was included in the data entry forms, and the system makes use of *typical* data in its assumptions to replace more detailed data required for the design process. Examples of this include film heat transfer coefficient assumptions (which are accurate enough for a preliminary design), utility greenhouse gas emission factors and built-in calculations of dew points for humid gas streams.

3. Must explain selection/design logic to the user: this will educate the user in the methods employed by the system thereby reducing/avoiding user confusion or mistrust.

This has been achieved as the system gives reasons for why equipment is not chosen for use. This is highlighted by the case study testing.

4. Must allow easy dissemination into the industrial domain: different users are likely to run various operating systems (Apple OS, Linux, Windows etc) meaning the software must be multi-platform compatible.

This was achieved by writing the system in the Java programming language which can run without further compilation or cost on all common operating systems.

5. Must allow a comparison of various technologies: this will educate users as to the benefits of each type of technology (when appropriate).

This was achieved as highlighted by case study testing. For example, case study 2 and case study 5 show that an organic Rankine cycle and heat exchanger were possible for waste heat recovery. Here, preliminary designs of both systems are provided, allowing the user to view the benefits and drawbacks of each design.

6. Must give accurate results: results from this software must be comparable with other modelling tools (to be validated by case studies).

This was demonstrated via case study testing. The results of the knowledge-based system were shown to be comparable with those of published results, common process modelling software and the results of experienced waste heat recovery consultants.

The KBS has also exhibited some advantages when compared to common process modelling tools and the work of consultants. Firstly, the time taken to input data and run the KBS is of the order of minutes, whereas this would be of the order of hours (or days) when using other methods. Secondly, in some case studies, the KBS found solutions that were better than those suggested by published results and that had not been previously considered by the authors.

7. Must include a variety of waste heat recovery techniques: this will allow a wide range of possible process conditions to be accommodated.

This was achieved, as four key types of heat recovery equipment were included, covering a wide range of possible scenarios. However, there is scope to expand the equipment database and include further technologies such as absorption heat pumps (for cooling) and other novel technologies as they develop.

8. Must include technologically viable results: results must be meaningful on an industrial scale. Technologies requiring significant further R&D should not be included

This has been achieved as each of the included technologies were rigorously analysed and selected according to this constraint.

9. Must include economically viable results: only technologies which have been proven to achieve economically viable results will be considered. Technologies incurring typical pay back times of greater than 5 years (under

economic conditions at the time of writing) will be considered non-economical

This has been achieved as this was, again, was a constraint in selecting the chosen technologies. Furthermore, each of the designed technologies in case study testing shown anticipated payback times of less than 5 years.

Overall, the system has achieved what was set out in the scope of the system and can therefore be deemed a success. Case study testing has validated the system as a viable alternative to expert consultancy and existing modelling software as, broadly, in terms of equipment specification and cost, the KBS produced the same results as the other methods. Furthermore, the preliminary designs generated were generally within 5% of those from the other sources.

In certain cases, the KBS suggested alternative technologies that were more viable (economically and/or practically) than those considered by the authors of published case studies, hence validating the educational aims of the software. In all cases, system operating time (data input, and processing of results) was of the order of minutes, whereas studies by consultants or the use of existing modelling packages would be significantly more time-consuming (of the order of hours or days). This highlights the ability of the system to be used as a rapid optioneering tool for investigation of waste heat recovery technologies.

However, various improvements could be made the system as considered in Section 7.2.

7.2. Recommendations for Future Work

Future work should focus on improving the system and addressing the drawbacks highlighted by testing. The main improvement required is to the method for obtaining capital cost estimates as the accuracy of cost factor methods employed is highly subjective, particularly considering that various sources were used for each of the technologies. Also, the missing capital cost data for some types of equipment must be found.

In order to improve this, significant input will be required from current manufacturers of these technologies. However, this is a difficult task given the wide range of technologies featured in the system knowledge-base, but it would considerably improve the validity of the economic results of the system, which in turn would considerably improve the system overall.

Other suggested improvements include expansion of the equipment database of the system to include *all* available (on an industrial scale) options. In this work, only the *best available* (according to constraints in the system scope) were considered for selection due to time constraints of the project. However, future iterations of the system should include other technologies such as absorption heat pumps, thermal vapour recompression and *specialist* heat exchangers (scraped surface, for example), all of which can provide useful solutions in certain circumstances. A regular review of emerging equipment should also be considered with a view to expanding the equipment database as future technologies develop on the industrial scale.

7.2.1. Commercialisation of the Knowledge-Based System

Commercialisation of the software is key to its overall success. Funding will be sought to build an online infrastructure with which to easily disseminate the software. This will allow further testing of the system, which will allow iterative improvements of the software.

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Appendices

Appendix I

The following is an example of some of the "data check" rules employed by KBS in order to ensure that no inconsistency exists in the user data input. This particular example refers to the "data check" rules for the data input to the liquid-liquid heat exchanger module and is only a small example of the total number of rules in the code,

IF (any data box is empty)

DATA CHECK RESULT = NEGATIVE. DISPLAY: "At least one text input field is empty"

IF (sourceTin > 260)

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Low-grade waste heat recovery only. Source temperature exceeds upper limit, 260^{o}C"

IF (sourceTtarget ≥ sourceTin)

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Source target temperature cannot exceed inlet temperature"

IF ((sourcemflow ≤ 0) OR (sinkmflow ≤ 0))

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Mass flow rates must be greater than 0"

IF ((sourcePin ≤ 0) OR (sinkPin ≤0))

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Source & sink pressure (absolute pressure) must be greater than 0"

IF (sinkTtagrget ≤ sinkTin)

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Sink target temperature must be greater than inlet temperature"

IF (planthours ≤ 0)

DATA CHECK RESULT = NEGATIVE. DISPLAY: "Error: Plant hours of operation must be greater than 0"

IF (all conditions ≠NEGATIVE)

DATA CHECK RESULT = POSITIVE. DISPLAY: "Data input OK. Proceed"

Note: sourceTin denotes the source inlet temperature, sourceTtarget denotes the source target temperature, sourcemflow denotes the source mass flowrate, sinkmflow denotes the sink mass flowrate, sourcePin denotes the source inlet pressure, sinkPin denotes the sink inlet pressure, sinkTin denotes the sink inlet temperature, sinkTtarget denotes the sink target temperature, planthours denotes the plants hours of operation per year.

A screenshot showing an example of a negative data check result is shown in Figure A1.1, and a screenshot showing an example of a positive data check result is shown in Figure A1.2.

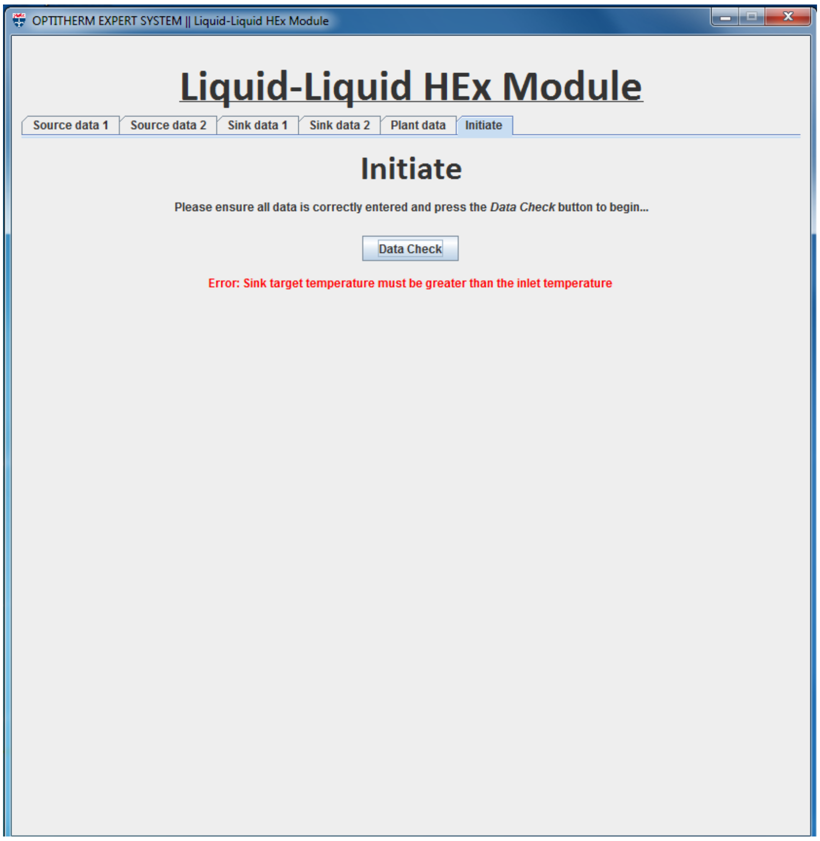


Figure A1.1. Example of a negative data check result

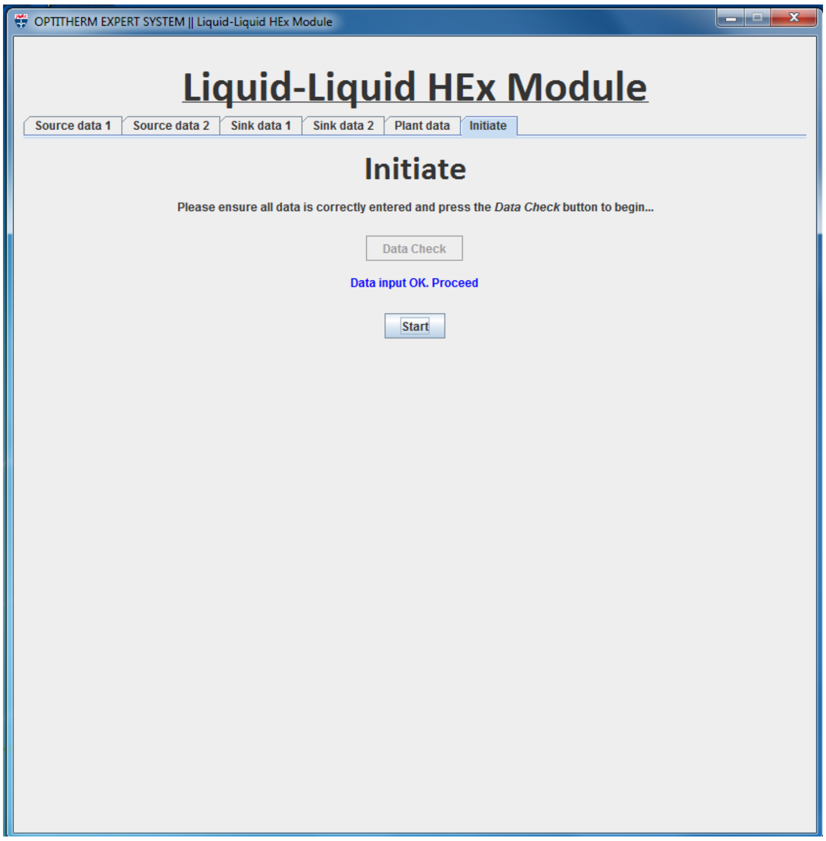


Figure A1.2. Example of a positive data check result

Appendix II

The following is an example of the data acquired from Alfa Laval (2013) regarding plate sizes for plate and frame heat exchangers.

M6

Plate Heat Exchanger

Applications
General heating and cooling duties. Heating by means of steam.

Standard design
The plate heat exchanger consists of a pack of corrugated metal plates with portholes for the passage of the two fluids between which heat transfer will take place.

The plate pack is assembled between a fix frame plate and a movable pressure plate and compressed by tightening bolts. The plates are fitted with a gasket which seals the interplate channel and directs the fluids into alternate channels. The number of plates is determined by the flow rate, physical properties of the fluids, pressure drop and temperature program. The plate corrugations promote fluid turbulence and support the plates against differential pressure.

The frame plate and the pressure plate are suspended from an upper carrying bar and located by a lower guiding bar, both of which are fixed to a support column.

Connections are located in the frame plate or, if either or both fluids make more than a single pass within the unit, in the frame and pressure plates.

Typical capacities
Liquid flow rate
Up to 16 kg/s (250 gpm), depending on media, permitted pressue drop and temperature program.

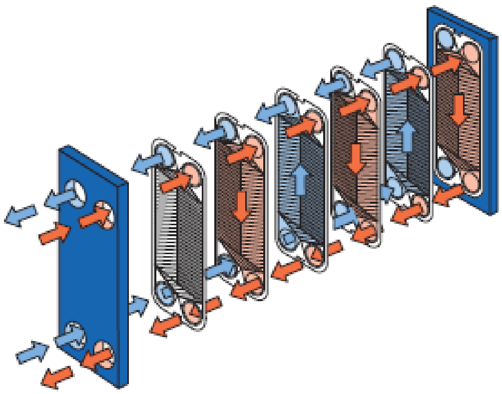
Water heating by steam
300 to 800 kW

Plate types
M6, M6-M and M6-MD

Frame types
FM, FG and FD

Working principle
Channels are formed between the plates and the corner ports are arranged so that the two media flow through alternate channels. The heat is transferred through the plate between the channels, and complete counter-current flow is created for highest possible efficiency. The corrugation of the plates provides the passage between the plates, supports each plate against the adjacent one and enhances the turbulence, resulting in efficient heat transfer.

M6-FG



Flow principle of a plate heat exchanger

Figure A2.1a. Example data from Alfa Laval (2013): Gasketted plate heat exchanger (part 1)

STANDARD MATERIALS

Frame plate
Mild steel, Epoxy painted

Nozzles
Carbon steel
Metal lined: Stainless steel, Titanium, Alloy 254 SMO, Alloy C276
Rubber lined: Nitrile, EPDM

Plates
Stainless steel: Alloy 316, Alloy 304. Alloy 254 SMO, Alloy C276, Titanium

Gaskets
Nitrile, EPDM, Viton®
Other grades and material available on request.

TECHNICAL DATA

Pressure vessel codes, PED, ASME, pvcALS™
Mechanical design pressure (g) / temperature

FM	pvcALS™	1.0 MPa / 180°C
FG	PED	1.6 MPa / 180°C
FG	ASME	162 psig / 482°F
FG	pvcALS™	1.6 MPa / 180°C
FD	PED, pvcALS™	2.5 MPa / 180°C
FD	ASME	351 psig / 482°F

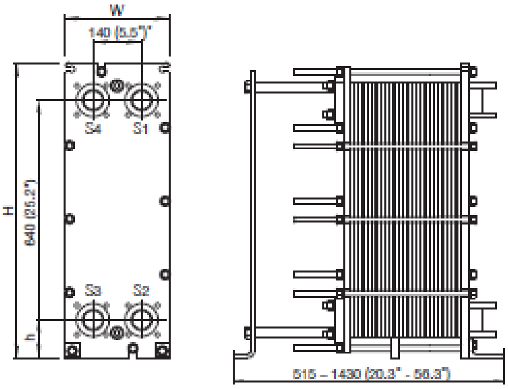
Connections
Pipe connections (not for frame type FD)

	Size:	
Straight threaded	50 mm	ISO G2"
Tapered threaded	50 mm	ISO R2", NPT2"
Straight weld	50 mm	
Threaded inlet port	50 mm	ISO G2"

Flange connections

		Size:	
FM	pvcALS™	50 mm	DIN/GB/GOST PN10, ASME Cl. 150, JIS 10K
FG	PED	50 mm	DIN PN16, ASME Cl. 150
FG	ASME	2"	ASME Cl. 150
FG	pvcALS™	50 mm	DIN/GB/GOST PN16, ASME Cl. 150, JIS 16K
FD	PED	50 mm	DIN PN25, ASME Cl. 300
FD	ASME	2"	ASME Cl. 300
FD	ALS	50 mm	DIN, GB, GOST PN25, JIS 20K

Dimensions



* Displacement of some connection types occur.

Measurements mm (inch)

Type	H	W	h
M6-FM	920 (36.2")	320 (12.6")	140 (5.5")
M6-FG	920 (36.2")	320 (12.6")	140 (5.5")
M6-FD	940 (37.0")	330 (13.0")	150 (5.9")

The number of tightening bolts may vary depending on pressure rating.

Maximum heat transfer surface
38 m² (400 sq. ft)

Particulars required for quotation
- Flow rates or heat load
- Temperature program
- Physical properties of liquids in question (if not water)
- Desired working pressure
- Maximum permitted pressure drop
- Available steam pressure

PCT00115EN 1203 Alfa Laval reserves the right to change specifications without prior notification.

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Figure A2.1b. Example data from Alfa Laval (2013): Gasketted plate heat exchanger (part 2)

The following is an example of the *typical* overall heat transfer data provided by ESDU via the Best Practice Programme (2000). This example is for a welded plate heat exchanger.

Table 3.6.9 – U and C Values for Welded Plate Heat Exchangers (Courtesy of ESDU)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot-side Fluid								
			Low Pressure Gas (<1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	Process Water	Low Viscosity Organic Liquid	High Viscosity Liquid	Condensing Steam	Condensing Hydrocarbon	Condensing Hydrocarbon with Inert Gas
30000	Low Pressure Gas (<1 bar)	U (W/m²K)	83	133	NA	238	176	134	246	191	170
		C (£/(W/K))	2.33	1.8		1.67	1.96	1.95	0.934	1.01	1.96
	Medium Pressure Gas (20 bar)	U (W/m²K)	129	503	NA	1262	819	774	1384	850	440
		C (£/(W/K))	2.1	0.733		0.43	0.54	0.61	0.494	0.634	0.52
	High Pressure Gas (150 bar)	U (W/m²K)	NA	NA	NA	NA	NA	NA	NA	NA	NA
		C (£/(W/K))									
	Treated Cooling Water	U (W/m²K)	210	1068	NA	9100	4268	2067	5465	1518	600
		C (£/(W/K))	1.53	0.42		0.147	0.26	0.26	0.18	0.274	0.55
	Low Viscosity Organic Liquid	U (W/m²K)	141	1060	NA	3570	3763	1075	3652	1366	600
		C (£/(W/K))	1.82	0.37		0.21	0.194	0.367	0.214	0.297	0.55
	High Viscosity Liquid	U (W/m²K)	131	298	NA	484	413	292	1626	403	300
		C (£/(W/K))	1.9	0.87		0.52	0.584	0.824	0.287	0.54	0.87
	Boiling Water	U (W/m²K)	220	1100	NA	9000	4500	2000	5500	1500	600
		C (£/(W/K))	1.51	0.42		0.15	0.25	0.26	0.18	0.28	0.55
	Boiling Organic Liquid	U (W/m²K)	160	900	NA	5000	3500	1200	4000	1000	500
		C (£/(W/K))	1.65	0.39		0.19	0.20	0.37	0.26	0.37	0.53
100000	Low Pressure Gas (<1 bar)	U (W/m²K)	65	141	NA	230	167	141	246	190	170
		C (£/(W/K))	3.84	1.33		1.42	1.74	2.0	1.06	0.92	1.75
	Medium Pressure Gas (2O bar)	U (W/m²K)	137	534	NA	1354	940	776	1342	834	440
		C (£/(W/K))	1.38	0.462		0.303	0.34	0.41	0.238	0.307	0.51
	High Pressure Gas (150 bar)	U (W/m²K)	NA	NA	NA	NA	NA	NA	NA	NA	NA
		C (£/(W/K))									
	Treated Cooling Water	U (W/m²K)	223	1112	NA	9420	4858	2106	5071	1518	600
		C (£/(W/K))	1.37	0.28		0.062	0.108	0.13	0.085	0.186	0.40
	Low Viscosity Organic Liquid	U (W/m²K)	157	722	NA	4140	3794	1277	3712	1367	600
		C (£/(W/K))	1.71	0.36		0.118	0.101	0.19	0.108	0.161	0.40
	High Viscosity Liquid	U (W/m²K)	133	355	NA	530	415	302	1617	452	300
		C (£/(W/K))	1.67	0.592		0.42	0.538	0.515	0.158	0.389	0.52
	Boiling Water	U (W/m²K)	220	1100	NA	9000	4500	2000	5500	1500	600
		C (£/(W/K))	1.37	0.28		0.063	0.13	0.15	0.09	0.17	0.40
	Boiling Organic Liquid	U (W/m²K)	160	900	NA	5000	3500	1200	4000	1000	500
		C (£/(W/K))	1.71	0.32		0.12	0.14	0.19	0.13	0.30	0.43

Figure A2.1b. Example data from ESDU (via Best Practice Programme, 2000) regarding *typical* overall heat transfer coefficients of various types of heat exchangers

Appendix III

An example of the thermophysical working fluid data by ASHRAE used in the KBS knowledge-base is shown below in Figure A2.1. This example is for the R-600 working fluid.

Thermophysical Properties of Refrigerants **30.47**

Refrigerant 600 (*n*-Butane) Properties of Saturated Liquid and Saturated Vapor

Temp.,[a] °C	Pressure, MPa	Density, kg/m³	Volume, m³/kg	Enthalpy, kJ/kg		Entropy, kJ/(kg·K)		Specific Heat c_p, kJ/(kg·K)		c_p/c_v	Velocity of Sound, m/s		Viscosity, µPa·s		Thermal Cond., mW/(m·K)		Surface Tension, mN/m	Temp., °C
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor		
-100	0.00016	699.3	150.44	-13.65	450.85	0.0318	2.7144	2.013	1.231	1.132	1592	167.4	792.2	4.30	161.6	6.83	28.03	-100
-95	0.00028	694.6	91.606	-3.57	457.02	0.0892	2.6746	2.021	1.247	1.130	1563	169.6	719.0	4.43	159.4	7.12	27.33	-95
-90	0.00045	689.9	57.588	6.56	463.27	0.1453	2.6389	2.029	1.263	1.128	1534	171.8	655.9	4.55	157.2	7.43	26.64	-90
-85	0.00072	685.2	37.271	16.73	469.58	0.2000	2.6069	2.038	1.279	1.127	1505	174.0	601.2	4.68	154.9	7.73	25.94	-85
-80	0.00111	680.5	24.773	26.94	475.97	0.2536	2.5784	2.048	1.295	1.125	1477	176.1	553.2	4.81	152.6	8.05	25.26	-80
-75	0.00168	675.8	16.873	37.21	482.41	0.3061	2.5529	2.058	1.312	1.124	1449	178.1	510.9	4.93	150.2	8.38	24.57	-75
-70	0.00247	671.0	11.752	47.53	488.92	0.3575	2.5303	2.069	1.330	1.122	1420	180.1	473.4	5.06	147.9	8.71	23.89	-70
-65	0.00355	666.2	8.3565	57.90	495.50	0.4080	2.5103	2.081	1.347	1.121	1392	182.1	439.9	5.18	145.5	9.05	23.22	-65
-60	0.00501	661.4	6.0558	68.34	502.13	0.4575	2.4926	2.094	1.366	1.120	1364	184.0	409.8	5.31	143.1	9.40	22.54	-60
-55	0.00695	656.6	4.4659	78.85	508.82	0.5062	2.4772	2.108	1.385	1.119	1336	185.8	382.8	5.43	140.8	9.75	21.88	-55
-50	0.00947	651.7	3.3470	89.42	515.56	0.5541	2.4638	2.122	1.404	1.118	1309	187.5	358.3	5.55	138.4	10.11	21.21	-50
-45	0.01270	646.8	2.5462	100.07	522.35	0.6013	2.4522	2.137	1.425	1.118	1281	189.2	336.1	5.68	136.0	10.48	20.55	-45
-40	0.01679	641.9	1.9638	110.80	529.19	0.6478	2.4423	2.153	1.446	1.117	1253	190.8	315.8	5.80	133.6	10.86	19.90	-40
-35	0.02190	636.9	1.5341	121.62	536.08	0.6937	2.4340	2.170	1.468	1.117	1226	192.3	297.3	5.92	131.3	11.25	19.25	-35
-30	0.02821	631.9	1.2127	132.52	543.01	0.7389	2.4271	2.188	1.490	1.116	1198	193.8	280.3	6.04	128.9	11.64	18.60	-30
-25	0.03591	626.8	0.96911	143.51	549.98	0.7836	2.4216	2.206	1.514	1.116	1171	195.1	264.7	6.16	126.6	12.05	17.96	-25
-20	0.04521	621.7	0.78237	154.60	556.98	0.8278	2.4173	2.226	1.538	1.116	1144	196.3	250.3	6.28	124.3	12.46	17.32	-20
-15	0.05635	616.6	0.63759	165.79	564.02	0.8715	2.4141	2.246	1.563	1.117	1116	197.5	237.0	6.41	122.0	12.88	16.69	-15
-10	0.06955	611.4	0.52415	177.08	571.08	0.9147	2.4120	2.267	1.589	1.117	1089	198.5	224.7	6.53	119.8	13.30	16.06	-10
-5	0.08509	606.1	0.43441	188.48	578.17	0.9576	2.4108	2.289	1.616	1.118	1062	199.4	213.2	6.65	117.5	13.74	15.44	-5
-0.49[b]	0.10132	601.3	0.36910	198.87	584.58	0.9959	2.4105	2.310	1.641	1.119	1038	200.1	203.5	6.76	115.5	14.14	14.88	-0.49
0	0.10323	600.7	0.36275	200.00	585.27	1.0000	2.4105	2.312	1.644	1.119	1035	200.2	202.5	6.77	115.3	14.19	14.82	0
2	0.11127	598.6	0.33818	204.64	588.12	1.0169	2.4106	2.321	1.655	1.120	1024	200.5	198.4	6.82	114.4	14.37	14.58	2
4	0.11980	596.4	0.31562	209.30	590.97	1.0337	2.4108	2.331	1.667	1.120	1014	200.7	194.4	6.87	113.6	14.55	14.33	4
6	0.12882	594.2	0.29488	213.98	593.82	1.0505	2.4112	2.341	1.678	1.121	1003	201.0	190.6	6.91	112.7	14.74	14.09	6
8	0.13837	592.0	0.27578	218.68	596.67	1.0672	2.4116	2.350	1.690	1.122	992	201.2	186.8	6.96	111.8	14.93	13.85	8
10	0.14845	589.8	0.25817	223.40	599.53	1.0838	2.4122	2.360	1.702	1.122	981	201.4	183.2	7.01	111.0	15.11	13.60	10
12	0.15909	587.6	0.24192	228.13	602.38	1.1005	2.4129	2.371	1.715	1.123	971	201.6	179.6	7.06	110.1	15.31	13.36	12
14	0.17031	585.4	0.22691	232.89	605.24	1.1170	2.4137	2.381	1.727	1.124	960	201.7	176.1	7.11	109.3	15.50	13.12	14
16	0.18213	583.1	0.21302	237.68	608.09	1.1335	2.4146	2.391	1.740	1.125	949	201.9	172.7	7.16	108.4	15.69	12.88	16
18	0.19457	580.9	0.20016	242.48	610.95	1.1500	2.4156	2.402	1.752	1.126	938	202.0	169.4	7.21	107.6	15.89	12.65	18
20	0.20765	578.6	0.18823	247.30	613.80	1.1665	2.4167	2.413	1.765	1.127	928	202.0	166.2	7.26	106.7	16.09	12.41	20
22	0.22139	576.3	0.17717	252.15	616.66	1.1829	2.4179	2.424	1.778	1.128	917	202.1	163.0	7.31	105.9	16.29	12.17	22
24	0.23582	574.0	0.16688	257.02	619.51	1.1992	2.4191	2.435	1.792	1.129	906	202.1	159.9	7.36	105.1	16.49	11.94	24
26	0.25095	571.7	0.15732	261.91	622.36	1.2155	2.4205	2.446	1.805	1.130	895	202.2	156.9	7.41	104.3	16.70	11.70	26
28	0.26680	569.3	0.14842	266.82	625.21	1.2318	2.4219	2.458	1.819	1.131	885	202.1	153.9	7.47	103.5	16.90	11.47	28
30	0.28341	567.0	0.14012	271.76	628.06	1.2481	2.4234	2.470	1.833	1.133	874	202.1	151.1	7.52	102.7	17.11	11.24	30
32	0.30079	564.6	0.13238	276.72	630.91	1.2643	2.4250	2.481	1.847	1.134	863	202.0	148.2	7.57	101.9	17.33	11.00	32
34	0.31897	562.2	0.12516	281.71	633.75	1.2805	2.4266	2.494	1.862	1.136	853	201.9	145.5	7.62	101.1	17.54	10.77	34
36	0.33796	559.8	0.11841	286.72	636.59	1.2966	2.4283	2.506	1.876	1.137	842	201.8	142.8	7.68	100.3	17.76	10.54	36
38	0.35779	557.4	0.11209	291.76	639.42	1.3127	2.4301	2.518	1.891	1.139	831	201.7	140.2	7.73	99.5	17.98	10.32	38
40	0.37849	554.9	0.10618	296.82	642.25	1.3288	2.4319	2.531	1.906	1.141	820	201.5	137.6	7.79	98.7	18.21	10.09	40
42	0.40007	552.4	0.10065	301.90	645.08	1.3449	2.4338	2.544	1.922	1.143	810	201.3	135.0	7.84	97.9	18.43	9.86	42
44	0.42256	550.0	0.09545	307.02	647.90	1.3609	2.4358	2.557	1.937	1.145	799	201.1	132.5	7.90	97.2	18.66	9.64	44
46	0.44599	547.4	0.09058	312.15	650.71	1.3769	2.4378	2.571	1.953	1.147	788	200.8	130.1	7.95	96.4	18.90	9.41	46
48	0.47038	544.9	0.08600	317.32	653.52	1.3929	2.4398	2.585	1.970	1.149	777	200.5	127.7	8.01	95.7	19.14	9.19	48
50	0.49575	542.3	0.08170	322.51	656.32	1.4089	2.4419	2.598	1.986	1.151	767	200.2	125.4	8.07	94.9	19.38	8.97	50
55	0.56365	535.8	0.07201	335.62	663.28	1.4488	2.4473	2.635	2.029	1.158	740	199.2	119.7	8.22	93.1	20.00	8.42	55
60	0.63824	529.1	0.06366	348.91	670.19	1.4885	2.4529	2.673	2.075	1.165	713	198.1	114.3	8.38	91.3	20.64	7.87	60
65	0.71991	522.3	0.05642	362.39	677.02	1.5282	2.4587	2.713	2.123	1.173	685	196.7	109.1	8.54	89.5	21.32	7.34	65
70	0.80908	515.2	0.05012	376.06	683.77	1.5679	2.4646	2.756	2.174	1.183	658	195.1	104.1	8.71	87.8	22.03	6.81	70
75	0.90616	507.9	0.04462	389.95	690.41	1.6075	2.4705	2.802	2.229	1.194	631	193.3	99.3	8.89	86.1	22.77	6.29	75
80	1.0116	500.4	0.03978	404.06	696.94	1.6471	2.4765	2.851	2.288	1.207	603	191.2	94.7	9.08	84.5	23.56	5.78	80
85	1.1258	492.6	0.03552	418.40	703.32	1.6868	2.4824	2.905	2.353	1.223	575	188.8	90.2	9.29	82.9	24.39	5.28	85
90	1.2493	484.5	0.03175	433.00	709.53	1.7266	2.4881	2.964	2.425	1.241	546	186.1	85.8	9.51	81.3	25.28	4.79	90
95	1.3825	476.0	0.02840	447.87	715.53	1.7665	2.4936	3.029	2.506	1.263	518	183.1	81.6	9.75	79.7	26.23	4.31	95
100	1.5259	467.1	0.02541	463.03	721.29	1.8066	2.4987	3.102	2.599	1.290	488	179.8	77.4	10.01	78.2	27.26	3.84	100
105	1.6801	457.8	0.02273	478.51	726.75	1.8469	2.5034	3.186	2.708	1.324	458	176.1	73.3	10.29	76.8	28.37	3.38	105
110	1.8456	447.9	0.02032	494.36	731.87	1.8876	2.5075	3.285	2.841	1.366	428	172.0	69.3	10.61	75.3	29.60	2.93	110
115	2.0230	437.3	0.01813	510.61	736.55	1.9287	2.5108	3.403	3.004	1.420	396	167.5	65.4	10.96	73.9	30.96	2.50	115
120	2.2131	425.9	0.01615	527.34	740.69	1.9704	2.5131	3.552	3.213	1.492	364	162.5	61.4	11.37	72.5	32.50	2.08	120
125	2.4166	413.4	0.01432	544.65	744.15	2.0129	2.5140	3.748	3.493	1.592	331	157.0	57.4	11.83	71.1	34.29	1.68	125
130	2.6344	399.6	0.01264	562.68	746.70	2.0566	2.5130	4.023	3.891	1.739	296	150.8	53.4	12.39	69.8	36.44	1.30	130
135	2.8675	383.7	0.01105	581.69	747.97	2.1020	2.5094	4.450	4.511	1.973	260	144.1	49.2	13.07	68.4	39.16	0.94	135
140	3.1172	364.7	0.00953	602.17	747.29	2.1502	2.5015	5.227	5.631	2.405	222	136.5	44.7	13.97	67.3	42.94	0.60	140
145	3.3853	339.9	0.00801	625.32	743.11	2.2041	2.4858	7.147	8.349	3.462	182	128.0	39.6	15.27	66.7	49.21	0.31	145
150	3.6746	297.7	0.00621	656.27	729.01	2.2755	2.4474	19.80	25.55	10.14	136	117.8	32.5	17.87	73.0	67.32	0.06	150
151.98[c]	3.7960	228.0	0.00439	693.91	693.91	2.3631	2.3631	∞	∞	∞	0	0.0	—	—	∞	∞	0.00	151.98

[a]Temperatures on ITS-90 scale [b]Normal boiling point [c]Critical point

Figure A2.1. ASHRAE thermophysical data for R-600. (ASHRAE, 2009)

Appendix IV

This Appendix displays an example of Java code used to write the software. This short section of code is the routine to calculate the heat balance in heat exchanger design which is shown in Figures 4.14 and 4.15, Section 4.4.2, where only sensible heat is transferred. The two figures are revisited here as follows:

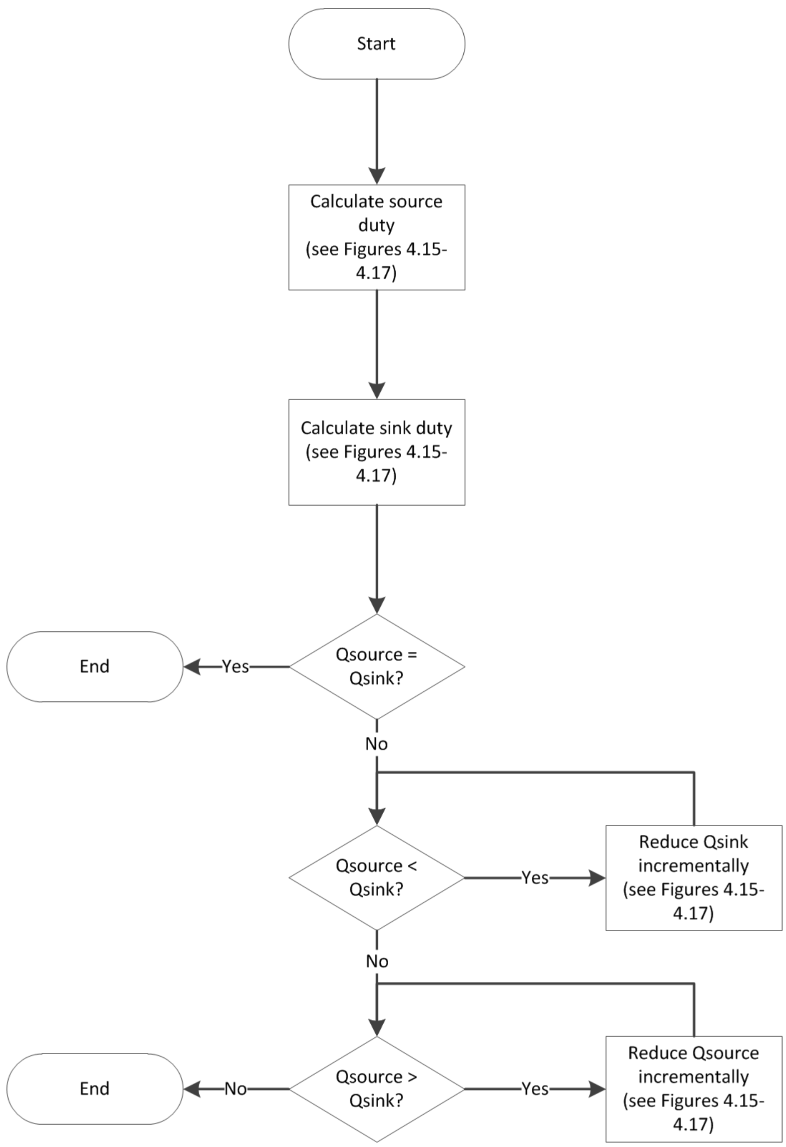
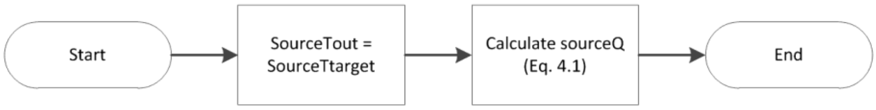
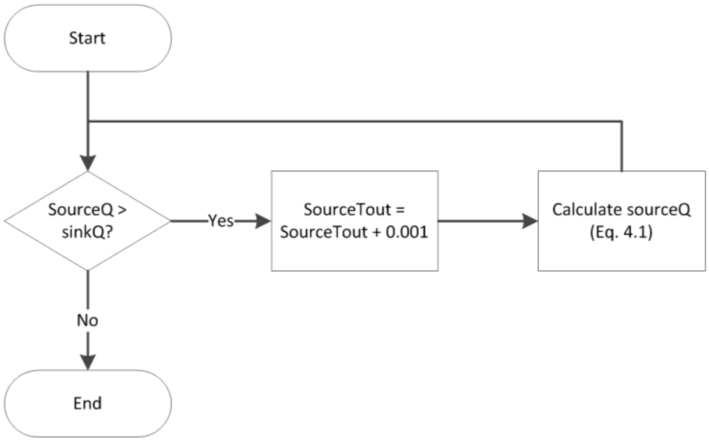


Figure A4.1. Routine to calculate the heat balance in heat exchanger design (previously Figure 4.14)

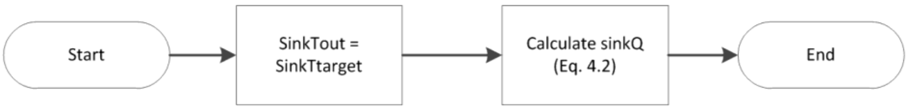
Heat source duty calculation:



Heat source incremental duty decrease routine:



Heat sink duty calculation:



Heat sink incremental duty decrease routine:

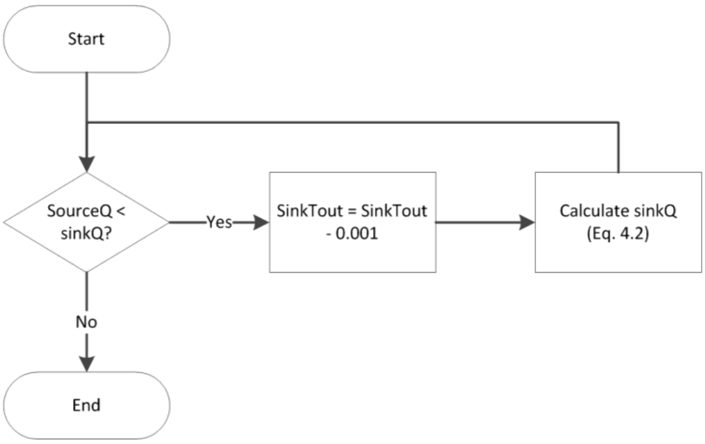


Figure A2.2. Routing to calculate heating duties and reduce duty incrementally when no phase change occurs

The corresponding Java code is as follows (note: comments not relating to the operation of the routine are displayed in green, proceeded by "//"):

START OF CODE

```
double sourcemflow, sourceCp, sourceTin, sourceTtarget; //defining source variables that will be provided by user data

String a = SOURCEMFLOW.getText(); //instructing the program to get the data from the text box entitled "SOURECMFLOW", the source mass flow rate
String b = SOURCECP.getText(); //instructing the program to get the data from the text box entitled "SOURCECP", the source specific heat capacity
String c = SOURCETIN.getText(); //instructing the program to get the data from the text box entitled "SOURCETIN", the source temperature
String d = SOURCETTARGET.getText(); //instructing the program to get the data from the text box entitle "SOURCETTARGET", the source target temperature

sourcemflow = Double.parseDouble(a); //setting the value to that of the string above
sourceCp = Double.parseDouble(b); //setting the value to that of the string above
source Tin = Double.parseDouble(c); //setting the value to that of the string above
sourceTtarget = Double.parseDouble(d); //setting the value to that of the string above

double sinkmflow, sinkCp, sinkTin, sinkTtarget; //defining sink variables that will be provided by user data

String e = SINKMFLOW.getText(); //instructing the program to get the data from the text box entitled "SINKMFLOW", the sink mass flow rate
String f = SINKCP.getText(); //instructing the program to get the data from the text box entitled "SINKCP", the sink specific heat capacity
String g = SINKTIN.getText(); //instructing the program to get the data from the text box entitled "SINKTIN", the sink temperature
String h = SINKTTARGET.getText(); //instructing the program to get the data from the text box entitle "SINKTTARGET", the sink target temperature

sourcemflow = Double.parseDouble(e); //setting the value to that of the string above
sourceCp = Double.parseDouble(f); //setting the value to that of the string above
source Tin = Double.parseDouble(g); //setting the value to that of the string above
sourceTtarget = Double.parseDouble(h); //setting the value to that of the string above

double sourceQ, sinkQ; //defining the source and sink duty variables
double sourceTout, sinkTout; //defining the source and sink outlet temperature variables
```

```
sourceTout = sourceTtarget; //setting the initial value of the source outlet temperature as equal to the target temperature
sinkTout = sinkTtarget; //setting the initial value of the sink outlet temperature as equal to the target temperature

sourceQ = sourceCp * sourcemflow * (sourceTin - sourceTout); //calculating the source duty

sinkQ = sinkCp * sinkmflow * (sinkTout - sinkTin); //calculating the sink duty

while (sourceQ < sinkQ)
	{sinkTout = sinkTout - 0.001;
	sinkQ = sinkCp * sinkmflow * (sinkTout - sinkTin);} //loop to reduce sink duty incrementally

while (sinkQ < sourceQ)
	{sourceTout = sourceTout + 0.001;
	sourceQ = sourceCp * sourcemflow * (sourceTin - sourceTout);} //loop to reduce sink duty incrementally

//At this stage: the heat balance is equal and realistic stream temperatures are defined
```

<u>END OF CODE</u>

Appendix V

Screenshots from case study 2 (Section 6.2):

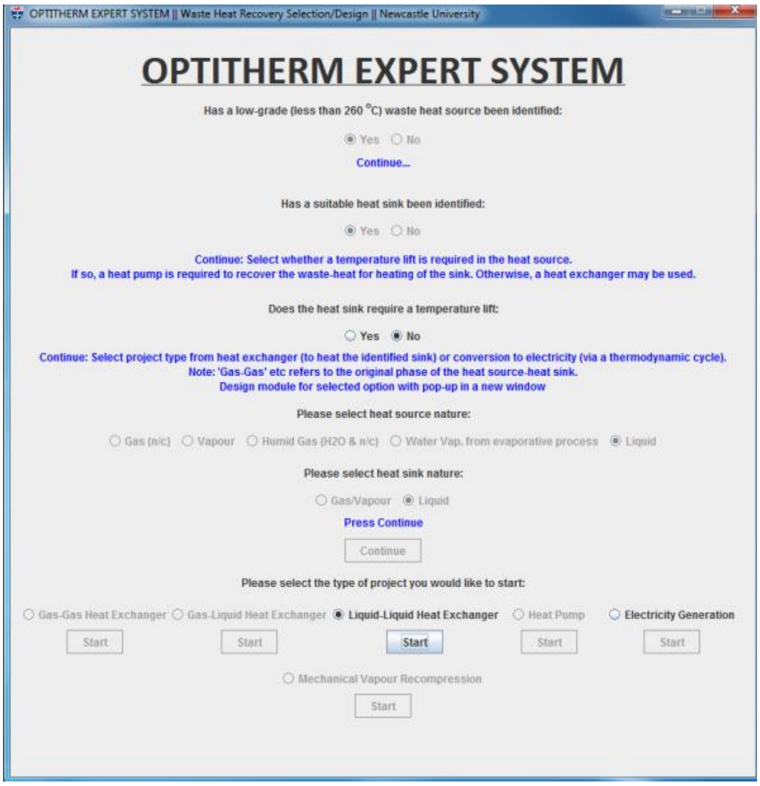


Figure A5.1. Initial user questions for case study

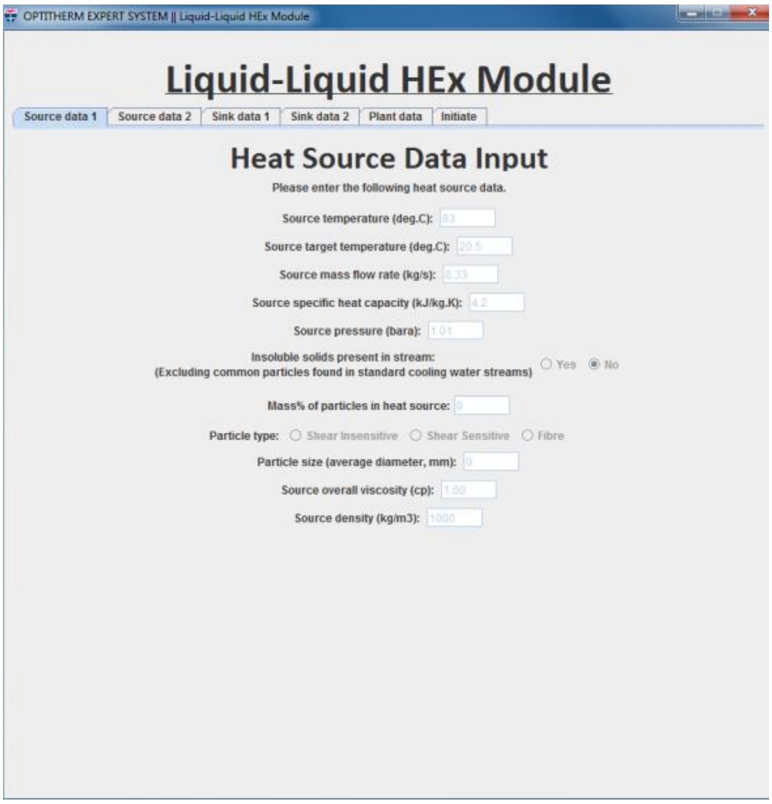


Figure A5.2. Data input for case study (heat exchanger module)

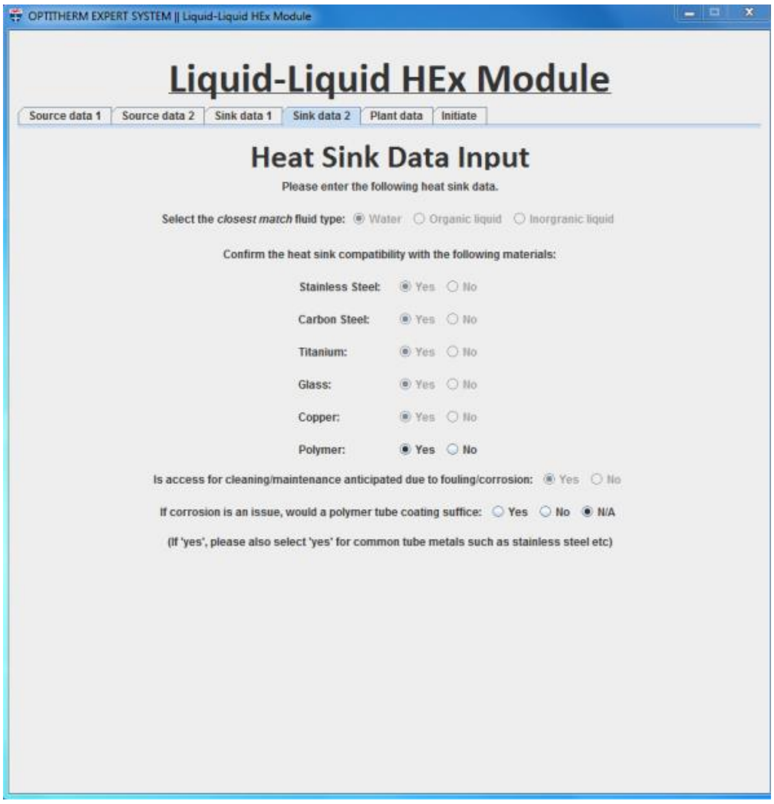


Figure A5.3. Further data input for case study (heat exchanger module)

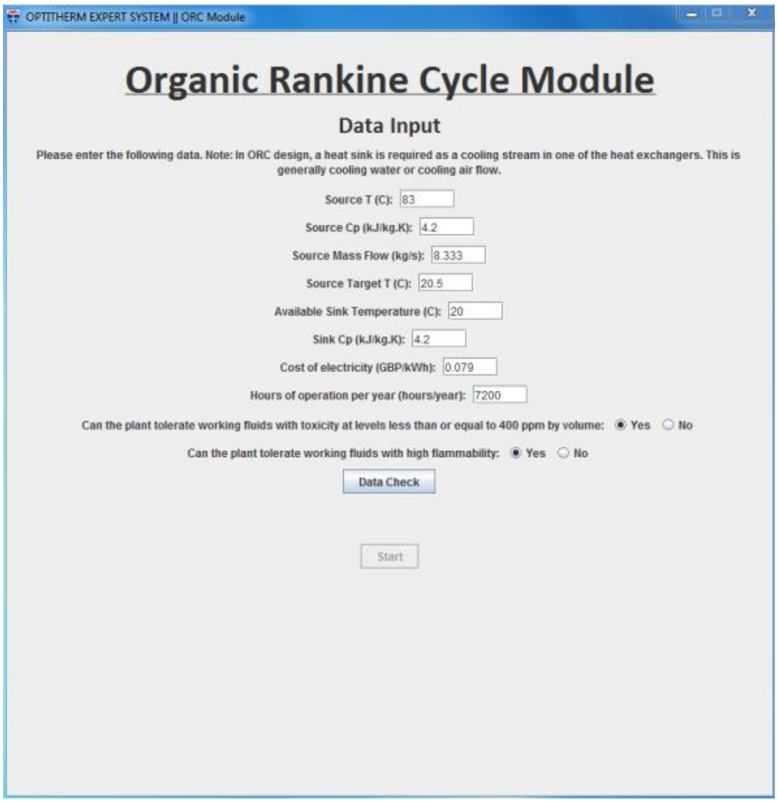


Figure A5.4. Further data input for case study (ORC module)

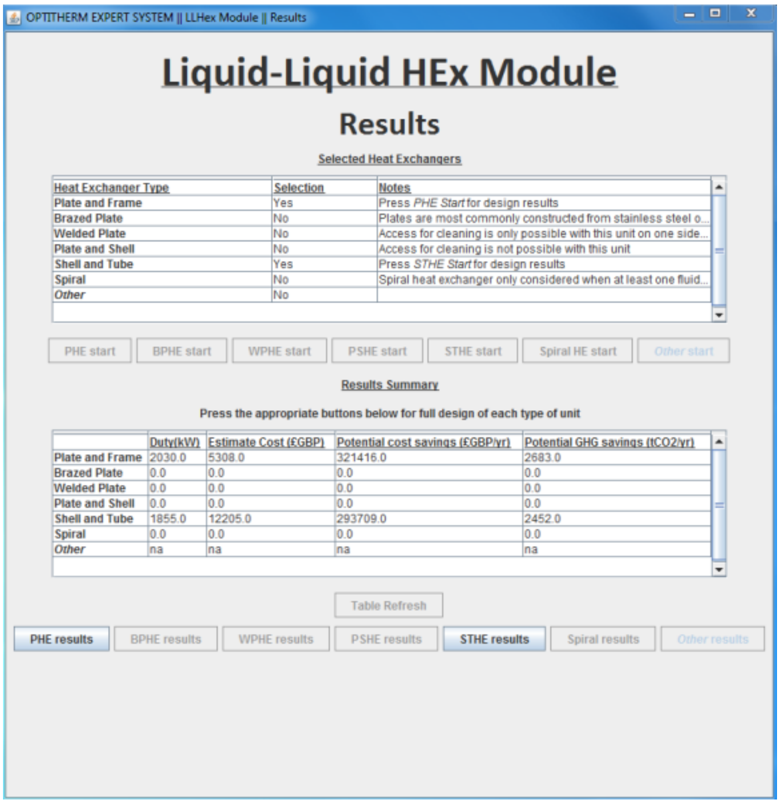


Figure A5.5. Heat exchanger selection results for case study

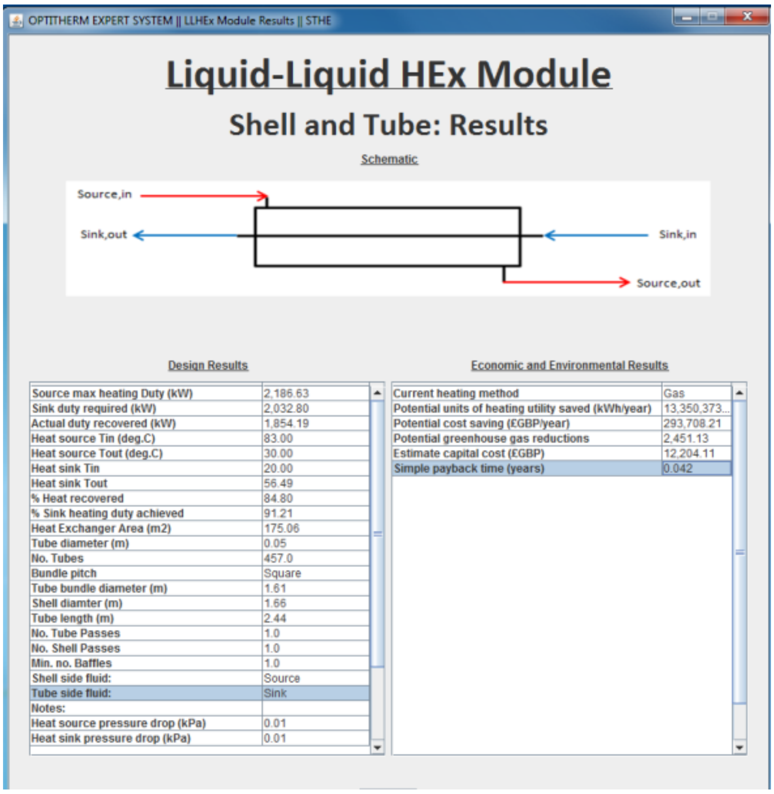


Figure A5.6. Shell and tube heat exchanger design results for case study

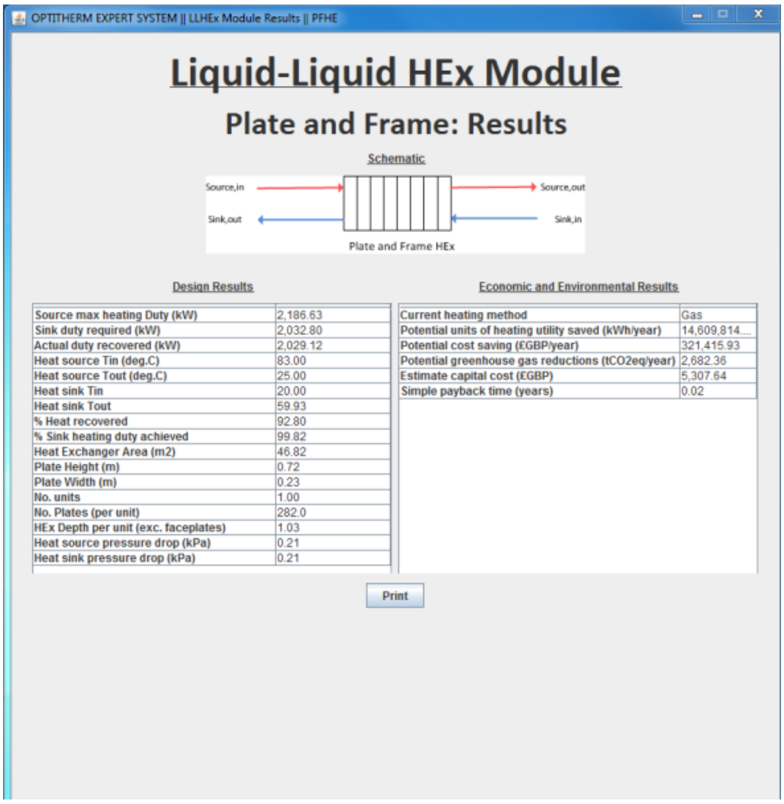


Figure A5.7. Plate and frame heat exchanger design results for case study

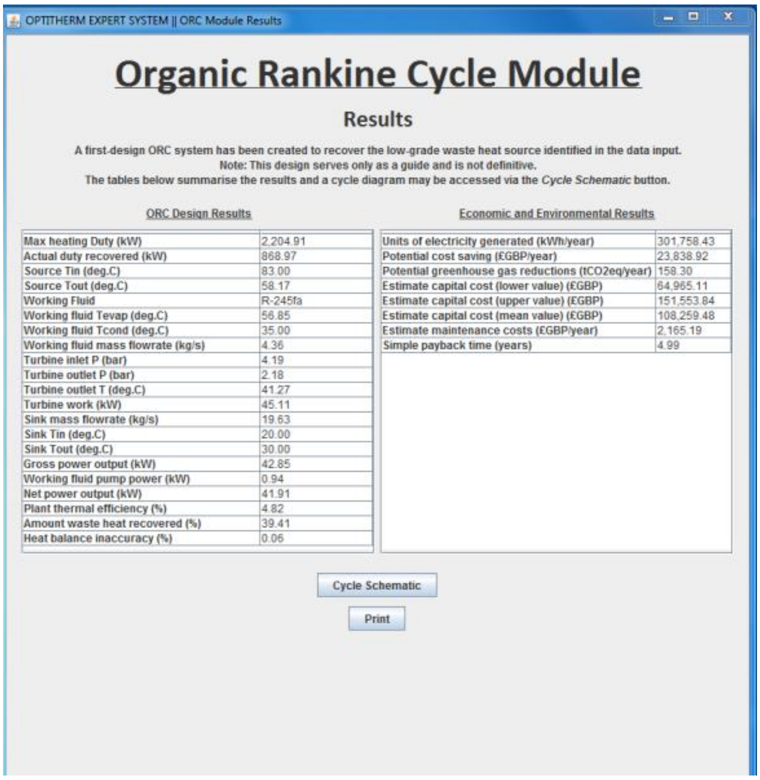


Figure A5.8. Organic Rankine cycle design results for case study

Screenshots from case study 3 (Section 6.3):

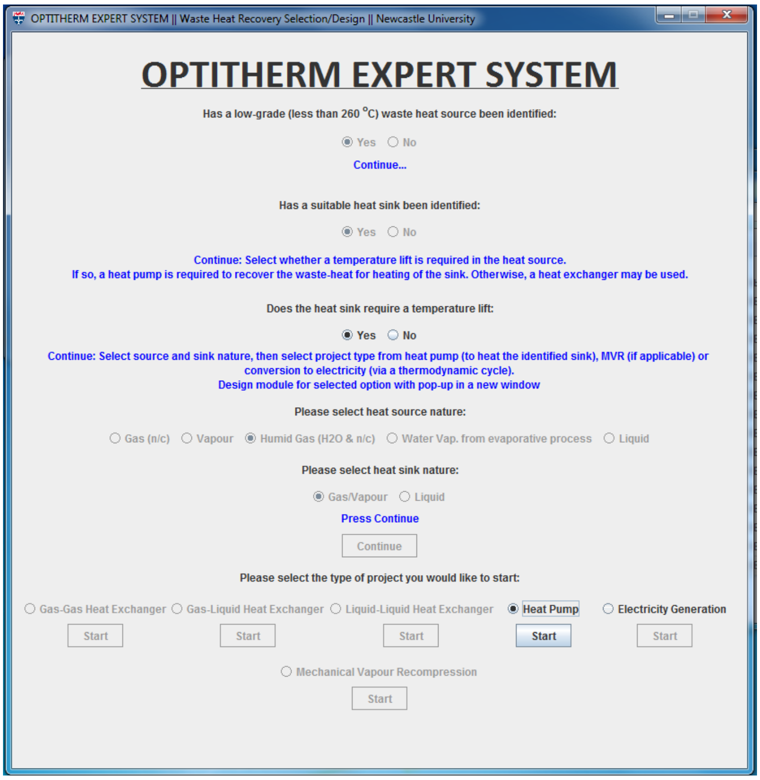


Figure A5.9. Initial questions for case study

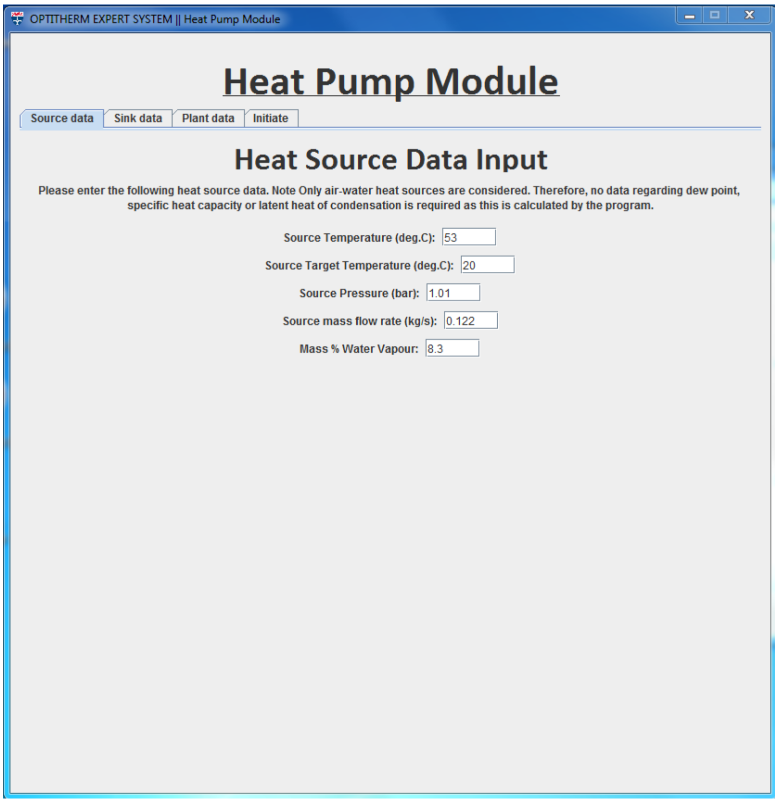


Figure A5.10. Data entry for case study

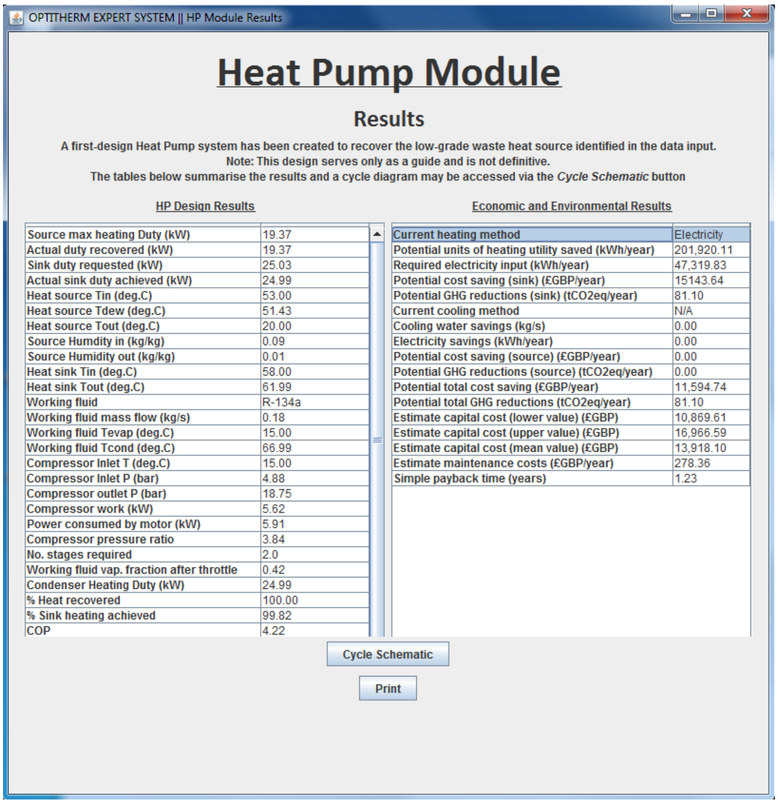


Figure A5.11. Heat pump design results for case study

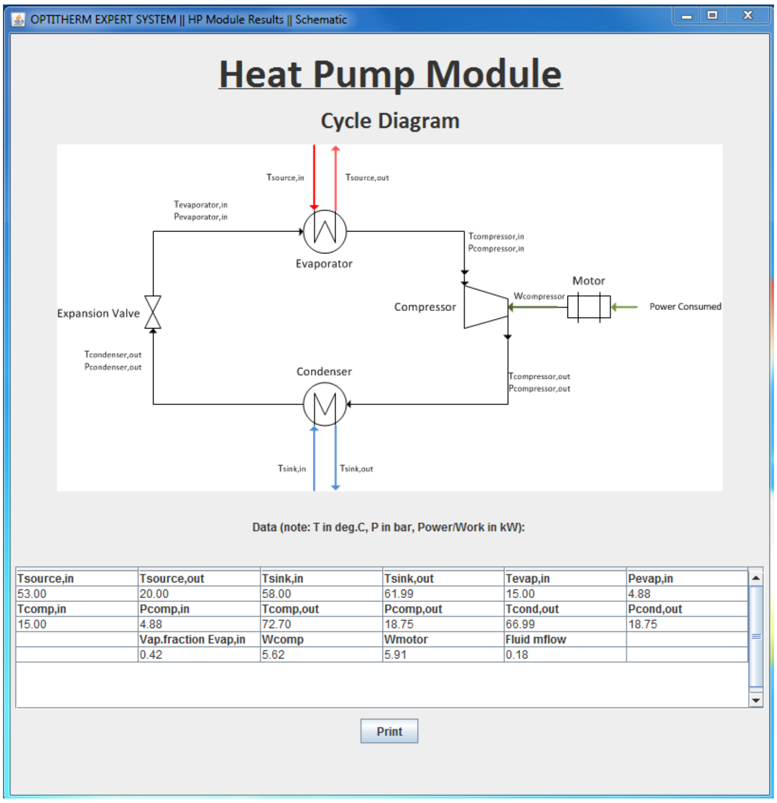


Figure A5.11. Heat pump cycle diagram for case study

Screenshots from case study 4 (Section 6.4):

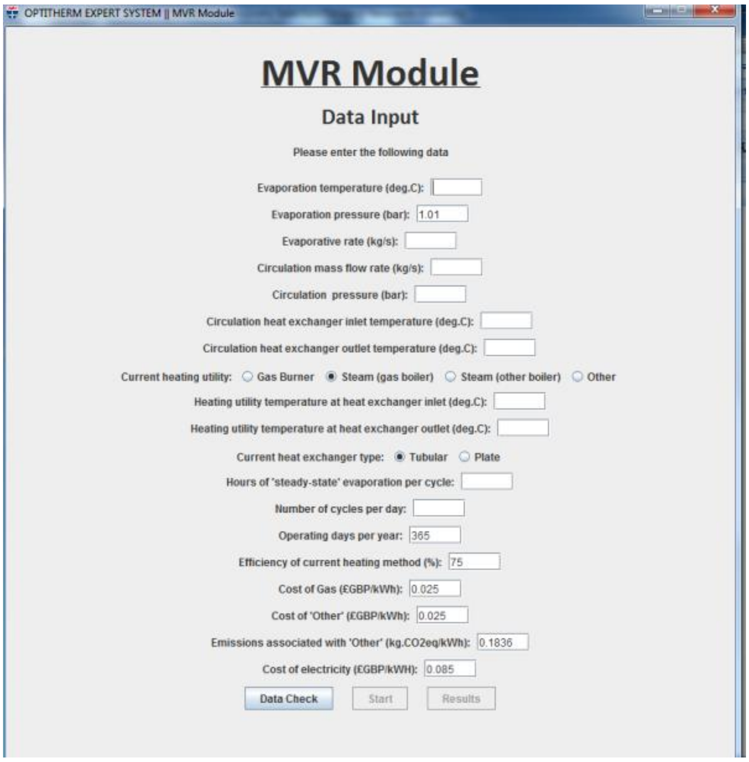


Figure A5.12. Data entry for case study

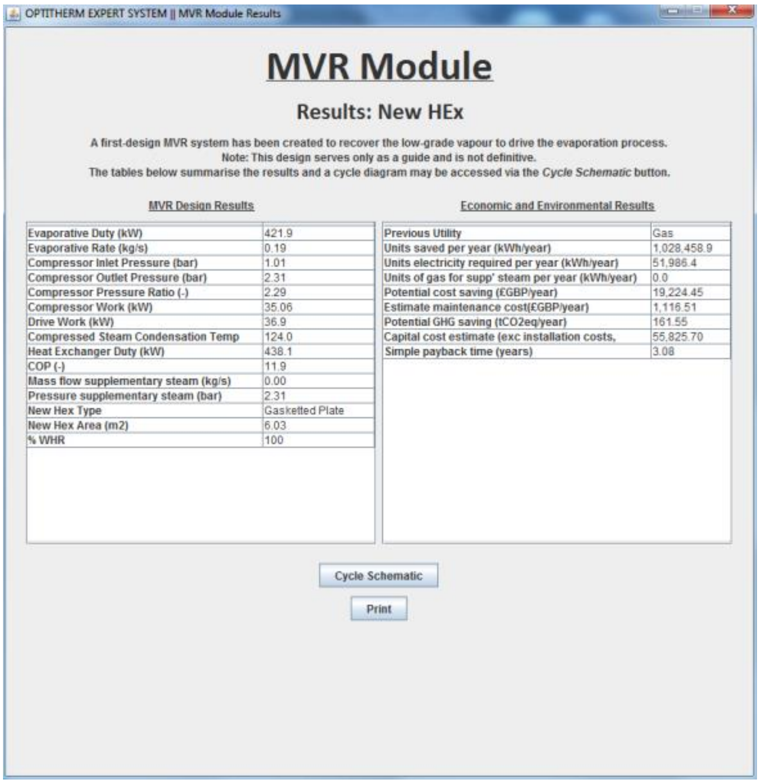


Figure A5.13. MVR design results for case study

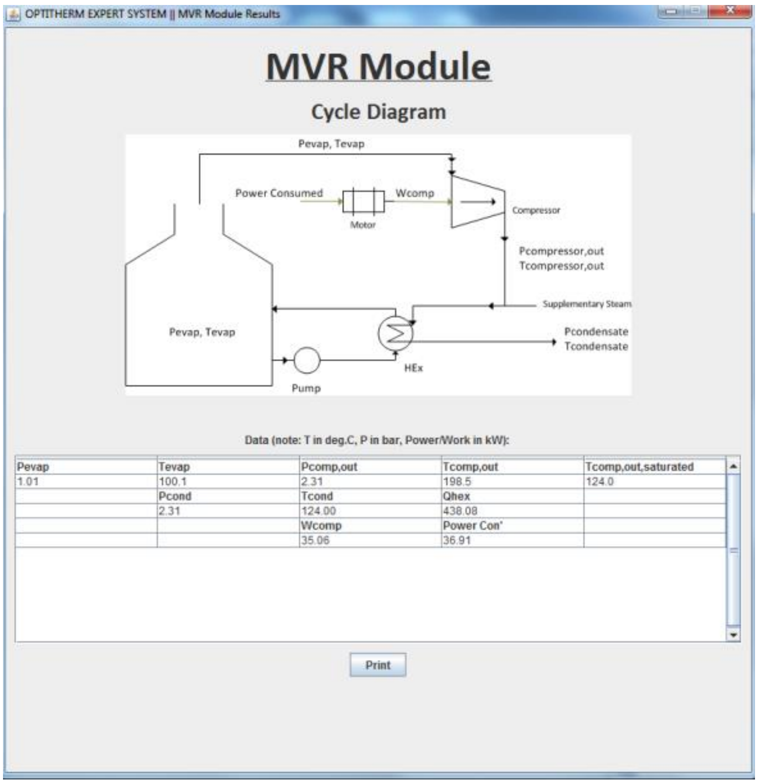


Figure A5.14. MVR (open) cycle diagram for case study

Screenshots from case study 5 (Section 6.5):

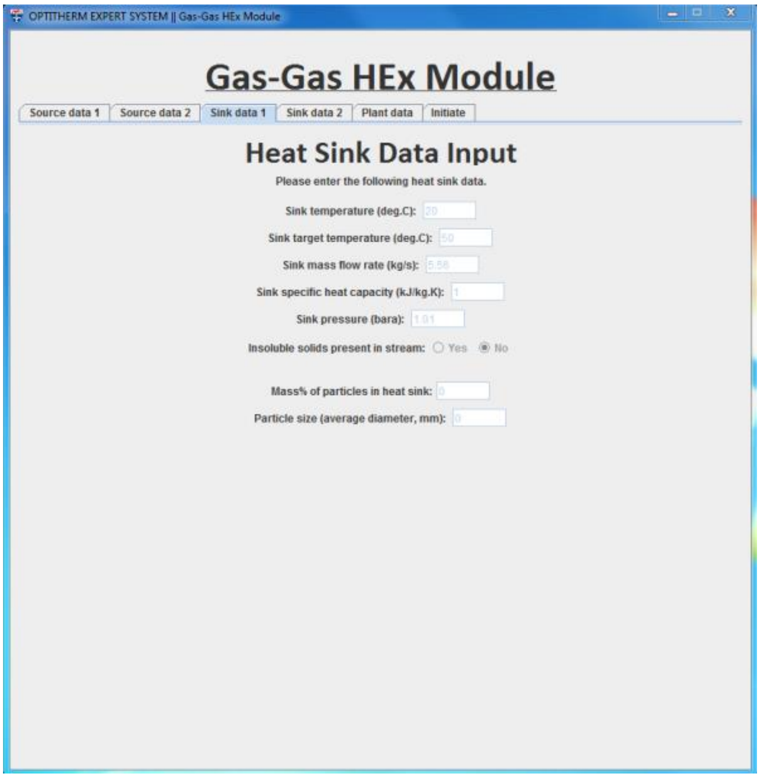


Figure A5.15. Heat exchanger data entry for case study

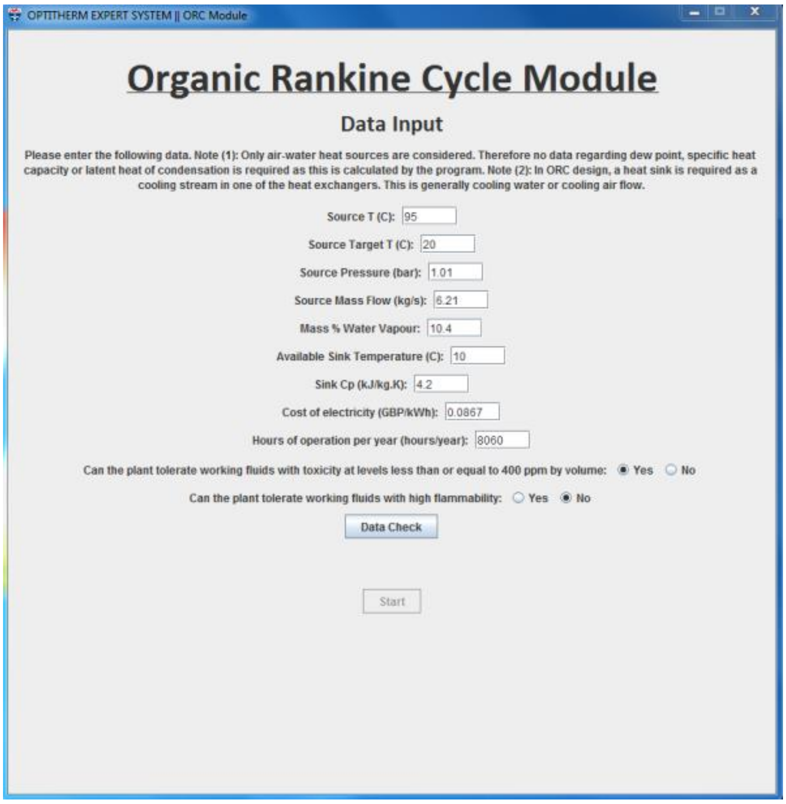


Figure A5.16. ORC data entry for case study

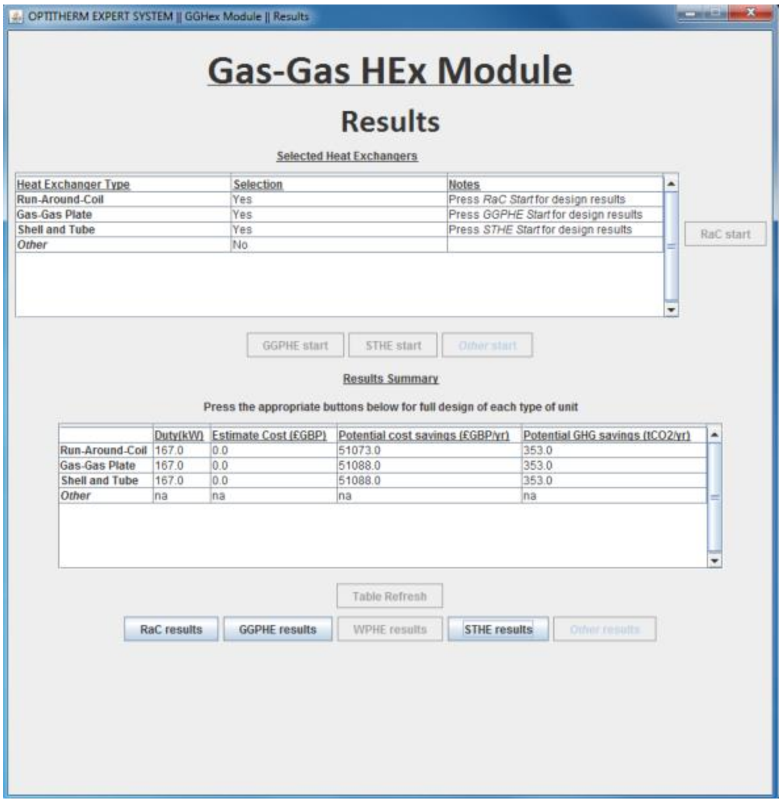


Figure A5.16. Heat exchanger selection results for case study

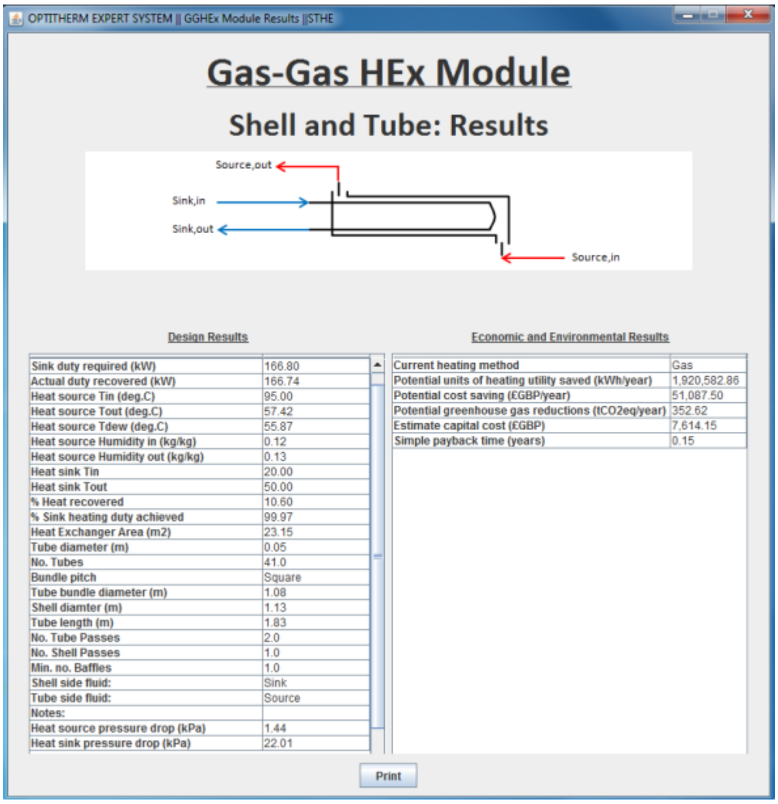
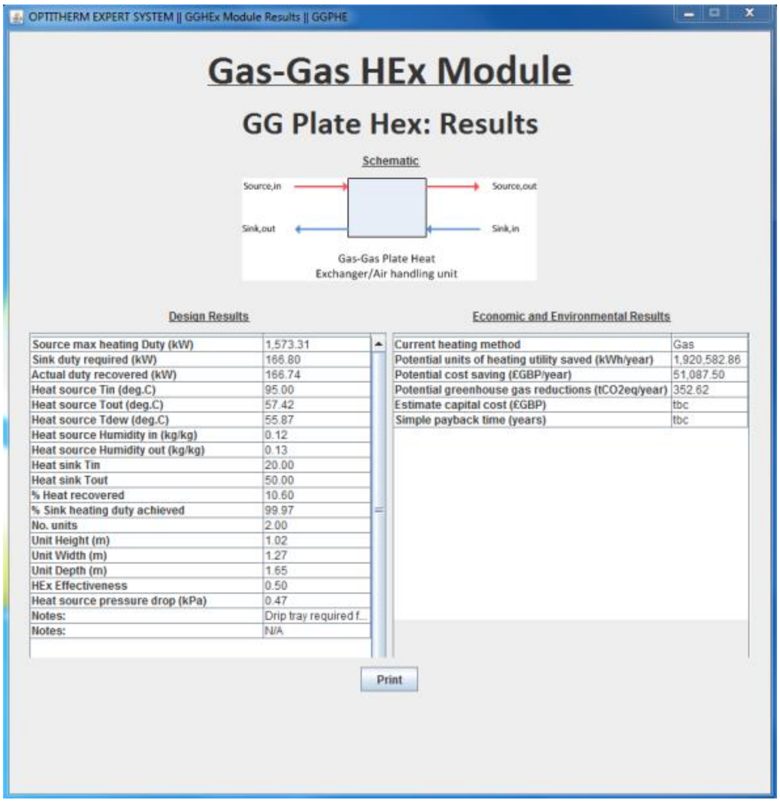


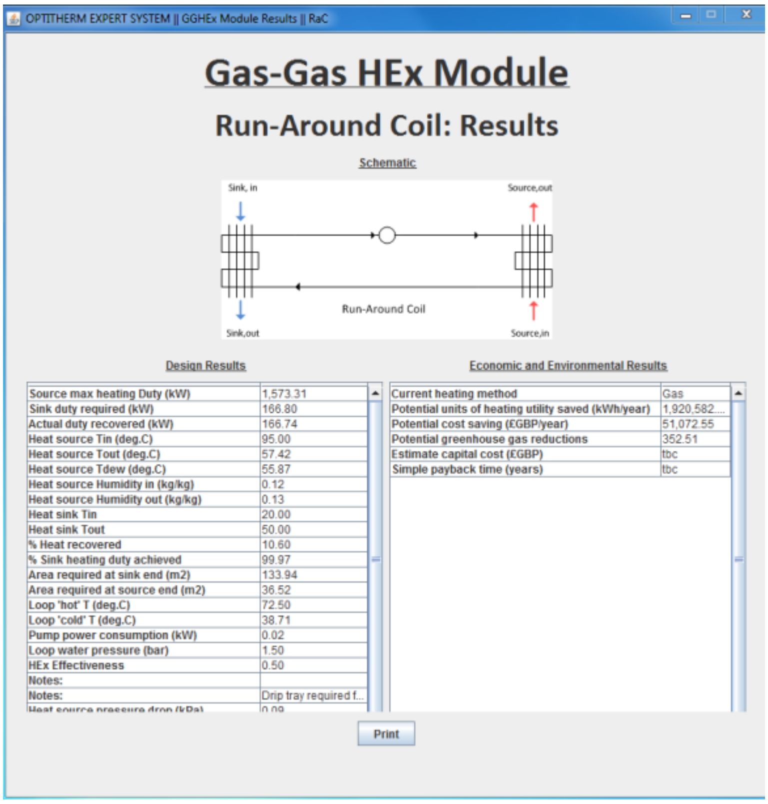
Figure A5.17. Shell and tube heat exchanger design results for case study



Source max heating Duty (kW)	1,573.31
Sink duty required (kW)	166.80
Actual duty recovered (kW)	166.74
Heat source Tin (deg.C)	95.00
Heat source Tout (deg.C)	57.42
Heat source Tdew (deg.C)	55.87
Heat source Humidity in (kg/kg)	0.12
Heat source Humidity out (kg/kg)	0.13
Heat sink Tin	20.00
Heat sink Tout	50.00
% Heat recovered	10.60
% Sink heating duty achieved	99.97
No. units	2.00
Unit Height (m)	1.02
Unit Width (m)	1.27
Unit Depth (m)	1.65
HEx Effectiveness	0.50
Heat source pressure drop (kPa)	0.47
Notes:	Drip tray required f...
Notes:	N/A

Current heating method	Gas
Potential units of heating utility saved (kWh/year)	1,920,582.86
Potential cost saving (£GBP/year)	51,087.50
Potential greenhouse gas reductions (tCO2eq/year)	352.62
Estimate capital cost (£GBP)	tbc
Simple payback time (years)	tbc

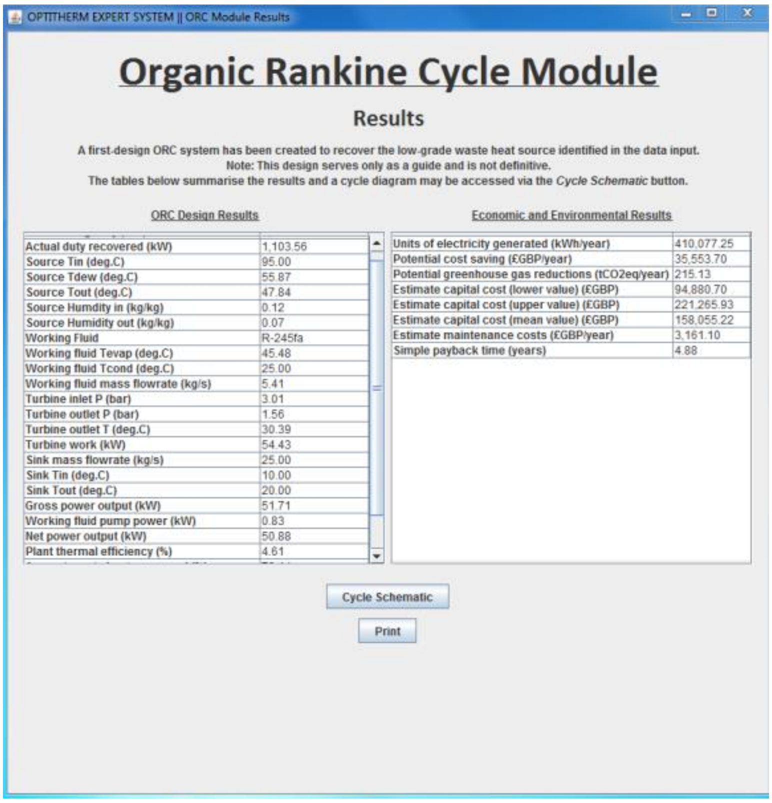
Figure A5.18. GG plate heat exchanger design results for case study



Source max heating Duty (kW)	1,573.31
Sink duty required (kW)	166.80
Actual duty recovered (kW)	166.74
Heat source Tin (deg.C)	95.00
Heat source Tout (deg.C)	57.42
Heat source Tdew (deg.C)	55.87
Heat source Humidity in (kg/kg)	0.12
Heat source Humidity out (kg/kg)	0.13
Heat sink Tin	20.00
Heat sink Tout	50.00
% Heat recovered	10.60
% Sink heating duty achieved	99.97
Area required at sink end (m2)	133.94
Area required at source end (m2)	36.52
Loop 'hot' T (deg.C)	72.50
Loop 'cold' T (deg.C)	38.71
Pump power consumption (kW)	0.02
Loop water pressure (bar)	1.50
HEx Effectiveness	0.50
Notes:	
Notes:	Drip tray required f...

Current heating method	Gas
Potential units of heating utility saved (kWh/year)	1,920,582...
Potential cost saving (£GBP/year)	51,072.55
Potential greenhouse gas reductions	352.51
Estimate capital cost (£GBP)	tbc
Simple payback time (years)	tbc

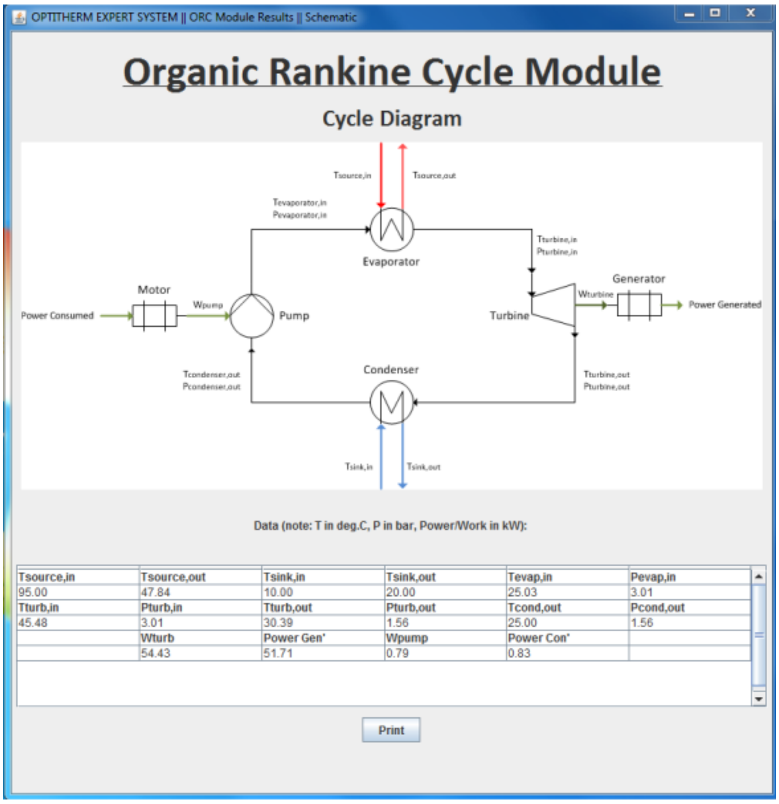
Figure A5.18. Run-around-coil heat exchanger design results for case study



Actual duty recovered (kW)	1,103.56
Source Tin (deg.C)	95.00
Source Tdew (deg.C)	55.87
Source Tout (deg.C)	47.84
Source Humdity in (kg/kg)	0.12
Source Humidity out (kg/kg)	0.07
Working Fluid	R-245fa
Working fluid Tevap (deg.C)	45.48
Working fluid Tcond (deg.C)	25.00
Working fluid mass flowrate (kg/s)	5.41
Turbine inlet P (bar)	3.01
Turbine outlet P (bar)	1.56
Turbine outlet T (deg.C)	30.39
Turbine work (kW)	54.43
Sink mass flowrate (kg/s)	25.00
Sink Tin (deg.C)	10.00
Sink Tout (deg.C)	20.00
Gross power output (kW)	51.71
Working fluid pump power (kW)	0.83
Net power output (kW)	50.88
Plant thermal efficiency (%)	4.61

Units of electricity generated (kWh/year)	410,077.25
Potential cost saving (£GBP/year)	35,553.70
Potential greenhouse gas reductions (tCO2eq/year)	215.13
Estimate capital cost (lower value) (£GBP)	94,880.70
Estimate capital cost (upper value) (£GBP)	221,265.93
Estimate capital cost (mean value) (£GBP)	158,055.22
Estimate maintenance costs (£GBP/year)	3,161.10
Simple payback time (years)	4.88

Figure A5.19. ORC design results for case study



Tsource,in	Tsource,out	Tsink,in	Tsink,out	Tevap,in	Pevap,in
95.00	47.84	10.00	20.00	25.03	3.01
Tturb,in	Pturb,in	Tturb,out	Pturb,out	Tcond,out	Pcond,out
45.48	3.01	30.39	1.56	25.00	1.56
	Wturb	Power Gen'	Wpump	Power Con'	
	54.43	51.71	0.79	0.83	

Figure A5.20. ORC cycle diagram results for case study

END