OPTIMISING THERMAL ENERGY RECOVERY, UTILISATION AND MANAGEMENT IN THE PROCESS INDUSTRIES

MATHEW CHIDIEBERE ANEKE

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MATHEW CHIDIEBERE ANEKE

A thesis submitted in partial fulfilment of the requirements of the University of Northumbria at Newcastle for the degree of Doctor of Philosophy

Research undertaken in the School of Built and Natural Environment in collaboration with Brunel University, Newcastle University, United Biscuits, Flo-Mech Ltd, Beedes Ltd and Chemistry Innovation Network (KTN)

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Abstract

The persistent increase in the price of energy, the clamour to preserve our environment from the harmful effects of the anthropogenic release of greenhouse gases from the combustion of fossil fuels and the need to conserve these rapidly depleting fuels has resulted in the need for the deployment of industry best practices in energy conservation through energy efficiency improvement processes like the waste heat recovery technique.

In 2006, it was estimated that approximately 20.66% of energy in the UK is consumed by industry as end-user, with the process industries (chemical industries, metal and steel industries, food and drink industries) consuming about 407 TWh, 2010 value stands at 320.28 TWh (approximately 18.35%). Due to the high number of food and drink industries in the UK, these are estimated to consume about 36% of this energy with a waste heat recovery potential of 2.8 TWh.

This work presents the importance of waste heat recovery in the process industries in general, and in the UK food industry in particular, with emphasis on the fryer section of the crisps manufacturing process, which has been identified as one of the energy-intensive food industries with high waste heat recovery potential.

The work proposes the use of a dual heat source ORC system for the recovery and conversion of the waste heat from the fryer section of a crisps manufacturing plant to electricity. The result, obtained through modelling and simulation, shows that the proposed technology can produce about 92% of the daily peak electricity need of the plant which is currently 216 kW. Also, the economic analysis shows that the proposed technology is viable (even at an inflation rate of 5.03% and discounted rate of 6%), with a payback period of approximately three years and net present value of over £2.2 million if the prices of electricity and carbon is at an average value of £0.16 and £13.77 respectively throughout the 30 years service life of the plant. The life cycle assessment study shows that the proposed technology can reduce the CO_2 emission by 139,580 kg/year if the electricity produced is used to displace that which would have been produced from a conventional coal-fired power plant.

Keywords: Waste Heat Recovery, Energy Efficiency, Organic Rankine Cycle, Life Cycle Assessment, Carbon Emission Reduction, Entropy Generation.

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Glossary

Subscripts

cf	cooling fluid
in	inlet
min	minimum
out	outlet
s	side
trans	transfer
uh	useful heat
wf	working fluid
wh	waste heat

Symbols

h	Specific enthalpy, kJ/kg
P	Pressure, bar
q or Q	Heat, kJ
s	Specific entropy, kJ/kg-K
T	Temperature, K or ^{o}C
U	Internal Energy, kJ
UA	Thermal Conductance, kW/K
W	Work
Δhs	Enthalpy change in an isentropic process, kJ/kg
ΔP	Pressure drop, bar
ΔS_{cv}	Rate of change of entropy, kJ/s-K

ΔT Temperature difference K or ^{o}C

ϕ Entropy generation rate, kJ/s-K

Abbreviations

ABC Air Bottoming Cycle

AC Absorption Chillers

AGMD Air Gap Membrane Distillation

AR Absorption Refrigeration

AR Absorption Refrigeration

COP Coefficient of Performance

DCMD Direct Contact Membrane Distillation

ECOP Exergetic Coefficient of Performance

ED Electro-Dialysis

EUF Energy Utilization Factor

HPP High Pressure Pump

ICE Internal Combustion Engine

MD Membrane Distillation

MDK Model Development Kit

MED Multi Effect Distillation

MSF Multi Stage Flash

mtoe Million Tonnes of Oil Equivalent

NEPA National Environmental Policy Act

NTU Number of Transfer Units

OEC Ormat® Energy Converter

OECD Organization for Economic Cooperation and Development

ORC	Organic Rankine Cycle
OTEC	Ocean Thermal Energy Conversion
PHE	Plate Heat Exchanger
PSE	Process Simulation Environment
RC	Rankine Cycle
REG	Recovered Energy Generation
RHE	Refrigerant Heat Exchanger
RO	Reverse Osmosis
SGMD	Sweep Gas Membrane Distillation
SHE	Solution Heat Exchanger
VCR	Vapour Compression Refrigeration
VMD	Vacuum Membrane Distillation

Dedication

I dedicate this work to my late dad, Mr Godwin Aneke, who lived to see the beginning of this PhD programme but was never there to witness its successful completion. Dad, we will always miss you and our prayer is that you will eternally rest in the bosom of the Almighty God.

Acknowledgements

I wish to acknowledge my supervisors Prof. Brian Agnew and Prof. Chris Underwood for their guidance, support and advice throughout my PhD programme. I also wish to acknowledge my family members Emmanuel, Henry, Norah, Stella, Jane, Gloria and my Mum, Mrs Roseline Aneke, for their prayers and encouragement throughout the period of this work. Furthermore, I would also like to acknowledge my fellow research partners from Newcastle University, Brunel University and the industrial partners for providing necessary data and an ideal collaborative environment which helped to make this research project a sucess.

Author’s Declaration

I declare that the work contained in this thesis has not been submitted for any other award and that it is my own work. I also confirm that this work fully acknowledges opinions, ideas and contributions from the work of others. The work was done in collaboration with Brunel University UK, Newcastle University UK, United Biscuits, Flo-Mech Ltd, Beedes Ltd and Chemistry Innovation Network (KTN).

There is no ethical clearance requirement for this thesis.

Name: Mathew Chidiebere Aneke

Signature: …………………………

Date: 1st August, 2012

Chapter One

This chapter covers the industrial energy demand and the need for efficient energy use and recovery of waste energy in the process industries with emphasis on the food, drink and chemical processing industries. It also introduces in detail the motivation for carrying out the research, the aims of the project, the methodology adopted in the work and the structure of the thesis.

1 Introduction

1.1 Energy and Industry

Industrialization and the world population are the two major factors that drive global energy demand. Thus, since these two factors are always on the increase, this means that world energy demand will continuously be on the increase also.

Amongst the different sources of energy (fossil fuels, wind, solar, hydro, geothermal and so on), fossil fuels still remain the most widely used. This is the result of their level of commercialization and technological advancement in their extraction. In 2009, the international energy agency (IEA) estimated that the total global energy consumption was about 508 x 10^{18} J (IEA, 2011). Out of this quantity, about 80% was produced through the burning of fossil fuels (IEA, 2011). This is close to the estimate given by US Energy Information Administration (EIA) who estimated that the world energy consumption in 2009 was 531 x 10^{18} J (EIA, 2010) with fossil fuels making up about 86% of this value. The projection for 2035 was estimated to be about

812 x 10^{18} J (EIA, 2011), with fossil fuels still playing a dominant role, although there is likely to be a remarkable improvement in the use of renewable energy (Figures 1–1 and 1–2). This shows that fossil fuels will still play a dominant role in energy generation and it will likely continue to dominate for many years to come.

A considerable quantity of total energy consumption in developed countries is used by industry. However, the percentage of energy use by industry differs from one country to another. As can be estimated from Table 1-1, industry accounted for 21% of the total national final energy consumption in the UK in 2006. The figures for the EU, the USA and the global average stand at 24.2%, 30% and 35% respectively (Eurostat, 2009, EIA, 2010, IEA, 2008).

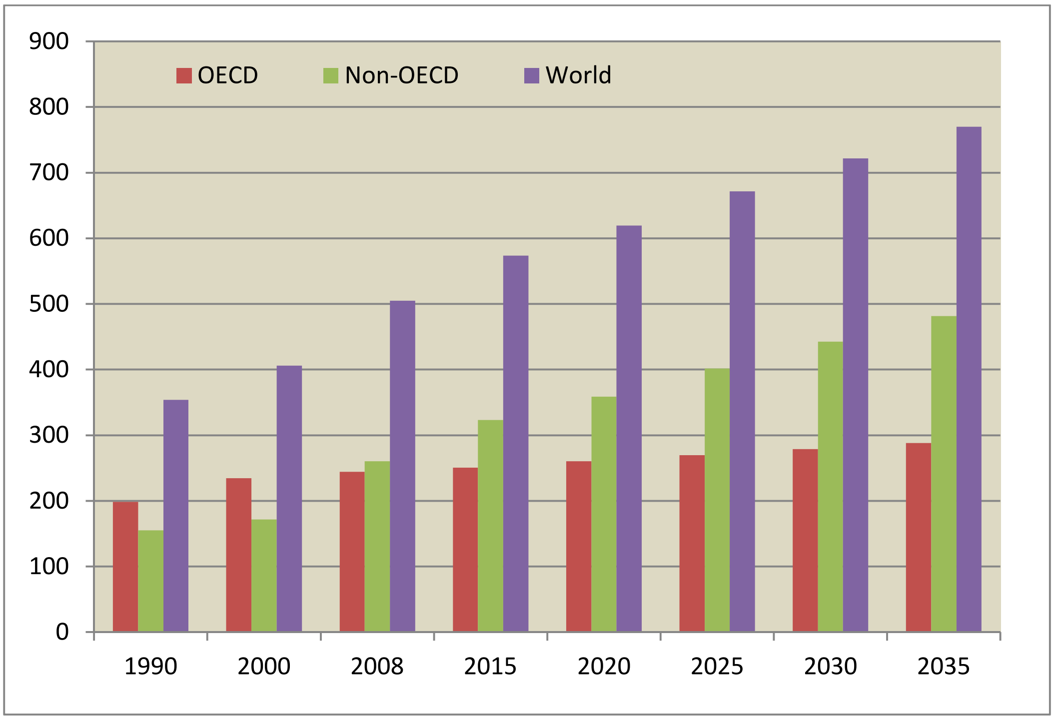


Figure 1–1: Energy Usage Projection (in quadrillion Btu) (EIA, 2011)

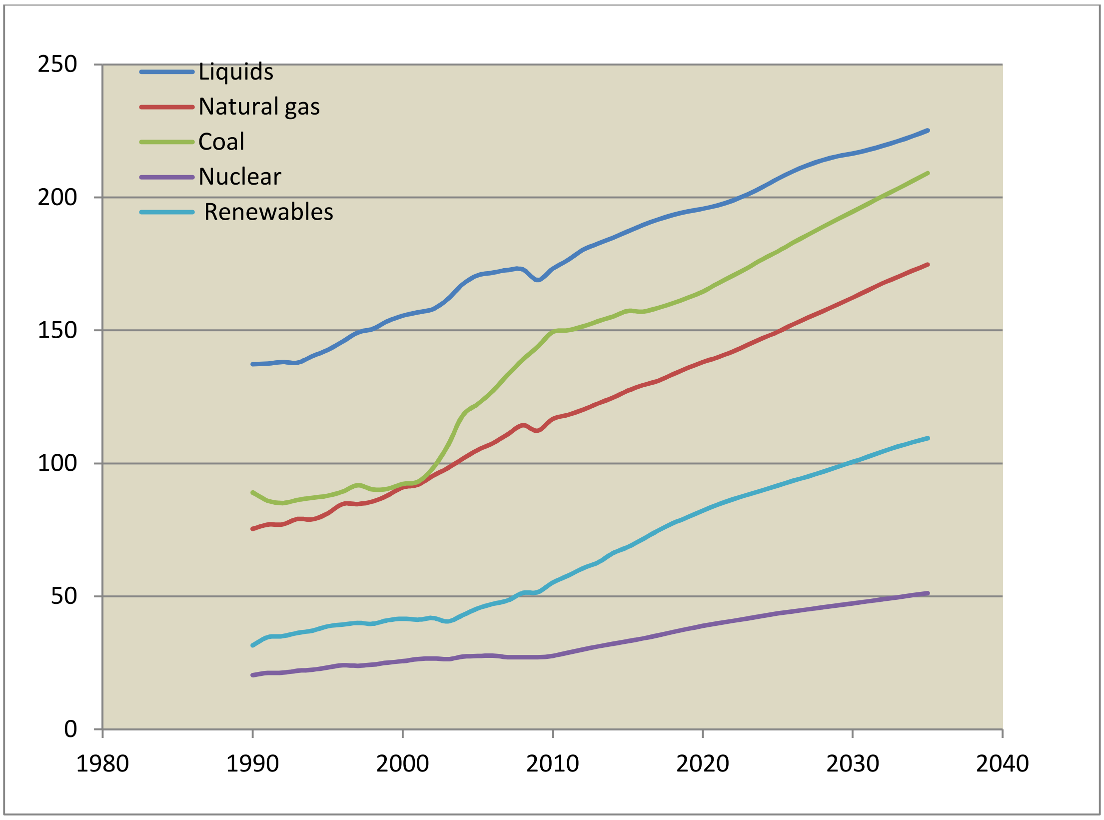


Figure 1–2: Energy Usage by Fuel (quadrillion Btu) (EIA, 2011)

Table 1–1: **Final Energy Consumption in the UK in 2006 (DBERR, 2007a, DBERR, 2007b)**

Energy	Consumption (in Mtonne of oil equivalent) by the sector stated				
	Industry	Domestic	Transport	Services*	Total
Coal and manufactured fuels	1.9	0.6			2.6
Natural gas	12.4	31.3		9.0	52.8
Oil	7.2	3.3	59.0	1.5	71.0
Electricity	10.0	10.0	0.7	8.7	29.5
Renewables and heat	1.0	0.3		0.6	1.9
Total	32.6	45.6	59.8	19.9	157.8

* Includes agriculture

In rapidly emerging economies like China, industries represent the largest energy consumer, with up to 70.8% of the national total energy consumption in 2005, of which the iron and steel subsector represented 22.8% (Liao *et al.*, 2007). This is also true in Taiwan, where the industrial sector accounted for about 50.7% of national total energy consumption in 2004 (Chan *et al.*, 2007). Studies carried out by the International energy agency in 2004 showed that energy-intensive industries (i.e. chemical, iron and steel, electronics/electrical, cement, textiles and pulp and paper) accounted for about 67% of total commercial industrial energy consumption (IEA, 2007).

In the UK, the trend in industrial energy use has changed significantly between 1970 and 2010 as shown in Figure 1-3. From this, it can be observed that the percentage of energy usage by industry is decreasing over time. This shows the impact of the use of energy conservation measures and energy-saving technologies following the oil crises of the 1970s and the shift from energy-intensive heavy industries to high technology and sophisticated service industries in developed countries. It is also as a result of the shift in the siting of most energy-intensive industries from developed to developing countries due to cheap labour, lower tax, and lower standards of health and safety and carbon emission regulations.

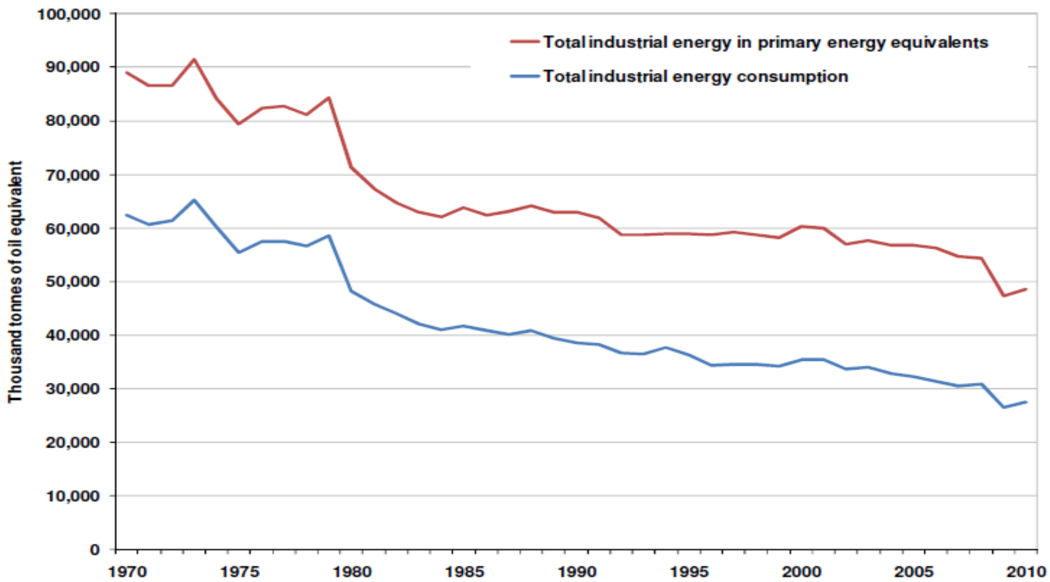


Figure 1–3: Total Industrial Energy Consumption, UK, 1970–2010 (DECC, 2011b)

Despite the changes in industrial energy usage with time, the use of different fuels for energy generation within the industrial sector has also changed over time. For example, Table 1–2 presents the energy consumption by industry in the UK from 1970 to 2010. From the table it can be seen that the total energy consumption decreased between 1970 and 2010, which demonstrates the implementation of more energy-efficient processes. However, it can also be observed that the majority of energy consumption comes from fossil fuels, while energy production from other sources, such as the use of renewable and heat energy are still under-utilized.

Fossil fuel is exhaustible and its usage as an energy source has been confirmed as the major single source of anthropogenic CO_2, a greenhouse gas believed to be responsible for causing global warming and ultimately climate change.

Recently, there has been global concern on the overdependence on fossil fuels for energy generation. This occurs not only as a result of their exhaustible nature but also because of their detrimental impact on the environment. These global concerns have led to clamour by the world leaders to develop alternative cleaner energy sources in order to abate the domineering and detrimental effect of using fossil fuels as an energy source.

Hence, as we aspire to develop alternative energy sources for cleaner energy generation, it is important in the interim to practise efficient use of energy so as to reduce the quantity of fossil fuels consumed. This can be achieved by ***avoiding energy wastage, as well as by practising efficient recovery (recycling) of wasted energy***.

The first law of thermodynamics states that energy can neither be created nor destroyed but can be changed from one form to another.

However, while the first law of thermodynamics tells us that the ***quantity*** of energy is unchanged, it does not tell us anything about its ***quality***. The concept of the quality of energy is treated in the second law of thermodynamics, which states that there is always an increase in entropy in any given process. It is this entropy generation that gives rise to unusable energy.

Almost all forms of energy which are unused within a system ultimately end up as thermal energy otherwise known as ***heat energy***. Heat energy is the most enduring or indestructible form of energy, and is not readily convertible into other forms of energy when compared with other energy forms such as mechanical or electrical energy. Considering the fact that heat energy is the most enduring or indestructible form of energy it then means that every other form of energy (as shown in Table 1–2) which is unused within a process will likely end up as heat energy. Hence, comparing the quantity of other forms of energy consumption (all except column 9 of Table 1–2) with the actual amount of heat consumed as energy (see column 9 of Table 1–2), bearing in mind that no thermal system is 100% efficient, we can conclude that there is a lot of heat energy unrecovered for energy production. This heat energy is, rather, released to the environment as waste heat energy. Therefore, in order to improve the energy efficiency of the system and reduce CO_2 emission, this waste heat energy has to be recovered. This brings us to the major motivation behind the execution of this project.

Table 1–2: Final Energy Consumption by Industry in the UK, 1970–2010 (DECC, 2011a)

Thousand tonnes of oil equivalent

	Coal	Coke and breeze	Other solid fuels	Coke oven gas	Town gas	Natural gas	Electricity	Heat sold	Renewables	Petroleum	Total
1970	12,681	9,655	209	1,164	1,778	1,788	6,275	..	..	28,397	**62,333**
1971	10,232	8,298	176	1,118	1,038	5,194	6,313	..	..	28,130	**60,746**
1972	7,675	7,832	252	1,111	1,154	8,136	6,292	..	..	28,674	**61,307**
1973	7,950	8,340	226	1,290	788	10,791	6,884	..	..	28,691	**65,149**
1974	7,290	7,167	201	975	494	12,320	6,517	..	..	24,968	**60,058**
1975	6,373	6,338	199	1,038	222	12,555	6,479	..	..	22,145	**55,444**
1976	5,902	7,129	131	1,091	68	14,237	6,950	..	..	21,966	**57,584**
1977	5,947	6,368	158	1,010	30	14,940	7,053	..	..	21,978	**57,574**
1978	5,627	5,932	179	899	15	15,149	7,222	..	..	21,570	**56,673**
1979	6,081	6,512	148	977	18	15,663	7,527	..	..	21,590	**58,564**
1980	5,083	3,335	133	642	13	15,258	6,854	..	..	16,938	**48,291**
1981	4,534	4,564	116	665	13	14,489	6,622	..	..	14,761	**45,776**
1982	4,668	4,083	144	605	8	14,588	6,353	..	..	13,530	**44,007**
1983	4,708	4,307	126	635	5	14,021	6,376	..	..	11,988	**42,191**
1984	3,796	4,408	68	537	5	14,686	6,758	..	..	10,859	**41,138**
1985	4,708	4,655	151	768	3	14,865	6,837	..	..	9,701	**41,702**
1986*(11)*	5,242	4,144	98	778	3	13,542	6,884	..	..	10,240	**40,931**
1987	4,048	4,660	80	821	3	14,137	8,005	..	..	8,456	**40,211**
1988	4,166	5,041	55	771	-	12,883	8,350	..	100	9,441	**40,807**
1989	4,489	4,286	30	613	-	12,515	8,550	..	102	8,820	**39,405**
1990	4,172	3,951	42	602	-	12,889	8,655	..	107	8,242	**38,660**
1991	4,270	3,691	14	570	-	12,311	8,563	..	109	8,729	**38,257**
1992	4,375	3,601	14	534	-	11,380	8,194	..	279	8,334	**36,711**
1993	3,553	3,613	7	560	-	11,521	8,328	..	266	8,592	**36,440**
1994	3,402	3,818	194	590	-	12,885	8,082	..	487	8,253	**37,711**
1995	2,840	3,750	184	576	-	12,680	8,654	..	526	7,066	**36,276**
1996	1,959	855	233	439	-	14,081	9,004	..	533	7,058	**34,470**
1997	1,963	787	249	457	-	14,754	9,189	..	532	6,315	**34,577**
1998	1,607	803	243	385	-	15,140	9,216	..	461	6,379	**34,512**
1999	1,353	820	215	205	-	15,203	9,542	1,086	283	5,374	**34,222**
2000	1,228	753	225	216	-	15,773	9,812	1,099	264	6,039	**35,506**
2001	1,195	719	210	154	-	15,464	9,573	1,001	243	6,611	**35,443**
2002	1,186	610	170	78	-	14,202	9,473r	1,321r	250	6,248	**33,764r**
2003	1,248	589	166r	53	-	14,292	9,396r	1,128	267	6,899	**34,074r**
2004	1,235	559	180r	67	-	13,238	9,584r	832	265	6,918	**32,912r**
2005	1,180	535	171	79	-	13,022	9,976	831	201	6,261r	**32,282r**
2006	1,164r	488	178	106	-	12,428	9,879r	809	213	6,080r	**31,423r**
2007	1,268r	513	177	101	-	11,466	9,785r	896	276	6,072r	**30,603r**
2008	1,296r	443	174	92r	-	11,925r	9,846r	1,021r	449r	5,567r	**30,852r**
2009	1,152r	332	152	49r	-	10,009r	8,671r	763r	447r	4,948r	**26,550r**
2010	1,135	289	163	97	-	10,487	8,985	841	484	4,972	**27,539**

1.2 Project Motivations

Waste heat is energy which is generated in a process by way of fuel combustion or chemical reaction and then dumped into the environment, even though it could still be reused for some useful and economic purposes (Khan, 2008).

Depending upon the process, waste heat can be rejected at virtually any temperature. This may range from temperatures as low as that of chilled water, to those as high as that of industrial furnaces, kilns and exhaust stacks. The most important property of heat is not necessarily the amount but its ***value*** or ***quality***. The higher the temperature of the waste heat, the higher the quality or value of the heat and thus the more economic the heat recovery process.

Thus, since waste heat is a useful source of energy, its ***efficient recovery***, ***utilization*** and ***management*** will not only help in the improvement of the energy efficiency of any given process, but also in the conservation of exhaustible natural energy resources (fossil fuels), and at the same time contribute to the preservation of our environment through its contribution to reducing CO_2 emissions. Furthermore, it will help in resource savings and improvement in profitability of any given process.

1.3 Waste Heat Energy Recovery

Waste heat energy recovery cuts across many industries, processes and techniques. As a result of the broad nature of the field, it will be pertinent to identify the area of interest as well as the technique to be adopted for any given process. In order for the technology to have the most impact on CO_2 emission reduction in the UK, it has to be implemented in a sector which promises a high waste heat recovery potential. However, its implementation in other sectors where waste heat is generated should also be encouraged.

1.3.1 Where Do We Recover Waste Heat Energy?

Energy consumption by individual sectors of the economy in the UK has changed substantially since 1980 (DBERR, 2007a). Although there have been rises by 68% for transport, 14% for the domestic sector and 6% for the service sector, energy consumption by industry fell by 33% due to a combination of structural changes and changes in energy utilization efficiency (DBERR, 2007a, DBERR, 2007b, DTI, 2002). Although industrial energy consumption is decreasing, there is still much room for improvement, especially in the area of waste heat recovery.

In the process industries, enormous amounts of hot flue gases are generated from boilers, kilns, ovens and furnaces. If some of this heat was to be recovered, a considerable amount of fuel and money could be saved.

In the UK, the process industries remain one of the major energy end users. In 2010, it was estimated that industries' energy use was 320.28 TWh which represents about 18.35% of the total energy end use in the UK (DECC, 2011a). Out of this, about 44.37% was consumed by the energy intensive industries (food, chemicals, paper and metal processing industries), with the food and drinks processing industries accounting for about 37.24 TWh, paper 27.77 TWh, chemicals 51.63 TWh and metals 25.47 TWh (DECC, 2011b). Associated with this high energy consumption is the generation of process heat which accounts for about 91.30 TWh with 40% as high grade heat while the remaining 60% are low grade (ERP, 2011). This low grade heat is mainly encountered in the chemical, paper and food industries where wider deployment of waste heat recovery could provide efficiency improvements. Considering the large number of food and drink processing industries in the UK, their heat recovery potential is estimated to be about 2.8 TWh (Reay and Morrell, 2007), which is about 1.2 TWh higher than that of chemical industries, 2.1 TWh higher than that of the metals industries and 2.46 TWh higher than the paper and pulp industry.

Thus the ***greatest energy recovery potential*** is in the ***food and drink*** and chemical processing industries, which serve as the main focus of this research; however, most of the results and outcomes will be generic and thus can be applied to other industrial sectors.

1.3.2 Why Do We Recover Waste Heat Energy?

The rises in the price of energy and government environmental regulations are the major driving forces which promote energy recovery.

In a real industrial plant, no matter how efficient or optimized a thermal component or plant process is, there is always bound to be energy loss in the form of heat from the units or the process. This low-grade heat energy is mostly dumped into the environment, thus giving rise to energy and resource loss.

One of the obvious ways of conserving resources and saving energy is to recover most of the low-grade heat energy contained in the effluent streams which are dissipated into the environment (Lamb, 1982). Resource savings from energy recovery from the exhaust gas, water and air streams from the process industries are, in most cases, becoming the largest benefits from investment in energy recovery equipments. Although energy recovery involves capital investment to set up the systems, studies have shown that a payback period of less of five years is achievable (Energy Technology Support Unit, 1986, Gottschalk, 1996). This shows that it is economically viable for industries to implement energy recovery systems.

Despite the saving in resources, stiff environmental regulations are also among the factors that drive energy recovery in process industries. Energy recovery is beneficial to the environment. Energy wastage had been linked to an increase in the emissions of CO_2, a major greenhouse gas believed to be responsible for the global warming which causes climate change. For example, for any useful energy unrecovered in the process industry, there is always additional thousands of tonnes of CO_2 released into the environment through the burning of fossil fuels while trying to generate energy to replace the wasted energy. Although CO_2 happens not to be the greenhouse gas with the highest global warming potential, its release into the environment in large quantities through the burning of fossil fuels is a matter of concern.

As a result of the potential environmental benefits, the UK government has identified energy recovery efficiency in its Energy White Paper, published in 2007, as one of the means of meeting the UK's Kyoto target to reduce greenhouse gas emissions by 12.5% from 1990 levels within the commitment period of 2008–2012, as well as meeting their new self-imposed targets of

26–34% reduction in CO_2 emissions by 2020, alongside an 80% reduction by 2050 as stated in the Climate Change Act (DECC, 2008).

Regardless of the economic and environmental benefits of waste heat recovery, it is also thermodynamically important to recover waste heat from the process industries. Thermodynamically, the recovery of waste heat helps not only to improve the energy utilization factor of the system, but also minimize its entropy generation. Entropy generation minimization is a relatively new thermodynamic principle based on the second law of thermodynamics. According to the second law of thermodynamics, the entropy of an isolated system can never decrease (Kotas, 1985).

$$\Delta S \geq 0 \tag{1-1}$$

For systems which interact with the environment, this increase in entropy causes irreversibility in the system, giving rise to loss of performance. This irreversibility or loss in performance occurs as the system generates entropy into the environment as a result of heat loss. The entropy can be reduced by reducing any energy loss in the system.

For example, if we consider a simple thermodynamic system where heat is generated to provide heating application while the unused heat is emitted into the environment as waste heat (see Figure 1–4):

Thermodynamically, the Energy Utilization Factor (EUF) of the system can be calculated from the first law as

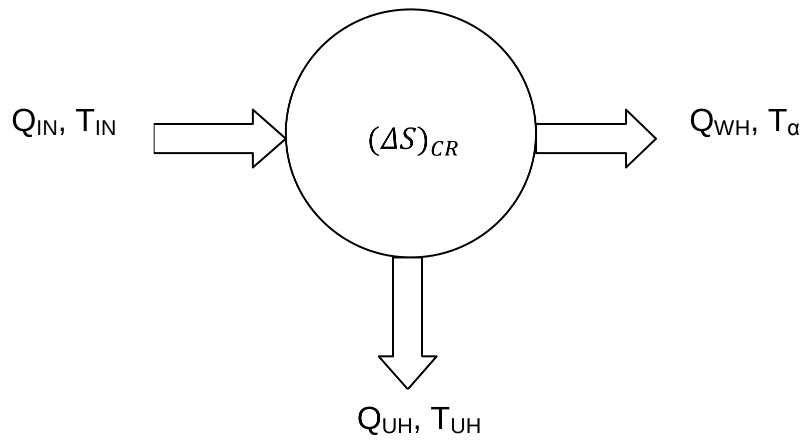


Figure 1–4: A Simple Thermodynamic System

$$EUF_1 = {Q_{UH}}/{Q_{IN}} \tag{1-2}$$

$$= {(Q_{IN} - Q_{WH})}/{Q_{IN}} \tag{1-3}$$

where

$Q_{UH} = \text{ useful heat at a reference temperature of } T_{UH}$

$Q_{IN} = \text{ heat input at a reference temperature of } T_{IN}$

$Q_{WH} = \text{waste heat emitted into the environment at ambient temperature } T_\propto$

From the second law of thermodynamics, the change in entropy of the entire system can be given as

$$dS_1 = (dS)_{CR} + (dS)_{IN} + (dS)_{UH} + (dS)_{WH} \geq 0 \tag{1-4}$$

The entropy terms in Equation 1-4 above can be written in terms of the various process parameters as follows:

$$dS_{IN} = -\frac{dQ_{IN}}{T_{IN}}$$

$$dS_{UH} = \frac{dQ_{UH}}{T_{UH}}$$

$$dS_{WH} = \frac{dQ_{WH}}{T_{\alpha}}$$

Substituting these into Equation 1-4 we have

$$dS_1 = (dS)_{CR} - \frac{dQ_{IN}}{T_{IN}} + \frac{dQ_{UH}}{T_{UH}} + \frac{dQ_{WH}}{T_{\alpha}} \geq 0 \qquad 1\text{-}5$$

Dividing by the time interval,

$$S_1 = \frac{dS_{CR}}{dt} - \frac{Q_{IN}}{T_{IN}} + \frac{Q_{UH}}{T_{UH}} + \frac{Q_{WH}}{T_{\alpha}} \geq 0 \qquad 1\text{-}6$$

The entropy generation by the system into the environment becomes

$$\frac{Q_{WH}}{T_{\alpha}} = -\frac{dS_{CR}}{dt} + \frac{Q_{IN}}{T_{IN}} - \frac{Q_{UH}}{T_{UH}} + S_1 \qquad 1\text{-}7$$

assuming the waste heat (Q_{WH}) is recovered and used for work production, as shown in Figure 1–5.

The change in the entropy of the new system becomes

$$dS_2 = (dS)_{CR} + (dS)_{IN} + (dS)_{UH} + (dS)_{EX} + (dS)_{W} \geq 0 \qquad 1\text{-}8$$

but

$$dS_W = 0$$

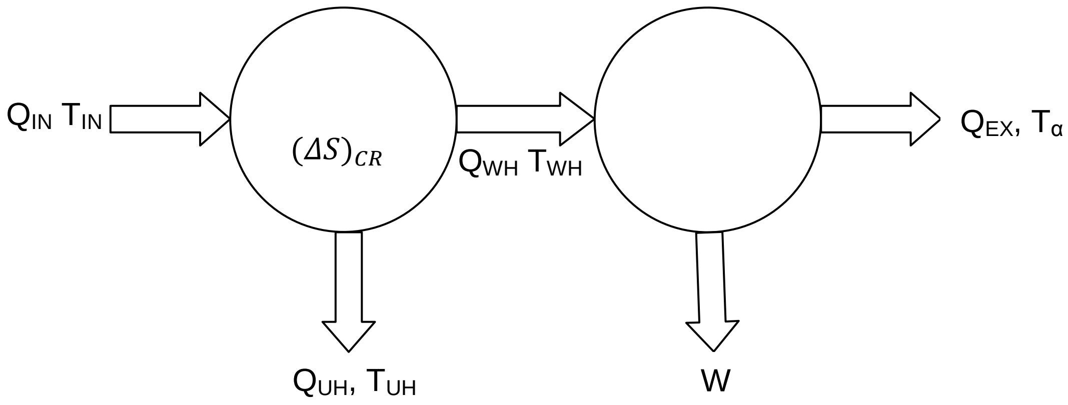


Figure 1–5: A Simple Thermodynamic System with Heat Recovery

Substituting the entropy values gives

$$dS_2 = (dS)_{CR} - \frac{dQ_{IN}}{T_{IN}} + \frac{dQ_{UH}}{T_{UH}} + \frac{dQ_{EX}}{T_\alpha} \geq 0 \tag{1-9}$$

Dividing by the time interval,

$$S_2 = \frac{dS_{CR}}{dt} - \frac{Q_{IN}}{T_{IN}} + \frac{Q_{UH}}{T_{UH}} + \frac{Q_{EX}}{T_\alpha} \geq 0 \tag{1-10}$$

The entropy generation by the second system into the environment becomes

$$\frac{Q_{EX}}{T_\alpha} = \frac{dS_{CR}}{dt} - \frac{Q_{IN}}{T_{IN}} + \frac{Q_{UH}}{T_{UH}} + S_2 \tag{1-11}$$

but

$$Q_{EX} < Q_{WH} \tag{1-12}$$

therefore

$$\frac{Q_{EX}}{T_\alpha} < \frac{Q_{WH}}{T_\alpha} \tag{1-13}$$

the EUF of the new system becomes

$$EUF_2 = {(Q_{UH} + W)}/{Q_{IN}} \qquad 1\text{-}14$$

$$EUF_2 > EUF_1 \qquad 1\text{-}15$$

Hence, from Equation 1-13, it can be seen that the inclusion of the waste heat recovery system reduces the system's generation of entropy into the environment. This helps to improve the EUF of the system as shown in Equation 1-14.

Therefore, from the above analysis, it can be concluded that waste heat recovery has economic, environmental and thermodynamic advantages and hence should be encouraged.

1.3.3 How Do We Recover Waste Heat Energy?

As we have established the "where and why" of recovering waste energy, the next question that remains to be answered is "how do we recover energy?" and that is the question this section seeks to address.

There are many ways to recover waste heat from the process industries. Some of them include:

- **Direct Heat Recovery**

In this process, the waste heat from the exhaust fluids is transferred to another fluid via a heat exchanger. For example, in Figure 1–6, the thermal energy contained within the products of combustion leaving the furnace is used to preheat the fresh air required for combustion. This approach is usually implemented when there is a need to use the recovered waste heat for other heating applications in the system.

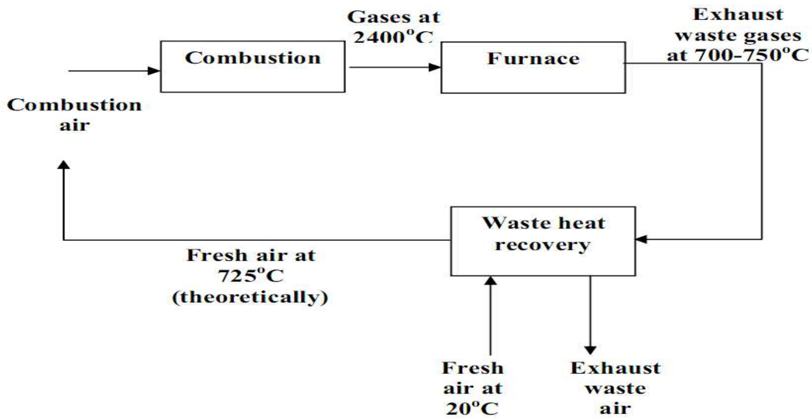


Figure 1–6: Direct Heat Recovery from Furnace Exhaust Gases

- **Cascading Heat Utilization**

This method is usually applicable in processes where many simultaneous processes are taking place, which require progressively reducing temperatures. This synchronization of demand may not always be achievable, and so thermal storage may also be required (Khan, 2008). Heat exchangers will be utilized where an exhaust fluid cannot be used directly. A typical flow diagram of a cascaded heat utilization system is shown in Figure 1–7.

- **Absorption Chillers**

In this system, a quantity of thermal energy at a high temperature level is used to produce a refrigerating or air conditioning effect. A typical flow diagram of the processes involved is shown in Figure 1–8.

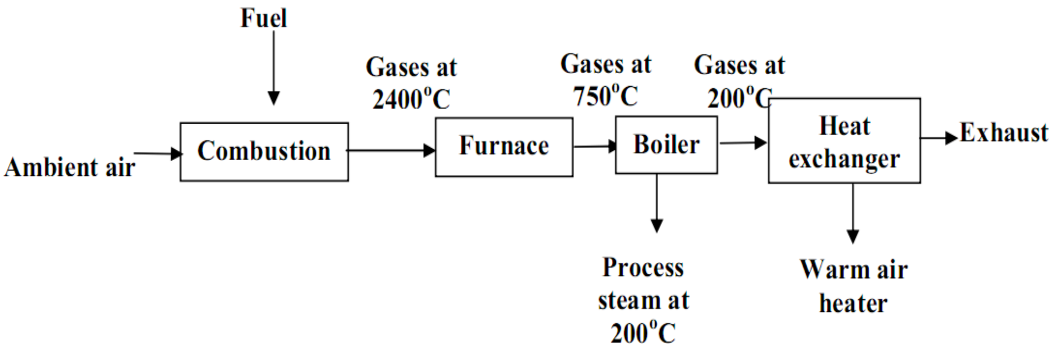


Figure 1–7: Cascading of Thermal Energy in an Industrial Process

The waste heat is used to vaporize the refrigerant from the solution contained in the generator. The remaining solution is sent back to the absorber while the evaporated refrigerant is passed to the condenser, where it is condensed, and then moved down to the evaporator via an expansion valve. There it is evaporated again to produce a cooling effect. The evaporated refrigerant is absorbed in the absorber by the solution and returned to the evaporator to restart the whole cycle. The refrigerant and solution cycle is a closed loop cycle.

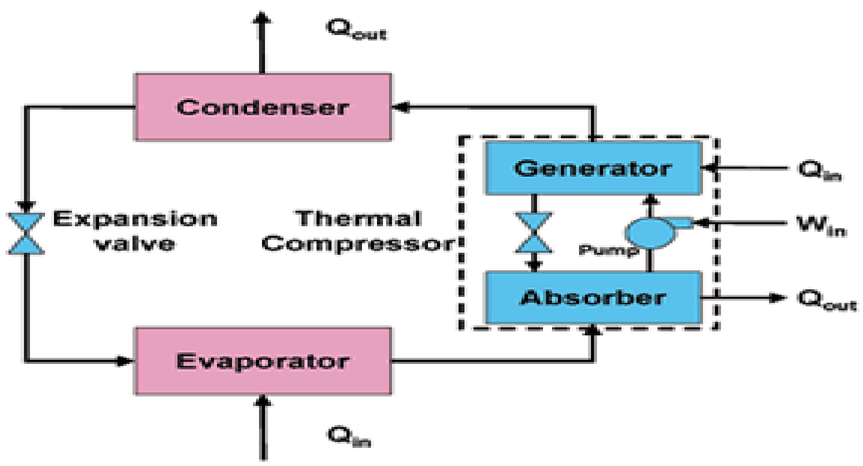


Figure 1–8: Absorption Chillers Process

- **Rankine Cycle (RC) / Organic Rankine Cycles Systems**

The Rankine Cycle system makes use of waste heat for the generation of work which can be used for power generation, reverse osmosis desalination or other applications. A typical process flow diagram of a conventional RC system is shown in Figure 1–9. In the process, waste heat from the process plant is passed through the evaporator (boiler) and used to vaporize a working fluid which is expanded in a turbine and used to generate power. The exit fluid from the turbine is condensed and pumped back to the evaporator to complete the cycle. Like the absorption chillers system, the working fluid cycle is a closed loop cycle. In an RC system, the working fluid is usually water. For low-grade heat recovery systems, the working fluid used is mainly an organic fluid or hydrocarbons. The main reason for this is that these always have a lower heat of vaporization than water (Figure 1–10) and hence can be vaporized more easily with low-grade heat. Such cycles with organic or hydrocarbon working fluids are known as **Organic Rankine Cycles (ORCs)**.

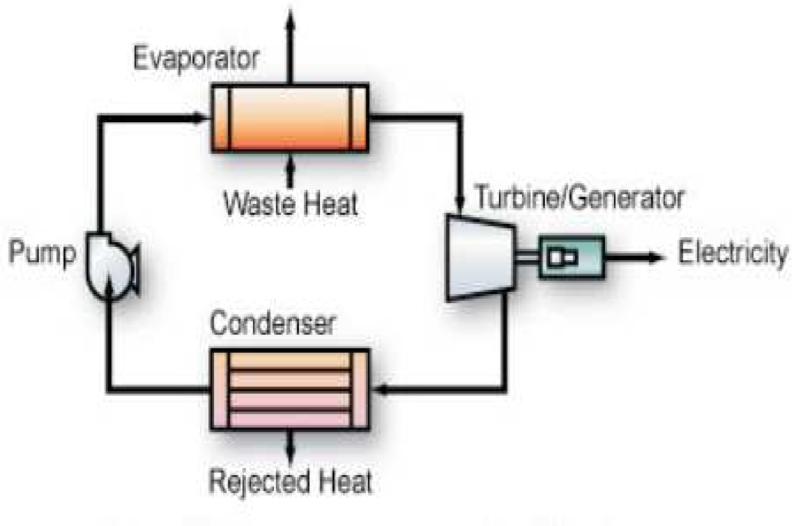


Figure 1–9: Rankine Cycle System

1.4 Waste Heat Energy Utilization

As mentioned earlier, heat energy from the process industries, otherwise known as low-grade heat, is always dumped into the environment instead of being utilized for energy production. Table 1–2 shows that the use of heat energy to generate further energy is still very much under-utilized when compared with other energy sources. Utilization of heat energy in industry is currently receiving close attention at both regional and national level. There are numerous programmes of energy conservation currently in progress in many countries (United Nations Industrial Development Organization, 2010, IEA, 2008) They include:

- legislative, physical and proportional frameworks to stimulate and support measures on industrial energy conservation;
- creation of appropriate financial, technical and organizational mechanisms to promote effective energy utilization;
- scientific and technological activities; and
- development of education and training programmes

Although, research and development efforts within the last three decades have resulted in numerous technologies, and innovative devices and better know-how and usage of energy in industries this has not been uniformly implemented globally and hence, governments has been advised to promote research and development in energy efficiency technologies (IEA, 2008).

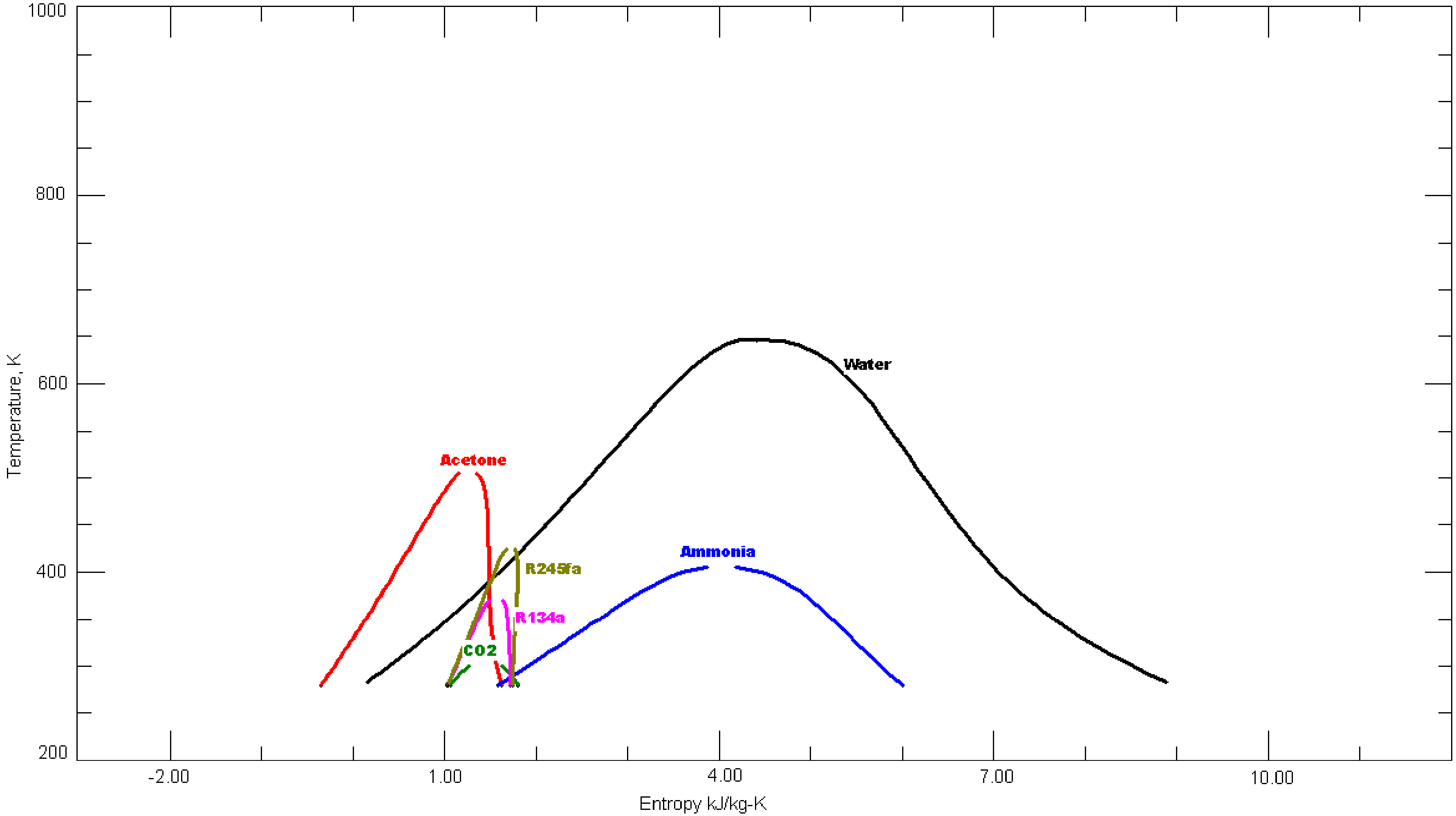


Figure 1–10: T-s Diagram Comparing some Organic Fluids with Water

The greatest progress in energy conservation has taken place in developed, industrialized countries (United Nations Industrial Development Organization, 2010). This is due to their rigorous attempts to reduce high energy consumption levels through the introduction and implementation of existing and new energy conservation technologies. Some developing countries like Brazil, Philippines and Korea have also had notable successes in energy conservation (Badr, 2008). It is therefore advisable that their experiences should be applied to other developing countries through increased technical cooperation.

As was mentioned earlier, energy conservation through improvement in energy-utilization efficiency has been identified by the UK government as a major approach in meeting their self-imposed target of 80% reduction in CO_2 emissions by 2050.

Through a combination of technological innovations in processes and equipment and systematic monitoring of actual consumption, it has been estimated that energy savings of up to 15% - 35% can be achieved in some industrial applications (United Nations Industrial Development Organization, 2010).

Research in many countries – developing and developed – has frequently shown that a given increment in useful energy could be achieved more cheaply by investing in energy-utilization efficiency rather than energy-resource development (Akbaba, 1999, Gottschalk, 1996).

Although energy savings produced through efficiency improvement are ultimately limited, for the foreseeable future such improvements must be considered as a possibly more attractive investment than direct energy-resource exploitation (Badr, 2008).

Hence, heat energy utilization in the process industries, as an example of energy savings produced through efficiency improvement, should be encouraged.

As we have established the need for heat energy utilization in the process industries, another major decision which is worth making is to consider the particular use to which the recovered energy will be put.

In order to utilize waste heat energy in any given process, there is a need for it first to be recovered. Any method of heat recovery technology adopted depends on the use of the recovered waste heat energy. The use of heat energy in a process plant depends on the energy needs of the plant, and this differs from one process to another. Since most heat recovery practices at present occur as retrofit projects, the use of the recovered heat energy can have a significant effect on the total installed cost of the heat recovery system. For example, if recovered heat is used to preheat combustion air feed to the burner, this will reduce the amount of fuel usage in the burner; however, it might also be detrimental if the burner turndown range was not originally designed to cope with the reduction in fuel consumption. Hence, this might result in changes to the burner, and this will definitely affect the economic analysis of the heat recovery system (Reay, 1980).

Despite the technical considerations in applying any given waste energy recovery technique or technology, on most occasions the decision to adopt any given waste heat recovery technique or technology is always at the discretion of top management and thus requires obtaining their support. Such decisions are always based on the energy audit of the process, the level of commercialization of the heat recovery process and the economics. However, good engineering expertise and judgement from a qualified expert can help in making such decisions.

Typical examples of use of waste heat in the process industries include: preheating of combustion air, space heating, preheating of boiler feed water or process integration, space cooling, electricity generation and so on. Hence, an idea of the use to which the waste heat will be put will help in selecting the best waste heat recovery technique for any given process. For example, in a process plant where process integration of the waste heat is the highest priority, the methods of **direct heat recovery** and **cascade heat utilization** are likely to be adopted, while in a process where there is no need to use the waste heat for process integration, then power generation or space cooling may be of higher priority, and thus the use of **Rankine cycles** and **absorption chillers** is likely to be considered.

1.5 Energy Management

Energy is one of the largest controllable costs in most organizations. Reductions in energy consumption will likely lead to reductions in plant operating costs. Thus, the benefits of energy conservation are reflected directly in an organization’s profitability while also making a contribution to global environmental improvement.

Apart from governmental regulations for energy efficiency improvement, the need to conserve energy particularly in industry and commerce is also strongly felt as energy costs takes up a substantial share of the overall operating costs. It is now apparent to organizations that in order to remain competitive, they have to cut their energy costs. Such an approach is cost-effective and the results can be immediate.

There are different drivers which attract the attention of companies to energy efficiency. These drivers can come from the company (i.e. from top managers, problems with quality, processes, production and resources) as well as from outside (i.e. government regulations, the market). However, no matter what the driver is, the energy efficiency technology applied to any process is almost always the sole decision of the top management. Ordinarily, energy is not usually a big cost factor in most companies, accounting for less than 3% of revenue. However, what has made energy a priority driving innovation across many organizations today is that it accounts for most of the company's enterprise carbon footprint, thus shifting energy use from a minor operating cost to a major environmental focus.

Although energy management strategy is beyond the scope of this research, it is worth mentioning that the implementation of any energy-efficiency investment in companies is often very low, and heavily influenced by the top managers' priorities, availability of capital, expected return on investment and so on.

Having seen the importance of implementing waste heat recovery in the UK food processing industries, it can be inferred that through the implementation and optimization of the waste heat recovery technologies in the process industries, the UK can be put on track in achieving their set CO_2 emission targets, while through the proper utilization and management of recovered waste energy, the problems associated with its improper use can be avoided.

This brings us to the topic of this PhD research which is: **"Optimizing Thermal Energy Recovery, Utilization and Management in the Process Industries"**.

The research is undertaken in collaboration with Brunel University, Newcastle University, United Biscuits, Flo-Mech. Ltd, Beedes Ltd and Chemistry Innovation Network (KTN). The process plant under investigation is the United Biscuits' KP Billingham plant used for potatoes crisps/chips production. The research will be concentrated on the frying process in the plant since it has the highest potential to emit waste heat.

1.6 Aim of the Project

As this research involves collaboration with the two other universities mentioned above, its generic aim is to investigate and develop methodologies for the optimum thermal (heat) energy recovery from the industrial waste streams of a food processing industry, as well as to improve the performance of some thermal unit equipment used in the existing plant in order to minimize entropy generation and reduce CO_2 emissions.

In order to achieve the generic aim, each partner was given some specific tasks. This research will concentrate on process modelling, simulation and optimization of the waste heat recovery techniques/technologies adopted.

1.7 Project Objectives

In order to achieve the project aim mentioned above, the following objectives have been adopted in this project:

- To conduct an extensive literature review of the state of the art techniques and technologies used for the recovery of waste heat from the process industries.

- Based on the literature review carried out, to consider the modification of the existing process flow diagram (see Figure 4–1) (in collaboration with Brunel University and the industrial partners) in order to improve plant energy performance without compromising product quality.

- To obtain the estimated waste heat stream composition, quality and quantity from the frying section of the KP Billingham plant (in collaboration with Brunel University and the industrial partners).

- Based on the data obtained from the above step, to select the heat recovery technology that may be used for waste heat recovery in the frying section of the KP Billingham plant in order to improve the energy utilization efficiency and thermal performance of the plant without compromising on the product quality.

- To develop IPSEpro steady state models of the selected waste heat energy recovery processes and validate the models using thermodynamic principles and industrial data from manufacturers.

- To conduct parametric studies with the validated models in order to determine the effect of some selected operating conditions on process performance.

- To conduct entropy generation analysis (second law analysis) of the unit operations of the validated models, to determine the components of the model that introduces the most entropy into the system.

- To carry out a life cycle assessment of the proposed models in order to establish their environmental friendliness and savings on greenhouse gas emissions.

- To perform an economic assessment of the developed models in order to establish their economic viability.

1.8 Structure of the Thesis

This thesis is presented in eight chapters. Each chapter considers a major aspect of the work and is made up of sections and sub-sections. The sections and sub-sections in each chapter are arranged in such a way (deemed suitable by the author) to convey the ideas to the readers in the most appropriate sequential order, in order to enhance understanding of the entire work.

This first chapter is the introduction to the work and covers industrial energy demand, with most emphasis on process industries. It has detailed the research motivation, which seeks to address the need for efficient heat energy recovery, utilization and management in the food and drink processing and chemical industries in the UK. It has also introduced the aims of the project, the methodology adopted and the structure.

The second chapter presents the literature review of some waste heat recovery techniques/technologies. Based on the literature review, the choice of the waste heat recovery technique deemed by the author to be the most useful for the process under study is adopted.

Chapter Three introduces the IPSEpro simulation software and its capabilities and limitations.

The fourth chapter gives the process description of the existing KP Billingham plant and the possibility of applying the state of the art technologies selected from the literature into the existing process in order to improve the EUF of the plant without jeopardizing the product quality.

In the fifth chapter, some thermodynamic theories relevant to the proposed waste heat recovery technique are presented and based on these theories; the model equations of the individual unit operations of the proposed waste heat recovery process are developed using IPSEpro simulation software (IPSEpro MDK).

The sixth chapter covers the modelling of the proposed waste heat recovery technique using the individual model equations developed in the previous chapter and the simulation of the model using the IPSEpro PSE tool. It also covers the entropy generation analysis of the process, as well as sensitivity analysis to ascertain the effect of the plant operating parameters.

In the seventh chapter, the life cycle analysis of the proposed process is carried out in order to establish the environmental and economical benefit of the project in terms of CO_2 emission reduction and profitability analysis respectively.

In the eighth chapter, conclusions are drawn based on the findings obtained in this research. Recommendations for future research are also presented.

Chapter Two

This chapter covers the review of the literature on the state of the art technologies adopted for the recovery of waste heat from the process industries.

2 Literature Review

2.1 State of the Art Waste Heat Recovery Techniques/ Applications

Most of the state of the art techniques used for waste heat recovery application were introduced in the previous chapter. In this chapter, a more detailed literature review is carried out on three retrofit waste heat recovery techniques/applications – Organic Rankine Cycles, Wastewater Desalination, and Absorption Chillers – in order to ascertain the technology that will be adopted in this research. The reason why these technologies were adopted is because there is no immediate need for process heat integration in the plant at present, and thus any waste heat application adopted should be focused on retrofit projects that will help to improve the EUF of the plant. Also, since this work is carried out in the food processing industry which makes use of electricity, refrigeration applications and pure water, the utilization of the waste heat produced for electricity generation, refrigeration application and water purification will be of the utmost importance.

2.1.1 Waste Heat for Power Generation

Electricity is essential and the most used form of energy all over the world both domestically and industrially. Most of the machines and process equipment used in the process industry make use of electricity. Hence, power generation from waste heat will be a welcome technology in this industry.

Amongst the low-grade waste heat to electricity generation technologies reviewed in the literature (Kalina cycles, Stirling cycles, and Organic Rankine Cycles), the Organic Rankine Cycle proves to be more favoured due to its technological advancement and maturity when compared to other cycles. A close competitor to ORC in terms of technological maturity is the Kalina cycle; however, it is highly complex and will likely be more expensive to develop, and also, unlike the ORC system which uses organic fluids and hydrocarbons, which have little or no impact on human health, Kalina cycles make use of an ammonia (poisonous gas)-water mixture as its working fluid, which may be dangerous to human health.

Based on the above facts, the ORC technology has been adopted in this project.

2.1.1.1 Organic Rankine Cycle (ORC)

As explained in section 1.3.3, ORC technology evolved from RC technology. The only difference between the two cycles is the nature of the working fluid used. The latter makes use of steam, whilst the former makes use of an organic fluid. This idea of using an organic fluid was first suggested as far back as 1823 (Leibowitz *et al*., 2006).

Although the cost of conventional steam RC seems to be lower than that of the ORC (Hettiarachchi *et al*., 2006), the ability of the latter to utilize low-temperature waste heat sources makes it a better alternative for low-grade temperature applications. Another major advantage of the ORC over the RC system in terms of the working fluid is that organic fluids tend to have a lower heat of vaporization than water (Figure 1–10) and become superheated more easily using low-grade heat sources, unlike the case of water where a high degree of superheating cannot be achieved using low-grade heat sources. As a result, there is likelihood that vapour droplets will be formed at the exit of an expansion turbine, thus causing erosion of the turbine blades.

A lot of research was presented in the literature on the use of ORC systems for converting waste heat energy to power.

Wei *et al*. (2006) carried out research on the performance analysis and optimization of ORC for waste heat recovery from exhaust heat, and concluded that the quality (temperature and mass flowrate) of waste heat affects system efficiency and net power of the system. They also concluded that output performance of the plant deteriorates under high ambient temperature. The exergy analysis also shows that the evaporator contributes most of the exergy in the system.

Invernizzi *et al.* (2007) presented their work on bottoming micro-Rankine cycles for micro-gas turbines using 16 different organic working fluids which were selected based on their thermal stability and thermodynamic properties. They concluded that working fluids with low molecular complexity tend to be more effective in cooling heat sources and thus tend to recover more waste heat from the exhaust.

Dai *et al.* (2009) found from their work on parametric optimization and comparative study of ORC for low-grade waste heat recovery that it does not always hold that an increase in the turbine inlet temperature will produce a corresponding increase in the turbine power output, especially with working fluids with a non-negative saturation vapour curve (i.e. isentropic and dry fluids).

Lemort *et al*. (2009) developed and validated a model of a scroll expander integrated into an ORC system. They suggested that displacement type machines such as scroll expanders are more appropriate for small-scale ORC units because they are characterized by lower flow rates, higher pressure ratios and much lower rotational speeds than turbo-machines.

Desai and Bandyopadhyay (2009) performed a process integration study of both basic and modified ORC using 16 different organic fluids. They concluded that ***dry fluids*** are the most preferred working medium for the ORC system, which utilizes low-grade heat sources. Their reason for selecting dry fluids was that they show high thermal efficiency and their post-expansion state is always superheated, thus enabling regeneration to improve thermal efficiency. They also found that the thermal efficiency of the ORC system can be improved significantly by simultaneous regeneration and turbine bleeding. They noted that the presence of non-condensable components in the working fluid such as air can pose technical problems related to heat transfer, and this can significantly have an adverse effect on the thermodynamic efficiency of the process.

Wang *et al*. (2010) proposed, designed, constructed and tested the performance of a low-temperature solar organic Rankine system using a rolling-piston R245fa expander, and found that their newly designed R245fa expander worked in a stable manner. They also found that the performance of an evacuated solar collector was better than that of a flat plate collector.

Schuster *et al*. (2010) found from their calculation that the efficiency of the ORC system could be improved by operating in the supercritical region; however, the result of their model was not validated through an experiment.

Yari and Mahmoudi (2010) carried out work on the utilization of exhaust waste heat from GT-MHR generator for power generation using ORCs, with R123 as the working fluid. They concluded that the efficiency of the system increased with the turbine inlet temperature at any given waste heat inlet temperature and turbine pressure ratio.

Gang *et al.* (2010) analysed a low-temperature solar thermal electric generator using a regenerative ORC, and found that for a given constant irradiation, evaporation temperature and environmental temperature, the collector efficiency decreased as the regenerative temperature increased. They also found that the optimum regenerative temperature at which ORC efficiency reached its maximum lay between the condensation and the evaporation temperatures. They concluded that the overall system efficiency was higher for the regenerative cycle than for the non-regenerative cycle.

Vaja and Gambarotta (2010) performed a comparison of three different working fluids (benzene (dry fluid), R11 (isentropic) and R134a (wet fluid)) when used in three different ORC cycle configurations as a bottoming cycle for an internal combustion engine (ICE). They found that fluids with a lower critical temperature caused an increase in the temperature difference between the exhaust gas and the working fluid in the evaporator, and hence gave rise to irreversibility, which had negative effects on system performance. They also established that the performance of dry fluids was always better than that of wet fluids with lower critical temperature.

Chacartegui *et al*. (2009) reviewed a combined system, where ORC with different organic working fluids was used as the bottoming cycle for a modern high efficiency gas turbine, like recuperative gas turbines. They concluded that the combined cycle based on the commercial gas turbine data and ORCs showed that ORCs were an interesting and competitive option when combined with high efficiency gas turbines with low exhaust temperature.

Saleh *et al*. (2007) carried out a thermodynamic screening of 31 pure component working fluids for ORCs, using the BACKBONE equation of state. They found that the thermal efficiency of wet fluids increased significantly when combining superheating with the regeneration system while that of the dry fluids decreased by superheating. They also observed that without the regeneration, the utilization of the available heat source was limited due to a high pinch point temperature.

Liu *et al*. (2002a) investigated the effects of working fluids on ORC for waste heat recovery. They found that the presence of hydrogen bonds in certain molecules such as water, ammonia, and ethanol resulted in wet fluids due to larger vaporizing enthalpy, and are thus regarded as inappropriate for ORC systems. They also concluded that the thermal efficiency for working fluids is a weak function of the critical temperature. Their findings also agree with that of Vaja and Gambarotta (2010), who concluded that the thermal efficiency was lower for working fluids with lower critical temperature. They confirmed that the maximum value of total heat recovery efficiency occurred at the appropriate evaporating temperature which lay between the inlet temperature of waste heat and the condensing temperature. They also found that the maximum value of total heat recovery efficiency increased with the inlet temperature of the waste heat; however, it could be decreased when a working fluid with lower critical temperature was used.

Hung *et al*. (2010) performed a study to investigate the suitability of 11 different organic working fluids for an ORC system used for the recovery of low-grade waste heat from a solar pond or an ocean thermal energy conversion (OTEC) system. They calculated the efficiency of the ORC system based on the assumption that the working fluid entered the turbine as saturated vapour. They found that the three factors of the fluid which had a major impact on the system performance of an ORC were the slope of the saturation curve, the specific heat, and the latent heat. They concluded that wet fluids with very steep saturated vapour curves in T-s diagram had a better overall performance in energy conversion efficiencies than that of dry fluids.

Wang and Zhao (2009) investigated the performance of a low-temperature solar-powered ORC system, using three different zeotropic compositions of organic fluid R245fa/R152a. They assumed that due to the inherent temperature glide associated with zeotropic mixtures during phase change, an internal heat exchanger (IHE) needed to be included in the system. They found that unlike pure fluids, isentropic zeotropic mixtures showed the lowest Rankine cycle efficiency. Their investigation also showed that an increase in thermal efficiency could be achieved by combining superheating with an IHE.

Mago *et al*. (2008) compared the performance of a regenerative ORC and a simple ORC system using four different dry organic fluids. They found that for each working fluid, the regenerative ORC showed a better thermal efficiency than the simple ORC. Their first and second law analysis also showed that the regenerative ORC reduces the system irreversibility and increased the second law efficiency. It also reduced the quantity of heat required to produce the same power. Their findings also confirmed the fact that dry fluids do not need to be superheated, since superheating reduces thermal efficiency and increases system irreversibility. They also found that the higher the boiling temperature of the dry organic fluid, the higher the thermal efficiency of the ORC.

Kaikko *et al*. (2009) compared the performance of an Air Bottoming Cycle (ABC) with that of an ORC (using toluene as the working fluid) when both are used as bottoming cycles for a small-scale (7.8 MW) gas turbine with an exhaust temperature of 534^{o}C and a large-scale (16.8 MW) diesel engine with an exhaust temperature of 400^{o}C. They found that under power generation mode, the ORC system always demonstrated a better performance than the ABC at exhaust temperature levels up to 680^{o}C, whilst the ABC dominated at higher temperatures. However, for the cogeneration of power and heat, the electric efficiency of both the ABC and the ORC are quite close to one another.

Angelino and Colonna di Paliano (1998) studied the use of multi-component working fluids for ORC systems, and concluded that the ORC represented an effective heat-conversion device in many energy fields, and that its performance could be improved by using multi-component zeotropic mixtures as the working media.

Quoilin (2007) performed an experimental study and modelling of a low-temperature organic Rankine Cycle for small scale cogeneration. Of all the organic fluids tested, he concluded that R123 was the best adapted for a hot source temperature between 100 and 200^{o}C. He also concluded that scroll expanders were better for small-scale units because of their robustness in two-phase flow conditions.

Doty and Shevgoor (2009) presented their research on improving the efficiency in the conversion of dual low- (from geothermal sources) and mid-grade (from concentrated solar power) heat sources using a dual heat source ORC system with isobutane as the working fluid. Their simulation result showed that such systems show a good economic advantage for reducing the cost of renewable energy.

Aneke *et al*. (2011b) developed a validated model of the Chena, Alaska, USA geothermal power plant using the IPSEpro simulation tool. They found that variations in the geothermal source temperature affect the power output of the plant, as well as the state of the working fluid in both the turbine inlet and condenser outlet. Hence, they advised that ORC systems should incorporate a great deal of control in order to avoid cavitation in the pump as well as the reduction in plant performance.

Apart from the simulation studies presented above, there are also real-life, commercially operating ORC plants designed for either waste heat recovery or geothermal application.

Alford (2005) and Nasir *et al*. (2004) presented a report on the ORMAT® Energy Converter (OEC) which generates electricity by using ORC technology to convert waste heat energy produced by a pair of gas turbines used to drive natural gas compressors, at the Neptune natural gas processing plant in Centerville, USA. The plant, which makes use of n-pentane as the working fluid, was installed in 2004 and was recognized as the first of its kind in the USA. It has an installation capacity of 4.5 MW. The plant has made the gas processing process to be self-sustainable, because interruptions in the purchased power, which happens to be the only source of power to the gas processing unit, no longer affect the gas processing facility. Furthermore, it has become a new source of revenue to the company, since the excess of electricity produced is sold to a local energy company.

Mettler (2006) presented a report on three Recovered Energy Generation (REG) power plants (similar to the OEC) each with a capacity of 5 MW_{net} developed by ORMAT to be used on the Alliance Pipeline, operated by an independent power producer in Western Canada, for the conversion of waste heat from the exhaust of existing gas turbines into electricity.

Like the Canadian company, there are other companies who entered into agreement with ORMAT. In 2006, an ORMAT subsidiary entered into a 20-

year power purchase agreement with Puget Sound Energy for the supply of power from a REG system located close to the Sumas compressor station of Northwest Pipeline, Inc. in Sumas, Washington (Mettler, 2006).

Apart from waste heat recovery from gas processing facilities, there are other processes where ORMAT has proved that the OEC is commercially feasible and economically viable. For example, Legmann and Citrin (2004) presented a report on the application of OEC for the conversion of low-temperature waste heat from the clinker cooler air of the HeidelbergCement manufacturing plant into electricity. The plant, which is located at Lengfurt, Germany, has been reported to meet not only HeidelbergCement's design criteria but also to cope automatically and seamlessly with wide fluctuations in heat source temperatures and flow (Legmann and Citrin, 2004).

Furthermore, this technology has also been implemented in the recovery of waste heat from ship exhausts. For example, Siemens and United Arab Shipping Company entered into an agreement for the former to provide waste heat recovery system for the latter's ships (Siemens-AG, 2009). The waste heat recovery system will use RC technology to convert the waste heat from the ship's exhaust to electricity. A similar fit with even higher recovery efficiency can be achieved using the ORC system.

ORC technology has also come of age in the world of electricity generation from geothermal heat sources. Holdmann (2007) presented a report on the performance of a 200 kW ORC geothermal power plant at Chena, in Alaska, USA, developed by the United Technologies Corporation. The plant uses R134a as the working fluid, with its heat source from a low-temperature geothermal at 73°C. The logged 3000 h performance report indicated that the plant produced about 578,550 kWh of electricity at 95% availability. This displaced about 44,500 US gallons of diesel fuel, which was formerly used for power generation before the introduction of the ORC system.

There are also some patented works on the use of the ORC system for power generation from waste heat. Sami (2010) patented a work on the use of ORC systems for power generation from a low-waste exhaust using refrigerant mixtures as the working fluid. Juchymenko (2009) also has a patent on the use of ORC for power generation in order to improve the energy efficiency of a process.

Other applications, some of which are still in the pilot phase, which make use of the ORC technology for electricity generation from low temperature heat sources, include: the 100 kW pilot plant known as Granex, which was developed by a team of researchers from the University of Newcastle, Australia Priority Research Centre for Energy and Granite Power Pty Ltd (Hamilton, 2009), the 5 kW pilot plant developed by Ener-G-Rotors for the conversion of low-grade heat to electricity (Lozanova, 2009), etc.

All the commercially operating ORC systems developed by ORMAT and discussed above make use of an indirect evaporation, in which the waste heat is used to heat the ORC fluid indirectly by first passing the heat to thermal oil or water, which is then used to preheat and vaporize the working fluid. This adds not only to the cost of the system but also to the irreversibility. In order to eliminate this, General Electric (GE) is currently working out ways of modifying the system to make use of direct evaporators

(Guillen, 2008), which they have identified will help to reduce the cost of ORC systems by 20%. Other ways of improving the economic advantage of the ORC cycle have also been identified. For example, Brasz *et al*. (2005) demonstrated that the cost of an ORC could be drastically reduced by adapting some of its hardware from the air conditioning equipment. This approach is currently being used by the United Technologies Corporation in close cooperation with the Carrier Corporation, under the trademark name PureCycle™200 (Brasz *et al*., 2005). The technology has been successfully applied in many commercially operating ORC power plants in the USA. They include: Chena Geothermal ORC power plant in Chena, Alaska, which makes use of a geothermal heat source (Holdmann, 2007); an ORC power plant in East-Hartford, Connecticut, which makes use of waste exhaust heat from a Pratt and Whitney FT12 gas turbine; an ORC plant in Austin, Texas, powered by heat from a landfill flare; and another at Danville, Illinois, powered by exhaust heat from three Jenbacher reciprocating engines. Another approach is to make use of a dual heat source (at different temperatures) to power a single ORC plant. This approach has been identified to have a greater economic advantage than two single ORC systems each powered by a single heat source (Aneke *et al*., 2011a).

From all the research on ORC systems for power generation reviewed in this thesis, it can be observed that the basic principle of the ORC system is the same. The only difference is the source of the waste heat, the configuration, the kind of expander and the nature of the working fluid used in the system.

2.1.1.2 Concluding Remarks

From the literature review conducted here, it can be concluded that the use of ORC for power generation from a low-grade waste heat source is technically and commercially feasible, as well as economically viable. Commercial ORC systems have been in the market since the beginning of the 1980's and have been providing waste heat recovery (WHR), biomass combined heat and power, geothermal and solar solutions (see Figure 2–1) in a broad range of power and temperature levels, as shown in Table 2-1.

From the data provided by the ORC manufacturers (see Figure 2–2), it can be inferred that the installed power and the number of plants in operation show an exponential growth over the years. This shows that the market has grown at a rapid pace ever since the first installation in the 1980s. However; there are still issues that are hindering the growth of the technology, which include the economics of scale of the process, the scepticism by some top decision-makers in the industry, and the neglect of waste heat recovery by both governments and key industry decision-makers. As rightly said by Aries, it is not easy to convince people to buy it (Alford, 2005). This particular problem is still persistent to date and is hindering the wide acceptance of the technology in the industrial sector. This is part of what prompted GE to embark on projects that would improve the economic advantage of ORC technology (Guillen, 2008).

Table 2–1: Main ORC Manufacturers (Quoilin and Lemort, 2009)

MANUFACTURER	APPLICATIONS	POWER RANGE	HEAT SOURCE TEMPERATURE	TECHNOLOGY
ORMAT, US	Geothermal, WHR, Solar	200 kWe–72 MWe	150–300^{o}C	Fluid:n-pentane
Turboden, Italy	CHP, Geothermal	200 kWe–2 MWe	100–300^{o}C	Fluids:OMTS, Solkatherm Axial turbine
Adoratec, Germany	CHP	315 kWe–1.6 MWe	300^{o}C	Fluid: OMTS
GMK, Germany	WHR, Geothermal, CHP	50 kWe–2 MWe	120–350^{o}C	300 rpm multi-stage axial turbine (KKK), Fluid: GL160 (GMK patented)
Koehler-Ziegler, Germany	CHP	70–200 kWe	150–270^{o}C	Fluid:Hydrocarbons, Screw expander
UTC, US	WHR, Geothermal	280 kWe	>93^{o}C	PureCycle
Cryostar	WHR, Geothermal	n/a	100–400^{o}C	Radial inflow turbine Fluids: R245fa, R134a
Freepower, UK	WHR	6 kWe–120 kWe	180–225^{o}C	
Tri-o-gen, Netherlands	WHR	160 kWe	>350^{o}C	Turbo-expander
Electratherm, US	WHR	50 kWe	>93^{o}C	Twin screw expander
Infinity Turbine, US	WHR	250 kWe	>80^{o}C	Fluid:R134a Radial Turbo expander

However, with more incentives and supporting policies from governments (such as allowing companies to sell their generated electricity to the national grid, as currently being practised by an ORMAT subsidiary in the US (Mettler, 2006), mandating process industries to implement waste heat recovery technologies, providing subsidies and no-interest loans to companies for the implementation of waste heat recovery projects), this technology will be highly welcomed in the process industries.

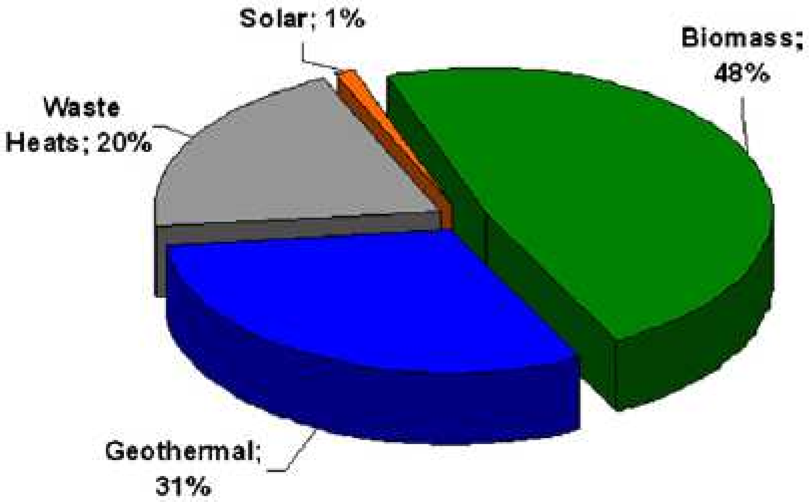


Figure 2–1: Share of Each Application in the ORC Market (Velez *et al*., 2012).

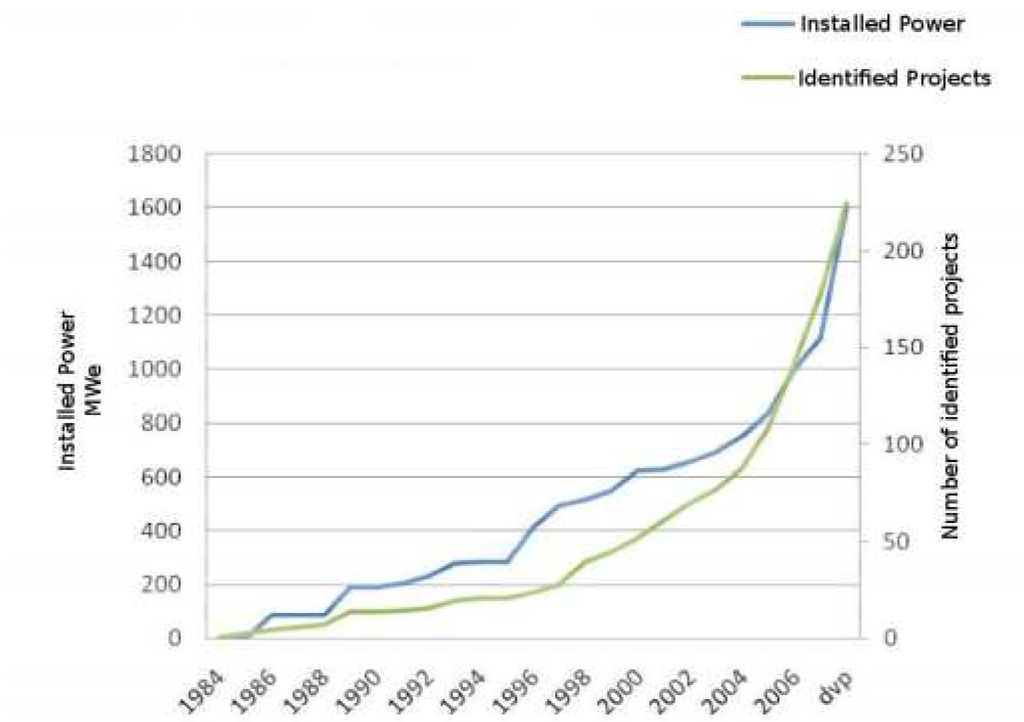


Figure 2–2: ORC Market Evolution (Velez *et al*., 2012)

2.1.2 Waste Heat for Wastewater Desalination

The importance of water for life can never be overemphasized. Water is an essential commodity in the day-to-day life of every individual. Water, like every other natural resource, is scarce and thus should be conserved. Its scarcity has been recognized as a major threat to humanity throughout the world (Fritzmann *et al*., 2007). Apart from its necessity in our individual lives, it is also useful in every process industry. In these industries, water helps in washing, mixing, dissolving, soaking, dilution, cooling, heating, separation, etc. Most of the time, the use of water reduces its quality and thus is referred to as ***wastewater***. This wastewater needs to be purified before it can be reused. There are many reasons why wastewater might be considered for reuse or recycling. This may be as a result of environmental protection, government legislation or economics (Judd and Jefferson, 2003).

It is a well-known fact that the reuse and recycling of wastewater conserves the supply of fresh water. Also, the unavailability or limited supply of fresh water in many parts of the world has also resulted in purification of low-quality seawater, wastewater, and brackish water for fresh water production.

Considering water reuse opportunities in industry, there is always a distinction between ***reclamation*** and ***recycling***. Reclamation is regarded as the recovery and treatment of water to make it available for reuse, while recycling is the recovery and reuse of water (whether or not subject to treatment) from a discrete operation (Judd and Jefferson, 2003). Water reclamation is believed to have been in existence for centuries; however, modern-day legislation dates as far back as 1956 in Japan, when the

Industrial Water Law was introduced to restrict the use of groundwater by the rapidly growing Japanese industries (Judd and Jefferson, 2003). This was followed by the introduction of the Federal Water Pollution Control Act of 1972 in the USA, and from there different countries and regions in the world started adopting policies for natural water protection.

In industry, the decision for water reuse or recycling is usually based on the reliability and cost-effectiveness of the process to provide water of the desired quality. Most of the time, the cost benefit is largely or wholly determined by statutory requirements; for example, some industries may have zero liquid discharge imposed upon their operation; hence, in such cases wastewater recovery or reuse is no longer an option but an absolute necessity. There are also cases where the decision is solely based on economics. In this scenario, the total cost of purification to produce water of the desired quality is always considered against the cost of freshwater supply and wastewater discharge.

In the industrial context, direct ***"closed loop"*** industrial water recycling is currently attracting greater interest and is being applied more often than the municipal system of water reclamation, especially in processes where other resources are recovered in addition to water (Judd and Jefferson, 2003). However, two major factors militate against its widespread application in some processes. Firstly, most industrial processes involve a number of individual operations that give rise to wastewaters of certain compositional ranges. These individual effluent streams are generally combined to produce wastewater whose resultant temporal variation in quality is immense, thus causing a significant challenge to any treatment process that is to provide water of a reliably high quality; and secondly, most of the conventional sewage treatment works have the capacity to treat industrial wastewater at a cost that is considered reasonable simply by blending with domestic water, which leads to the significant dampening of the effects of the broad temporal variation in industrial wastewater quality (Judd and Jefferson, 2003). These two mitigating factors are not likely to be a problem in water effluents from a potato crisps/chips manufacturing plant (which is the focus of this research), where there is little variation in process operations and in which water is mainly used for physical processes, such as the washing application, with little contamination level. Hence, the direct ***closed loop*** industrial water reclamation process will be considered in this research.

2.1.2.1 Industrial Water Use

Industry is believed to account for about a quarter of all water consumption (Judd and Jefferson, 2003). Water is to fish as it is to an industry, meaning that there is virtually no industry that does not require large volumes of water. There are various sources through which industries can obtain water in order to meet their high demand. This mainly depends on the location of the industry, and sometimes the statutory regulations in existence where the industry is located. Some industries abstract water from rivers and boreholes; however, the majority still get their water from public water supplies (especially in developed countries where there is an adequate pipe-borne water supply) which has been treated to potable quality standards. Despite the high potable standard, some industries require further treatment of the water in order to reduce the mineral and organic material content, according to the specific application to which it is to be put. Table 2–2 and 2–3 respectively show general water quality standards and the specific water quality requirements for various industries.

Table 2–2: General Industrial Water Quality Standards adapted from (Judd and Jefferson, 2003)

Class:	**3**	**4**	**5**	**6**	**7**	**8**	**9**
Type:	**Softened**	**Dealkalised**	**Deionised**	**Purified**	**Apyrogenic**	**High Purity**	**Ultrapure**
Conductivity, μS/cm			20	5	5	0.1	0.06
Resistivity, MΩ cm			0.05	0.2	0.2	10	18
TDS, mg/l			<10	<1	<1	0.5	0.005
pH			5.0 – 9.5	6.0 – 8.5	6.0 – 8.5	6.5 – 7.5	
Hardness, mg/l $CaCO_3$	<20		0.1	<0.1	<0.1		0.001
Alkalinity, mg/l $CaCO_3$		<30					0.001
Ions, mg/l							0.001
Silica, mg/l			0.5	0.1	0.1	<0.01	0.002
TSS, mg/l			<0.1	<0.1	<0.1	<0.1	ND
Turbidity, NTU			<0.5				
SDI			<5	<3	<3	<1	<0.5
Particle count, no./ml				1	1	1	0.1
COD, mg/l				<0.1	<0.1		
TOC, mg/l							0.05
Microorganisms, cfu/ml				<10	<1	<1	<1
Pyrogens, EU/ml					<0.25		<0.25

Many methods have been used for the purification of low-quality water, both on a commercial and a pilot scale. Most of these techniques make use of heat either directly or indirectly. Some of the methods include: multi-effect distillation (MED), multi-stage flash (MSF), electro-dialysis (ED), membrane distillation (MD), and reverse osmosis (RO).

Table 2–3: General Water Quality Requirements for Specific Applications (Judd and Jefferson, 2003)

Use	Industry	Application	Quality
Fire fighting	All		1 (natural)
Irrigation	Agriculture		1 (river)
Domestic	Offices	Drinking water	2 (potable)
	Hotels/catering	Down services	2/3
	Healthcare	Laundries	3
Steam raising	Process industries	Heating	3/4
		Steam stripping	3/4
		High-pressure steam	5
	Power generation	Turbine drive	8
Heat transfer	Manufacturing	Closed heating and cooling systems	3/4
	Process industries	Open recirculatory cooling systems	3/4
	Offices	Air conditioning	5
	Hotels/catering		
	Healthcare		
Process water	Heavy chemicals	Product washing	2/3
	Fine chemicals	Solvent	4
	Food/soft drinks	Bottle/container washing	5
	Brewing	Cleaning in place (CIP)	2
	Pharmaceuticals		6/7
	Metal finishing		5
	Photographic		5
	Laboratories		8
	Semiconductor	Ultrapure water	9
Product	Food/soft drinks	Product quality	4
	Brewing	Shelf life	2/4
	Pharmaceuticals	Parenterals	7
	Cosmetics	Lotions/liquids/topicals	5/6

However, since this project is based on waste heat recovery (WHR) systems, only processes that require the use of heat energy (MED, MSF, MD and RO), either directly or indirectly, are considered here.

2.1.2.2 Waste Heat Based Wastewater Purification Techniques

MED is among the traditional wastewater/seawater desalination methods. Its application to seawater purification dates as far back as the 19^{th} century (Van der Bruggen and Vandecasteele, 2002). Its principle is based on heat transport from condensing steam to seawater/wastewater in a series of stages or effects.

This is an example of a direct heat recovery system, and for waste heat recovery application, the waste heat can be used to produce steam (primary steam) which is passed through the first effect, where it is condensed, and in so doing evaporates the preheated seawater, thus giving rise to the secondary steam, which goes into the second effect, operated at a slightly lower temperature and pressure than the first (Van der Bruggen and Vandecasteele, 2002). The primary steam condensate is then sent back to the waste heat recovery heat exchanger. A schematic diagram showing the principle of the process is shown in Figure 2-3.

There are a lot of problems associated with this process, including: corrosion and scaling of oversaturated compounds on the heat transfer surfaces, which cause fouling and thus a reduction in heat exchanger effectiveness and process performance. As a result of this problem, the number of effects is limited by a maximum temperature of about 120^{o}C in the first effect, and the minimum temperature that allows heating of the incoming water in the last effect.

Hence, considering the low nature (heat content) of the waste heat obtainable in a food processing plant under study (120–164^{o}C) (Wu, 2009) and the high energy demand of the MED process, this technology will not be an appropriate one to adopt in this research.

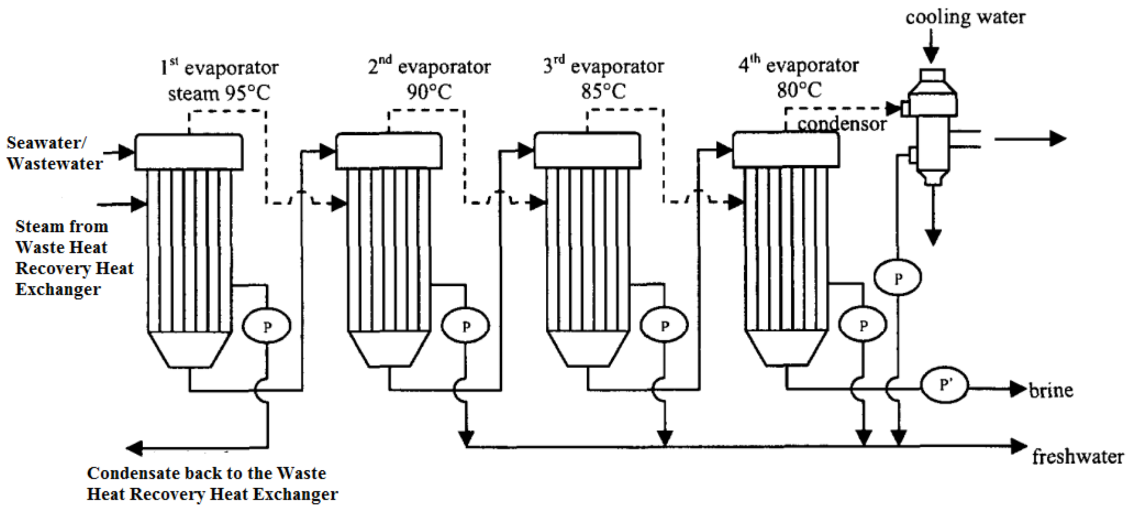


Figure 2–3: Principle of MED

The MSF process for seawater/wastewater desalination came into existence in the 1960s, and became the most common process for water purification due to its reliability and simplicity. Its principle is based on a series of flash chambers where steam is generated from saline feed water at a progressively reduced pressure, as shown in Figure 2-4 (Van der Bruggen and Vandecasteele, 2002). The steam generated in the chambers is condensed as a result of heat exchange within a series of closed pipes which contain the seawater to be desalinated, and in so doing the seawater is preheated as well. The condensed steam, which is the primary product, collects in the trays contained in the chamber.

In this process, heat exchange with the wastewater does not occur through heat transfer surfaces, hence there is a reduced risk of scaling. It is also easier to control corrosion as compared to MED. However, due to the indirect heat transfer, MSF has a lower performance and consumes more energy than MED (Van der Bruggen and Vandecasteele, 2002).

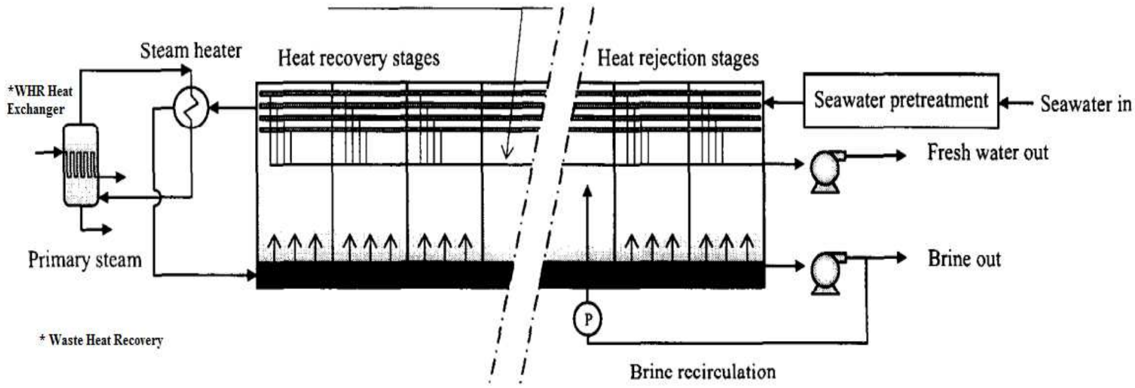


Figure 2–4: Principle of MSF

Furthermore, considering the low-grade nature of heat from the potato crisps processing plant under investigation, this technology will not be economic for a wastewater purification application.

MD is a relatively new technology. It is an evaporative process, using a porous hydrophobic membrane, which physically separates the aqueous liquid feed from the gaseous permeate based on the vapour pressure gradient (Wirth and Corinne, 2002). A typical flow diagram of an MD process is shown in Figure 2-5. The feed stream is sent to a heat exchanger where it is preheated using the heat from the vaporized permeate. The heated feed stream is then pumped through a heater where it is vaporized. The vaporized (permeate) part of the feed stream is drawn through the hydrophobic membrane using a vacuum pump. The permeate is then used to preheat the feed stream in the heat exchanger before being passed through a cooler where it is condensed. The retentate then leaves through the retentate channel.

The method used to achieve the vapour pressure is what characterizes the four different kinds of MD configurations (Drioli *et al.*, 2006). The most common arrangement is known as Direct Contact Membrane Distillation (DCMD) and it involves a direct contact between the condensing fluid and the membrane on the permeate side.

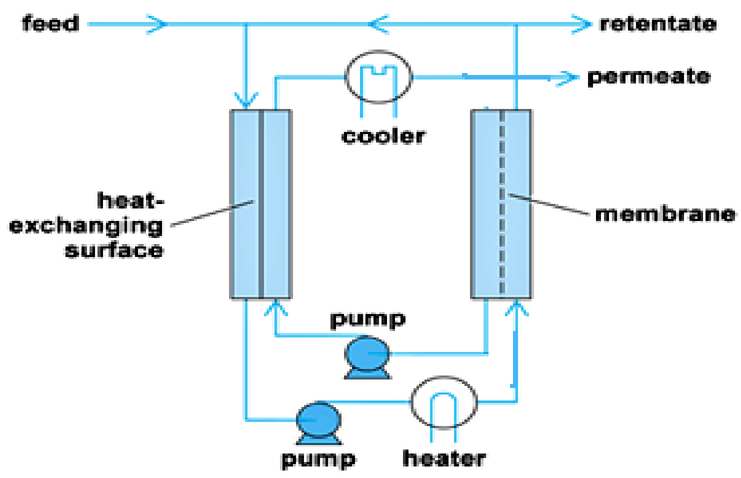


Figure 2–5: Principle of MD

Alternative methods differ in the way the vaporized fluid is recovered, and include the recovery of the vaporized solvent on the condensing surface separated from the membrane by an air gap (AGMD), vacuum (VMD), or removed by a sweep gas (SGMD). The driving force is linked to both the partial pressure gradient and the thermal gradient between the two sides of the membrane, and it is usually characterized by the vaporization of the more volatile compounds at the liquid/vapour interface and diffusion of the vapour through the membrane pores, according to the Knudsen Mechanism (Wirth and Corinne, 2002, Drioli *et al*., 2006). The nature of the driving force together with the hydrophobic character of the membrane allows – at least theoretically – the complete rejection of non-volatile solutes like macromolecules, colloidal species, ions etc. (Drioli *et al*., 2006). The required feed temperature varies from 30 to 50^oC, thus permitting the efficient recycling of low-grade or waste heat streams, as well as the use of alternative energy sources like solar, wind or geothermal. Unlike RO systems, it does not suffer from concentration polarization. It achieves high permeate recovery factors or retentate concentration (Drioli *et al*., 2006). However, it has some shortcomings when compared with RO systems such as lower permeate flux, higher energy consumption and higher costs. It is mainly implemented as a supplement for the treatment of the rejected water in the RO process (Liu *et al*., 2008, Drioli *et al*., 2006) because of its high separation performance (Xu *et al*., 2005).

The water purification technologies presented above involve the direct use of heat for desalination applications. However, heat can also be used indirectly for wastewater treatment. A typical example is the ORC-driven RO system.

The RO process is a general and widely applicable technique for separation, concentration or fractionation of inorganic or organic substances in aqueous or non-aqueous solutions, by letting the fluid mixture flow under pressure through an appropriate porous membrane, and withdrawing the membrane-permeated product (which is enriched in one or more constituents of the mixture) generally at atmospheric pressure and surrounding temperature

(Agrawal and Sourirajan, 1969). A schematic diagram of the RO desalination process showing the detailed parts of the membrane is shown in Figure 2–6.

The process involves no heating of the membrane and no phase change in the product recovery. The first successful RO application in water desalination technology was in brackish water desalination and the first large-scale plant was constructed in the late 1960s (Van der Bruggen and Vandecasteele, 2002). Ever since its first application in brackish water desalination, advances in membrane technology have led to higher permeability, which made RO systems become competitive with the classical distillation techniques. The RO process requires the feed stream to be pumped through a semi-permeable membrane with a pressure higher than the osmotic pressure of the feed stream (Van der Bruggen and Vandecasteele, 2002, Agrawal and Sourirajan, 1969, Fritzmann *et al*., 2007, Sherwood *et al*., 1965).

Currently RO is by far the most widespread type of membrane-based water desalination process, and it is capable of rejecting nearly all colloidal or dissolved matter from an aqueous solution, producing a brine concentrate and a permeate which consists of almost pure water (Fritzmann *et al*., 2007). It also consumes the least energy (in the form of electrical energy used to drive the high pressure pump (HPP)) when compared with the other desalination processes discussed previously (Fritzmann *et al*., 2007). The energy consumption is usually in the range of 0.4–7 kWh/m^3.

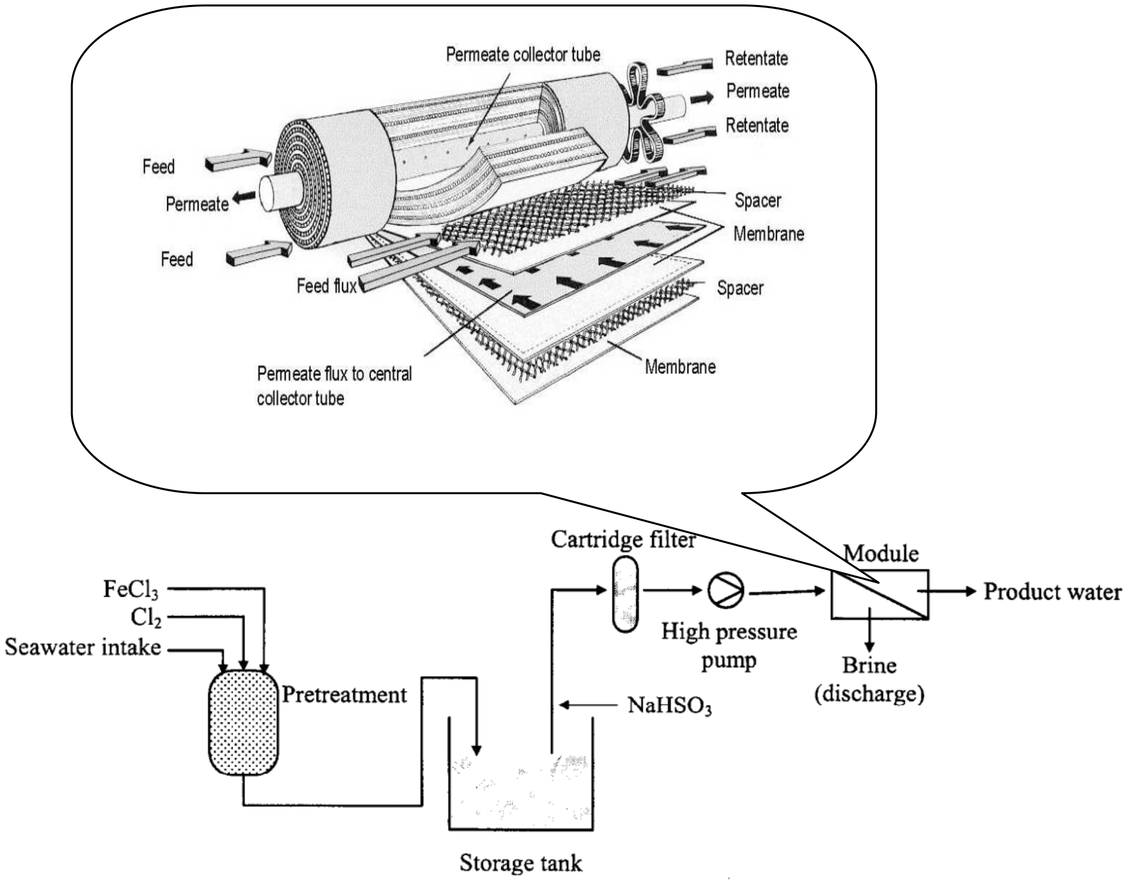


Figure 2–6: RO Desalination Principle

As a result of the low energy consumption, RO desalination is now the most widely applied desalination process in Europe (see Figure 2–7) as well as the least cost-intensive process for water desalination when compared with thermal distillation processes. The world's largest RO desalination plant, in Ashkelon, Israel, achieves a production water price of 0.53 US$/m^3. This achievement is attributed to the technological improvement of membranes, economy of scale, improvement of pre-treatment options and the application of the energy recovery option.

There are some limiting factors in RO desalination applications (see Figure 2-8). This includes fouling, scaling and membrane deterioration;

however, the effect of these factors has been seriously reduced and is still being reduced through some advancement in water pre-treatment technology.

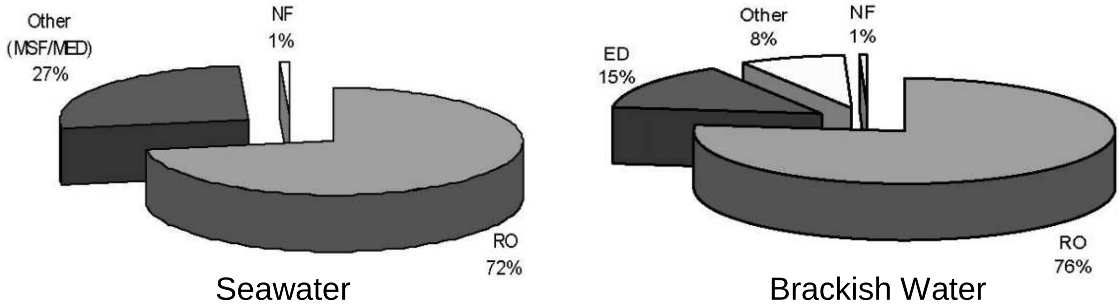


Figure 2–7: Market Share of the Different Desalination Technologies of Seawater and Brackish Water in Europe for Plants with Capacity of at least 700 m^3/d (Fritzmann *et al.*, 2007)

The different pre-treatment processes will not be considered in this research; however, more details can be found in (Fritzmann et al., 2007, Van der Bruggen and Vandecasteele, 2002, Chapman-Wilbert, 1993, Brehant et al., 2002, van Hoof et al., 2001, Rautenbach et al., 1997, Bonnelye et al., 2004, Van Houtte et al., 1998).

As mentioned earlier, the RO system makes use of energy in the form of electrical or mechanical energy to drive the HPP. Hence, it can be incorporated in a system capable of generating electrical or mechanical energy. Studies on the integration of RO with wind turbines have been carried out by some researchers (Van der Bruggen and Vandecasteele, 2002, Miranda and Infield, 2002, Liu *et al*., 2002b). Also, photovoltaic-powered RO systems have been investigated by different authors (Joyce *et* *al.*, 2001, Fiorenza *et al.*, 2002, Thomson and Infield, 2002). Most of the investigations were done in pilot scale.

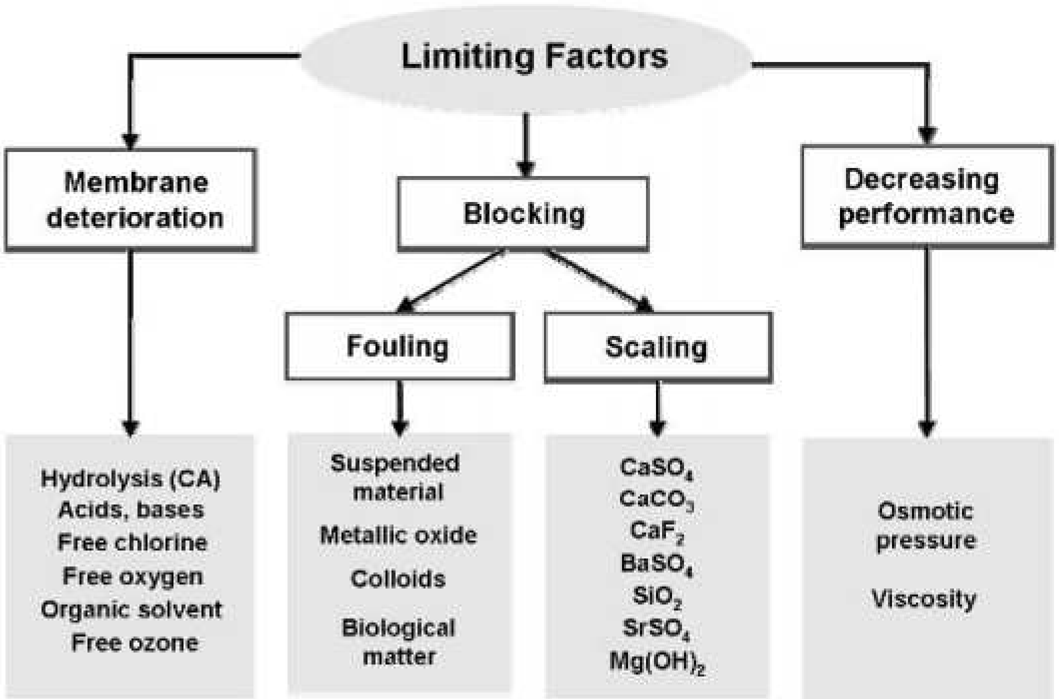


Figure 2–8: Factors Limiting RO Processes (Fritzmann *et al*., 2007)

Thus, since the RO system only needs an energy source (in the form of mechanical or electrical energy) to operate the HPP, this then means that any system which is capable of providing the required energy can be used to drive the RO process. As shown in section 2.1.1, ORC systems are capable of producing mechanical or electrical energy using heat energy, and thus can be used for water desalination by incorporating a RO process.

2.1.2.3 ORC Driven RO System for Wastewater Desalination Application

As mentioned earlier, ORC systems are capable of producing electricity or mechanical energy using different heat sources. Hence, the mechanical or electrical energy from the ORC can be used to power the HPP in order to achieve wastewater desalination in a food processing plant.

Some research has been done on the use of ORC systems to drive RO desalination systems, a few of which are presented below:

Manolakos *et al.* (2009) evaluated the performance of a small scale, low-temperature ORC system coupled with an RO desalination unit in a laboratory-scale experiment. They concluded that ORC could be effectively used to exploit low-temperature thermal sources (in the range of 40 to 70^{o}C) for the desalination of sea or brackish water through the RO process.

Kosmadakis *et al.* (2009) designed a two-stage ORC system for RO desalination process, using R245fa for the top cycle and R134a for the bottom cycle, and concluded that a two-stage ORC could be used efficiently to recuperate heat and produce fresh water.

Figure 2–9 shows the plant investigated by the author during a research visit to the Agricultural University of Athens. The plant, which is in a pilot scale, makes use of a solar-powered ORC unit to drive a RO desalination unit for water purification.

2.1.2.4 Concluding Remarks

From the review, it can be concluded that water purification using an ORC-driven RO desalination system is a low-energy process and is also economically viable. Hence, waste heat can be used to drive an ORC system which can be used either for power generation or water purification, depending on the most pressing need of the plant under investigation. Thus, using waste heat for water purification using the RO desalination technique can be seen as an indirect use of waste heat for wastewater purification.

Figure 2–9: Solar-Powered ORC Unit used to Drive RO Desalination Unit (Pilot Plant at the Agricultural University of Athens, Greece)

2.1.3 Waste Heat for Cooling or Refrigeration Applications

2.1.3.1 Absorption Chillers (ACs) or Refrigeration (AR)

As introduced in section 1.3.3, ACs systems make use of heat to achieve a cooling effect. In a typical ACs system (see Figure 1–8), the waste heat passing through the generator is used to vaporize the refrigerant contained in the transport medium. The refrigerant is passed through the condenser, where it is condensed, and then through the valve to lower the pressure

The low-pressure refrigerant is evaporated in the evaporator to produce a cooling effect. The vaporized refrigerant is reabsorbed in the absorber by the transport medium to form a solution. The solution is pumped back into the generator, and the cycle continues.

ACs can be classified as half effect, single effect, double effect, triple effect and multiple effect systems (ASHRAE, 2001, Zogg *et al*., 2005, Goodheart, 2000, Bailey, 2009). The classification depends on the number of cycles as well as the number of the four basic heat exchangers (generator, absorber, condenser, and evaporator) present in the entire cycle. ACs systems can also be either water-cooled or air-cooled (Liao, 2004).

There are two major kinds of refrigerant-sorbent mixture commercially used in ACs manufacturing. They include:

- **Ammonia – Water ($NH_3.H_2O$) Mixture**
- **Water – Lithium Bromide ($H_2O.LiBr$) Mixture**

As the name implies, the ammonia-water mixture ACs system makes use of ammonia as the refrigerant and water as the sorbent, while the water-lithium bromide mixture uses water as the refrigerant and lithium bromide solution as the sorbent.

Since NH_3, with a normal freezing point of -77.73^oC, serves as the refrigerant in the $NH_3.H_2O$ ACs system, this system can be used for refrigeration, air-conditioning application and freezing applications. However, in the $H_2O.LiBr$ ACs system, which uses H_2O as the refrigerant, with a normal freezing point of 0^oC, the system cannot be used for the freezing application, to avoid the freezing of H_2O in the system.

NH_3 is a poisonous gas and its use is highly regulated, especially in areas where human beings might be exposed to it. Because of the environmental unfriendliness of the use of ammonia, superior performance of ***H_2O.LiBr*** ACs system and the complexity of ***$NH_3.H_2O$*** cycle which occurs as a result of imperfect separation of NH_3 from H_2O, heat driven H2O.LiBr ACs system is adopted in this report .

Much research has been conducted on the use of heat energy for the production of refrigerating and air conditioning effects through H_2O.LiBr ACs.

Kececiler *et al.* (1999) conducted an experiment on the thermodynamic analysis of a H_2O.LiBr absorption refrigeration (AR) system powered with geothermal heat energy from the hot spring in Sivas, Turkey. Considering the very low temperature of the geothermal heat source, they concluded that it was more economical to use the geothermal heat for H_2O.LiBr AR systems for storing at $4–10^oC$ than to use it for electricity generation.

Tsoutsos *et al.* (2003) carried out an economic viability analysis of solar cooling (air-conditioning) in Greece using an H_2O.LiBr ACs system. They found that using this system helped to reduce the demand on electricity (usually generated using the hydro system) for the powering of a vapour compression air conditioning application, especially in the dry season. However, they did admit that considering the high initial investment cost of such projects, there was a need for legislations to promote its implementation.

Mittal *et al.* (2005) modelled a solar-powered H_2O.LiBr absorption air conditioning system using the weather conditions at Bahal (Haryana), India. Their simulation results showed that variations in the inlet hot water temperature to the generator affected the surface area of the system components (generator, evaporator, absorber and condenser). They also found that an increase in the hot water temperature increased the coefficient of performance (COP) and decreased the surface area of the system components and vice versa.

Rafferty (undated) found that the COP and capacity of an ACs system are mostly affected by the generator heat input conditions. He also confirmed that it could also be affected by other variables, such as the condenser and chilled water temperature and flow rates. He found that the performance of the basic H_2O.LiBr ACs system could be improved by operating the chillers' input stage at constant temperature rather than constant pressure, since the former tends to lower the thermodynamic irreversibility in the cycle.

Younes *et al.* (2005) carried out an optimal design and economic study of a solar air conditioning ACs system. They concluded that the machine would be very economical for application in Lebanon. Their analysis showed that for a 2110 kW capacity, the payback period was about six years.

Florides *et al.* (2002) designed and constructed a single stage H_2O.LiBr absorption machine. From their analysis, they found that the greater the difference between the absorber LiBr inlet and outlet percentage ratios, the smaller the mass circulating in the absorber. They also found that an increase in the surface area of the solution heat exchanger produces an increase in the efficiency of the system. They ultimately concluded that the cost of a typical H_2O.LiBr ACs system was about 3.2 times that of an equivalent electric chillers system. However, the former makes use of mainly waste heat (free energy) while the latter depends on electricity (non-free energy), which is mainly generated from fossil fuels which have harmful effects on the environment.

Elsafty and Al-Daini (2001) carried out an economic comparison between a solar-powered vapour absorption air-conditioning system and a vapour compression system in the Middle East. From their present worth comparison analysis, they found that the total cost of the vapour compression system was 11% lower than that of the single-effect vapour absorption system while that of the double-effect system was 45% less than that of the single-effect and 30% less than that of the vapour compression system. Also, their equivalent annual comparison showed that the total cost of vapour compression system was 6% lower that of the single-effect vapour absorption system while that of the double-effect system was 45% less than that of the single-effect and 37% less than that of the vapour compression system. They therefore concluded that the double-effect solar air conditioning system was a better alternative for air conditioning application in the Middle East.

Sumathy *et al.* (2001) developed a 100 kW two-stage ACs system integrated solar cooling and heating system in southern China, powered by low-temperature hot water ranging from 60 to 75^{o}C. From the preliminary operating data of the system, they found that the system was efficient and cost effective, and could achieve the same total COP as the conventional system with a cost reduction of about 50%.

Castro *et al.* (2002) developed a prototype of an air-cooled H_2O.LiBr ACs system, using solar energy to achieve a cooling effect. They used a prototype machine to validate their numerical simulation, which was then used to investigate the thermodynamic performance of the machine. From their analysis they found that a low value of internal mass flow caused an incomplete wetness of the tubes contained in the components, giving rise to poor heat and mass transfer, and hence poor performance of the machine.

Şencan *et al.* (2004) carried out an exergy analysis of an H_2O.LiBr absorption system, and found that the exergy losses and heat loads of the condenser and evaporator were less than those of the generator and absorber. They attributed the behaviour to the heat of mixing the solution, which is not present in pure fluids. From their simulation results, they found that the cooling and heating COP of the system increased slightly when the heat source temperature increased, while the exergetic efficiency decreased.

Abu-Ebin *et al.* (2009) conducted first and second law analysis of a 10 kW solar AR system and found that about 40% of the system exergy was lost in the generator, and that this tended to increase as the generator and evaporator temperatures increased and decreased respectively.

Liao *et al.* (2004) modelled and simulated an air-cooled AC integrated in a Combined Heat and Power (CHP) system. They found that the air-cooled AC was a feasible alternative, especially in applications where it is not necessary for the chilled water supply temperature to be too cold. Based on the simulations conducted, they proposed a control strategy (for both air-cooled and water-cooled systems) on how to avoid crystallization of such systems.

Maidment and Tozer (2001) carried out a theoretical analysis of the importance of using CHP systems in UK supermarkets. Their study showed that such projects could result in a payback period of less than seven years. They also found that it could offer significant primary energy/CO_2 savings when compared with CHP schemes based upon gas boiler and coal-derived electricity.

Aphornratana and Eames (1995) presented a second law thermodynamic analysis of a single effect AR cycle system. From their analysis, they found that the solution circulation ratio and the irreversibility associated with heat transfer in the evaporator played a significant role in determining the performance of the cycle. An increase in the circulation ratio resulted in an increase in the internal irreversibilities at the absorber and the generator, while an increase in solution heat exchanger effectiveness reduced the irreversibilities. However, an increase in effectiveness also increased the tendency of crystallization occurring in the system, which should be avoided. They therefore strongly proposed that in order to improve the cycle performance, the evaporator had to be considered first while the absorber might be considered second. They concluded that it was more thermodynamically efficient to operate absorption systems using low-temperature waste heat rather than high-temperature sources, due to the inherent crystallization problem associated with increasing the temperature of the generator.

Yoon and Kwon (1998) showed how to improve the crystallization limit of air-cooled double-effect $H_2O.LiBr$ ACs systems by using a new working solution: $H_2O.LiBr + HO(CH_2)_3OH$. Their simulation results showed that the new working fluid could achieve a crystallization limit of 8% higher than the conventional $H_2O.LiBr$ solution. They also found that the new working fluid may provide a COP of approximately 3% higher than the conventional $H_2O.LiBr$ solution.

Liao and Radermacher (2007) developed a novel temperature control strategy to effectively prevent the occurrence of crystallization in an air-cooled AC system. They claimed that the novel approach would automatically increase the chilled water temperature settings, or reduce the exhaust temperature accordingly, to make sure the system stayed within the safe operation zone.

Izquierdo *et al.* (2007) presented the results of the trials test carried out at La Poveda, Arganda del Rey, Madrid in August 2005 to investigate the performance of an air-cooled single effect $H_2O.LiBr$ AC system. During the test period, they noted that the hot water inlet temperature in the generator varied between 80 and 107^{o}C. They also found from their calculations that the cooling power declined with rising outdoor dry bulb temperatures, and at temperatures from 35 to 41.3^{o}C, the chilled water outlet temperature in the evaporator was found to climb to over 15^{o}C. They found that the average COP for the period stood at about 0.37.

Alva and González (2002) modelled and simulated an air-cooled solar assisted $H_2O.LiBr$ absorption air conditioning system for application in Puerto

Rico. They concluded that the results of the thermodynamic properties for the air-cooled absorption cycle were very close to those with cooling towers, especially during periods of high solar radiation. Also, the COP was found to decrease as the ambient temperature increased.

Kim and Ferreira (2009) investigated theoretically the performance of an air-cooled H_2O.LiBr AC to be combined with low-cost flat solar collector for solar air conditioning in extremely hot and dry regions. They used a dilute H_2O.LiBr solution to avoid the risk of crystallization even in extremely hot weather conditions. The results of their simulation showed that chillers would deliver water of around 7^oC with a COP of 0.37 from 90^oC hot water under ambient conditions of 35^oC. However, as the ambient temperature climbed to 50^oC, the chillers retained about 36% of the cooling power achieved at 35^oC ambient.

Vega *et al*. (2006) investigated the performance of H_2O.LiBr ACs operating with plate heat exchangers (PHE). They found that the use of PHE in the generator, condenser and solution heat exchanger gave a higher chilling capacity to volume ratio. They also found a COP as high as 0.8 when the ambient temperature was as low as 20^oC. At higher ambient temperatures of more than 30^oC, the COP was found to be 0.75; however, this occurred at the expense of higher heating temperature.

Kaynakli and Yamankaradeniz (2007) carried out thermodynamic analysis of a H_2O.LiBr AR system based on entropy generation through modelling and simulation. They evaluated variations in entropy generation and non-dimensional entropy generation in each component under different operating conditions, and found that a decrease in condenser and absorber temperatures and an increase in generator and evaporator temperatures brought about an increase in system performance. Their result also showed that the entropy generation in the refrigerant expansion valves, refrigerant heat exchanger and solution pump were negligible when compared to the total entropy generation in the entire system. The generator was found to have the greatest impact on the COP and the entropy generation, while the non-dimensional entropy generation in the generator, absorber and evaporator accounted for about 90% of entropy generation in the system. Hence, they advised that such components need to be designed to minimize entropy generation.

Sedighi *et al*. (2007) carried out an exergetic analysis and parametric study of H_2O.LiBr AR systems. They found that the solution heat exchanger (SHE) demonstrated a more significant impact on the system performance than the refrigerant heat exchanger (RHE). They also noticed that the RHE could cause crystallization when it increased the outlet temperature of the absorber. Also, a reduction in the cooling water temperature was found to improve the COP and exergetic COP (ECOP), and so did an increase in the evaporator temperature; however, the later caused a reduction in the ECOP of the system.

Lee and Sherif (2000) conducted thermodynamic analysis of an H_2O.LiBr absorption system for cooling and heating applications, by investigating the first and second law efficiencies of the system over a host of operating conditions. They stated clearly from their results that a lower cooling water temperature yielded both a higher cooling COP and higher exergetic efficiency. Also, they noted that increasing the heat source temperature could improve the cooling COP; however, a threshold value existed at which the COP of the system levelled off and then decreased. In the area of absorption system for heating, they found that increasing the heat source temperature would increase both the heating COP and the exergetic efficiency. However, this might be detrimental to the system operation since it might give rise to crystallization. They also found that increasing the temperature for the supply hot water would reduce the operating range of the system.

Tozer and James (1997) developed the concept of a universal law of cold generation systems for heat-powered refrigeration cycles by merging the direct and reverse Carnot cycle with Carnot theory, through the combination of driving and cooling cycles, which produced cooling from a combustion process. They analysed the application of direct-fired ACs and their integration into CHP systems. From their analysis, they concluded that direct fired AC was economically feasible.

Moné *et al.* (2000) carried out an economic feasibility study of CHP by combining ACs with commercially available gas turbines. They found that these systems were of potential benefit to consumers. They also observed that the amount of heating or cooling available from the rejected heat and available to the AC system depended on the mass flow rate of the exhaust gas, the temperature of the gas and the turbine size. The cooling capacity was also found to be more affected by the exhaust flow rate than the temperature.

Kaynakli and Kilic (2006) conducted a theoretical study on the effects of operating conditions on the performance of H_2O.LiBr AR systems. From their study, they found that variations in the operating temperatures and the effectiveness of heat exchangers affected the performance of the system. They concluded that the thermal load of the components and COP of the system increased with increasing generator and evaporator temperature and decreased with increasing condenser and absorber temperature. They also found that the SHE had more effect on the parameters of the system than the RHE.

Castro *et al.* (2007) modelled the components of the AC system (generator, evaporator, condenser and absorber) and validated the model with experimental results. From their evaluation, they found that within the allowable experimental error, the model was able to predict the component behaviour of the AC system. They proposed that one useful way of reducing the final size of the heat and mass exchange components (especially the absorber, generator and evaporator) was to improve their wetted area.

Izquierdo *et al.* (2003) investigated the limit caused by crystallization in the operation of an air-cooled solar-powered (using flat plate collectors) double-stage H_2O.LiBr ACs system and compared the performance with a single-stage system. They found that the efficiency gain of the double-stage over the single-stage system increased as the condensation temperature increased. Their analysis also showed that the single-stage system could not operate at condensation temperatures higher than 40^oC as a result of the occurrence of crystallization in the system, while the double stage systems could perform up to a condensation temperature of 53^{o}C.

From the literature review, it can be observed that the basic principle of operation of the different classes of the AC system is basically the same; what determines the effect to be applied for any particular heat source is the quality of that source (ASHRAE, 2001) and the available cooling medium. Also it can be observed that most of the AC systems reported in the literature are driven by solar-powered systems; however, any waste heat source can be used.

Apart from the cooling technologies presented above, which make use of waste heat directly, cooling can also be achieved using systems which make use of it indirectly. A typical example is ORC-driven vapour compression refrigeration (VCR) systems, also known as electrical chillers or refrigerators.

VCR systems are heat pumps which make use of mechanical or electrical energy to drive a compressor, used to pressurise a vaporized refrigerant. The refrigerant in the vapour phase is passed to the condenser where it is condensed. The condensed refrigerant is passed through a throttle valve where its pressure is suddenly decreased, causing it to flash into a wet vapour. The wet vapour is then completely vaporized in the evaporator in order to obtain a cooling effect and the cycle continues. A typical schematic diagram is shown in Figure 2-10. These are very efficient systems, with very high COP when compared with AR systems. They are a well-developed and matured technology, and are also free from some operational problems associated with LiBr.H_2O based ACs system like crystallization. Furthermore, research has shown that electrical chillers or refrigerators are always cheaper than AR systems of similar capacity (Elsafty and Al-Daini, 2001, Florides *et al*., 2002).

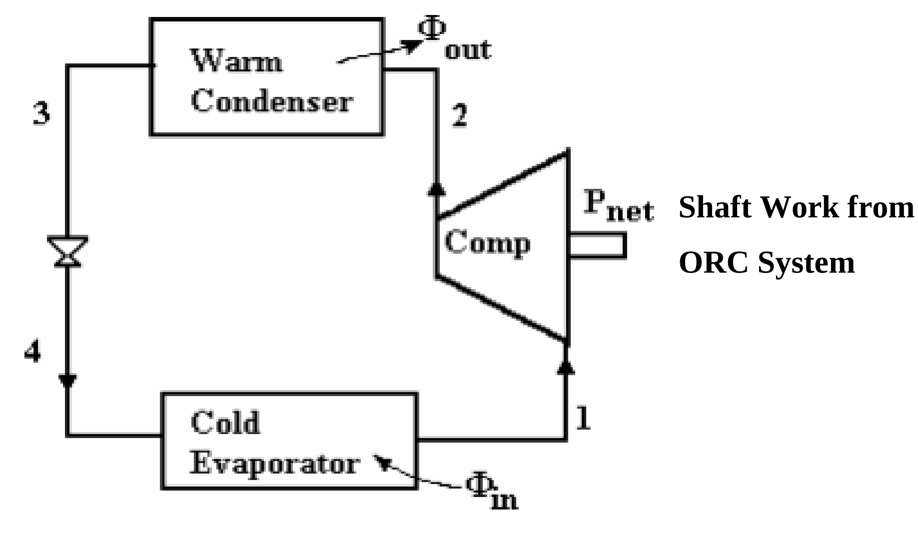


Figure 2–10: Vapour Compression Refrigeration (VCR) Cycle

Based on the above advantages, an ORC-driven VCR system and its performance are worth comparing with an AC/AR system, where both are driven by the same waste heat energy.

A theoretical study carried out by the author shows that within the chosen design constraints, the ORC-driven VCR system gives a better thermodynamic performance in terms of COP and second law efficiency than a single-effect AR system, where both are driven from the same waste heat; even above the breakeven pressure where the AR system gives a better COP, its second law efficiency was still lower than that of an ORC-driven VCR system (Aneke *et al.*, 2012a), which is a paradox.

Table 2–4: List of Some Thermally-Driven ACs in Europe (Henning, 2005)

WORKING FLUID	SORPTION MATERIAL	DEVELOPER(S)	HEAT SOURCE TEMPERATURE	FEATURES
Water	Lithium-bromide	Company: Rotartica; Research centre: Ikerlan (both Spain)	70–95°C	Rotating Absorber; very low temperatures on HXs
Water	Lithium-Bromide	Company: EAW; Research centre: ILK Dresden (both Germany)	80–90°C	Market available system (cooling capacity > 15 kW)
Water	Lithium-Bromide	Company: Phönix Sonnenwärme; Research centre: ZAE Bayern; Technical University Berlin (all Germany)	70–95°C	Good part load behaviour; compact design; prototypes in operation
Water	Lithium-Bromide	Polytechnic Univ. Catalunya (Spain)	75–95°C	Directly air cooled; still in research status
Water	Silica gel	Company: Sortech; Research Centre: Fraunofer Institute (ISE) (both Germany)	65–95°C	Compact design; no mechanical moving parts; prototypes in operation
Water	Lithium-Chloride	Company: Climatewell; Solar Energy Research Centre (both Sweden)	70–100°C	High efficient storage included
Water	Sodium-Sulfide	Company: Sweat; Research Centre: ECN (both Netherlands)	80–90°C	High efficient (long term) storage; modular system, modular operation
Ammonia	Water	Company: Aosol; Research Centre: INETI (both Portugal)	100–120°C	Standard components; dry air cooling
Ammonia	Water	Research Institute Joanneum Research (Austria)	80–110°C	Prototype in operation; adjustable to different applications; low temperatures possible
Ammonia	Water	University of Applied Science, Stuttgart, Germany	70–120°C	No solution pump; still in research status

2.1.3.2 Concluding Remarks

It can be observed that the use of heat for cooling operation is technically feasible. The economic viability of such systems when compared with other cooling applications depends mainly on the value of the heat source (i.e. whether it is free or not) (Henning, 2005).

AC is still the dominating technology for thermally-driven chillers. About 59% of installed thermally-driven cooling applications in Europe use AC (Henning, 2005). They are available on the market in a wide range of capacities and are usually designed for different applications (see Table 2-4).

The basic cycle, where for each unit mass of refrigerant which evaporates in the evaporator one unit mass of refrigerant has to be desorbed from the refrigerant-sorbent solution, is known as the ***single-effect cycle***. Such cycles usually operate in the temperature range of 80–100 ^{o}C and achieve a COP of about 0.7. Other effects adapted from the basic single-effect system such as half, double, triple and multiple-effect systems are also applicable; however, some of them have not been implemented commercially (e.g. triple and multiple-effect systems).

An ORC-driven VCR system has been shown to have a better second law efficiency than an AR system for a given waste heat source (Aneke *et al*., 2012a). Apart from that, the ORC part of the ORC-driven VCR system may also be used for power generation when there is no need for a cooling application, thus having more operational time, which gives rise to a greater return on investment, unlike the AR system which can only be used for cooling applications. In order words, the ORC-driven VCR system is more versatile and will produce a better return on investment.

2.2 Overall Conclusions from the Literature Review

Based on the literature review carried out, this work will focus on using the waste heat emitted from the crisps/chips manufacturing plant to drive an ORC system to generate power.

The reason why this has been adopted is because it has been proved from the literature review that other applications which might be of interest in a food processing plant, such as water purification and refrigeration applications can easily be achieved by incorporating RO or VCR respectively into the ORC unit. Another reason this has been adopted is because electricity is the most used form of energy and hence should be of utmost importance to the process plant.

2.2.1 Original Contribution to Knowledge

A critical review of the literature presented in this thesis shows that the state of the art technology adopted in this work (ORC system for power generation; which can easily be adapted to achieve other purposes like RO wastewater desalination or VCR application) is not a new technology; however, its application in the food processing industry in general and in the UK crisps manufacturing process in particular, for the conversion of waste heat from the fryer section to electricity, has never been carried out before.

Hence, this project brings a new insight into the possibility of utilizing the waste heat from the fryer section of a potato crisps/chips manufacturing plant for power generation using ORC technology. Again, this project has contributed to knowledge by introducing the concept of a dual heat source ORC system, as can be seen in later chapters in this thesis.

Furthermore, this project has also contributed to knowledge through journal publications which will serve as research material for future engineers and professionals who are researching in a similar field. Some of the relevant publications are listed below:

Aneke, M., Agnew, B. and Underwood, C. (2011) Performance Analysis of Chena Binary Geothermal Power Plant, *Applied Thermal Engineering*, 31, 1825–1832.

Aneke, M., Agnew, B. and Underwood, C. (2011) Power Generation through the use of Waste Heat Energy from Process Industries: a greener approach to reducing CO_2 emission and global warming in Nigeria, Proceedings of the FUTO 2011 Renewable & Alternative Energy Conference, Nigeria.

Aneke, M., Agnew, B. and Underwood, C. (2011) Approximate Analysis of the economic advantage of a dual source ORC system over two single ORC systems in the conversion of dual low and mid grade heat energy to electricity, *EUEC Journal,* USA.

Aneke, M., Agnew, B., Underwood, C., Wu, H. and Masheiti, S. (2012) Power generation from waste heat in a food processing application, *Applied Thermal Engineering*, 36, 171–180.

Aneke, M., Agnew, B, Underwood, C and Menkiti, M. (2012) Thermodynamic Analysis of Alternative Refrigeration Cycles Driven from Waste Heat in a

Food Processing Application, *International Journal of Refrigeration,*35,1349 - 1358.

Chapter Three

This chapter presents the software used in this work. It covers the software selection process, as well as the capabilities and limitations of the selected software.

3 Modelling and Simulation Tool

3.1 Modelling and Simulation Software

Since this work is purely based on modelling and simulation, the selection of a good process modelling and simulation tool is of utmost important to the success of this work. This research is also energy-based, and the software selected for it will have to be energy-based software which has a good commercial and industrial reputation.

Among the process simulation software considered, IPSEpro simulation software (SimTech, 2008) developed by SimTechnologies plc was adopted for this work. Its selection was purely based on its flexibility, capability, industrial recognition and high level of commercialization in the energy modelling and simulation industry.

3.1.1 IPSEpro Simulation Software

IPSEpro is a highly flexible and comprehensive environment for modelling and analysing processes in energy engineering, chemical engineering and many other related areas (SimTech, 2008). It is a modular-mode as well as an equation-oriented process simulator and has been designed to solve problems represented by a network of components from a standard library or from libraries created by the user. Unlike some other process simulators in which the models appear as a black box, IPSEpro allows the user the freedom of either modifying or creating an entirely new underlying model equation in order to represent a component. This flexibility can be achieved in two levels: the component level and the process level.

- **Component Level**

  At component level, IPSEpro allows for unlimited flexibility in defining the characteristics of the component models that are used for modelling processes. This gives the user the capability to build component model libraries that exactly match his/her application requirements. This can be achieved by using the IPSEpro Model Development Kit (MDK) suite.

- **Process Level**

  IPSEpro allows the user the freedom to arrange the available components in order to represent a process scheme through the use of a graphical user interface, known as the Process Simulation Environment (PSE) suite, which substantially facilitates and accelerates the development of a process scheme and the presentation of the calculated results. In order to set up a process model in IPSEpro PSE, the user has to choose component icons from a library menu, place them in the project window and connect them appropriately. Numerical data and the results of the process calculations are entered and displayed directly in the project windows.

The IPSEpro system architecture, showing different component levels, is presented in Figure 3-1.

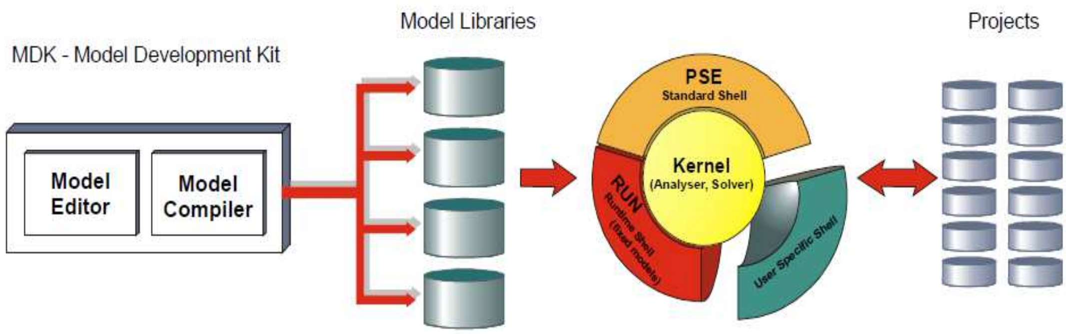


Figure 3–1: IPSEpro System Architecture

IPSEpro allows the user to simulate the behaviour of a single element of processes, parts of a process and the model of a complete plant. It uses robust algorithms which results in an extremely short calculation time.

In IPSEpro, the components of a process are generally known as objects. The objects are not necessarily pieces of equipment; they can also represent connections between components or chemical compositions.

In order to create a process model in IPSEpro, the component models from the model library are used. The component models are made from mathematical descriptions which use mathematical equations comprising items such as variables and parameters to represent the behaviour of the component. These component models are arranged appropriately in the PSE to form a process model.

IPSEpro uses three basic types of model to develop a process model. They include:

- Unit
- Connection
- Global

Units are known as nodes in the process model. They represent an actual individual piece of equipment in the process. A good example is heat exchanger, pump, turbine etc.

Connections are mainly used for information transfer between units in any given network structure or process model. They are mainly process or energy streams.

Globals are used to represent information that is shared by an undefined number of objects. Examples of global include: chemical composition, fuel composition etc.

Based on the definitions presented above, it can be deduced that only up to two units can reference a connection while an unlimited number of objects can reference a global.

The individual unit models from a model library are represented by graphic icons. These icons can be placed on the drawing area of the PSE and be connected to build up a process scheme, which is generally known as the process flowsheet. Since what matters when it comes to system solutions are the mathematical equations that define each unit icon, the shape and size of these icons can be chosen arbitrarily, since they do not affect the system solution.

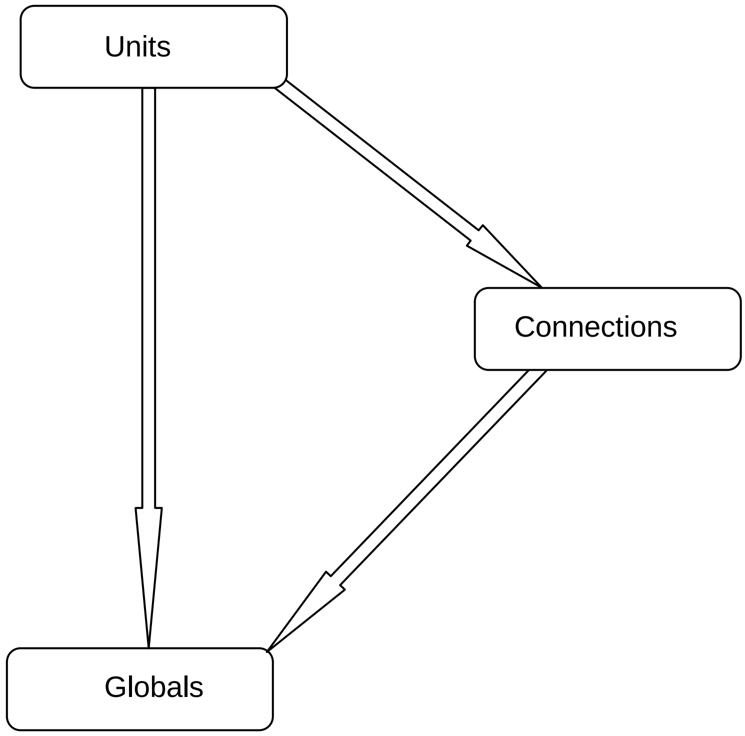


Figure 3–2: Hierarchy of the Model Classes

3.1.2 Modelling an Object Using IPSEpro MDK

IPSEpro MDK gives the user the flexibility of developing a new model of a component/object or modifying an existing one. It basically houses the mathematical equations which determine the behaviour of the component together with the ***test conditions***. The test conditions are mathematical expressions which test the physical validity of the solution obtained when the mathematical equations of the model are solved during simulation. If the solution obtained is not valid during the simulation in PSE, the test condition triggers a warning which informs the user that the solution is not valid. These capabilities provide the user with the advantage of being able to develop a model to suit his/her process as well as to maintain the validity of the model. The models of the components developed in MDK can be compiled together as a model library. The model library in MDK is used for developing the process models in IPSEpro PSE. A screen shot of a typical MDK model of a unit is shown in Figure 3-3.

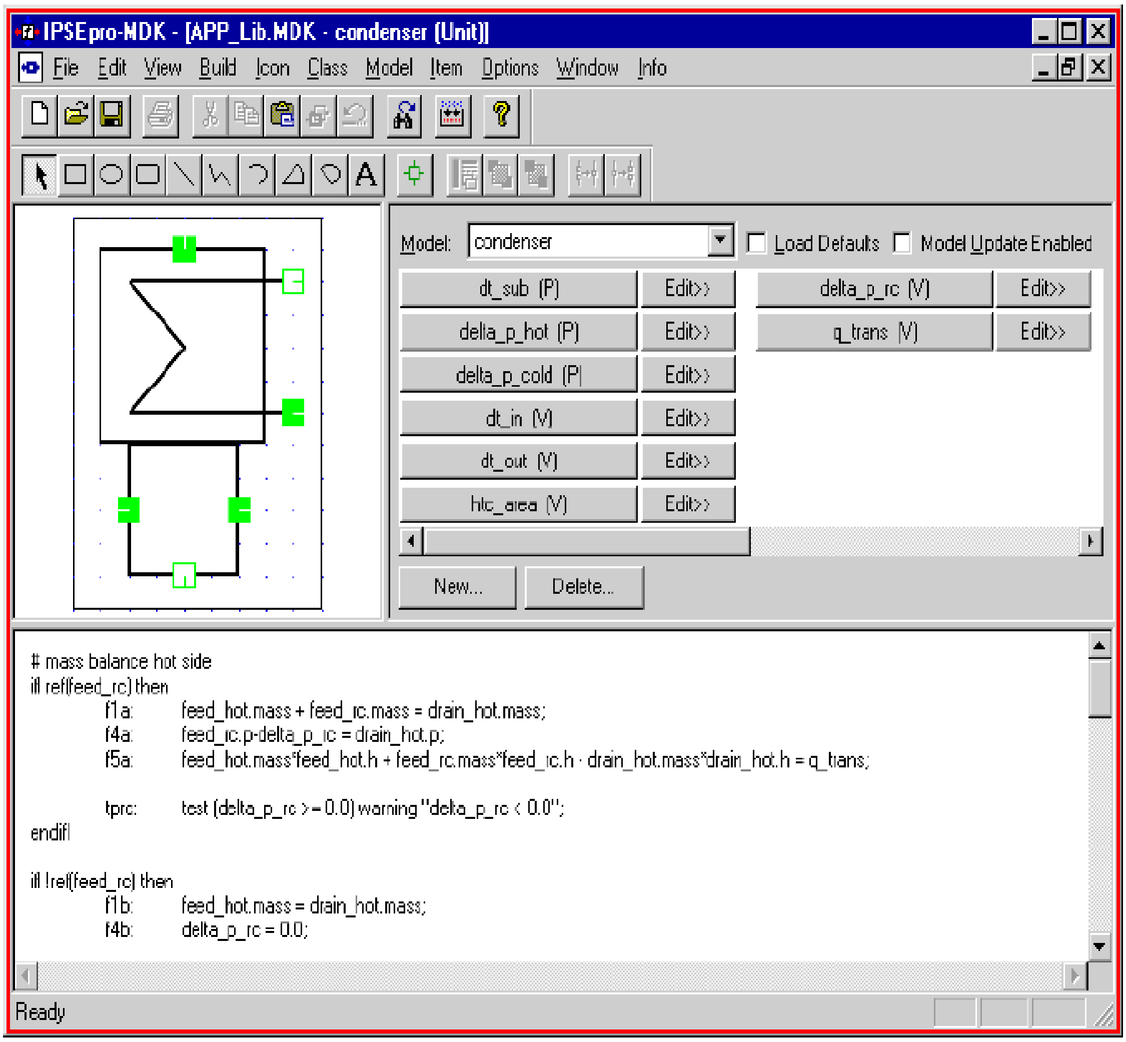


Figure 3–3: MDK Model of a Condenser Showing the Model Icon & Equations

3.1.3 Modelling a Process Using IPSEpro PSE

As mentioned above, process modelling in IPSEpro is carried out using the PSE suite. In IPSEpro, a process is referred to as a system that has the following:

- There are one or more objects or process components.
- The objects are connected in a defined way.
- The behaviour of each object can be formulated mathematically.
- The overall behaviour of the process is determined by the behaviour of the objects that compose the process and by the connections which link these objects.

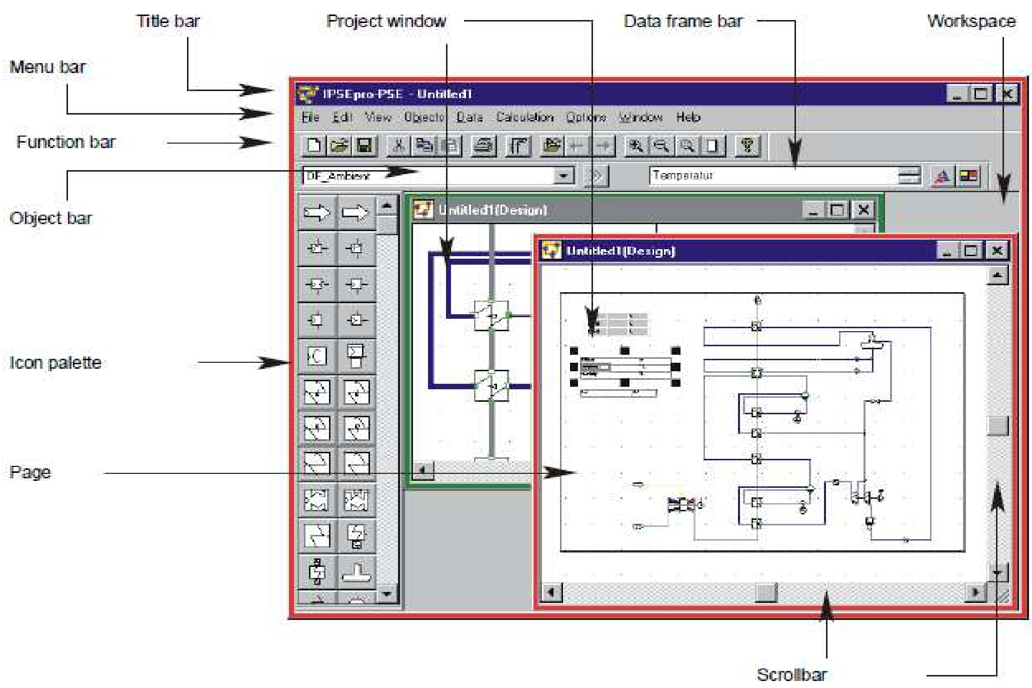


Figure 3–4: PSE Screen

A typical example of a PSE screen of a process model showing the process flowsheet is shown in Figure 3-4.

For detailed description of how to use the IPSEpro PSE and MDK suite for process modelling, the reader is referred to the manufacturers manual in (SimTech, 2008).

Chapter Four

This chapter presents the details of the processes involved in potato crisps manufacturing, as well as the energy usage of the process with more emphasis on the waste heat recovery potential in the fryer section of the plant.

4 Potato Crisp/Chip Manufacturing Process

As already mentioned in section 1.3.1, there is a high potential for waste heat recovery from the food processing industries in the UK. Although this research will concentrate on the potato crisps/chips manufacturing industry, its outcome will be applicable to other industries where waste heat energy is emitted.

Commercial crisp manufacturing is a well-established market both in the UK and worldwide. In 2005, crisp consumption in the UK was estimated to be as high as 10^{10} packets, which represents more than half of the crisps sold in the European Union (Applesnapz, 2005). Despite many campaigns by some health organizations against the consumption of crisps, crisp consumption has always been on the increase.

Associated with this increase in crisp consumption is an increase in energy consumption during crisp manufacturing. Crisp manufacturing is an energy-intensive process (Hardcastle and Ward, 1984) which involves a series of recipes in order to transform the raw potato from the farm into the finished product. The sequential processes involved in crisp production include destoning, washing, peeling, drum washing and inspection, slicing, cold washing, hot washing, dewatering, frying and inspection, flavouring and packaging. The process flow diagram of the crisp manufacturing process at the KP Billingham plant, which serves as the plant for our case study, is shown in Figure 4-1. The description of the individual processes taking place in the plant is as given below.

- **Bulk hopper and incline elevator**

  The bulk hopper holds approximately 1.5 crates of potatoes. It is fed from the crate tipper and provides a store of potatoes for the line. The speed of this conveyor is variable and can be altered to deal with changing feed rates. The bulk hopper is set to run at a speed that allows the line to run constantly without producing any gaps in the process.

- **Destoning and Washing**

  In this unit operation, stones are separated from the potatoes using water. To accomplish this, water is recirculated by using pumps from the tank base up the central tube into a dish. The potatoes fall into the water dish and are rotated around the dish and onto the exit conveyor by the water wave and the effects of buoyancy created by the water flow down the central tube. Stones that are present fall down the central tube and onto the stone removal conveyor belt.

- **Peeling**

  The potatoes exiting the destoner go into the peeler where they are peeled using a peeler disc which has a gritted surface with three undulations. As the disc rotates, the potatoes spin in a corkscrew effect and rub against the drum. The peelings are washed down into a drainage channel below the peeler by the peeler water.

- **Drum wash and inspection**

  This removes any leftover peel and/or roots. Also, the potatoes are inspected in order to sort by size for the slicing operation.

- **Slicing**

  The sorted potatoes are sent to the slicer where they are sliced. The slicer head consists of eight separate slicing head shoes and knives which are used to slice the potatoes as they pass in a smooth and uninterrupted manner.

- **Cold wash rinse unit**

  After the slicing operation, the potatoes are sent to the cold wash unit where they are rinsed with cold water to remove starch solids. The water is destarched and reused on the precleaning units on the process line.

- **Hot wash system**

  The hot wash system works similarly to the cold wash, but instead of removing starch from the slices, the hot wash removes sugars or picric and other water soluble solids from the slice.

- **Dewatering**

  Before being sent to the fryer for the frying operation, the washed crisp slices are drained off carefully using fans to remove the adherent water, as dry slices fry better and faster than wet ones. Also, the frying oil can go bad more rapidly when used to fry slices that are too wet.

- **Frying**

  The drained slices are then sent to the fryer where they are fried with oil at a temperature of approximately 170^{o}C. The fryer holds approximately 5.5 tonnes of oil. It has a main drain valve and four low-level valves. The diagram of the fryer section of the plant is shown in Figure 4-2.

- **Inspection, Flavouring and Packaging**

  After draining off the adherent oil and removing slices which appear bad, salt and other flavourings such as onion, cheese, etc, are added to the still warm products. The flavoured products are cooled and then vacuum packed.

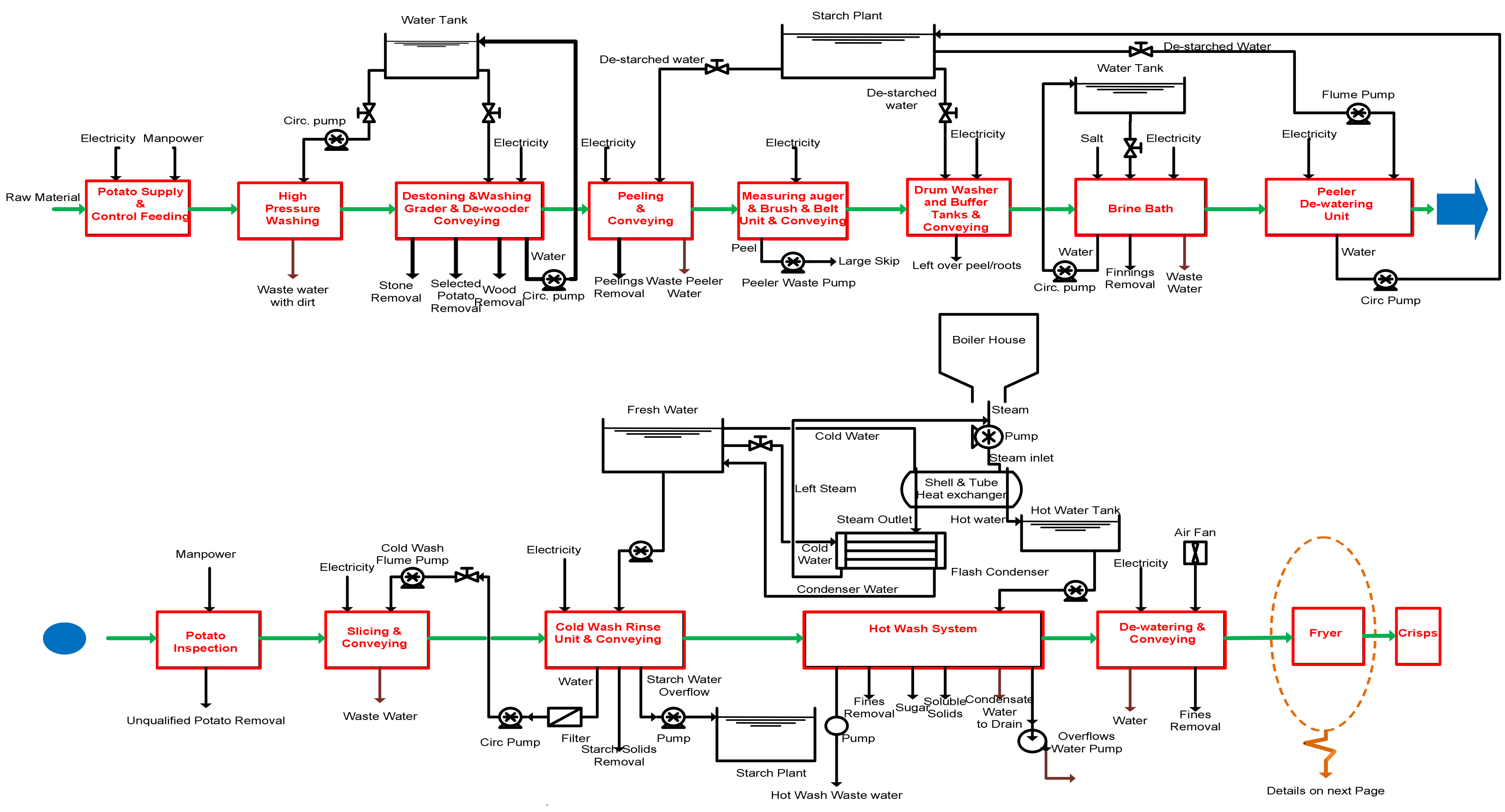


Figure 4–1: Process Flow Diagram of the Potato Crisp/Chip Manufacturing Line at KP Billingham Plant

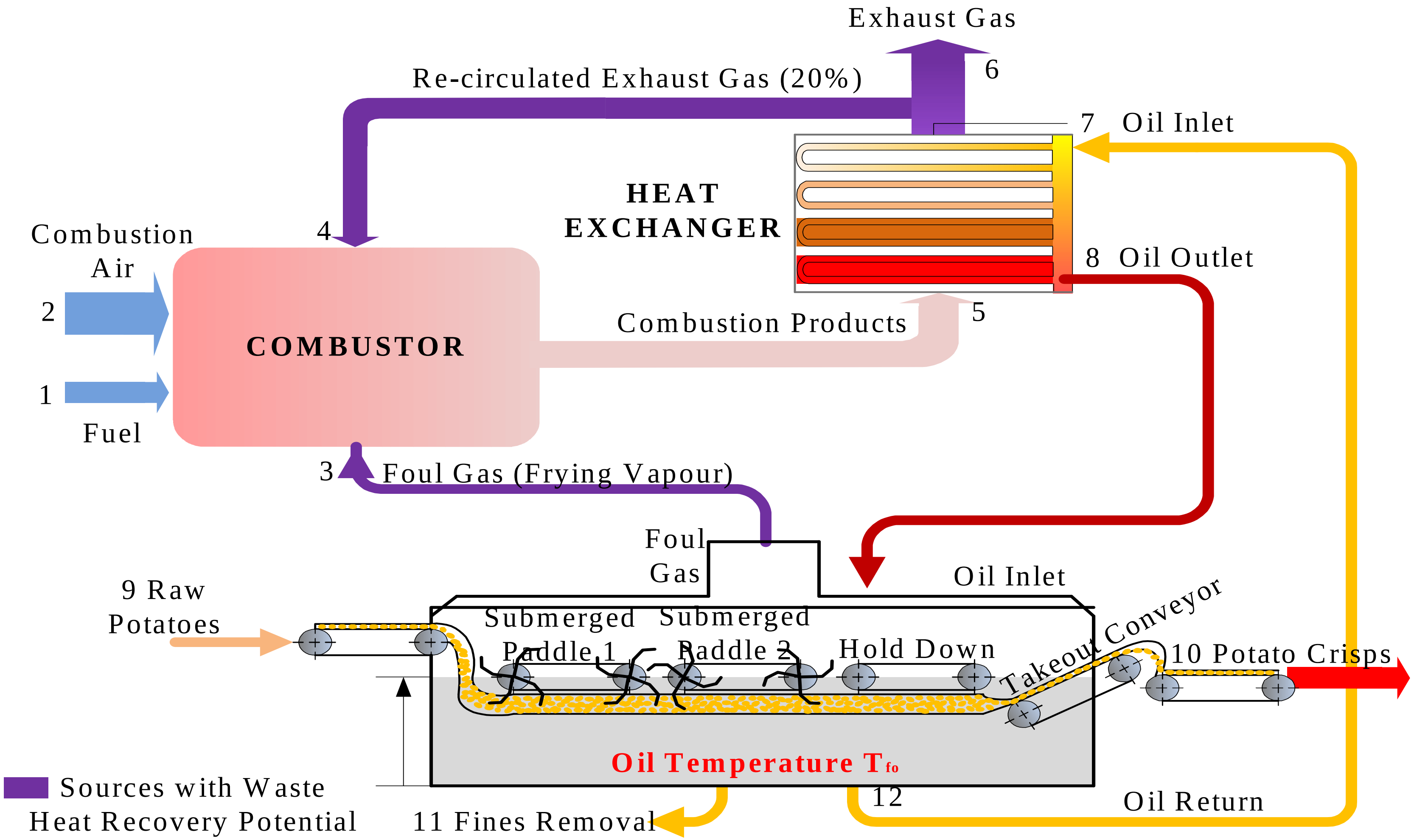


Figure 4–2: Fryer Section of the Crisp Manufacturing System

4.1 Energy Consumption in Crisp Manufacturing Process

As mentioned above, crisp manufacturing is an energy-intensive process. Processes such as the hot washing and peeling require hot water and steam respectively, while the packing hall is required to be heated. Space heating accounts for about 20% of the total energy and is produced using boilers, which makes use of boiler fuels (Hardcastle and Ward, 1984).

The frying operation is the most energy-consuming unit operation and consumes more than 65% of the total energy use (in the form of electrical energy). This comprises the electrical energy used in driving the conveyor systems, foul gas fans, pumps and other auxiliaries. Figure 4-3 shows a typical day's hourly electrical energy consumption in the KP Billingham crisp manufacturing plant. The graph shows that the average hourly electricity usage stands at 65.90 kW for the processes up to the fryer and 124.40 kW in the fryer. This results in a total daily average electricity usage of 190.30 kW.

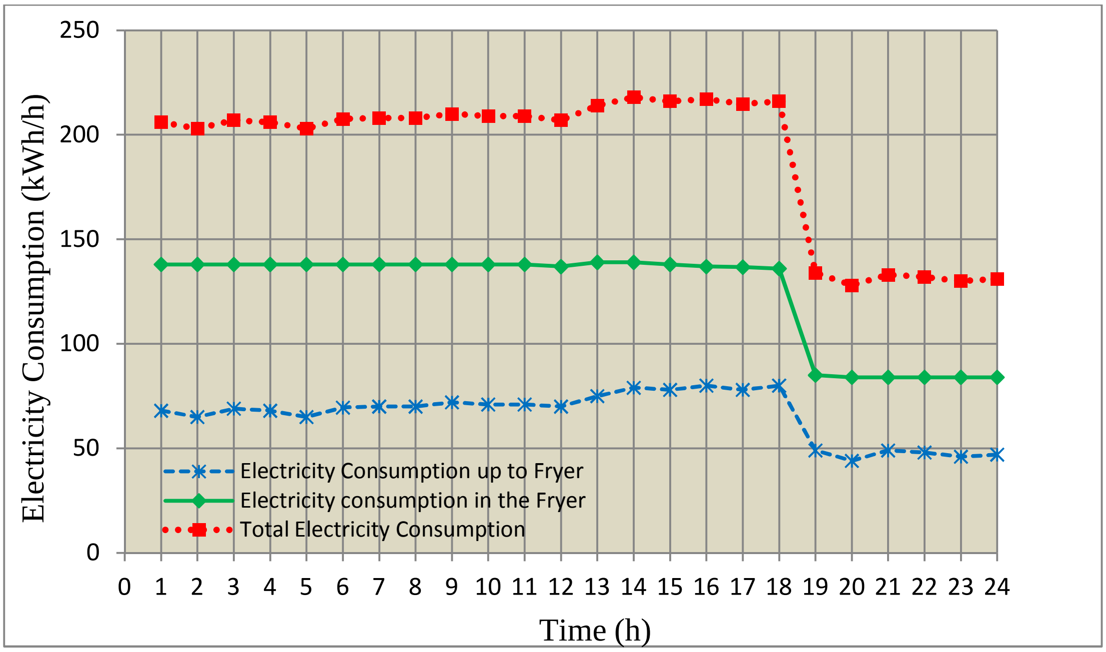


Figure 4–3: Electricity Consumption in Crisp Manufacturing

4.2 Waste Heat Recovery Potential in Crisp Manufacturing Process

As shown in Figure 4-2, crisp frying involves passing the crisps through hot cooking oil. The oil is heated using heat from the combustion chamber. After heating the oil, the heat is dumped from the process into the environment through the industrial stack. Also, during the cooking operation, some hot polluted air (foul gas) is emitted from the fryer. In some crisp manufacturing processes surveyed, the hot polluted air is recycled back into the combustion chamber, while in others; they are sent to the stack directly and are then emitted into the environment. Whatever the method adopted by any crisp manufacturing operation, it is observed that the quality of heat emitted as effluent from the stack and from the foul gas from the fryer is high. With the effluent heat in a temperature range of 120 to 212^{o}C (Aneke *et al*, 2012b), it can be economically recovered and used to drive any of the state of the art heat recovery processes adopted in this thesis to the benefit of the process, the environment and the plant owner.

The use of this waste heat energy, identified in this crisp manufacturing plant under investigation, to drive an ORC system for power generation, RO wastewater desalination or VCR system for refrigeration application will not only help in saving resources, it will also help to reduce CO_2 emissions and ultimately contribute to a reduction in global warming.

This project focuses on the theoretical study of using the waste heat from the fryer section of the plant for power generation, using ORC technology. The study is carried out through modelling and simulation using the IPSEpro Process Simulation tool.

Chapter Five

This chapter presents the basic thermodynamic theory behind the ORC system and the actual modelling of ORC system components. The modelling is carried out using the MDK tool in the IPSEpro process simulation software.

5 Thermodynamics of ORC System and IPSEpro MDK Modelling of ORC Unit Operations

5.1 Thermodynamic Properties

There are some basic terms used to represent any thermodynamic system. These are usually regarded as thermodynamic variables or properties. The main thermodynamic variables are generally classified into two, namely:

- Extensive Properties
- Intensive Properties

Extensive properties are those properties which are dependent on system size, while intensive properties are those which are not dependent on system size. Some properties of a system occur solely as intensive, while there are others which are intensive with a corresponding extensive property.

Table 5-1 shows some thermodynamic properties, with some having a corresponding extensive property.

Table 5–1: Some Thermodynamic Properties

Property	**Extensive Property Name**	**Symbol & SI Unit**	**Intensive Property Name**	**Symbol & SI Unit**	**Flow Property**	**Symbol & SI Unit**
Mass	Mass	$\mathrm{m}\ [\mathrm{kg}]$	Specific Mass	No Unit	Mass Flow rate	$\dot{\mathrm{m}}\ [\frac{\mathrm{kg}}{\mathrm{s}}]$
Temperature	×	×	Temperature	$\mathrm{T}\ [°\mathrm{C}\ \mathrm{or}\ \mathrm{K}]$	×	
Pressure	×	×	Pressure	$\mathrm{P}\ [\mathrm{Pa}]$	×	
Volume	Volume	$\mathrm{V}\ [\mathrm{m}^3]$	Specific volume	$\mathrm{v}\ [\frac{\mathrm{m}^3}{\mathrm{kg}}]$	Volumetric Flow rate	$\dot{\mathrm{V}}\ [\frac{\mathrm{m}^3}{\mathrm{s}}]$
Enthalpy	Enthalpy	$\mathrm{H}\ [\mathrm{J}] = \mathrm{U} + \mathrm{P} * \mathrm{V}$	Specific enthalpy	$\mathrm{h}\ [\frac{\mathrm{J}}{\mathrm{kg}}]$	Enthalpy (heat) flow rate	$\dot{\mathrm{H}}\ [\frac{\mathrm{J}}{\mathrm{s}}\ \mathrm{or}\ \mathrm{W}]$
Internal energy	Internal energy	$\mathrm{U}\ [\mathrm{J}]$	Specific internal energy	$\mathrm{u}\ [\frac{\mathrm{J}}{\mathrm{kg}}]$	Internal energy flow rate	$\dot{\mathrm{U}}\ \left[\frac{\mathrm{J}}{\mathrm{s}}\ \mathrm{or}\ \mathrm{W}\right]$
Entropy	Entropy	$\mathrm{S}\ \left[\frac{\mathrm{J}}{\mathrm{K}}\right]$	Specific entropy	$\mathrm{s}\ [\frac{\mathrm{J}}{\mathrm{kg}}\mathrm{K}]$		

× **Not Applicable**

5.2 Thermodynamic Processes

The change in the state of any thermodynamic system occurs as a result of a sequence of events known as thermodynamic processes. There are several thermodynamic processes which include:

- **Isothermal Process:** a process which occurs at constant temperature and is always maintained with the addition or removal of heat from a heat source or sinks respectively. A typical example of an isothermal process is the evaporation or condensation of a pure fluid.

- **Isobaric Process:** a constant pressure process.

- **Isometric/Isochoric Process:** a constant volume process.

- **Adiabatic Process:** a thermodynamic process in which there is neither heat addition nor removal from the system.

- **Isentropic Process:** a process in which the entropy of the system is maintained as constant. It is also referred to as a reversible adiabatic process.

- **Isenthalpic Process:** a constant enthalpy process.

5.3 Thermodynamic Laws

There are two basic laws governing any thermodynamic system. They are known as the first and second laws of thermodynamics.

5.3.1 First Law of Thermodynamics

The first law of thermodynamics, also known as the law of conservation of energy, states that the increase in the energy of any given system equals the amount of work provided to the system, plus the total amount of heat provided to the system, plus the net enthalpy flow entering/leaving the system (Quoilin, 2008). For any given system (as represented by Figure 5-1) the first law can be written mathematically as

$$\dot{\mathrm{U}}_{\mathrm{tot}} = \sum_{\mathrm{j}} \dot{\mathrm{Q}}_{\mathrm{j}} + \sum_{\mathrm{j}} \dot{\mathrm{W}}_{\mathrm{j}} + \sum_{\mathrm{j}} \dot{\mathrm{H}}_{\mathrm{tot,j}} \tag{5-1}$$

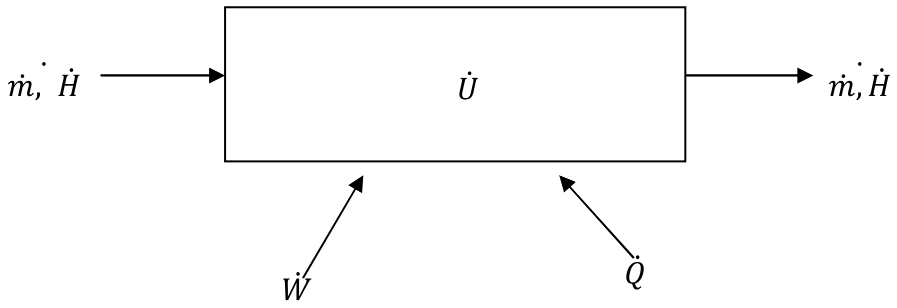


Figure 5–1: System

where $\dot{Q}, \dot{W}$ and $\dot{H}$ are the heat fluxes, the power and the enthalpy flows provided to the system.

NB: Any flow directed toward the system is considered as positive while that leaving the system is considered negative.

5.3.2 Second Law of Thermodynamics

While the first law only tells us about the conversion of energy from one form to another with no recourse to the direction of flow, the second law of thermodynamics explains the phenomenon of irreversibility in thermodynamic systems. It states that:

- Heat generally cannot spontaneously flow from a lower temperature to a higher temperature.
- It is impossible to convert heat completely to work.
- Two gases placed in an isolated chamber will be mixed uniformly throughout but will not separate completely once mixed.

The second law of thermodynamics is based on the Clausius theorem, which states that for any reversible cycle,

$$\int {}^{\delta Q_{rev}}\!/_{T} = 0 \qquad 5\text{-}2$$

5.4 Reversible Cycle

A typical thermodynamic cycle which can be used to illustrate the Clausius theorem is the Carnot cycle. The Carnot cycle is an ideal cycle in which all the processes are regarded as being reversible. The P-v diagram of a typical Carnot cycle is shown in Figure 5-2.

The thermodynamic processes encountered in this system together with the process equations are explained below:

- **Process 1–2:** This is an isothermal expansion process in which heat is added to the process from the surroundings at a constant temperature. The process is governed by the thermodynamic equation shown below:

$$q_h = -w = nRT_1 \ln\left({}^{V_2}\!/_{V_1}\right) > 0 \qquad 5\text{-}3$$

- **Process 2–3:** This is an adiabatic expansion process where there is no heat transfer across the system boundary but there is production of work.

$$q = 0 \qquad 5\text{-}4$$

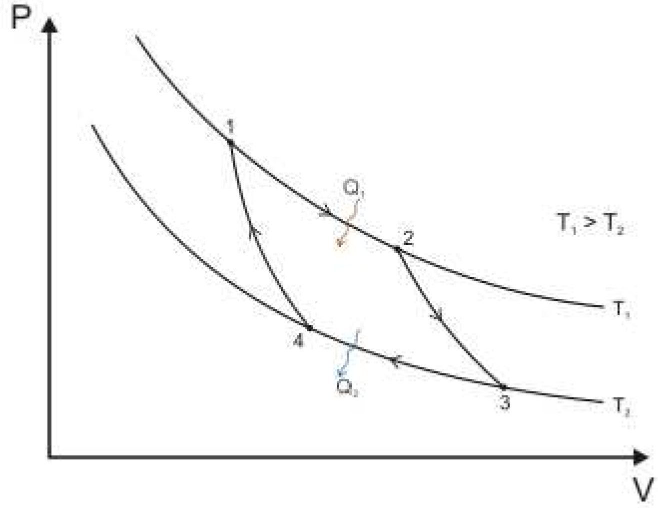


Figure 5–2: P-v diagram of a Carnot Cycle

- **Process 3–4:** This is an isothermal compression process in which heat is rejected from the system to the surrounding at a constant temperature. The governing equation is given as:

$$q_c = -w = nRT_2 \ln\left({}^{V_4}\!/_{V_3}\right) < 0 \qquad 5\text{-}5$$

- **Process 4–1:** This is an adiabatic process in which no heat is transferred across the system boundary but work is provided to the system.

$$q = 0 \qquad 5\text{-}6$$

From the first law of thermodynamics, it can be deduced that the amount of work obtained from a Carnot cycle can be given as

$$w = q_h - q_c \qquad 5\text{-}7$$

The thermal efficiency of the system is given as

$$\eta_{carnot} = {}^{w}/_{q_h} = 1 - {}^{q_c}/_{q_h} \qquad 5\text{-}8$$

Since it is assumed that the Carnot cycle is a purely reversible cycle (no irreversibility) this efficiency shown in Equation 5-8 is the maximum efficiency that can be achieved with a given heat source and sink temperature.

Since q_h and q_c are purely functions of t_h and t_c respectively, Kelvin proposed a new temperature scale known as Kelvin (K) in order to achieve the following relationship for a reversible heat transfer system.

$$ {}^{q_c}/_{q_h} = {}^{t_c[K]}/_{t_h[K]} = {}^{T_c}/_{T_h} \qquad 5\text{-}9$$

Based on the above corollary, the efficiency of a Carnot engine can be rewritten as follows:

$$\eta_{carnot} = 1 - {}^{T_c}/_{T_h} \qquad 5\text{-}10$$

Hence, for the particular case of the Carnot cycle, the Clausius theorem can be expressed as:

$$\int {}^{\delta Q_{rev}}/_{T} = {}^{q_h}/_{T_h} - {}^{q_c}/_{T_c} = 0 \qquad 5\text{-}11$$

5.5 Irreversible Cycle

No real thermodynamic cycle is reversible. Irreversibility introduces entropy generation into the system and this lowers the work output, which, according to Equation 5-7, brings about an increase in the quantity of heat rejected at the sink. In other words, for an irreversible thermodynamic cycle:

$$\mathrm{w} < \mathrm{w}_{\mathrm{rev}} \text{ and } \mathrm{q}_{\mathrm{c}} > \mathrm{q}_{\mathrm{c,rev}}$$

This implies that:

$$\eta = 1 - {q_c}/{q_h} < \eta_{carnot} = 1 - {q_{c,rev}}/{q_h}$$

This validates the fact that the efficiency of the Carnot cycle is the maximum efficiency that any given cycle operating between the same temperature reservoirs can attain.

$$\int {\delta Q}/{T} = {q_h}/{T_h} - {q_c}/{T_c} < \int {\delta Q_{rev}}/{T}$$

This leads to the Clausius inequality:

$$\int {\delta \mathrm{Q}}/{\mathrm{T}} < 0 \qquad 5\text{-}12$$

5.6 Thermodynamic Processes of ORC System

ORC allows heat recovery from low-temperature sources such as industrial waste heat, geothermal heat, solar ponds, etc. It uses the low temperature heat to obtain useful work that can be used to generate electricity. The working principle of the cycle has been explained previously in section 2.1.1.1.

There are four major thermodynamic processes involved in an ideal ORC system, as shown in Figures 5-3 and 5-4.

- **Process 1–4**: this is an isobaric (constant pressure) heat addition to the working fluid contained in the heat exchanger. This process can be divided into three zones depending on the nature of the process. These zones include: preheating (1–2), evaporation (2–3) and superheating (3–4).

- **Process 4–5**: This is an isentropic expansion process in which it is assumed that there is no heat transfer (adiabatic), friction losses or fluid leakage in the expander.

- **Process 5–8**: This is also an isobaric heat removal process from the working fluid contained in the heat exchanger, otherwise known as the condenser. As in the process 1–4, this heat transfer process can also be subdivided into three zones, which include: de-superheating (5–6), condensation (6–7) and subcooling (7–8).

However, in a real ORC system there are irreversibilities which lower the cycle efficiency, as proved earlier through the Clausius theorem for irreversible cycles. The irreversibility occurs mainly during the following processes:

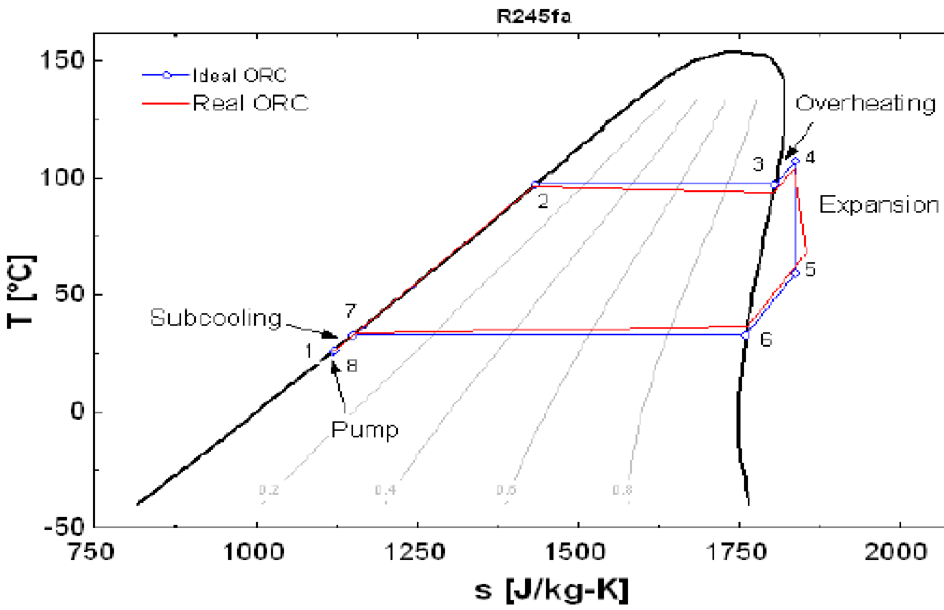


Figure 5–3: T-s Diagram of Ideal/Real ORC System (Quoilin, 2008)

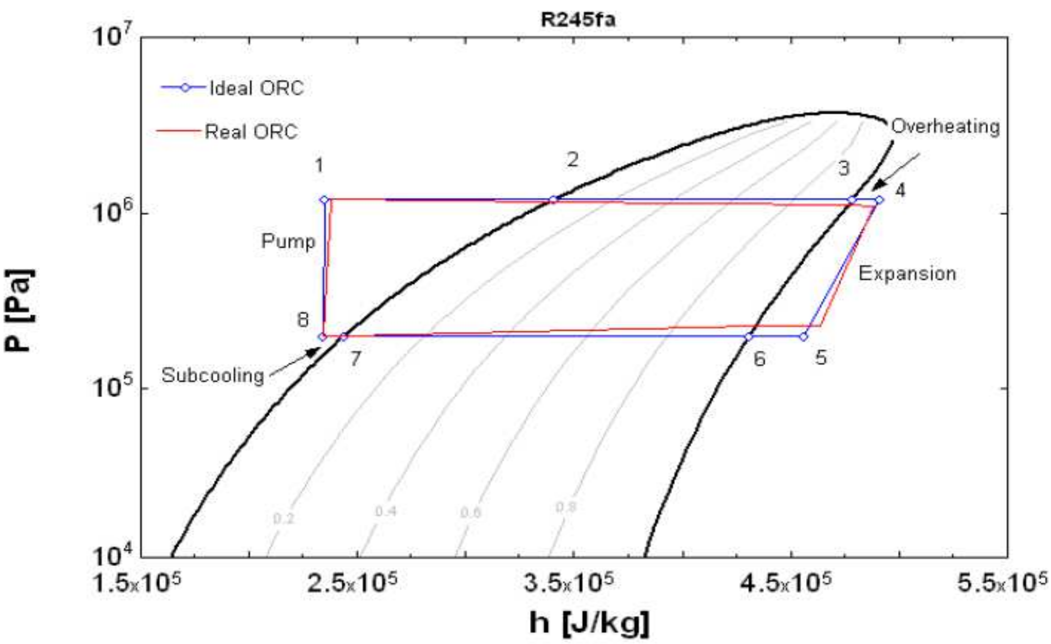


Figure 5–4: P-h Diagram of Ideal/Real ORC System (Quoilin, 2008)

- **Expansion**: in real ORC systems, the expansion process is never isentropic; hence, only part of the energy recovered from the pressure difference is transformed into useful work while the remaining part is lost as heat in the system. Also, the presence of leakages and friction losses can result in a loss in the efficiency of the turbine. Turbine efficiency is defined by comparison with an isentropic expansion.

- **Heat Exchange**: the exchange of heat in both the evaporation and the condensation sections of the process never occur at constant pressure as assumed in the ideal ORC system. The presence of pressure drops in the heat exchangers causes an increase in the pumping power and thus lowers the power obtained from a real ORC system.

- **Pumping**: The electro-mechanical losses and internal leakage lead to irreversibility which transforms part of the useful work to heat, thus reducing the overall efficiency of the real ORC cycle.

Hence, in order to develop a model of a real ORC system, all these losses have to be accounted for. In this project, the losses were lumped into the efficiency parameter in each of the individual unit operations.

5.7 Steady State Modelling of the Proposed ORC Unit Operations in IPSEpro MDK

As mentioned in section 3.1.2, the high level of flexibility in the IPSEpro simulation software allows the user to develop models of components or unit operation of a process plant using the IPSEpro MDK tool. This capability is implemented in this section to develop the model of the unit components which make up the ORC system proposed in this work.

A typical ORC system is made up of the following major components or unit operations:

- Evaporator
- Condenser
- Turbine/Expander
- Pump
- Motor
- Working Fluid Enthalpy Parameter

5.7.1 Evaporator (LMTD Method)

The function of the evaporator is to preheat and vaporize the organic working fluid (wf) (regarded as cold fluid) using the waste heat energy (wh). This involves heat transfer from the hotter fluid (waste heat) to the colder fluid (organic fluid) through a heat transfer area/surface.

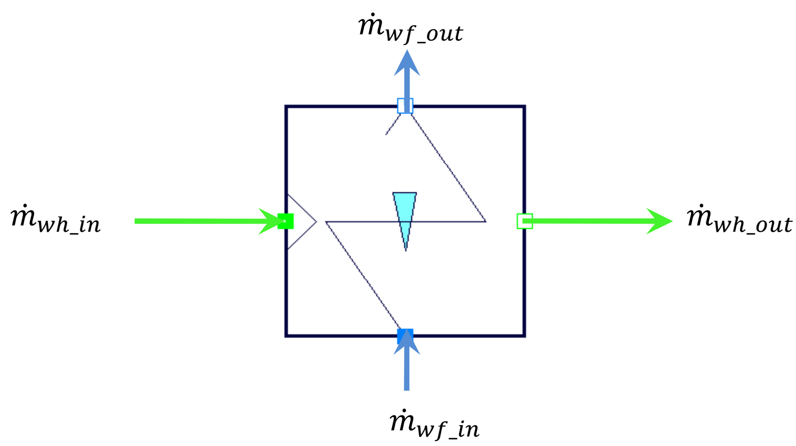


Figure 5–5: Evaporator

The evaporator is modelled as a counter-current heat exchanger system with the following model equations:

- **Mass Balance Equations**

$$\dot{m}_{wf\_in} = \dot{m}_{wf\_out} \tag{5-13}$$

$$\dot{m}_{wh\_in} = \dot{m}_{wh\_out} \tag{5-14}$$

- **Pressure Drops**

$$P_{wf\_in} - \Delta P_{wfs} = P_{wf\_out} \tag{5-15}$$

$$P_{wh\_in} - \Delta P_{whs} = P_{wh\_out} \tag{5-16}$$

- **Energy Balance**

$$\dot{m}_{wf\_in}\left(h_{wf\_in} - h_{wf\_out}\right) + q_{-trans} = 0 \tag{5-17}$$

$$\dot{m}_{wh\_in}\left(h_{wh\_in} - h_{wh\_out}\right) - q_{-trans} = 0 \tag{5-18}$$

- **Heat Exchanger Approaches**

$$T_{wh\_out} - \Delta T_{cold\_end} = T_{wf\_in} \tag{5-19}$$

$$T_{wh\_in} - \Delta T_{hot\_end} = T_{wf\_out} \tag{5-20}$$

- **UA Value**

if

$$\left|\frac{\Delta T_{cold\_end}}{\Delta T_{hot\_end}}\right| \,||\, \left|\frac{\Delta T_{hot\_end}}{\Delta T_{cold\_end}}\right| \geq 1.2 \tag{5-21}$$

then

$$q_{\_trans} * \ln\left(\frac{\Delta T_{cold\_end}}{\Delta T_{hot\_end}}\right) / (\Delta T_{cold\_end} - \Delta T_{hot\_end}) = UA \tag{5-22}$$

else 5-23

$$q_{\_trans} * \left(\frac{2}{\Delta T_{cold\_end} + \Delta T_{hot\_end}}\right) = UA \quad (5\text{-}24)$$

- **Model Parameters**

ΔP_{wfs} Pressure drop of the working fluid side of the heat exchanger

ΔP_{whs} Pressure drop of the waste heat side of the heat exchanger

- **Model Variables**

ΔT_{cold_end} Temperature difference at the cold end of the heat exchanger

ΔT_{hot_end} Temperature difference at the hot end of the heat exchanger

UA Heat exchanger thermal conductance

$q_{\_trans}$ Heat transfer rate

5.7.2 Condenser (LMTD Method)

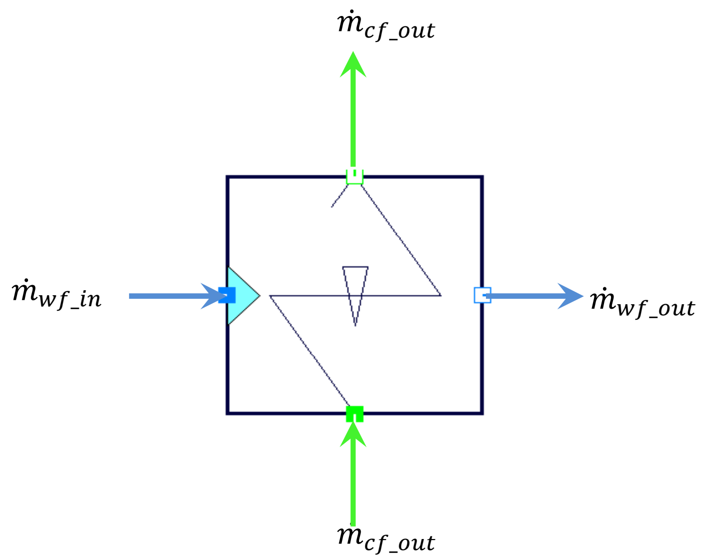


Figure 5–6: Condenser

- **Mass Balance Equations**

$$\dot{\mathrm{m}}_{\mathrm{wf\_in}} = \dot{\mathrm{m}}_{\mathrm{wf\_out}} \qquad 5\text{-}25$$

$$\dot{m}_{cf\_in} = \dot{m}_{cf\_out} \qquad 5\text{-}26$$

- **Pressure Drops**

$$\mathrm{P}_{\mathrm{wf\_in}} - \Delta\mathrm{P}_{\mathrm{wfs}} = \mathrm{P}_{\mathrm{wf\_out}} \qquad 5\text{-}27$$

$$\mathrm{P}_{\mathrm{cf\_in}} - \Delta\mathrm{P}_{\mathrm{cfs}} = \mathrm{P}_{\mathrm{cf\_out}} \qquad 5\text{-}28$$

- **Energy Balance**

$$\dot{\mathrm{m}}_{\mathrm{wf\_in}}\left(\mathrm{h}_{\mathrm{wf\_in}} - \mathrm{h}_{\mathrm{wf\_out}}\right) - \mathrm{q}_{\mathrm{-trans}} = 0 \qquad 5\text{-}29$$

$$\dot{\mathrm{m}}_{\mathrm{cf\_in}}\left(\mathrm{h}_{\mathrm{cf\_in}} - \mathrm{h}_{\mathrm{cf\_out}}\right) + \mathrm{q}_{\mathrm{-trans}} = 0 \qquad 5\text{-}30$$

- **Heat Exchanger Approaches**

$$\mathrm{T}_{\mathrm{wh\_out}} - \Delta\mathrm{T}_{\mathrm{cold\_end}} = \mathrm{T}_{\mathrm{wf\_in}} \qquad 5\text{-}31$$

$$\mathrm{T}_{\mathrm{wh\_in}} - \Delta\mathrm{T}_{\mathrm{hot\_end}} = \mathrm{T}_{\mathrm{wf\_out}} \qquad 5\text{-}32$$

- **UA Value**

if

$$\left|\frac{\Delta T_{cold\_end}}{\Delta T_{hot\_end}}\right| \,||\, \left|\frac{\Delta T_{hot\_end}}{\Delta T_{cold\_end}}\right| \geq 1.2 \qquad 5\text{-}33$$

then

$$\mathrm{q}_{\mathrm{\_trans}} * \ln\left(\frac{\Delta\mathrm{T}_{\mathrm{cold\_end}}}{\Delta\mathrm{T}_{\mathrm{hot\_end}}}\right) / (\Delta\mathrm{T}_{\mathrm{cold\_end}} - \Delta\mathrm{T}_{\mathrm{hot\_end}}) = \mathrm{UA} \qquad 5\text{-}34$$

else

$$q_{\_trans} * \left(\frac{2}{\Delta T_{cold\_end} + \Delta T_{hot\_end}}\right) = UA \qquad 5\text{-}35$$

- **Model Parameters**

ΔP_{wfs} Pressure drop of the working fluid side of the heat exchanger

ΔP_{whs} Pressure drop of the waste heat side of the heat exchanger

- **Model Variables**

ΔT_{cold_end} Temperature difference at the cold end of the heat exchanger

ΔT_{hot_end} Temperature difference at the hot end of the heat exchanger

UA Heat exchanger thermal conductance

$q_{\_trans}$ Heat transfer rate

5.7.3 Turbine/Expander

The turbine expands the vaporized working fluid, and depressurizes it to produce power which is transmitted to the generator via the shaft. There are four levels of turbine modelling which can be implemented in IPSEpro simulation software:

- Modelling based on the mechanical and isentropic efficiency of the turbine.

- Modelling based on turbine efficiency characteristics expressed in terms of relative isentropic efficiency as a function of relative mass flow.

- Modelling based on turbine characteristics expressed as a function of the pressure number.
- Modelling based on turbine characteristics expressed in terms of isentropic efficiency as a function of the ratio of blade velocity to flow velocity.

The first method uses the mechanical and isentropic efficiency of the turbine at the design point together with the properties of the working fluid to estimate the power output of the turbine. The last three methods require the performance characteristics of the turbine to be obtained through experimental set-up. The last three methods are very useful in determining the off-design performance of the turbine as well as in estimating the turbine efficiency at partial load condition.

In this research, the first method has been adopted. Although it is not the most appropriate considering the fact that the system will not operate only at full load condition, however, it has been adopted because of some constraints encountered during the course of this research work which mainly arise as a result of the unwillingness on the part of ORC turbine manufacturers to give out information on the performance characteristics of their ORC turbines for reasons of confidentiality. In order to overcome this constraint, some turbine efficiency data published in the literature were used (Turboden, 2011, Drbal *et al*., 1996, Aneke *et al*., 2011b).

Based on the above constraints, the turbine has been modelled using isentropic and mechanical efficiency data of an R245fa ORC turbine developed by United Technologies Corporation which was used to build the

Chena (Alaska, USA) geothermal ORC binary power plant (Holdmann, 2007). This author has validated the efficiency claim by UTC in one of his publications through modelling and simulation using the IPSEpro Process Simulation Software (Aneke *et al*., 2011b); however, this is only verified at the design point.

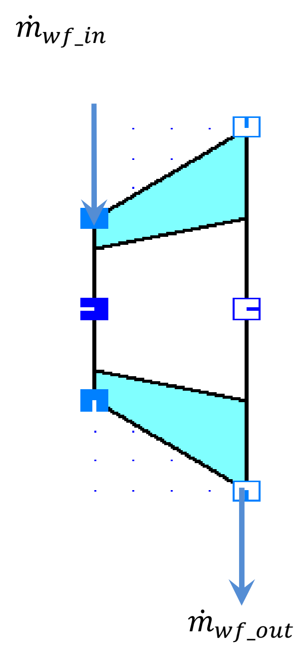


Figure 5–7: Turbine/Expander

- **Mass Balance Equations**

$$\dot{m}_{wf\_in} = \dot{m}_{wf\_out} \tag{5-36}$$

In IPSEpro, the specific entropy of the working fluid at the inlet to the turbines is calculated using the specific enthalpy code (IAPWS-IF97, 1997).

$$s_{wf\_in} = X_{wf\_out}.f\_s\_Ph(P_{wf\_out}, h_{wf_{in}} - \Delta h_s) \quad \text{5-37}$$

$$h_{wf\_in} - h_{wf\_out} = \Delta h_s * \eta_s \quad \text{5-38}$$

- **Energy Balance (Power Production)**

$$(h_{wf\_in} - h_{wf\_out}) * \eta_{mt} * \dot{m}_{wf\_in} = \text{Power Output} \quad \text{5-39}$$

- **Model Parameters**

η_{mt} mechanical efficiency of turbine

- **Model Variables**

η_s isentropic efficiency

Δh_s isentropic enthalpy difference

5.7.4 Pump

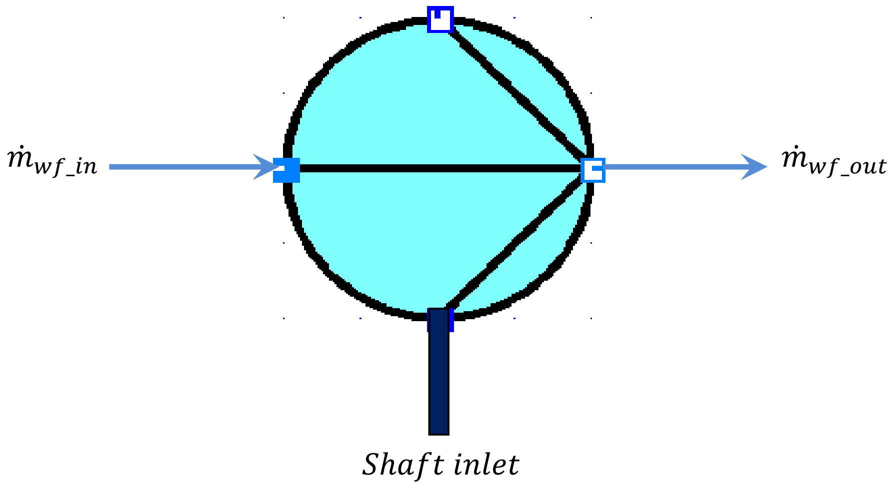


Figure 5–8: Pump

- **Mass Balance**

$$\dot{m}_{wf\_in} = \dot{m}_{wf\_out} \tag{5-40}$$

In IPSEpro, the specific entropy of the working fluid at the inlet to the turbines is calculated using the specific enthalpy code (IAPWS-IF97, 1997).

$$s_{wf\_in} = X_{wf\_out}.f\_s\_Ph(P_{wf\_out}, h_{wf_{in}} + (h_{wf\_out} - h_{wf\_in)}) * \eta_p) \tag{5-41}$$

- **Energy Balance (Power Requirement from the Shaft)**

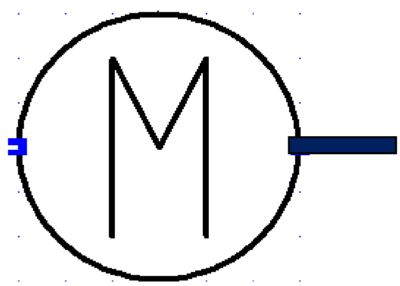
$$(h_{wf\_out} - h_{wf\_in}) * \dot{m}_{wf\_in}/\eta_{mp} = \text{Power Input} \tag{5-42}$$

- **Model Parameters**

η_{mp} mechanical efficiency of pump

η_p pump efficiency

5.7.5 Motor



Shaft outlet

Figure 5–9: Motor

- **Energy Balance Equation (Power to the Shaft)**

$$\text{Power Input} * \eta_e * \eta_{mm} \;=\; \text{Power Output} \tag{5-43}$$

- **Model Parameters**

η_{mm}	mechanical efficiency of motor
η_e	electrical efficiency

- **Model Variable**

Power Input	electrical power input

5.7.6 Working Fluid Enthalpy Parameter

In power cycle modelling, it is very important to determine the state of the working fluid as it goes through the cycle. This helps the design engineer to resolve operational problems such as turbine blade erosion or pump cavitation (Aneke *et al*., 2011b) which are caused by passing wet fluids through the expander and vapour through the pump respectively. The state of the working fluid as it goes through the cycle is determined using the enthalpy parameter function as defined below:

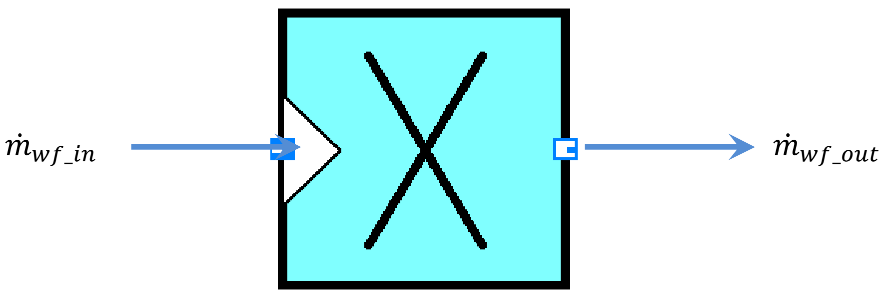


Figure 5–10: Enthalpy Parameter Box

- **Mass Balance Equation**

$$\dot{m}_{wf\_in} = \dot{m}_{wf\_out} \tag{5-44}$$

- **Pressure Balance Equation**

$$P_{wf\_in} = P_{wf\_out} \tag{5-45}$$

- **Energy Balance**

$$h_{wf\_in} = h_{wf\_out} \tag{5-46}$$

$$\frac{\left(h_{wf\_in} - X_{wf_{in}}.fh\_px\left(P_{wf\_in}, 0\right)\right)}{\left(X_{wf_{in}}.fh\_Px\left(P_{wf\_in}, 1\right) - X_{wf_{in}}.fh\_Px\left(P_{wf\_in}, 0\right)\right)} = x^{*} \tag{5-47}$$

- **Model Variable**

x^{*}	working fluid enthalpy parameter

From the definition of x^{*}, it can be seen that x^{*}<0 when the working fluid is at the sub-cooled phase, x^{*}=0 when the working fluid is saturated liquid, 0< x^{*}<1 when the working fluid is at the two-phase region, x^{*}=1 when the working fluid is saturated vapour, and x^{*}>1 when the working fluid is at superheated phase.

The screen shots showing the implementation of the model equations for the individual unit operations using the IPSEpro MDK tool are shown in Appendix A.

Chapter Six

In this chapter, the standalone ORC system capable of converting the waste heat from the fryer section of the crisp manufacturing plant were developed in the IPSEpro PSE tool, using the individual unit operations modelled in the previous chapter.

6 Proposed ORC System Utilizing Waste Heat from the Fryer for Power Generation

From Figure 4-2 it can be seen that there are two major sources of waste heat from the fryer section of the KP Billingham crisp manufacturing plant. These include:

- Waste heat from the foul gas from the fryer
- Waste heat from the exhaust gas to the stack

These individual waste heat sources identified above are capable of generating electricity using the ORC system. However, the choice of an ORC system configuration capable of utilizing the waste heat source in the most economical way for power generation is very important.

The use of a single ORC system for each of the waste heat sources or other configurations like the reheat cycle has been found to be less economical in terms of payback period than using a dual heat source ORC system capable of using both waste heat sources simultaneously in a single ORC system (Aneke *et al*., 2011a). Apart from being more economical, the dual-source ORC system has also been found to generate less entropy than every other configuration which the author has investigated, such as two single ORC systems, reheat ORC system, etc. (Aneke *et al*., 2012b).

Therefore, based on the above facts, the standalone ORC system proposed in this work for the conversion of the waste heat from the fryer section of the KP Billingham crisp manufacturing plant is the ***dual heat source ORC system****.*

The schematic diagram of the proposed system is shown in Figure 6-1*.*

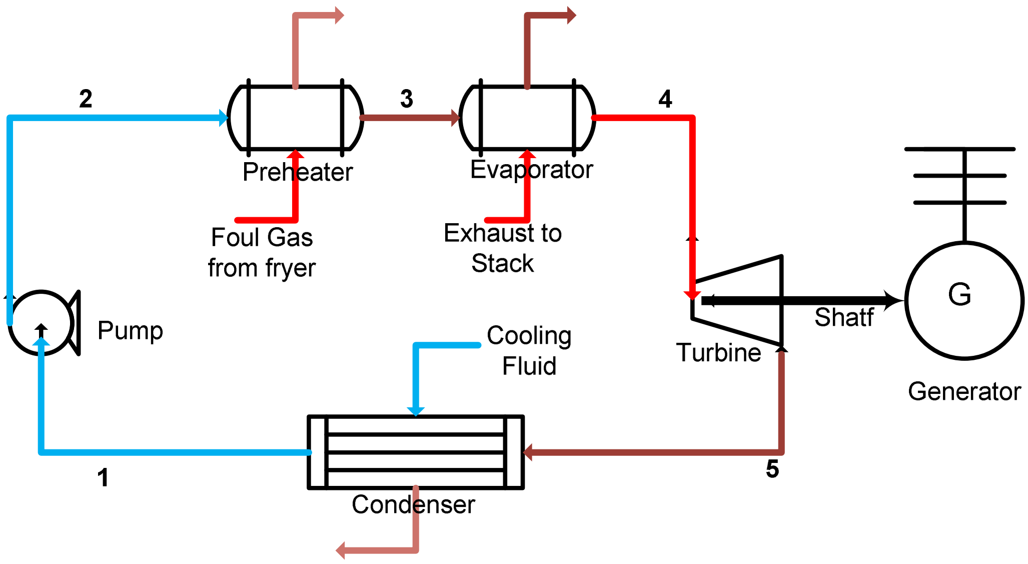


Figure 6–1: Proposed Dual Heat Source ORC System

6.1 Process Description of the Proposed ORC System

In the process, the liquid working fluid from the condenser (state 1) is pumped to the preheater (state 2) where the lower-temperature (120^{o}C) waste heat from the fryer is initially used to preheat the working fluid to a high-temperature sub-cooled liquid (state 3). The high-temperature liquid working fluid is passed from the first preheater to the preheater/evaporator unit, where the waste heat from the exhaust to stack is used to preheat/evaporate the working fluid to the saturated vapour phase (state 4). The working fluid at the saturated vapour phase is passed through the turbine, where it is expanded to produce work. The work is transmitted to the generator via the connecting shaft to generate electricity. The working fluid exiting from the turbine (state 5) is sent to the condenser where heat is rejected from the working fluid to the condensing fluid. The loss of heat in the condenser causes the condensation of the working fluid back to the liquid phase to complete the cycle.

6.2 Waste Heat Quality

6.2.1 Waste Heat from Fryer Foul Gas

The waste heat from the fryer foul gas is at a temperature of 120^{o}C, a mass flow rate of 3.172 kg/s, and a mass composition of 60.6% H_2O, 29.9% N_2 and 9.5% O_2.

6.2.2 Waste Heat from the Exhaust Stack

The waste heat from the exhaust stack is at a temperature of 164^{o}C, a mass flow rate of 10.51 kg/s and a mass composition of 5% CO_2, 41.1% H_2O, 50.6% N_2 and 3.3% O_2.

6.3 Building the IPSEpro Model of an ORC System Using IPSEpro PSE

The developed MDK models presented above were put together in the IPSEpro PSE module in order to develop the entire ORC model.

The model is developed using the following model parameter:

- **Expander Parameters**

Isentropic efficiency 0.80

Mechanical efficiency 0.98

- **Pump Parameters**

Efficiency 0.71

- **Motor Parameters**

Efficiency 0.72

- **Heat Exchanger Parameters**

Pressure drop in both the hot side and cold side 0.1 bar

- **Working Fluid Selection**

There is much literature on the use of different working fluids for developing ORC systems. Based on the waste heat source temperature (120–164^{o}C) and the literature and interactions with ORC manufacturers, the most suitable working fluid for this work is R245fa. The reason for this is because it has better thermodynamic properties than its major industrial rival R134a at the waste heat conditions under consideration.

The thermo-physical properties of the working fluid are shown in Table 6-1.

Table 6–1: Thermo-physical Properties of R245fa

Physical Properties of Refrigerant	**R245fa**
Environmental Classification	HFC
Ozone-depletion class	3
Global Warming Potential (CO_2 = 1)	950
Ozone Depletion Potential (CFC 11 = 1)	0
Molecular Weight	134.05
Boiling Point (101.325 kPa, C)	15.30
Critical Pressure (kPa)	3.64×10^3
Critical Temperature (C)	154.05
Critical Density (kg/m^3)	517
Liquid Density (21.11^{o}C, kg/m^3)	1339
Vapour Density (kg/m^3)	5.92
Heat of Vaporization (kg, kJ/kg)	196.83
Specific Heat Liquid (21.11^{o}C, kJ/kg.K)	1.38
Specific Heat Vapour (101.325 kPa, kJ/kg.K)	0.912
NEPA Classification	2/1/0

- **Condenser Cooling Fluid**

There are generally two types of condenser cooling mode, namely:

- Water Cooled Condenser
- Air Cooled Condenser

Water-cooled condensers are always better than air-cooled condensers, because water has better thermal properties than air. As a result of this, the condenser proposed for use in this project is a water-cooled condenser.

The condenser cooling water is modelled using the average annual weather conditions at Billingham, Cleveland, UK, which is the location of the crisp manufacturing plant. The most important weather property which will likely affect the plant operating condition is the ambient temperature.

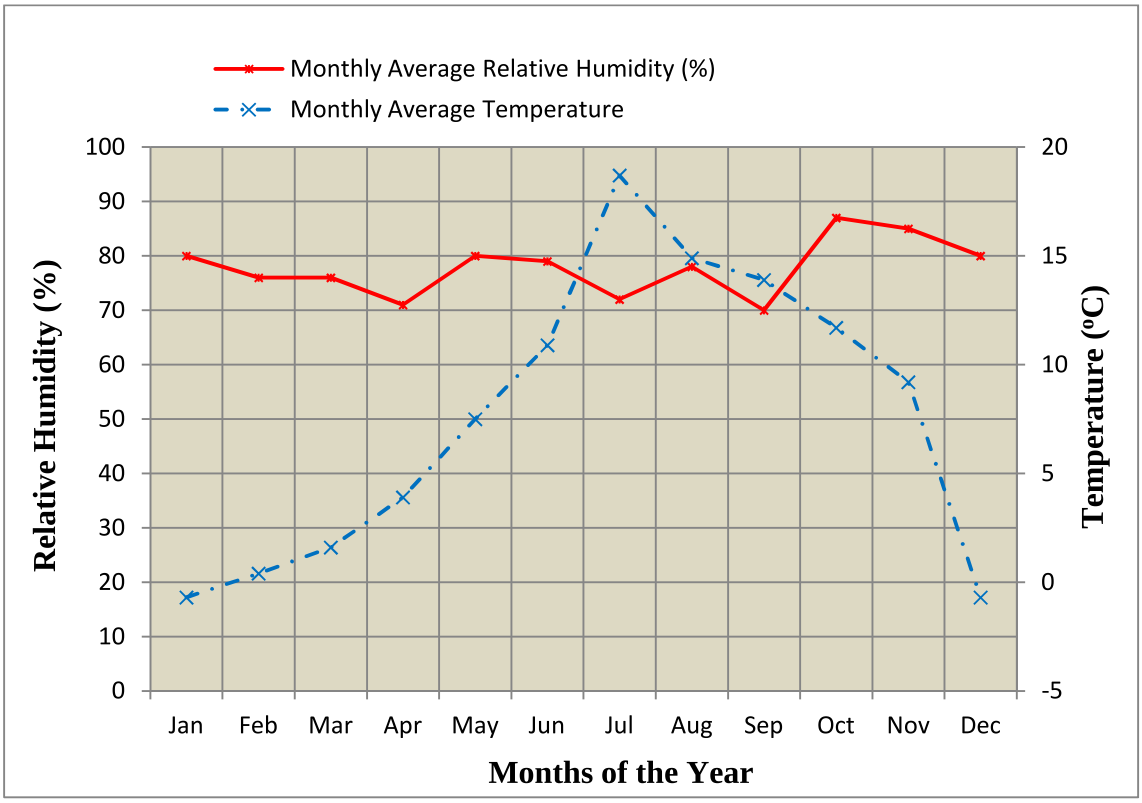


Figure 6–2: Monthly Average Weather Conditions in Cleveland in 2010

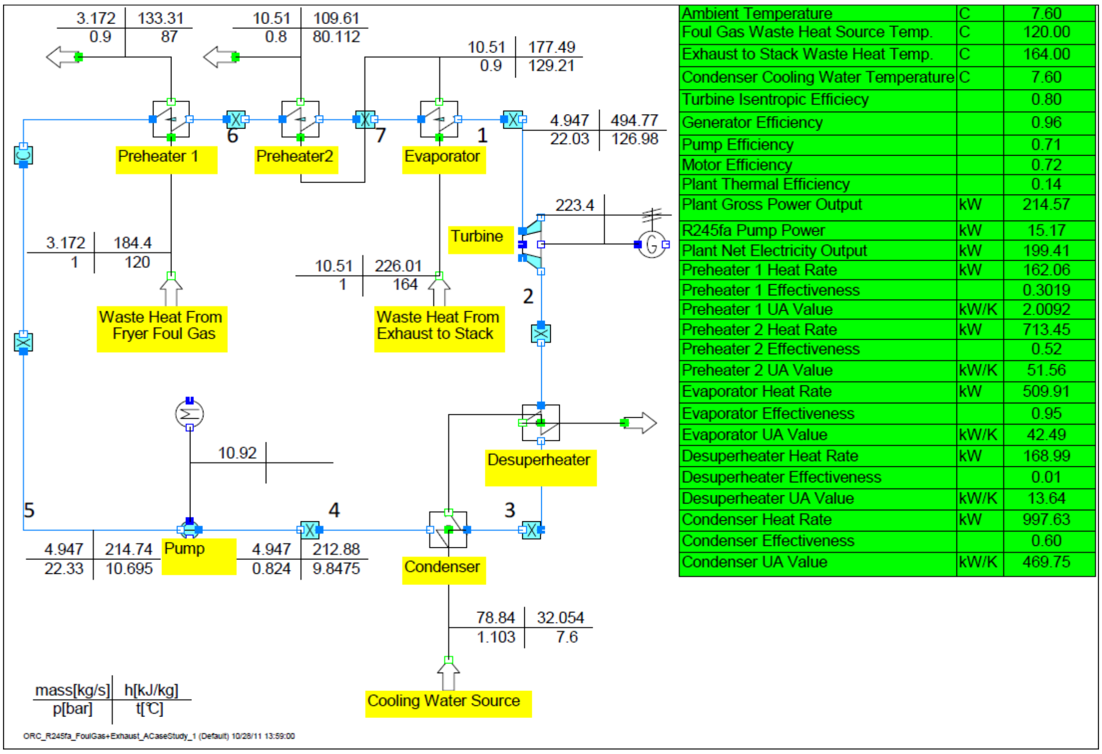
The values of the monthly average ambient temperature are shown in Figure 6-2 (Newlands Weather Station, 2011). The plant is modelled using the annual average ambient temperature of 7.60^{o}C.

Based on the parameters presented above, the IPSEpro PSE models of a dual heat source ORC system were developed using the following design constraints:

- The pinch point temperature difference in the heat exchangers is assumed to be equal to 2^oC.
- The condenser is water cooled with a source temperature of 7.6^oC.
- The working fluid is at the saturated vapour state at the entrance to the turbine.
- To avoid persistent fouling of the evaporator's surface (which adds to the cleaning cost) which might occur as a result of condensation of oil contained in the effluent waste streams, the waste heat must leave the evaporator as saturated vapour after being used to preheat and evaporate the working fluid.
- The pressure at the inlet to the turbine is set to achieve optimum power output and thermal efficiency from the ORC system while maintaining the design constraints.

6.3.1 Proposed Water Cooled Dual Heat Source ORC System

The IPSEpro PSE model of the proposed dual heat source ORC system using water as the condenser cooling fluid is shown in Figure 6-3 . The T-Q diagram of the preheater 2/evaporator, de-superheater/condenser and the T-s diagram of R245fa working fluid at design point are shown in Figures 6-4, 6-5 and 6-6 respectively. The thermodynamic processes exhibited by the working fluid are shown in Table 6-2.



Ambient Temperature	C	7.60
Foul Gas Waste Heat Source Temp.	C	120.00
Exhaust to Stack Waste Heat Temp.	C	164.00
Condenser Cooling Water Temperature	C	7.60
Turbine Isentropic Efficiecy		0.80
Generator Efficiency		0.96
Pump Efficiency		0.71
Motor Efficiency		0.72
Plant Thermal Efficiency		0.14
Plant Gross Power Output	kW	214.57
R245fa Pump Power	kW	15.17
Plant Net Electricity Output	kW	199.41
Preheater 1 Heat Rate	kW	162.06
Preheater 1 Effectiveness		0.3019
Preheater 1 UA Value	kW/K	2.0092
Preheater 2 Heat Rate	kW	713.45
Preheater 2 Effectiveness		0.52
Preheater 2 UA Value	kW/K	51.56
Evaporator Heat Rate	kW	509.91
Evaporator Effectiveness		0.95
Evaporator UA Value	kW/K	42.49
Desuperheater Heat Rate	kW	168.99
Desuperheater Effectiveness		0.01
Desuperheater UA Value	kW/K	13.64
Condenser Heat Rate	kW	997.63
Condenser Effectiveness		0.60
Condenser UA Value	kW/K	469.75

Figure 6–3: IPSEpro PSE Model of Water Cooled Dual Heat Source ORC System

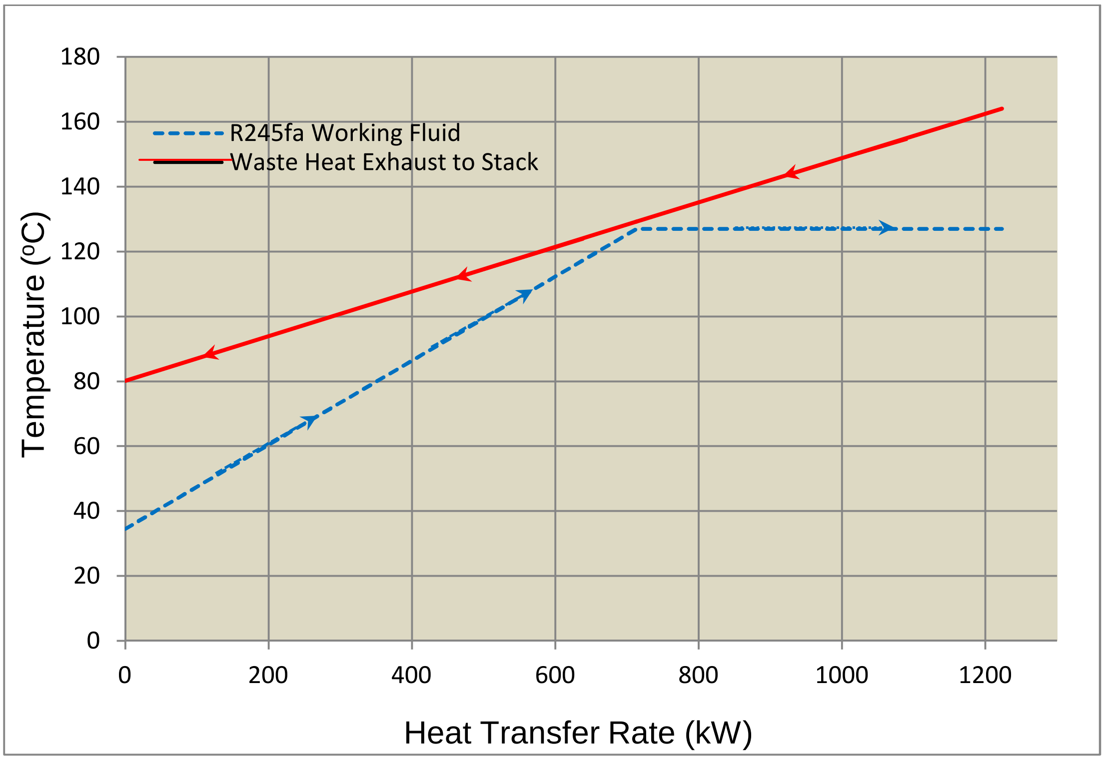


Figure 6–4: Preheater 2/Evaporator T-Q Diagram

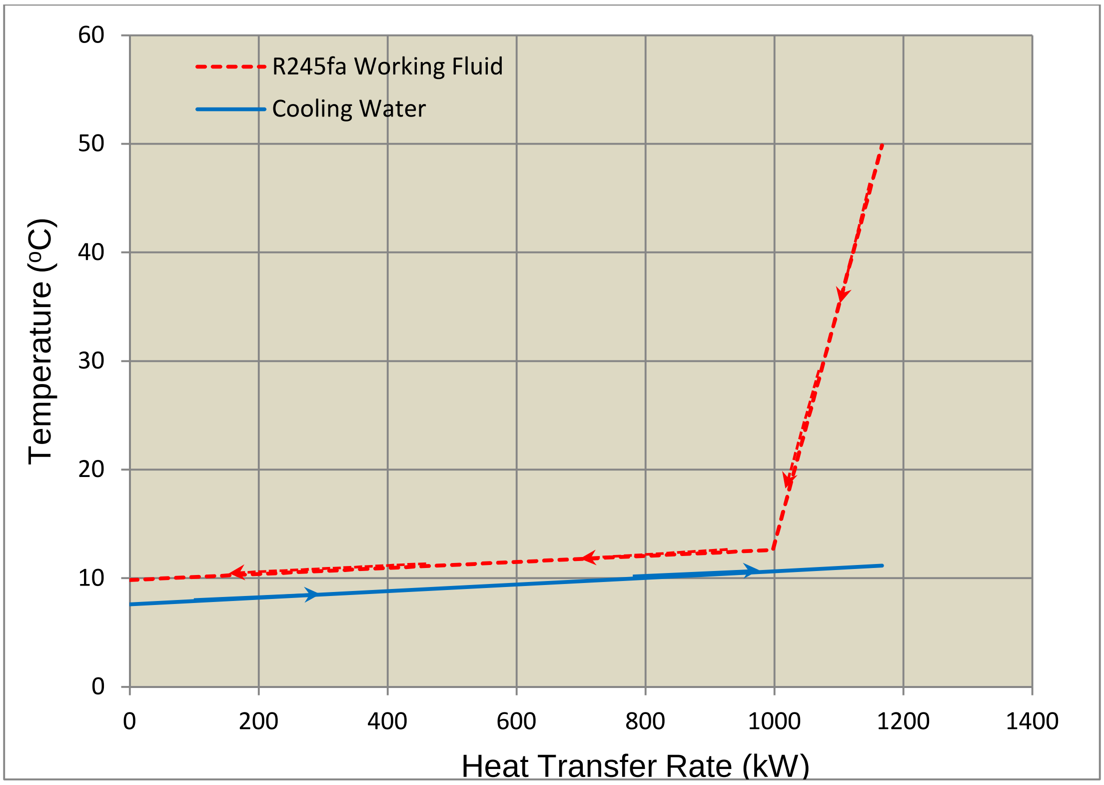


Figure 6–5: De-superheater/Condenser T-Q Diagram

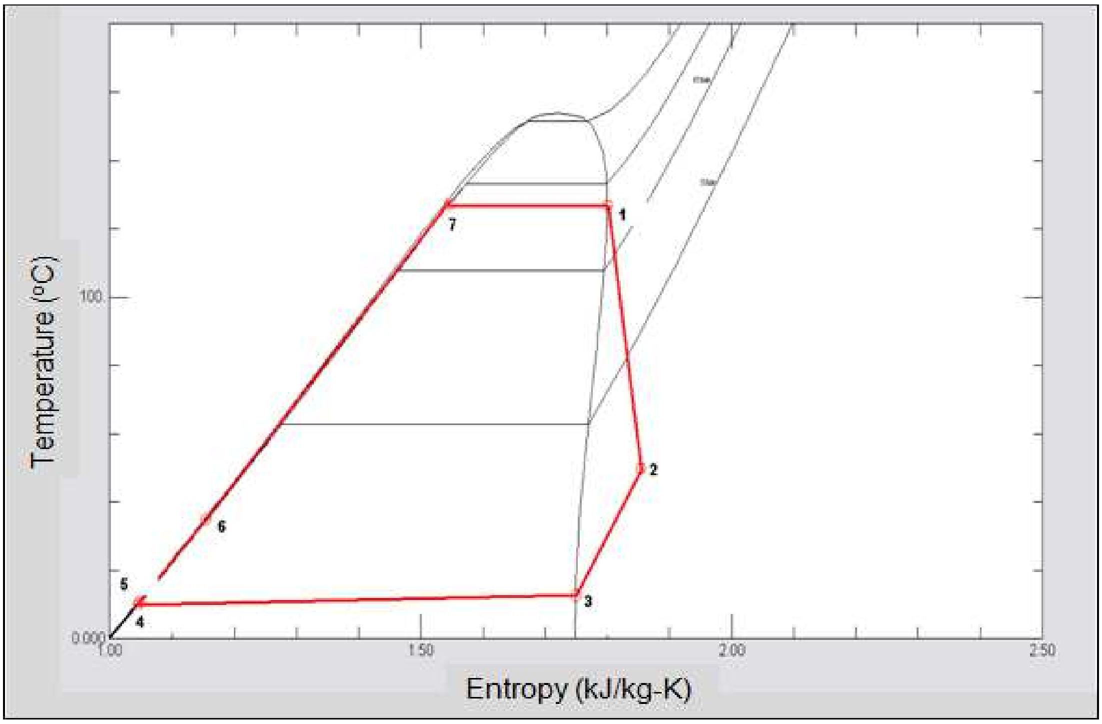


Figure 6–6: T-s Diagram of R245fa Working Fluid at Design Conditions

Table 6–2: Thermodynamic Processes Exhibited by the R245fa Working Fluid

States	Process
1–2	The high-pressure vaporized working fluid from the evaporator is expanded in the turbine to produce work.
2–3	The low-pressure working fluid vapour from the turbine is passed through the de-superheater where heat is removed from the working fluid (in order to turn it into saturated vapour) using the condenser cooling water.
3–4	The low-pressure saturated working fluid from the de-superheater is further cooled in the condenser to saturated liquid state.
4–5	The low pressure saturated liquid working fluid is pumped to high pressure using the pump.
5–6	The high pressure liquid working fluid is preheated in the first preheater using waste heat from the fryer foul gas.
6–7	The preheated high pressure working fluid from the first preheater is further heated to saturated liquid state using the waste heat from the exhaust to stack.
7–1	The high pressure saturated liquid working fluid is further evaporated in the evaporator (to complete the cycle) using the waste heat from the exhaust to stack.

6.4 Process Calculations

The power plant thermal efficiency $\eta_{thermal}$, heat exchangers (condenser and evaporator) NTU value, and effectiveness ε, were determined using the standard equations presented below:

The thermal efficiency of the plant, $\eta_{thermal}$ was calculated using Equation 6-1

$$\eta_{\mathrm{thermal}} = {(\mathrm{W_G} - \mathrm{W_P})}/{\mathrm{Q_{input}}} \qquad 6\text{-}1$$

where

$\mathrm{W_G}$ = Total power produced in the generator

$\mathrm{W_P}$ = Total power consumed by the pump

Q_{input} = Total energy input from the waste heat source to the system

The Number of Transfer Units (NTU) of the heat exchangers was calculated using Equation 6–2

$$\mathrm{NTU} = {\mathrm{UA}}/{\mathrm{C_{min}}} \qquad 6\text{-}2$$

where

C_{min} = Smaller heat capacity rate of the fluid that pass through the heat exchanger

In a heat exchanger which involves a phase-change of one of the fluids, the effective specific heat for the phase-changing fluid is infinity; therefore, C_{min} is the heat capacity of the non-phase changing fluid (waste heat and water) for both the evaporator and the condenser (Shah and Sekulic, 1998).

The effectiveness (ε) of the heat exchangers was calculated using Equation 6-3 shown below:

$$\varepsilon = {}^{Q}/_{Q_{max}} \qquad 6\text{-}3$$

where

$Q =$ Actual heat transfer rate in the heat exchanger

$Q_{max} =$ Maximum heat transfer rate in the heat exchanger

$$Q_{max} = C_{min}(T_{hin} - T_{cin}) \qquad 6\text{-}4$$

where

$T_{hin} =$ Inlet temperature of the hot fluid to the heat exchanger

$T_{cin} =$ Inlet temperature of the cold fluid to the heat exchanger

6.4.1 Entropy Generation Analysis of the Proposed Model

Entropy generation is a measure of dissipated useful energy and degradation of the performance of an engineering system (Demirel, 2002). It accounts for the extent of irreversibility present in such a system. This section shows the entropy generation analysis of the dual heat source ORC configurations proposed in this research.

Generally, the entropy balance relations for a control volume is given as

$$\Delta\dot{S}_{cv} = \sum\frac{\dot{Q}}{T} + \sum\dot{m}_{in}s_{in} - \sum\dot{m}_{out}s_{out} + \Phi \quad \text{6-5}$$

where

$\Delta\dot{S}_{cv}$ = rate of change of entropy of the system, kJ/s − K

$\dot{Q}$ = heat transfer rate, kJ/s

T = temperature, K

$\dot{m}$ = mass flowrate, kg/s

s = specific entropy, kJ/kg − K

Φ = entropy generation rate, kJ/s − K

For a general steady state flow process, $\Delta\dot{S}_{cv} = 0$, then Equation 6-5 reduces to

$$\Phi = \sum\dot{m}_{out}s_{out} - \sum\dot{m}_{in}s_{in} - \sum\frac{\dot{Q}}{T} \quad \text{6-6}$$

Applying the steady state entropy generation equation (Equation 6-6) to the dual heat source ORC configurations presented in this work.

6.4.1.1 Evaluation of the Entropy Generation for Different Components of the Proposed ORC Model

The ORC system presented in this work contains heat exchangers (preheater, evaporator and condenser), expander (turbine) and pump. The entropy generation in each of the components is evaluated as shown below using the steady state entropy generation equation (Equation 6-6)

- **Heat Exchangers (preheater, evaporator, de-superheater and condenser)**

Applying the entropy generation equation to the above heat exchanger, we have:

For the hot side

$$\Phi_{hot} = \dot{m}_{hout}s_{out} - \dot{m}_{hin}s_{in} \qquad 6\text{-}7$$

For the cold side

$$\Phi_{cold} = \dot{m}_{cout}s_{out} - \dot{m}_{cin}s_{in} \qquad 6\text{-}8$$

Therefore, for the overall heat exchanger

$$\Phi_{hex} = \Phi_{hot} + \Phi_{cold} \qquad 6\text{-}9$$

$$\Phi_{hex} = \dot{m}_{hout}s_{out} + \dot{m}_{cout}s_{out} - \dot{m}_{hin}s_{in} - \dot{m}_{cin}s_{in} \qquad 6\text{-}10$$

- **Expander (Turbine)**

Since expanders do not transfer heat, the heat transfer term of Equation 6-6 becomes zero and thus the entropy generation equation reduces to

$$\Phi_{exp} = \dot{m}_{out}s_{out} - \dot{m}_{in}s_{in} \qquad 6\text{-}11$$

- **Pump**

Similarly to expanders, pumps do not transfer heat, and the heat transfer term is therefore zero, thus the entropy generation equation reduces to

$$\Phi_p = \dot{m}_{out}s_{out} - \dot{m}_{in}s_{in} \tag{6-12}$$

The total entropy generation is given by

$$\Phi_{tot} = \Phi_{pre/eva} + \Phi_{con} + \Phi_{exp} + \Phi_p \tag{6-13}$$

6.5 Results and Discussion

The result of the simulation presented in Table 6-2 shows that the waste heat from the crisp manufacturing plant can be used to produce up to 199.40 kW of electricity at an efficiency of 14.39% while operating at the plant nominal design point. This value is more than the daily average electricity usage by the plant which currently stands at about 190.30 kW (see section 4.1). It also meets about 92% of the peak daily electricity usage, which was 216 kW at 18 hrs (see Figure 4-3). From the entropy generation analysis carried out, Preheater 2 is found to generate the most entropy in the system. It accounts for about 35.67% of the entropy, while the pump generates the least entropy in the system.

In a real system, there is always a deviation in the nominal design point during the day-to-day operation of the plant. This may arise due to variations in the ambient weather conditions or other disturbances in the system such as variations in the waste heat source temperature and mass flow rate.

Table 6–3: Table of Simulation Results

Operating Conditions	Unit	Value
Total Heat Input Rate	kW	1385.42
Gross Power Output	kW	214.57
Pump Power Consumption	kW	15.17
Net Power Output	kW	199.40
Plant Thermal Efficiency	%	14.39
Preheater 1 UA value	kW/K	2.01
Preheater 1 NTU		0.41
Preheater 1 Effectiveness		0.30
Preheater 2 UA value	kW/K	51.56
Preheater 2 NTU		6.66
Preheater 2 Effectiveness		0.52
Evaporator UA Value	kW/K	42.49
Evaporator NTU		2.90
Evaporator Effectiveness		0.95
De-superheater UA Value	kW/K	13.64
De-superheater NTU		3.01
De-superheater Effectiveness		0.01
Condenser UA value	kW/K	469.75
Condenser NTU		1.42
Condenser Effectiveness		0.06
Ambient Temperature	^{o}C	7.60
Cooling Water Inlet Temperature	^{o}C	7.60
Cooling Water Outlet Temperature	^{o}C	11.15
Cooling Water Mass Flow rate	kg/s	78.84
R245fa Working Fluid Mass Flow rate	kg/s	4.95
Entropy Generation Analysis		
Preheater 1	kW/K	0.25
Preheater 2	kW/K	0.56
Evaporator	kW/K	0.45
Turbine	kW/K	0.16
De-superheater	kW/K	0.10
Condenser	kW/K	0.04
Pump	kW/K	0.01

Any variations in the waste heat temperature, mass flow rate or cooling water temperature will definitely have a significant impact on the operation of the proposed ORC system. This influence is examined in the next section through sensitivity analysis of the proposed ORC system.

6.6 Sensitivity Analysis of the Proposed ORC System

Real systems are always influenced by variations in some of the plant operating parameters. These variations usually cause disturbance in the plant operating conditions and hence make plant performance deviate from the nominal design conditions.

In the case of this project, there are three major operating conditions which will likely influence the plant performance. They include: changes in the waste heat temperature and mass flow rate, as well as changes in the cooling water temperature (ambient conditions).

In order to carry out this sensitivity analysis to reflect real plant conditions, it is assumed that the plant is designed, fabricated and commissioned based on the nominal (base load) design point parameters (Aneke *et al*., 2011b). Based on this assumption, some of the plant's operating parameters were fixed at the same as that of the nominal (base load) design point. Using the base design operating conditions, all the components of the plant were fixed in size using the approach proposed by Price and Hassani (2002). This was achieved by fixing the pinch point temperature of the heat exchangers at the same value as that of the base load (Price and Hassani, 2002), as well as fixing the isentropic and mechanical efficiency of the turbine at the same value as that of the base load condition (Aneke *et al*., 2011b).

Hence, by fixing the sizes of the components using the above mentioned criteria, the performance of the plant during variations in waste heat temperature, mass flow rate and cooling water temperature is obtained as follows.

6.6.1 Effect of Variation in Foul Gas and Exhaust to Stack Temperature at Fixed Cooling Water Temperature and Condenser Pressure

Although the plant was designed with foul gas and exhaust to stack temperature of 120^{o}C and 164^{o}C respectively, however, in a real system, there is every tendency that the waste heat temperatures mentioned above will deviate from the base load condition. In this case study, the waste heat temperature from the foul gas was assumed to vary from 110 to 130^{o}C while that of the exhaust to stack was assumed to vary from 156 to 168^{o}C.

Figure 6–7 shows the graph depicting the effect of variation in the waste heat temperature on the net power output of the dual heat source ORC system. The graph shows that for any given waste heat temperature from the exhaust to stack, the net power output from the plant increases as the foul gas temperature increases. This occurs because as the foul gas temperature increases for any given exhaust to stack temperature; more working fluid is vaporized in the evaporator (i.e. more working fluid is used in the system) (see Figure 6–8) and this gives rise to more power output.

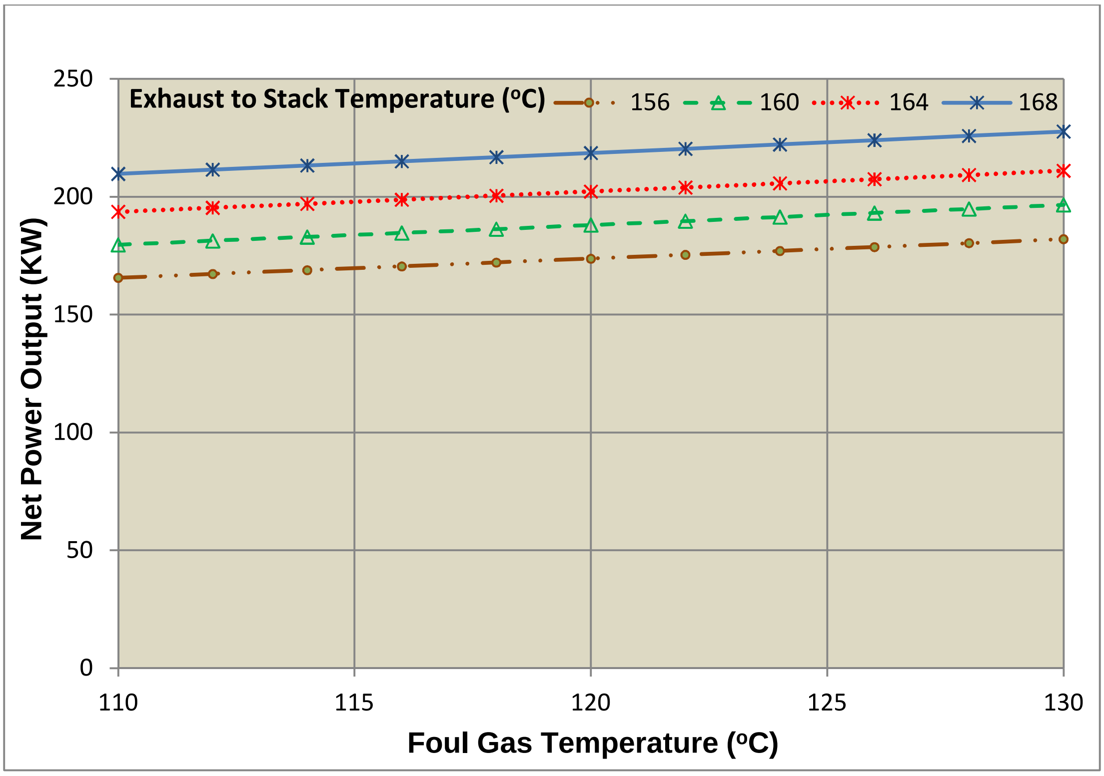


Figure 6–7: Effect of Variation in Waste Heat Temperature on Net Power Output

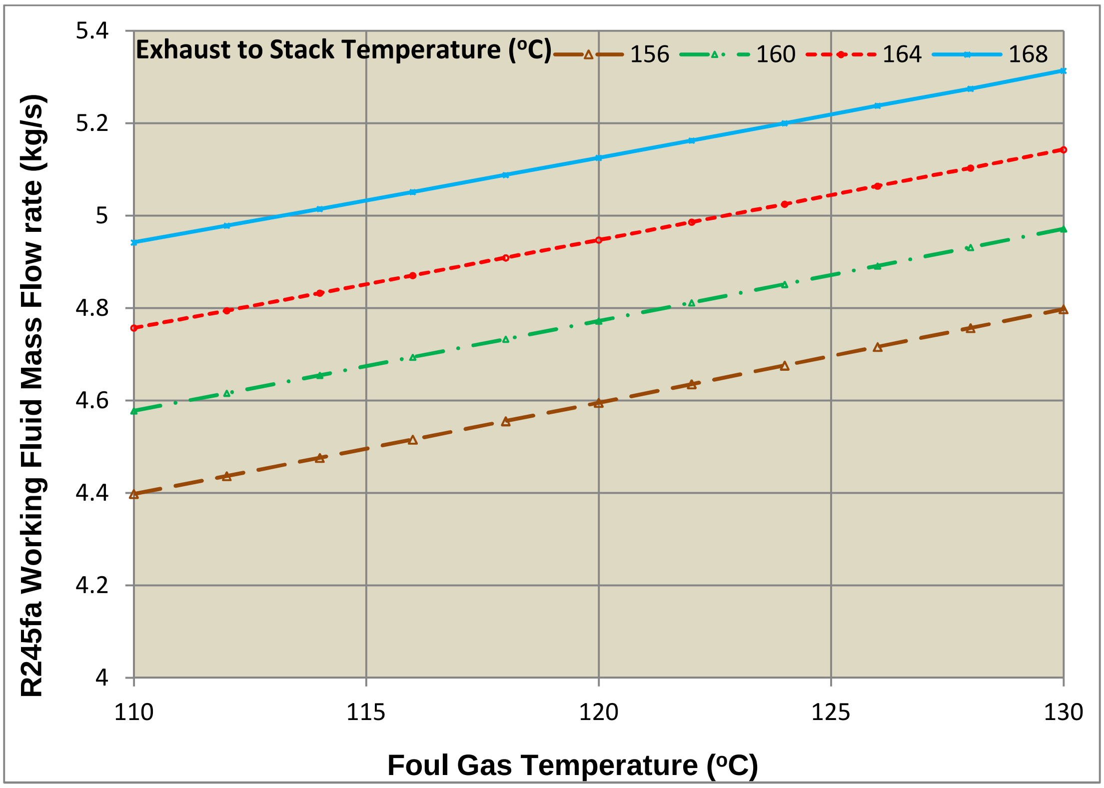


Figure 6–8: Effect of Variation in Waste Heat Temperature on Working Fluid Mass Flow Rate

Figure 6-9 shows the effect of the variation in the foul gas temperature on the cooling water demand for a given exhaust to stack waste heat temperature. The graph shows that there is an increase in cooling water demand when the waste heat temperature increases. The reason for this is because when the waste heat temperature increases, more working fluid is used in the system and thus in order to cool the working fluid in the condenser, more cooling water will be required.

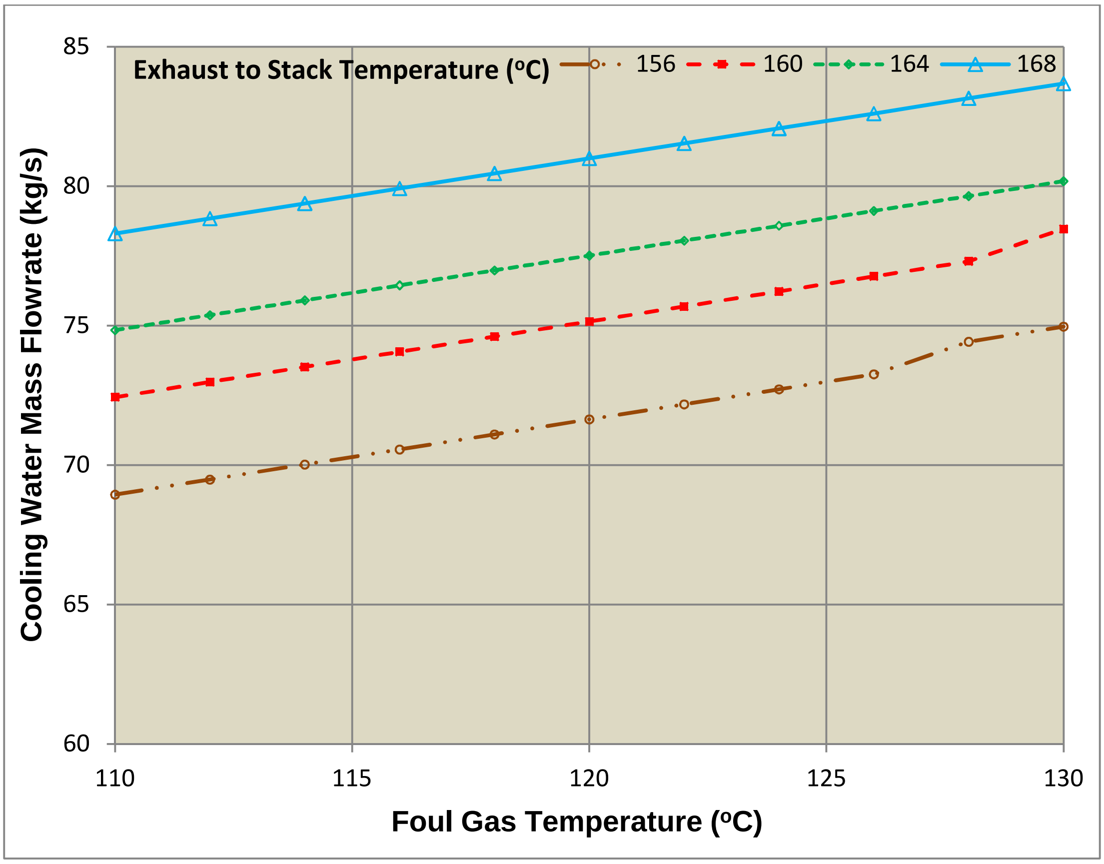


Figure 6–9: Effect of Variation in Waste Heat Temperature on Cooling Water Mass Flow Rate

6.6.2 Effect of Variation in Foul Gas and Exhaust to Stack Mass Flow Rate at Fixed Cooling Water Temperature and Condenser Pressure

Apart from the waste heat source temperature, there is every tendency that the mass flow rate of the waste heat source will vary considerably, and this variation will affect the performance of the ORC power plant under investigation. In order to carry out this investigation, the waste heat source temperature and the cooling water source temperature are fixed at values similar to the design conditions. The size of the components of the plant was fixed as explained earlier in section 6.5.

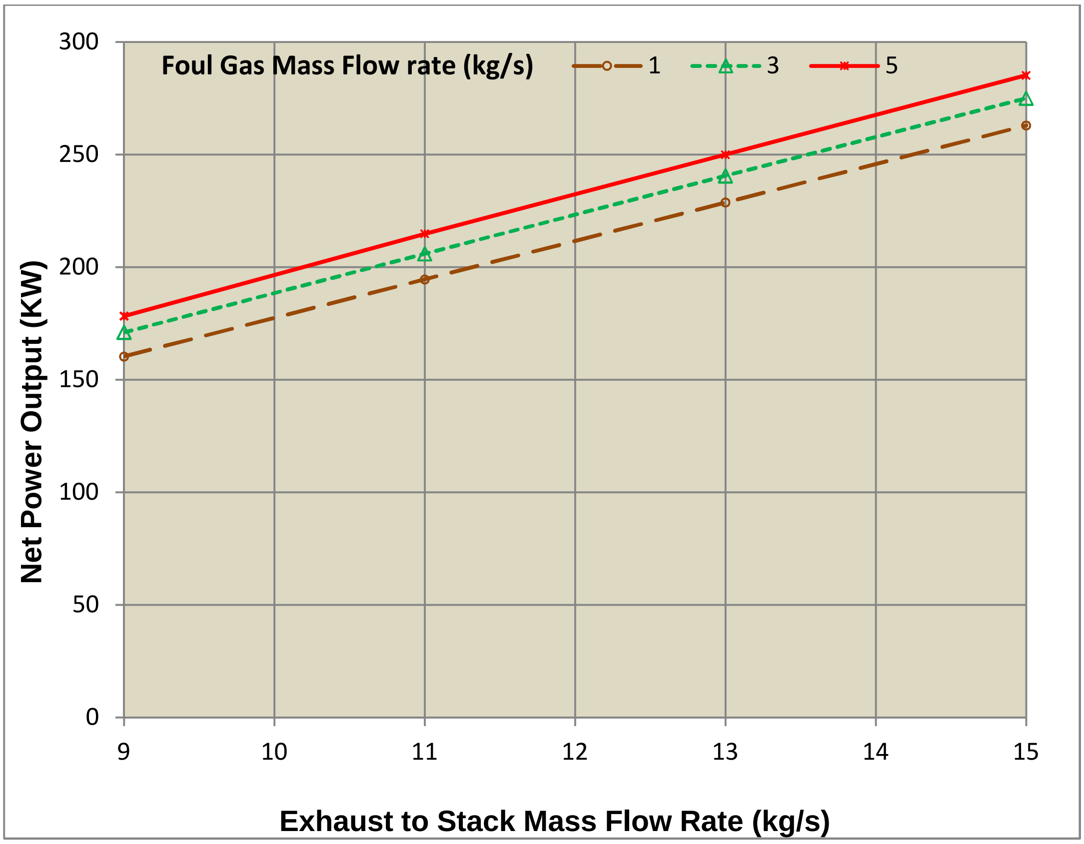


Figure 6–10: Effect of Variation in Waste Heat Mass Flow Rate on Net Power Output

Figure 6-10 shows the effect of changes in the waste heat source mass flow rate on the net power output of the proposed dual heat source ORC plant. The graph shows that an increase in the exhaust to stack mass flow rate for any given foul gas mass flow rate causes an increase in the net power output of the plant. The reason is because increase in waste heat source mass causes an increase in the working fluid mass flow rate (see Figure 6-11) through the cycle, and this gives rise to an increase in the amount of power

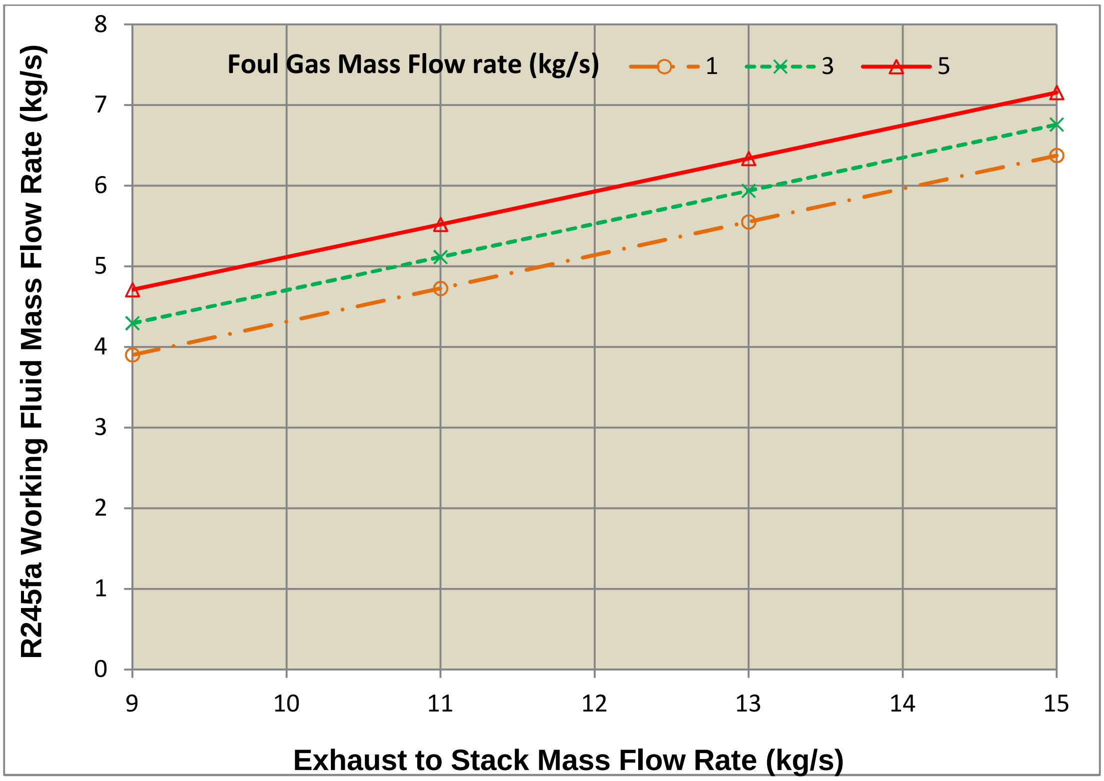


Figure 6–11: Effect of Variation in Waste Heat Mass Flow Rate on Working Fluid Requirement

produced by the working fluid. Figure 6-12 shows the effect of variation in waste heat mass flow rate on the cooling water requirement in the condenser. The graph shows that an increase in waste heat source mass causes a corresponding increase in cooling water requirement. This is because an increase in the waste heat source causes an increase in working fluid requirement in the system and in order to condense the working fluid, more cooling water is needed in the system.

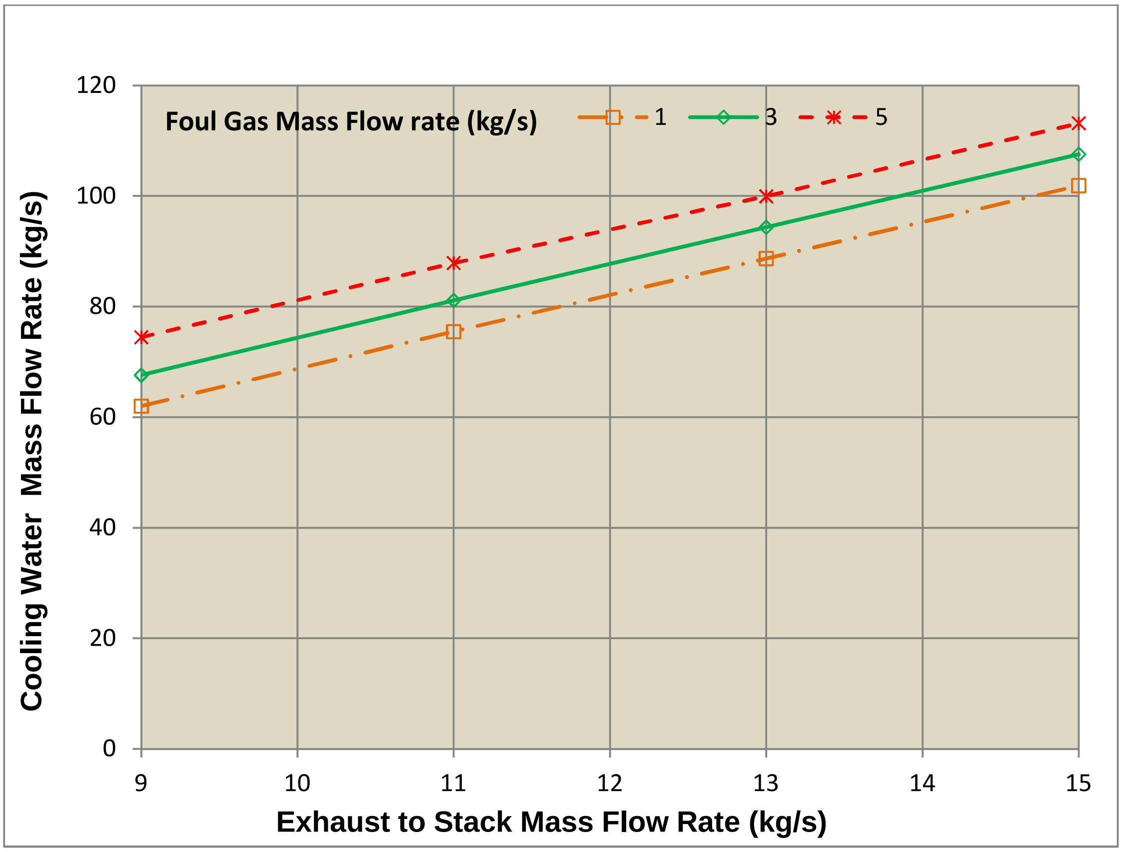


Figure 6–12: Effect of Variation in Waste Heat Mass Flow Rate on Cooling Water Requirement

6.6.3 Effect of Variation in Cooling Water Temperature (Ambient Temperature) on Net Power Output

Figure 6-13 shows the effect of variation in cooling water temperature on the net power output of the proposed dual heat source ORC power plant. The graph shows that an increase in the cooling water temperature will give rise to a corresponding decrease in the net power output of the plant. Hence this shows that the plant will give a better performance in the winter than in the summer, since the former is characterized by a lower ambient temperature than the latter.

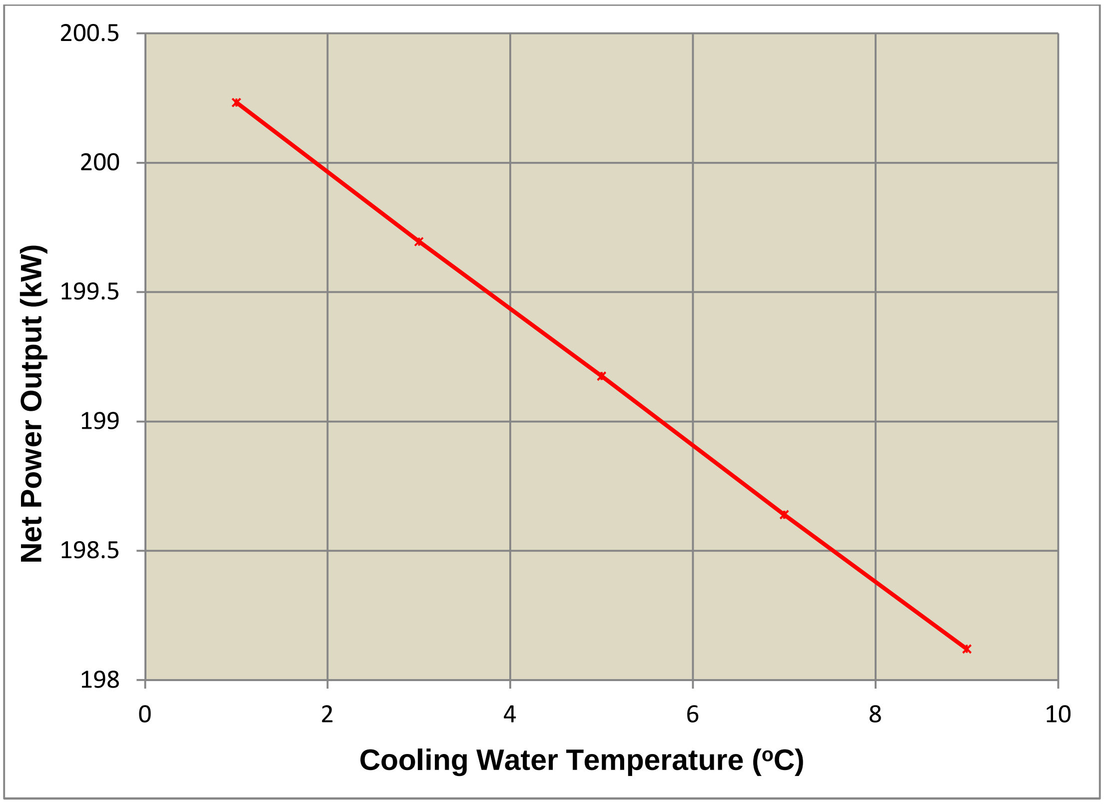


Figure 6–13: Effect of Variation in Cooling Water Temperature on Net Power Output

6.6.4 Effect of Variation in the Turbine Isentropic and Mechanical Efficiency on Net Power Output

As mentioned in section 5.7.3, ORC turbine isentropic and mechanical efficiency claim by some ORC manufacturers remains controversial. Many ORC manufacturers refuse to disclose performance characteristics of their turbine. Isentropic efficiency claims as high as 90% has been published in the literature (Turboden, 2011, Drbal *et al*., 1996). Although this claim might be true when the turbine is operating at its best efficiency design point under full load condition however, in real life situations, the system will operate at off design conditions and studies have shown that at off design conditions, the efficiency of the turbine tends to reduce. Hence, it is worth investigating what effect this reduction in turbine efficiency will have in the overall performance of the ORC system. This parametric study is carried out by varying the efficiency of the turbine from 40% to 95% while maintaining other process parameters at the nominal design condition. The simulation result is shown in Figure 6–14. As expected, the result shows that the net power output decreases as the turbine efficiency decreases and vice versa.

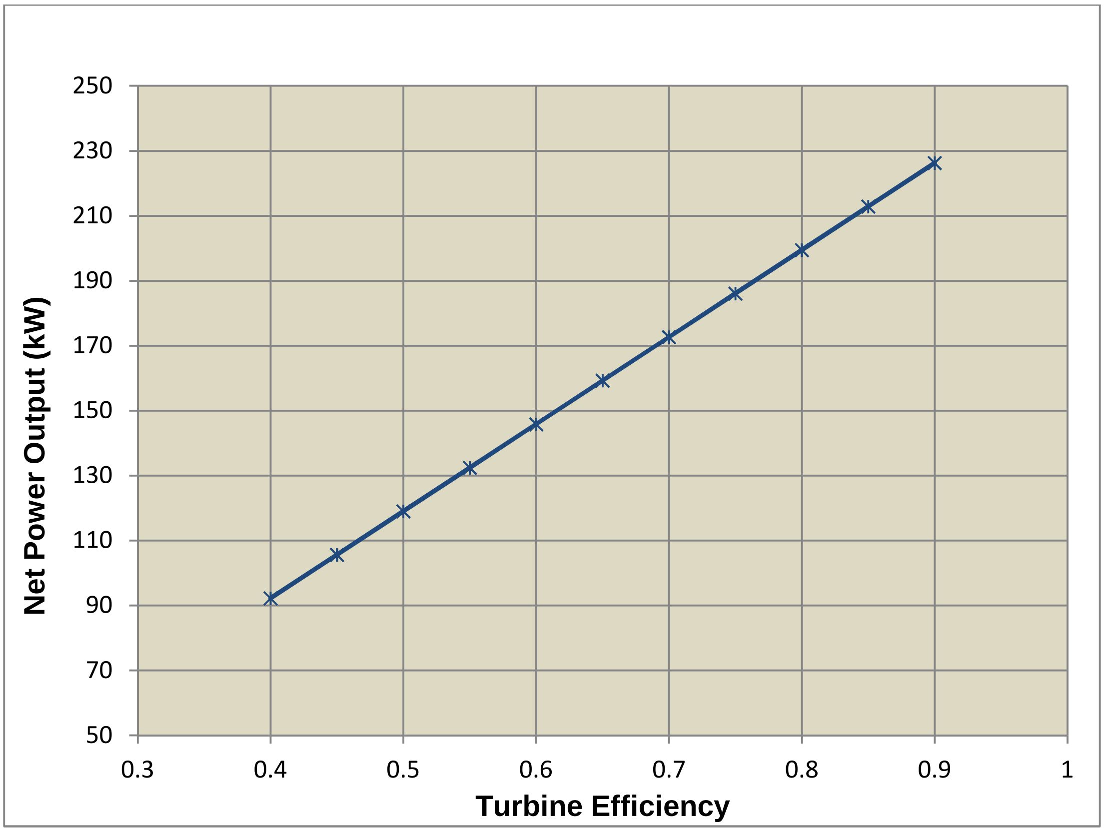


Figure 6–14: Effect of Variation in Turbine Efficiency on Net Power Output

6.7 Concluding Remarks

In this chapter, it has been established through thermodynamic modelling and simulation using the IPSEpro simulation tool that the waste heat from the foul gas from the fryer and exhaust to stack can be effectively utilized for power generation using a dual heat source ORC system, as proposed in this work.

The theoretical study shows that up to 199 kW of electricity and an efficiency of about 14% can be obtained when the plant is operating at nominal design conditions. Through the parametric/sensitivity analysis carried out, it was established that changes in the nominal (base load) conditions, such as variations in the waste heat source temperature and mass flow rate, as well as variations in the ambient conditions, will cause the plant to deviate from its design operating conditions and this will affect the entire behaviour of the plant in terms of power output, cooling water requirement, and working fluid demand. Hence, in order to minimize the effect of the variation in plant operating conditions, there is a need to incorporate adequate control systems in the plant module. However, this is not within the scope of this research.

Chapter Seven

In this chapter, the Life Cycle Assessment (LCA) of the proposed waste heat recovery technology is carried out, to establish the environmental impact (in terms of carbon emission reduction) of the inclusion of the proposed dual heat source ORC system in the frying section of the crisp manufacturing plant under study together with its economical importance.

7 Evaluation of the Environmental Impact and Economic Viability of the Proposed Technology

Almost all technologies have an impact on the environment, which may be positive or negative. The evaluation of this impact helps to establish the environmental sustainability of any proposed technology, and also to inform the decision-makers on whether to accept the use of the technology or pick some possible alternatives. A sustainable energy system is one which balances energy production and consumption with minimal negative impact on the environment (Poeschl *et al*., 2011). The multi-criteria decision-making procedure for any sustainable energy system includes: technology (primary energy ratio), economy (investment cost), environment (CO_2 emission) and society (job creation).

Since the aim of this project is to recover waste heat energy and minimize CO_2 emission from the crisp manufacturing plant under study (see section 1.6), the only criteria considered here are the environmental and economic ones, with emphasis on the study of CO_2 emissions reduction and economic viability of the proposed system. This study is carried out using the Life Cycle Assessment (LCA) approach.

7.1 Life Cycle Assessment (LCA)

LCA is a powerful technique used to evaluate the impact of any technology or product on the environment in order to establish its sustainability. It is a crucial tool for a better understanding of how a technology may reduce the environmental footprint (Piemonte *et al*., 2011). The standard LCA evaluates the whole life cycle of a product from 'cradle to grave' (i.e. from raw material extraction to decommissioning) (Pehnt, 2007, Frick *et al*., 2010, Piemonte *et al*., 2011, Parliamentary Office of Science and Technology Postnote, 2006).

However, there may be other system boundaries which can be studied depending on the scope of the work; for example, the system boundaries may be from raw material to production ('cradle to gate') (Piemonte *et al*., 2011).

Since the plant under study will serve as a retrofit installation to an existing plant, and since the carbon footprint of the proposed ORC technology from the cradle to the installation as a retrofit to the existing crisp manufacturing plant has to be covered by the ORC plant manufacturer, the part considered in this work which should be of concern to the crisp manufacturing plant will be from the operation phase to the decommissioning phase of the proposed ORC plant. Based on this, a new system boundary is defined in this study. The LCA covered in this study will be from 'gate to grave' (i.e. from the 'use phase' to the decommissioning phase).

7.1.1 LCA Procedure

In compliance with ISO 14040 (2006) and ISO 14044 (2006), the LCA is achieved through four distinct phases, which include (Parliamentary Office of

Science and Technology Postnote, 2006, Piemonte *et al*., 2011, Frick *et al*., 2010, Poeschl *et al*., 2011):

- Goal and scope of the study
- Inventory analysis
- Impact assessment
- Interpretation/improvements

7.1.1.1 Goal and Scope Definition

The goal of this LCA is to assess the environmental impact (in terms of CO_2 emissions) of the use of the low-grade waste heat from the fryer section of the crisp manufacturing plant under study to generate power, using the dual heat source ORC system proposed in this project. The environmental effects are analysed in reference to CO_2 emission savings that would not occur if the electricity was generated using other conventional means, such as fossil fuel or energy mix technologies.

The functional unit adopted in this study is the net power output of the plant at base load which is equal to 199.40 kW (see Table 6-3). In order to account for the plant downtime due to overhaul, maintenance and varying power output due to changing ambient conditions, the plant is assumed to be in continuous operation at full load for about 7000 h/year. The plant life is assumed to be 30 years.

7.1.1.2 Life Cycle Inventory (LCI)

The inventory is the most objective result of the LCA study, referring mainly to measures of mass and energy. It also covers the raw material and energy consumption and the emission of solid liquid and gaseous wastes. However, it does not say anything about the environmental impact of a particular emitted substance. The ‘carbon footprint’ is one main output from the LCI step.

For this process under study, the energy and material flow into the proposed dual heat source ORC plant includes the waste heat energy and mass from the fryer section of the crisp manufacturing plant, while the energy outflow from the system boundary includes the heat lost at the condenser and power output from the generator.

7.1.1.3 Life Cycle Impact Assessment (LCIA)

This phase of the LCA involves the analysis of the LCI phase outputs to determine their impacts. The impact considered in this work is the CO_2 emission.

The CO_2 emissions from the plant without the addition of the dual heat ORC system are about 0.526 kg/s (estimated from section 6.2.2). This will produce CO_2 emissions from the exhaust to stack of about 13,255,200 kg/year (~13,255 t/year). This gives an emission of about 397,656,000 kg of CO_2 if the plant operates for 30 years without the proposed dual heat source ORC system. With the inclusion of the dual heat ORC system proposed in this work, which generated 199.40 kW of electricity at base load (with zero operational CO_2 emission) using the waste heat emitted from the crisps manufacturing plant, the annual electricity obtained would be about 1.3958 GWh/year. If this electricity is fed back to the grid or reused in the plant, it will displace electricity which would have been obtained using other conventional methods of electricity generation.

The benefits of integrating the dual heat source ORC system into the plant are presented in different scenarios (depending on the source of grid electricity to be displaced) as shown below:

- **Scenario 1: Conventional Coal Power Plant**

Data obtained from the literature shows that emission from coal fired power plant in UK is about 1,000 gCO_2eq/kWh (Parliamentary Office of Science and Technology Postnote, 2006). Using the electricity generated from the dual heat source ORC system under study to displace that from conventional coal power plant would reduce CO_2 emissions by 1,395,800 kg/year. This would reduce the total CO_2 emissions from the plant from 13,255,200 kg/year to 11,859,400 kg/year (~ 11% reduction). This would give a CO_2 saving of 41,874,000 kg throughout the life of the plant.

- **Scenario 2: Oil Fuel Power Plant**

The average carbon footprint of an oil-fired electricity generation plant in the UK is about 650 gCO_2/kWh (Parliamentary Office of Science and Technology Postnote, 2006). Using the electricity generated from the dual heat source ORC system under study to displace that from an oil-fired power plant would reduce CO_2 emissions by 907,270 kg/year. This would reduce the plant CO_2 emissions from 13,255,200 kg/year to 12,347,930 kg/year (~7% reduction).

This would give a CO_2 saving of 27,218,100 kg throughout the life of the plant.

- **Scenario 3: Gas Power Plant**

Gas-powered electricity generation in the UK has a carbon footprint of around 500 gCO_2eq/kWh) (Parliamentary Office of Science and Technology Postnote, 2006). Displacing this with the electricity from the dual heat source ORC system under investigation would give a CO_2 reduction of 697,900 kg/year. This would result in the reduction of the CO_2 emission from the crisp manufacturing plant from 13,255,200 kg/year to 12,557,300 kg/year (~5% reduction), with a CO_2 saving of 20,937,000 kg throughout the life cycle of the plant.

- **Scenario 4: Energy Mix Power Plant**

Most of the time, the electricity feed to the grid system is achieved using an energy mix. A typical example of energy mix obtained from the updated UK energy projection shows an energy mix of 30% coal, 17% nuclear power, 37% natural gas, 11% renewable, 3% imports, 0.57% oil and 0.85% pumped storage, with CO_2 emissions of 394 g/kWh (Gibbins *et al*., 2006). Using the electricity generated from the dual heat source ORC system under investigation to displace the electricity generated through the energy mix would give a CO_2 reduction of 549,945.20 kg/year. This would reduce the CO_2 emission of the plant from 13,255,200 kg/year to 12,705,254.80 kg/year (~4% reduction), with a CO_2 saving of 16,498,356 kg of throughout its life cycle.

7.1.1.4 Interpretation

From the analysis, it can be found from the LCIA that it is beneficial environmentally to use the waste heat from the crisp manufacturing plant to generate power using the dual heat source ORC cycle. As expected, the worst case scenario occurs when the waste heat is not utilized at all and is allowed to be emitted into the environment, while the best case scenario occurs when the electricity generated from the dual heat source ORC system is used to displace the electricity supplied from a conventional coal-fired power plant. The different benefits in CO_2 reduction through the use of the proposed technology is shown at a glance in Figure 7–1.

Although this analysis is focused on the reduction of CO_2 emissions, the inclusion of the dual heat source ORC plant in the system will also help to reduce heat pollution caused by emission of the high-temperature waste heat into the environment.

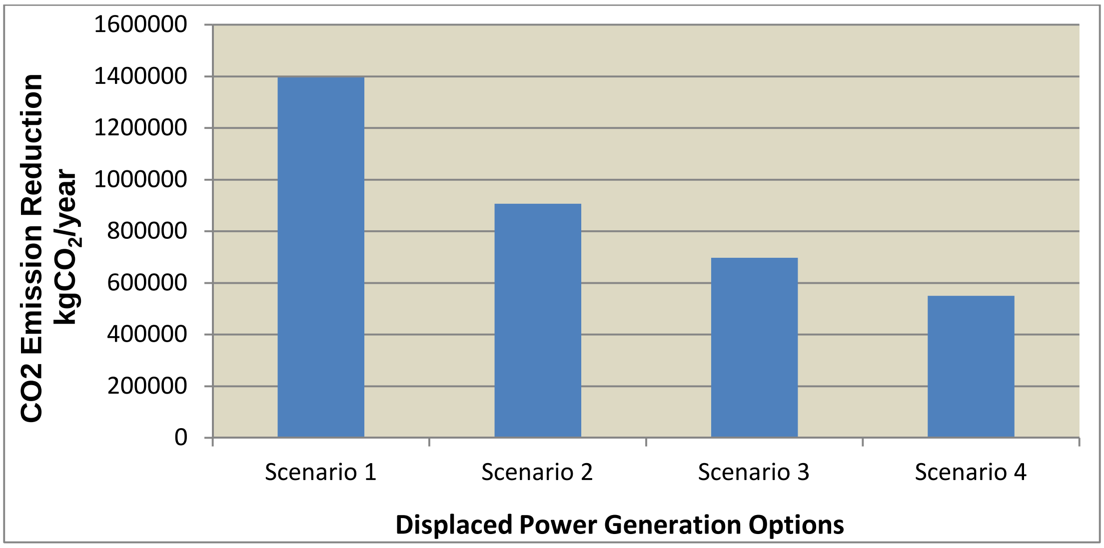


Figure 7–1: CO_2 Emission Reduction Associated with Electricity Production using Waste Heat from the Fryer Section of the Crisp Manufacturing Plant

7.2 Concluding Remarks: Environmental Benefits

The LCA study indicates that there are environmental benefits of utilizing the waste heat from the fryer section of the crisp manufacturing plant to generate power using the dual heat source ORC system proposed in this work.

Although there is no adequate data from the literature to carry out the environmental impact in terms of CO_2 emission associated with decommissioning and maintenance of the proposed ORC plant, experience drawn from a similar published work on enhanced geothermal power plant using an ORC system (Frick *et al*., 2010), in which the decommissioning operation was not considered, implied that the environmental impact (in terms of CO_2 emission) during the decommissioning operation might be negligible.

7.3 Economic Analysis of the Proposed Model

From the previous section, it was established that the proposed project is environmentally viable. However, the environmental viability alone does not give much justification to decision-makers to approve whether a project is worth execution or not. In this present era of environmental sustainability, an attractive project is one which is both environmentally and economically friendly. Hence, to fully establish whether a project is worth execution, it is customary to evaluate its environmental as well as its economic viability.

In engineering economy, the economic analysis of any proposed project is evaluated through profitability assessment study. As a common practice, it is always useful not to rely solely on a single investment analysis criterion, as the information that can be drawn from each one of them may support the decision of different types of investors (Rentizelas *et al*., 2009). There are many profitability assessment methods established in the literature; however, only four of them which are commonly used are considered in this thesis. They include:

- **Payback Period (PBP)**
- **Net Present Value (NPV)**
- **Average Rate of Return (ARR)**
- **Net Benefit Cost Ratio (NBCR)**

7.3.1 Payback Period

As the name implies, the payback period is the length of time it takes for the initial investment to be repaid out of the net cash inflows from the project. During decision making, it is a common practice to accept investments with a lower payback period than those with a higher one. In payback period calculation, the time value of money is usually disregarded. It is the most easily used economic analysis tool, especially in energy efficiency projects. It is also easy to apply; however, it is regarded as the method of analysis with the greatest limitations, because of its inability to account for the time value of money, risks, financing and so on.

Mathematically, the payback period is defined as:

$$\mathrm{TCI} = \sum_{\mathrm{j}=1}^{\mathrm{PBP}} \mathrm{NCF_j} \qquad \text{7-1}$$

where

TCI：Total Capital Investment

NCF：Net Cash Flow

7.3.2 Net Present Value

Net present value calculates all incomes and outgoings in the economic life of the project (Ziher and Poredos, 2005). The net present value can be either positive or negative. A positive NPV indicates a net gain corresponding to the cash flow, while a negative present value indicates a net loss. Projects with negative NPV are usually rejected, while those with the highest NPV among various alternatives are always given the highest preference.

This is always welcomed as a better alternative than the payback period because it puts into consideration the time value of money and other risk indicators associated with a project. This time value is accommodated in the calculation using the discounted rate parameter r, which is defined as the rate of return that could be earned on an investment in the financial markets with similar risk. Its value may be constant or variable throughout the life of the project. The calculation with a variable discounted rate value is always considered to be close to a real-life scenario, since the discounted rate varies in real-life investment.

In mathematical terms, the NPV is defined as

$$\mathrm{NPV} = \sum_{j=1}^{t} \mathrm{NCF_j} \Big/ (1+\mathrm{r})^{j} - \mathrm{TCI} \qquad 7\text{-}2$$

where

TCI：Total Capital Investment

NCF：Net Cash Flow

t : Time

r： Discount rate (the rate of return that could be earned)

7.3.3 Average Rate of Return

This is defined as the ratio of the average annual net profit to the total capital investment. A project with a positive and higher value of ARR is better than one with a lower value.

Mathematically this can be represented as

$$\text{ARR} = \overline{\text{NP}}/_{\text{TCI}} \qquad 7\text{-}3$$

where

$\overline{\text{NP}}$：The average annual Net Profit

7.3.4 Net-Benefit Cost Ratio (NBCR)

As the name implies, the benefit cost ratio is the ratio of the net present value of future net cash flow to the initial investment over its entire life. It is also known as profitability index (PI). A NBCR value of unity is logically the lowest acceptable measure of the index. Any value lower than unity would indicate that the project's NPV is less than the initial investment.

NBCR can be written mathematically as

$$\mathrm{NBCR} = {}^{\mathrm{NPV}}/_{\mathrm{TCI}} \qquad 7\text{-}4$$

7.3.5 Evaluation of Costs and Revenues

From the individual profitability assessment methods presented above, it can be seen that they all require the evaluation of the cost and revenue parameters.

The cost parameters generally consist of the initial investment cost, operation cost and maintenance cost of the plant throughout the life of the project, while the revenue parameters include all forms of cash inflow into the project, as a result of the sale of products generated from the project or savings made through the execution of the project.

In this economic evaluation, the different costs associated with the project were estimated using data obtained from the manufacturer (where available). In the absence of manufacturer data, data obtained from similar projects published in the literature were used. In cases where the capacity of the reference project and year of execution is different from the one under study, the cost is updated using either regression analysis, the sixth-tenth rule (see Equation 7-5) or the Chemical Engineering Plant Cost Index (CEPCI) (see Equation 7-6) equations respectively.

$$C_n = C_o \left({}^{P_n}/_{P_o} \right)^e \qquad 7\text{-}5$$

where

C_n : cost of equipment to be estimated

C_o : known cost of existing equipment

P_n : Capacity of new equipment

P_o : Capacity of existing quipment

e : exponent with values ranging from 0.2 to 1 (0.6 is the most common)

$$C_n = C_o \left({CEPCI_n}/{CEPCI_o} \right) \tag{7-6}$$

where

$CEPCI_n$: Index value at present time

$CEPCI_o$: Index value at time of original purchase

7.3.5.1 Cost Estimation

Proper cost estimation is important in all aspects of a project. In most engineering practice, the estimation of costs receives much more attention than revenue. For this project, where the proposed dual heat source ORC system would come as a complete module from the vendor/manufacturer and would be installed as a retrofit to an existing plant, the costs, which are of significant importance, can be classified into two major groups, namely:

- **Direct Costs**
- **Indirect Costs**

Direct Costs

Direct costs are those costs that are directly associated with the purchase and installation of equipment: They can be subdivided into the following:

- **Purchase Equipment Costs**
- **Installation Costs**
- **Costs of Material and Labour**

Estimation of Purchased Equipment Costs

As the name implies, these are the costs of the purchased equipment. In the case of this project, it is the cost of the proposed dual heat source ORC system.

Due to the inability to obtain cost data from the manufacturer (Infinity Turbine®) for the particular ORC size proposed in this study, the cost of the proposed system is estimated through regression analysis, using the cost–size relationship generated from the price list of various ORC sizes produced by Infinity Turbine®. However, the cost does not include that of the R245fa working fluid proposed for this design. The price list from the manufacturer and the cost estimation using regression plot are attached in Appendix B.

The cost–size relationship obtained from the regression analysis gives

$$C_n(\mathrm{US\$}) = 2075.7\, P_n + 24004 \qquad 7\text{-}7$$

Applying Equation 7-7 and substituting the size of the proposed ORC models $P_n = 199.40 \text{ kW}$ gives the cost of the proposed ORC system as US$437,898.58 (£282,223.89).

Estimation of Cost of R245fa Working Fluid used in the Proposed Dual Heat Source ORC Model

The price list for the ORC systems provided by the ORC manufacturer (Infinity Turbine®) does not include the price of the R245fa working fluid. The cost of this is obtained from R245fa refrigerant vendor (an extract of the chat with the vendor is shown in Appendix C). This is given as US$9.85/kg (£6.35/kg), with a shipping cost to the UK of US$1300/container (£837.84/container). Each container contains 14 cylinders, each of 1,000 kg. This gives a shipping cost of US$0.09/kg (£0.06/kg). Hence, the total cost of the R245fa refrigerant including shipping is given as US$9.94/kg (£6.41/kg). From information obtained from one of the ORC manufacturers, an ORC with capacity of about 200 kW would require a refrigerant charge of about 250 kg (Brasz and Zyhowski, 2005). Based on this, the cost of refrigerant for the proposed ORC system would be US$2,485.00 (£1601.57).

Estimation of Combined Installation, Material and Labour Costs

The installation cost of the ORC system, as a retrofit project to an existing plant, is estimated using correlation obtained from a similar project in which an ORC system was integrated as a retrofit project to an existing biomass plant to achieve a Combined Heat and Power (CHP) application at Lienz, Austria. From the study carried out by the ORC manufacturer (Turboden, Italy), it was found that the installation cost (including fittings) of a 1,000 kW_e

to the existing biomass plant was about 4.8% of the cost of the ORC module (Obernberger *et al*., 2002). Applying this relationship to the proposed system, the installation, material and labour costs of the proposed ORC system can be evaluated as US$20,928.98 (£13,488.64).

Estimation of Cost of the Control System

Integration of an ORC system to an existing plant and its adequate operation requires the use of control systems. In this project, the control system is given as a percentage of the ORC module, using a percentage value similar to that used in the similar project mentioned earlier (Obernberger *et al*., 2002). In this project it was found that the control system cost was about 2.65% of the ORC module cost. Applying a similar percentage to this project, the control system cost would be about $11,591.43 (£7,470.63).

Estimation of Grid Connection Cost

Generation of electricity from the ORC plant requires that the electricity be reused in the process or connected to the grid system. Assuming the power generated in this system is connected to the grid system, certain costs will be incurred in the project to achieve that. The cost of connecting the electricity produced to the grid varies with the location of the power plant. It is also a weak function of quantity of the power produced. Due to inadequate information on how to evaluate this cost, it is assumed to be the same as that for a reference project carried out by Turboden® in 2001 where an ORC plant was retrofitted to a biomass plant (Obernberger *et al*., 2002). The cost has been updated to the 2011 equivalent using the Chemical Engineering

Plant Cost Index (CEPCI) equation (Equation 7-6). Based on the above assumption, the cost is evaluated to be about £111,866.45.

Indirect Costs

Indirect costs include other costs such as the cost of freight to deliver the equipment to the plant site with associated insurance and taxes. Also involved in this category is the cost of engineering, including salaries for projects and process engineers, procurement expenses and so on. These costs are usually very difficult to estimate. In this work, the indirect costs considered include the cost of freight and the engineering cost.

Estimation of Engineering Costs

Based on the correlation from the aforementioned real-life project in Lienz, Austria, the cost of engineering is estimated to be about 15% of the cost of the ORC module. Implementing a similar correlation in this project, the cost of engineering would amount to about US$65,684.79 (£42,333.58).

Estimation of Freight Costs, Import Duties, VAT and Taxes

Most of the major ORC manufacturers are based outside the UK. Hence there will be a need to include the cost of transportation, import duties and VAT in order to obtain a more realistic value of the total investment cost. From the cost analysis carried out so far, it can be seen that cost values from two major players in ORC manufacturing (Turboden®, Italy and Infinity Turbine®, USA) have been used. If the ORC system under study were imported from Italy, there would not be any charges for import duty and VAT; however, if it were imported from the USA, import duty and VAT have to be charged on the imported goods. In order to achieve a good economic analysis, the author has decided to portray the worst case scenario, in which the proposed plant is imported from the USA. Information obtained through conversation with a manufacturer shows that an ORC system with similar capacity weighs about 10,000 kg, while communication with shipping agents shows that the cost of shipping together with the export clearance charges would amount to about US$2,200.00 (£1417.89). This would give an overall cost for the imported ORC system of US$440,098.58 (£283,641.78). From the European Commission Taxation and Customs Database (EC, 2011) the import duty for electrical equipments imported into the UK from the USA is 2.7%. Also, the VAT on the imported plant would be 20%.

All the various costs which make up the total capital investment are shown in Table 7–1.

Estimation of the Operating and Maintenance Costs

Apart from the money spent in executing the proposed model, its operation would also require some maintenance and this would incur some extra cost. From experience developed over the years from real-life operations of ORC plants (Obernberger *et al*., 2002), ORC systems have been known to operate stably and require little maintenance. As a normal practice regarding maintenance, periodic weekly checks by the operator, as well as a routine one-to-two-day inspection once a year, are recommended by the manufacturer (Stowa, 2006). The maintenance cost of an ORC system, based on information obtained from the reference real-life ORC project mentioned above, is 2% of the total capital investment, while the administration and insurance cost amounts to about 0.7% of the total investment costs (Obernberger *et al*., 2002). The personnel cost, which includes the consulting cost is given as US$60/hr (£38.67/hr) (obtained from price list in Appendix A) for 400 hours per year. These various costs are tabulated in Table 7–2.

7.3.5.2 Revenue Estimation

The revenue accrual from the operation of the proposed plant includes that from the sale of the electricity produced and from carbon savings. The prices of electricity and carbon both vary over time. Hence, this profitability analysis will be carried out based on the sensitivity analysis for different prices of carbon and electricity.

The price of carbon for the different scenarios presented in previous chapter varied between €5 to €60/tCO_2 (£4.30 to £51.63), while the electricity price varied between £0.01 to £0.35/kWh.

7.3.6 Profitability Assessment: Sensitivity Analysis

The profitability assessment in this project is carried out using the economics tool in the IPSEpro simulation software. In order to carry out the analysis, all the cost factors estimated above were inputted into the software as appropriate. The profitability assessment was then carried out for different scenarios based on revenue considerations, in order to ascertain the economic viability of the proposed model.

Table 7–1: Estimation of the Total Investment Cost of the Proposed 199.40 kW Dual Heat Source ORC System

Cost Description	% of Purchased Equipment Cost	Amount USD	Amount GBP
Direct Costs			
Purchased Equipment Cost (ORC Module)	100.00	437,898.58	282,223.89
Cost of Charged R245fa Working Fluid	0.57	2,485.00	1,601.57
Cost of Installation, Material and Labour	4.78	20,928.98	13,488.64
Cost of Control System	2.65	11,591.43	7,470.63
Cost of Grid Connection	39.60	173,571.98	111,866.45
Indirect Costs			
Cost of Engineering	15.00	65,684.79	42,333.58
Cost of Freight	0.50	2,200.00	1,417.89
Import Duty & Tax	2.70	11,823.26	7,620.04
VAT = 20% (of ORC module + Import Duty & Tax)		89,944.37	57,968.79
Total Capital Investment		**816,128.39**	**525,990.59**

Exchange Rate GBP1 = USD1.5516= EUR1.1621[1]

[1] HMRC/UK Customs Monthly Exchange Rate (December 2011)

Table 7–2: Estimation of the Operating Cost of the Proposed Model

Cost Description	Amount	
	USD	**GBP**
Operating Costs		
Personnel Costs ($60/hr for 400 hrs per year)	24,000.00	15,467.90
Maintenance Costs (2% of Total Capital Investment)	16,322.56	10,519.81
Other Costs		
Administration and Insurance Costs (0.7% of Total Capital Investment)	5,712.90	3,681.93
Total	**46,035.46**	**29,669.64**

Exchange Rate GBP1 = USD1.5516 = EUR1.1621[2]

As pointed out earlier, the main sources of revenue in this project include revenue from the sale of generated electricity and revenue from carbon emission reduction. This profitability analysis is carried out in two main scenarios, namely:

- **Scenario 1**: Based only on revenue accrual from the sale of generated electricity;
- **Scenario 2**: Based on revenue accrual from the sale of generated electricity as well as from the reduction of carbon emissions.

[2] HMRC/UK Customs Monthly Exchange Rate (December 2011)

Scenario 1

The proposed plant generates about 199.4 kW of electricity from the waste heat and is assumed to operate for about 7000 h/year with a service life of 30 years. Based on the above conditions, the profitability analysis based on revenue accrual from the sale of electricity alone is calculated in IPSEpro economics software as follows:

Case 1

- **Profitability assessment based on revenue from the sale of generated electricity for a service life of 30 years at 0% discount rate on financing and 0% inflation rate.**

  In this case, the project is assumed to be financed solely from the equity of the company with 0% interest. The inflation rate is also assumed to be 0%.

Case 2

- **Profitability assessment based on revenue generated from the sale of generated electricity for a service life of 30 years at 6% discount rate on financing and 0% inflation rate.**

  In real-life projects, companies always borrow money to finance their projects. In this case, it is assumed that the company has borrowed money from the financial institutions or from the government carbon loan scheme (if in existence) at a reduced or subsidized interest rate of 6% (HM Treasury, 2011), with the inflation rate assumed to be negligible.

Case 3

- **Profitability assessment based on revenue generated from the sale of generated electricity for a service life of 30 years at 6% discount rate on financing and 5.03% inflation rate.**

  Apart from the discount (interest) rate on borrowed capital to finance a project, another parameter which also has a significant impact on economic analysis of a real-life project is the inflation rate. Inflation affects the cost of commodities and would hugely affect the operating, maintenance and other running costs of the project during its service life. In this case, the impact of the inflation rate on the profitability assessment of the project is investigated. This case is a more realistic economic analysis when compared with cases 1 and 2. For the purpose of the analysis, the inflation rate is assumed to be 5.03%.

The profitability assessment indexes for the three different cases explained above are shown in Figures 7–2, 7–3 and 7–4 below. The PBP analysis shows that for the three cases, when the electricity price is at about £0.05/kWh, the payback is greater than 12 years, which is not acceptable in any project. Thus, if the revenue is based only on the sale of generated electricity, this project will not be economically viable if the price of electricity falls to £0.05/kWh or below.

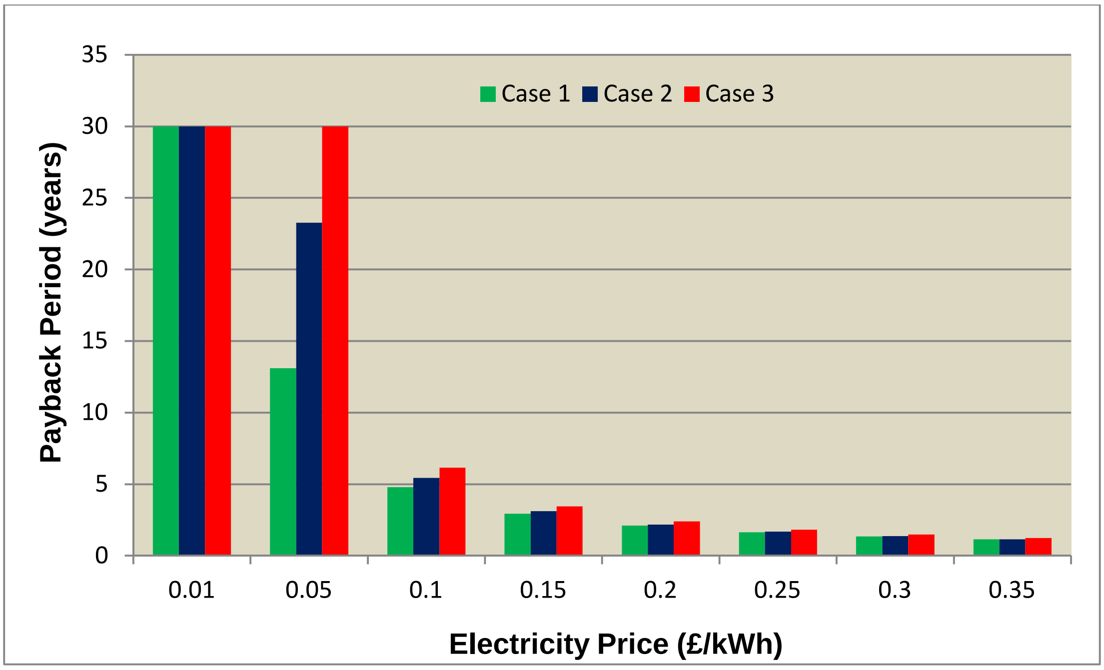


Figure 7–2: Comparison of the PBP for the Different Cases

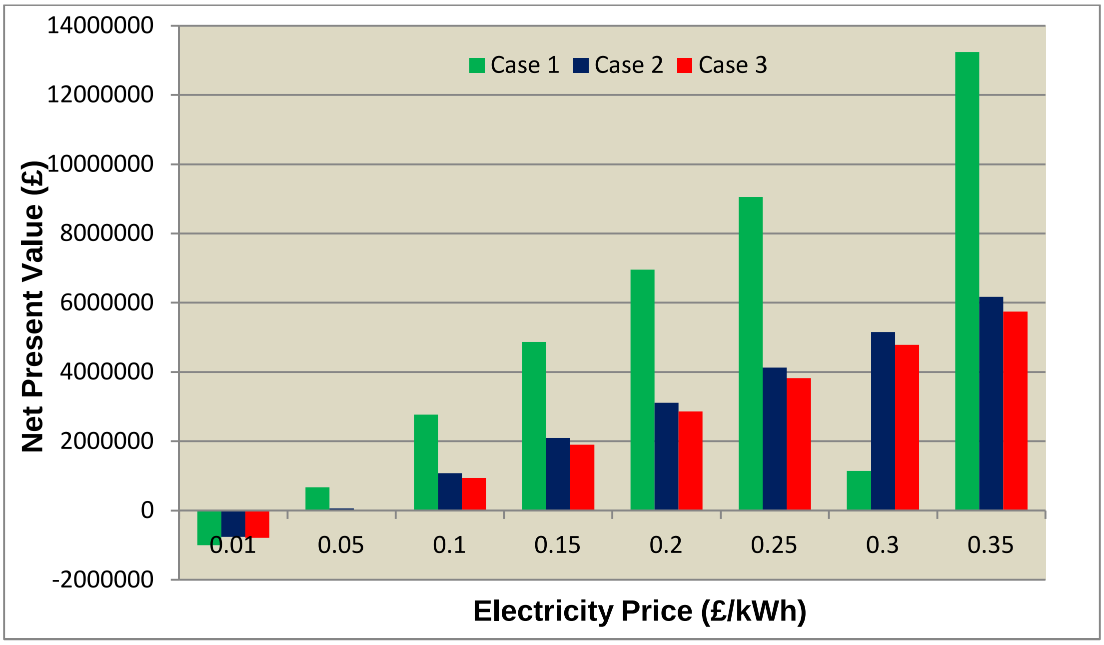


Figure 7–3: Comparison of the NPV for the Different Cases

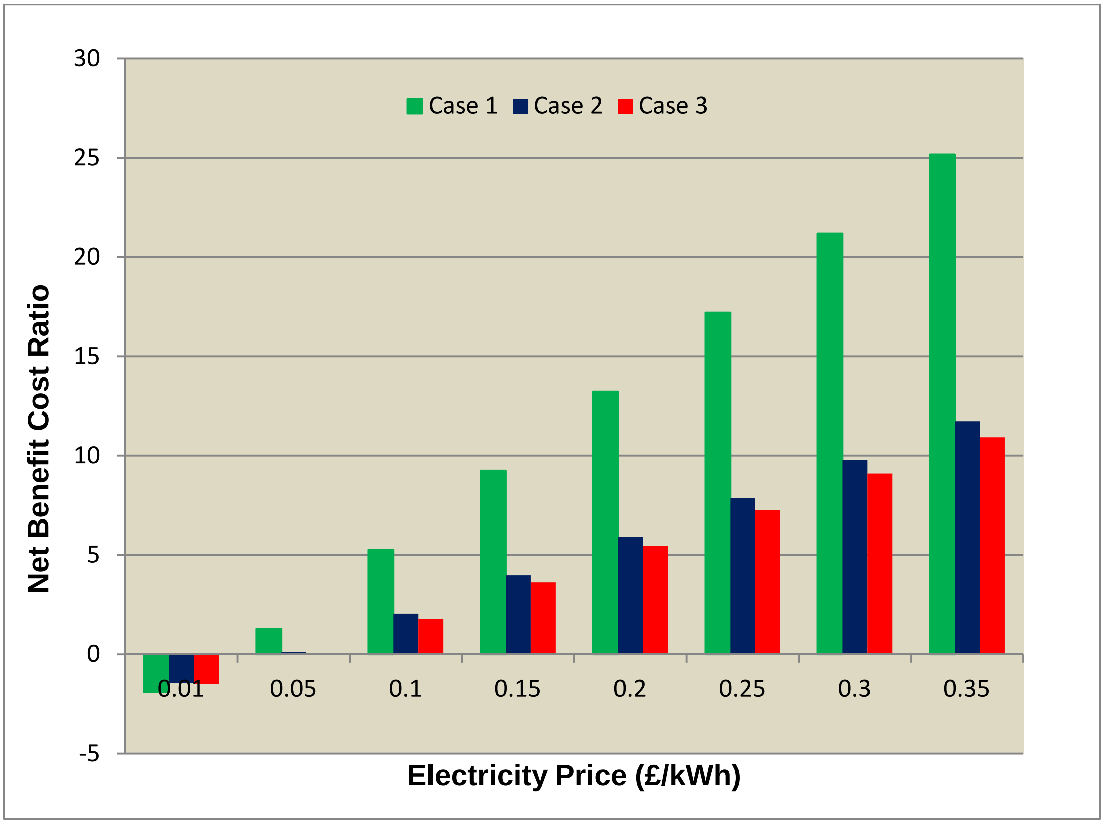


Figure 7–4: Comparison of the Net Cost Benefit Ratio for the Different Cases

Similarly, the NPV and the NBCR analysis also shows that the project is not economically viable if the price of electricity is lower than or equal to £0.05/kWh, as depicted in Figures 8-2 and 8-3 respectively, which show the values of NPV and NBCR lower than zero for the more realistic case (case 3).

At an electricity price within the range of £0.10 to £0.15/kWh (which is within the range of the current price of electricity in the UK), the PBP ranges from 6.16 to 3.44 years respectively for the more realistic case (case 3). This is outside the range acceptable to most companies, which are mainly interested in projects that will give them a payback period of three years at most. In order to achieve a payback period of ≤3 years, the electricity has to be sold for at least £0.20/kWh, which would give a PBP of 2.389 years with a net present value of £2,866,273.96.

However, since this project qualifies for carbon credit, the incorporation of revenue from the carbon credit in the economic analysis will likely make the project to still be economically viable at electricity prices even lower than £0.20/kWh.

The incorporation of the revenue from carbon reduction will be investigated in the next section.

Scenario 2

As a result of the Clean Development Mechanism (CDM) and Joint Implementation (JI) mechanism initiative by the United Nations, companies can earn carbon credit by investing in low-carbon technology. In the previous chapter, it was established that the proposed project would help in carbon offsetting in the crisp manufacturing plant under study. This would earn some carbon credit for the company and thus add to the revenue of the company. In this scenario, the additional benefit from earning carbon credit is investigated, by varying the price of carbon from £4.30 to £51.63 per metric tonne of carbon offset by the project. The analysis is carried out only for the realistic case.

Case 1

- **Profitability assessment based on revenue generated from the sale of generated electricity for a service life of 30 years at 6% discount rate on financing, 5.03% inflation rate and revenue from carbon credit.**

In this case, the profitability assessment is carried out by considering the revenue accrual from both the sale of the electricity generated and from carbon emission reduction. The electricity generated was assumed to be used to displace that which would have been produced from a coal-fired power plant. Thus, this project would create a carbon emission reduction of about 1395.8 tCO_2 per year. The graphs for the payback period and the net present value of the project are plotted and compared for different carbon prices as shown below.

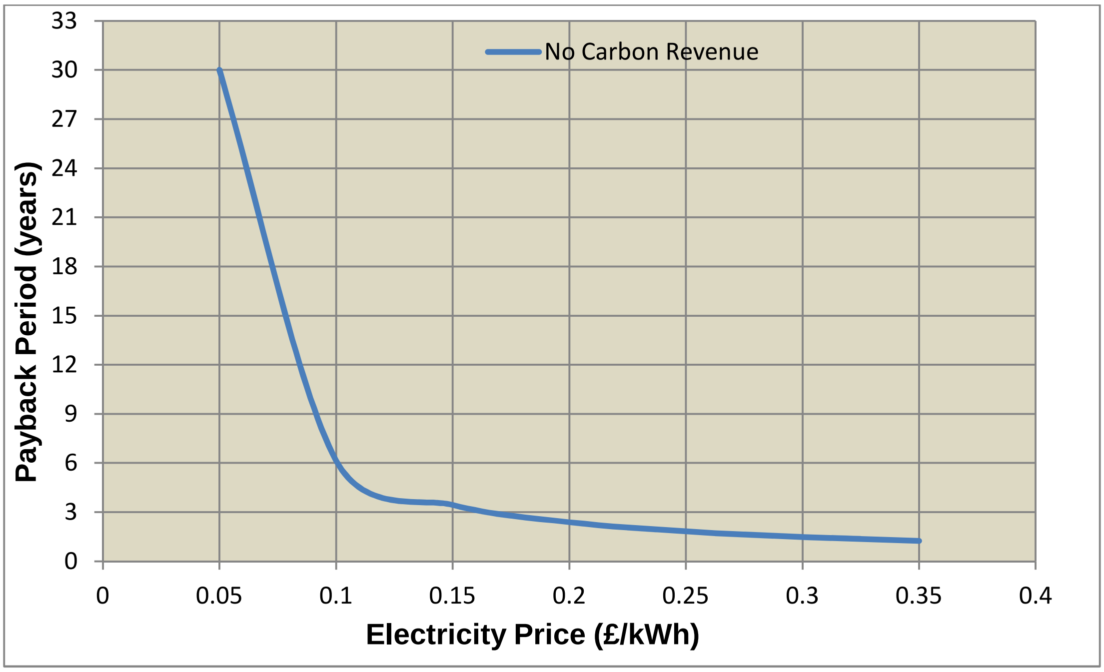


Figure 7–5: Payback Period for Scenario 2 Case 1 with No Revenue from Carbon Reduction

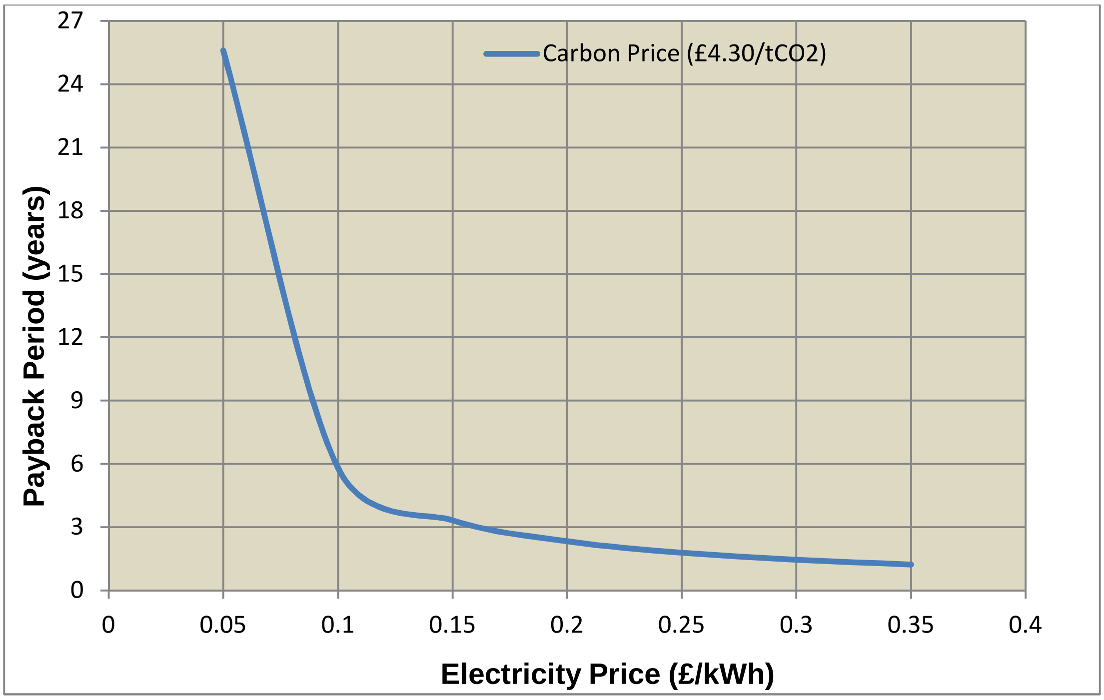


Figure 7–6: Payback Period for Scenario 2 Case 1 with Carbon Price of £4.30/tCO_2

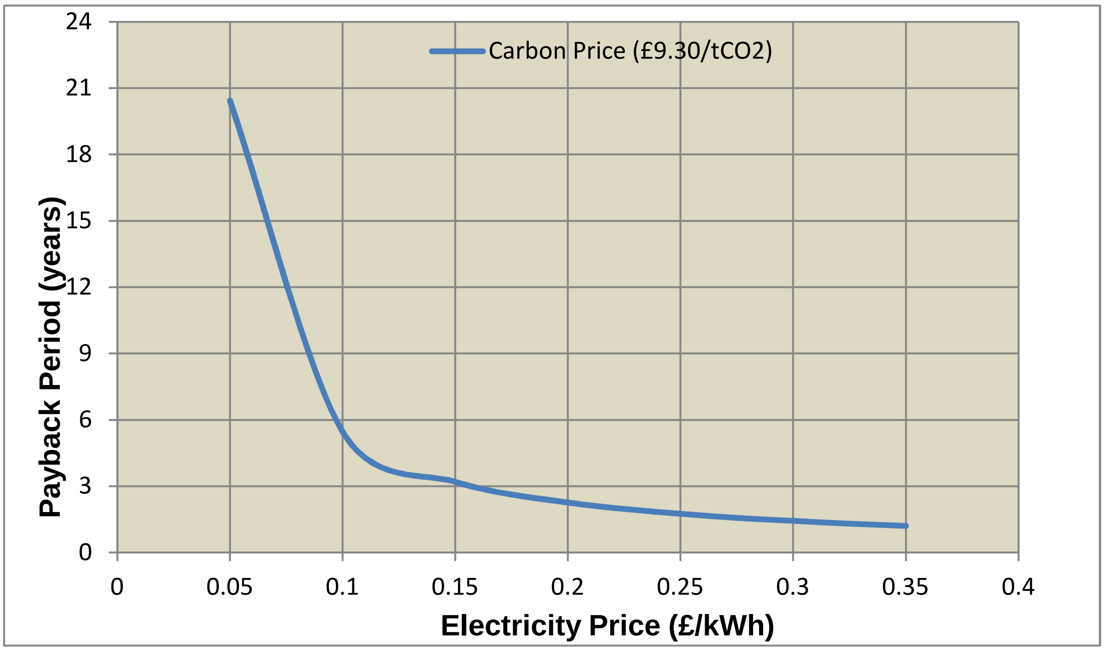


Figure 7–7: Payback Period for Scenario 2 Case 1 with Carbon Price of £9.30/tCO_2

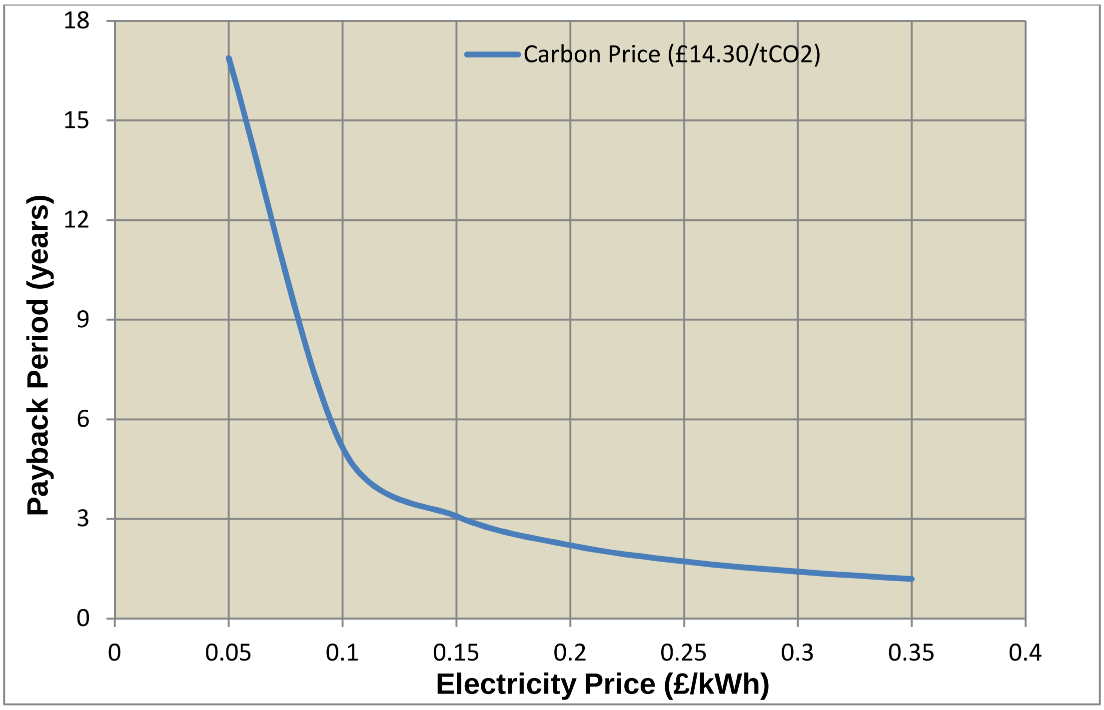


Figure 7–8: Payback Period for Scenario 2 Case 1 with Carbon Price of £14.30/tCO_2

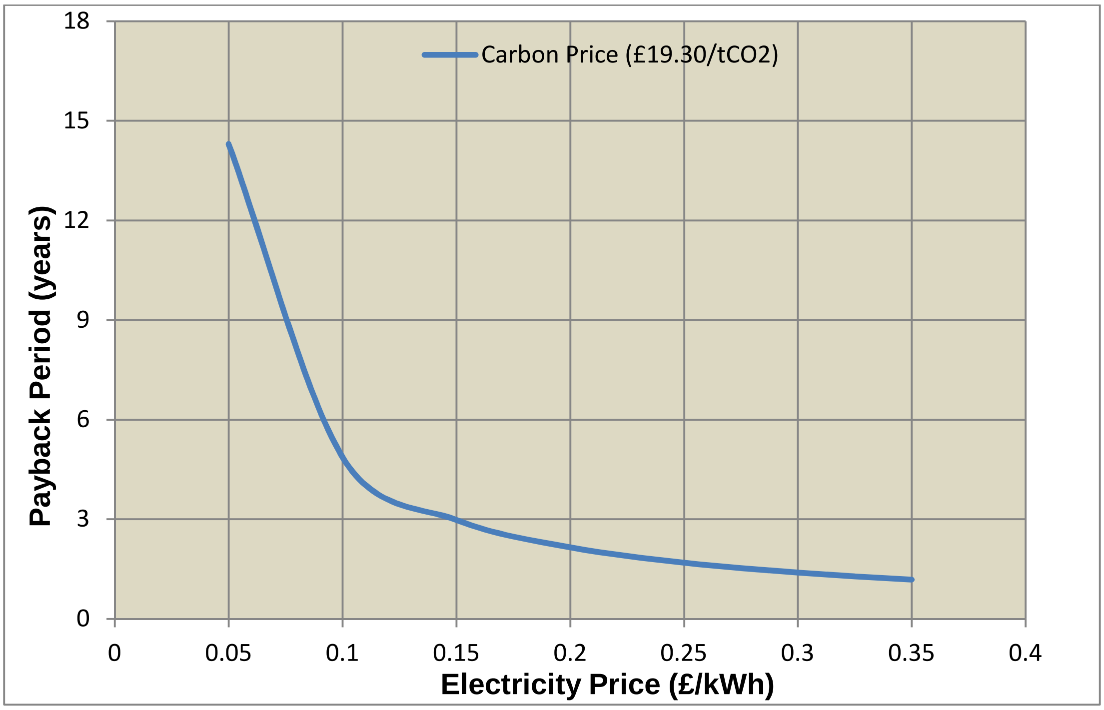


Figure 7–9: Payback Period for Scenario 2 Case 1 with Carbon Price of £19.30/tCO_2

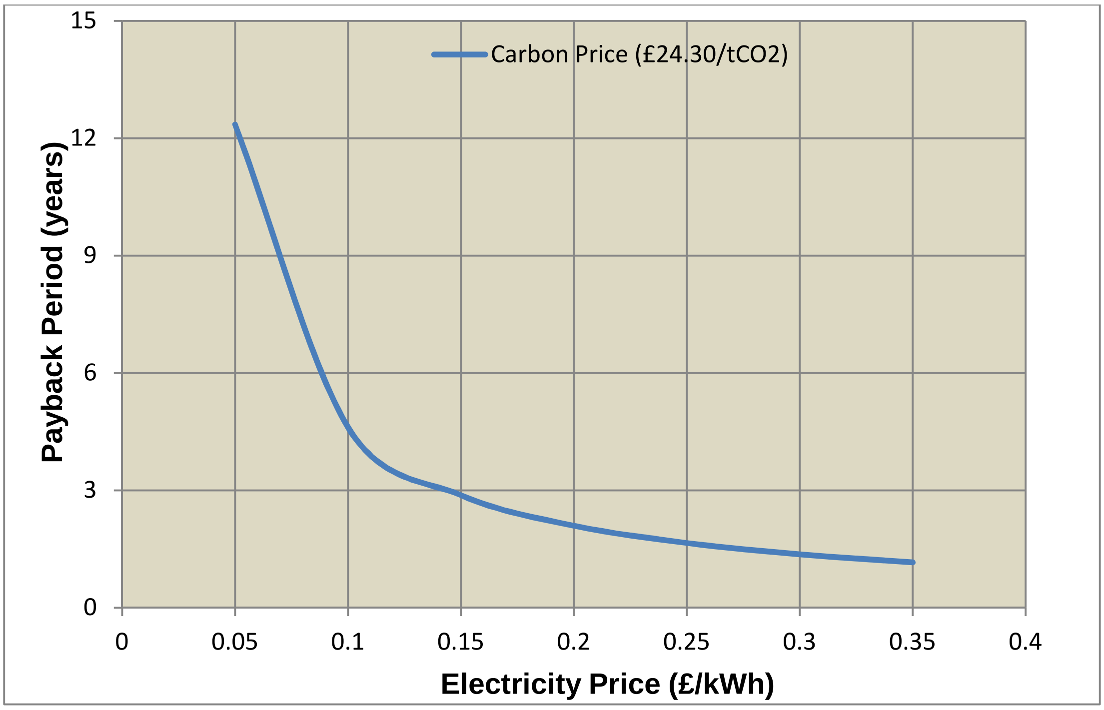


Figure 7–10: Payback Period for Scenario 2 Case 1 with Carbon Price of £24.30/tCO_2

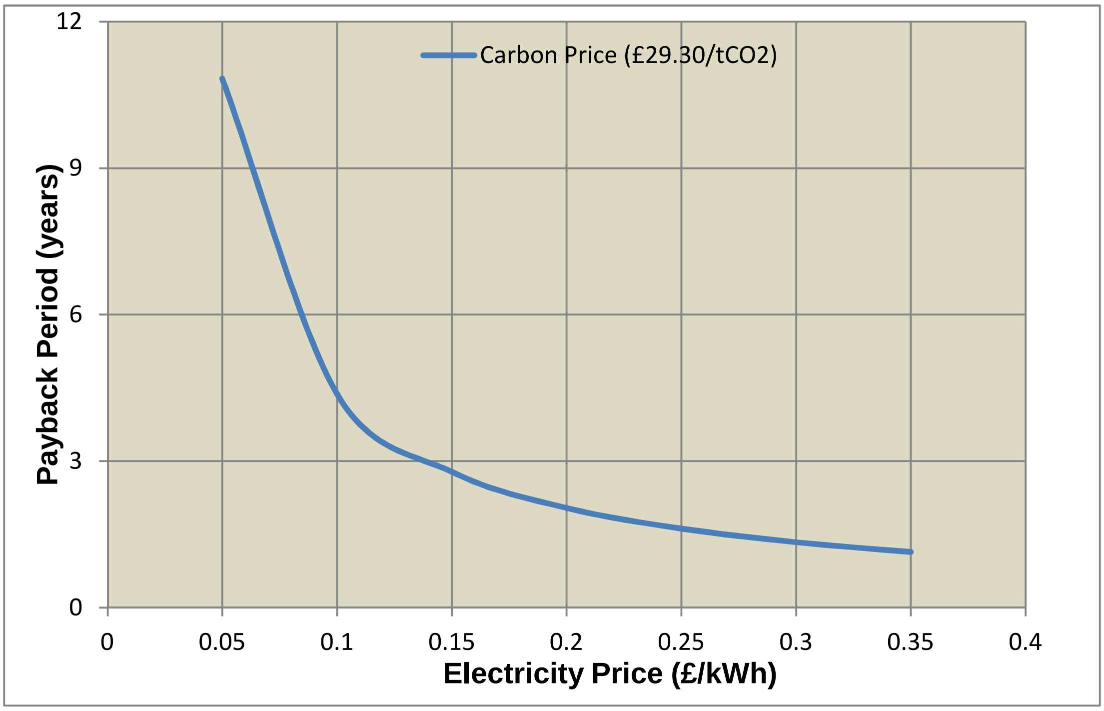


Figure 7–11: Payback Period for Scenario 2 Case 1 with Carbon Price of £29.30/tCO_2

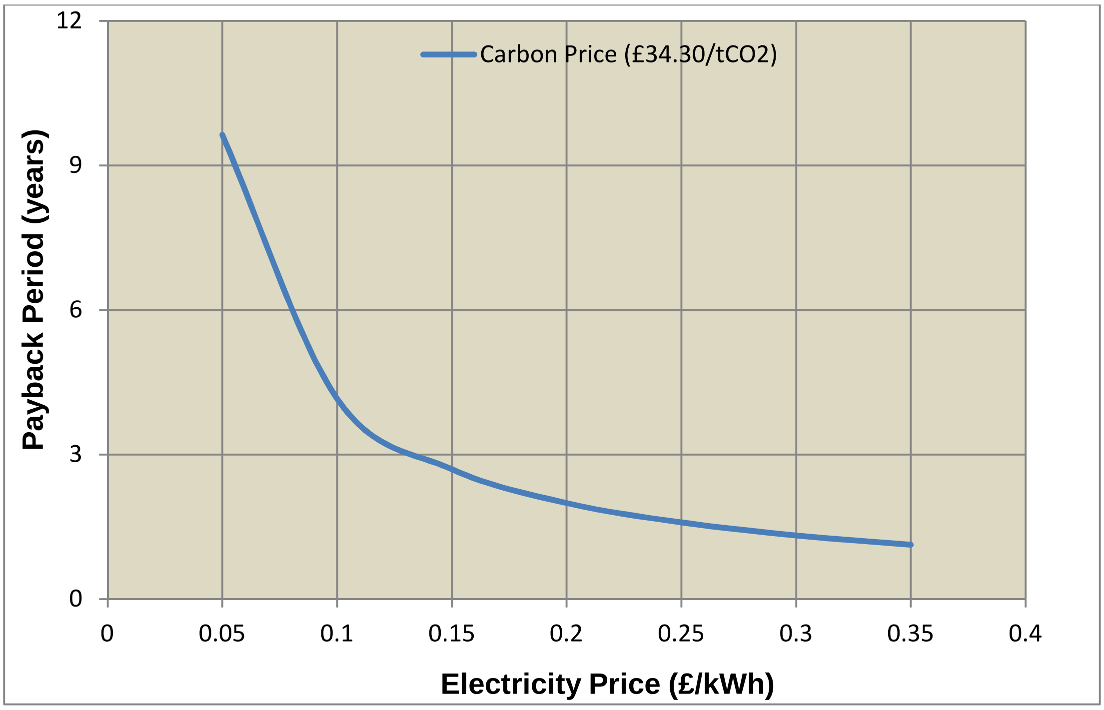


Figure 7–12: Payback Period for Scenario 2 Case 1 with Carbon Price of £34.30/tCO_2

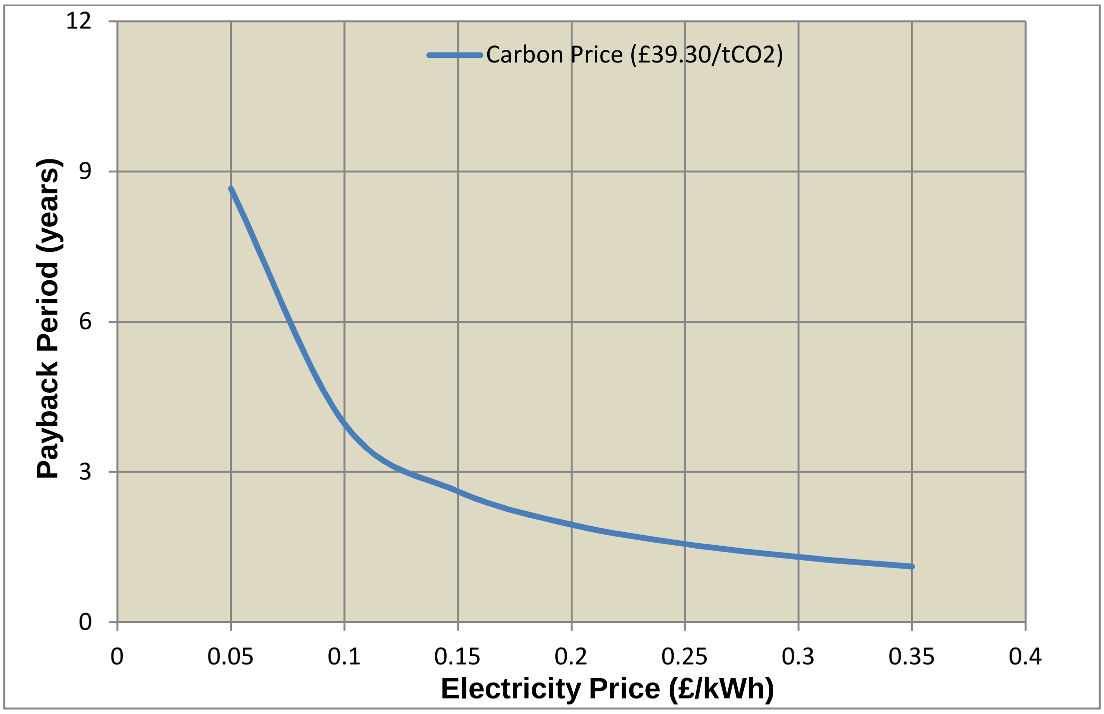


Figure 7–13: Payback Period for Scenario 2 Case 1 with Carbon Price of £39.30/tCO_2

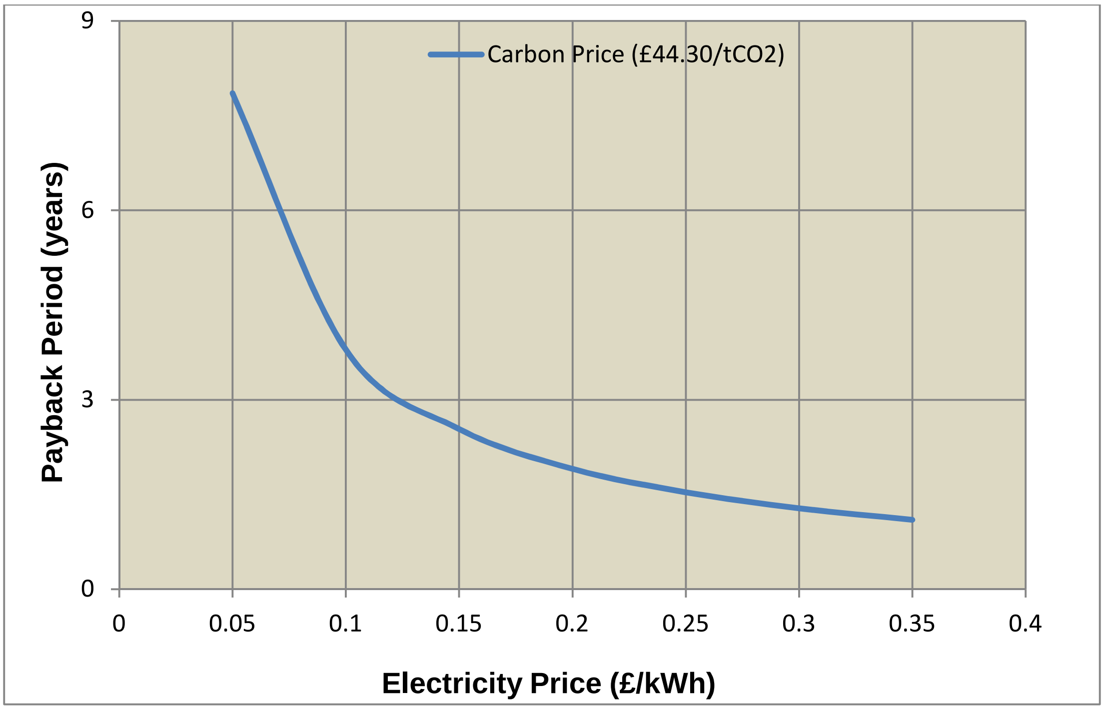


Figure 7–14: Payback Period for Scenario 2 Case 1 with Carbon Price of £44.30/tCO_2

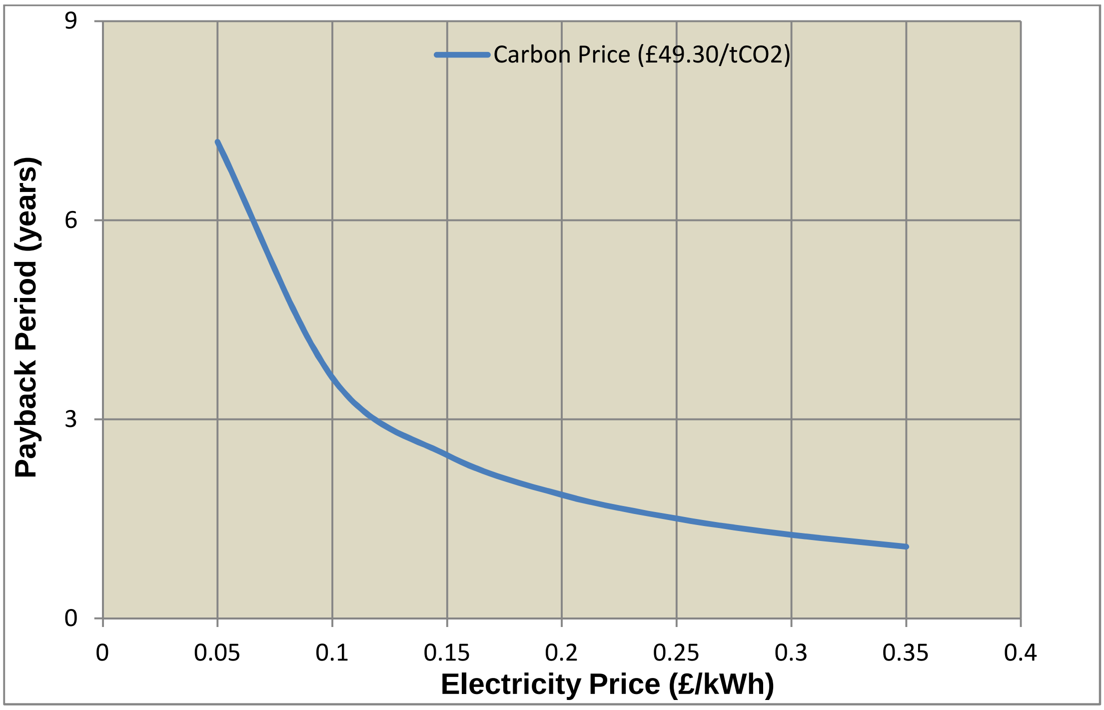


Figure 7–15: Payback Period for Scenario 2 Case 1 with Carbon Price of £49.30/tCO_2

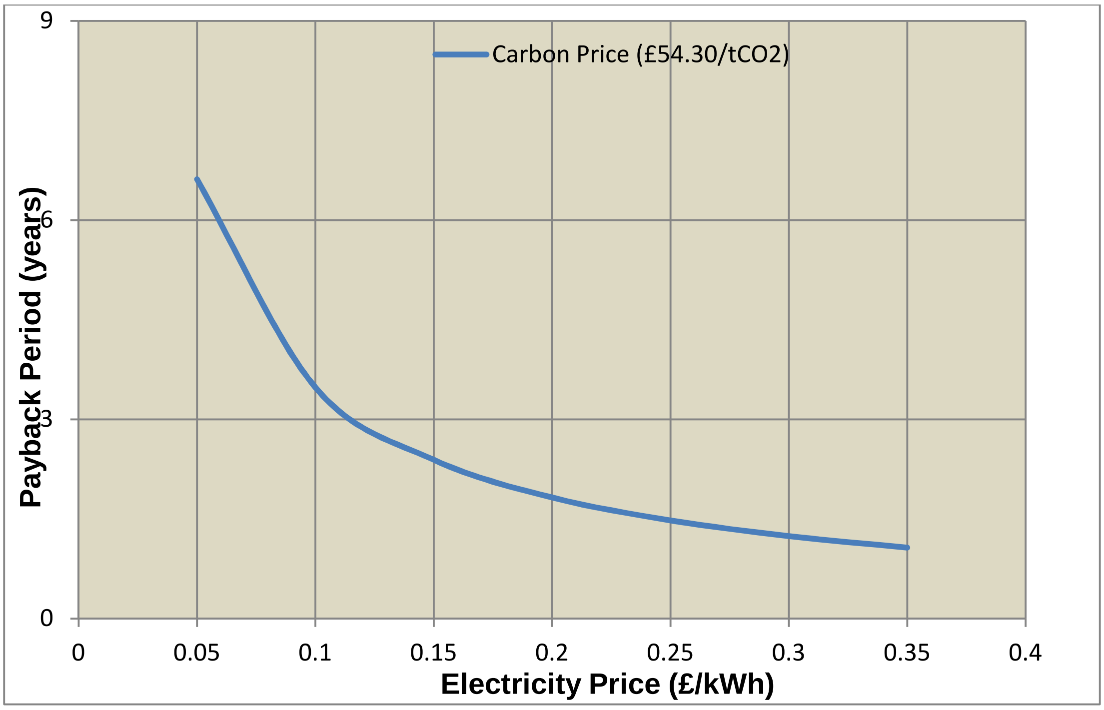


Figure 7–16: Payback Period for Scenario 2 Case 1 with Carbon Price of £54.30/tCO_2

From the graphs presented above, it can be observed that the price of carbon has a significant impact on the economic viability of the proposed project. Hence, since this project is a proposition for implementation in an existing crisp manufacturing plant, it will be of utmost importance to base the economic analysis on the most common profitability index usually used by industry.

From the industrial point of view, the most significant economic index used is the payback period of the project, and it is generally assumed that a project should have at most such a period of three years for it to be generally acceptable. Hence, if the proposed project is developed using borrowed capital at a subsidized interest rate of 6% and considering an inflation rate of 5.03% the project will be able to achieve a payback period of three years if an electricity price of at least £0.17/kWh (for the worst case scenario, i.e. if the carbon saving is not considered) is maintained. This price is higher than the average electricity price in the UK, which currently stands at about £0.16/kWh. However, with the consideration of revenue from the carbon credit, this project would be economically viable and would give a payback period of about three years with a carbon price as low as £4.30/tCO_2 and an electricity price of about 0.16/kWh. The carbon market is a volatile market. The price of carbon currently hovers in the neighbourhood of €16/tCO_2 (£13.77/tCO_2). At this price, the payback period of the proposed project would be 2.868 years if the electricity price is assumed to be the current UK average price of £0.16/kWh. Hence, based on the payback period, the project can be seen as economically viable. Another incentive that would promote investment in the project is if the government were to bring in regulations which would reduce the level of volatility in the carbon trading market. Such an incentive would provide some level of certainty for companies to embark on carbon-saving projects such as the particular one proposed in this study.

Furthermore, apart from the payback period, the project would also provide a new source of revenue to the plant, as can be seen from the net present value analysis carried out below.

The NPV analysis plot shown in Figure 7–17 shows that the economical viability of the project increases as the carbon and electricity prices increase. At the current UK average electricity price of £0.16/kWh and a carbon price of £13.77/tCO_2, the proposed project would provide an NPV of about £2,260,593.99. Furthermore, the NBCR shown in Figure 7–18 also shows that the proposed project is economically viable. At the average electricity price in the UK and a carbon price of about £13.77/tCO_2, the NBCR of the project is 4.297, which is an acceptable value.

7.4 Concluding Remarks: Economic Benefits

From the economic analysis carried out in this project, it has been established that the proposed project is economically viable when the borrowed capital is subsidized at an interest rate of about 6%, the electricity price is at the current UK average of about £0.16/kWh and the carbon price is about £13.77/tCO_2. At lower electricity and carbon prices, the PBP increases while the NPV and the NBCR of the project reduce. Comparison of the different cases in scenario 1 shows that the interest rate and inflation rate plays a significant role in the economic viability of the system. The economic viability of the project increases as the interest rate and inflation rate decreases and vice versa.

Furthermore, it is noteworthy to remember that the establishment of the Total Capital Investment in this project has been based on estimated data obtained from similar projects, and has been carried out as detailed as possible using the industry recognized cost estimation technique such as the Chemical Engineering Plant Cost Index. However, there may be differences in the estimated cost and the real cost although the author believes that the difference would be minimal, but if any significant change in the actual cost of the individual items that made up the total capital investment were to occur, there might be a big variation in the actual profitability index. As the plant has been assumed to operate for about 30 years, there may be significant variations in the price of electricity and carbon during the operation period.

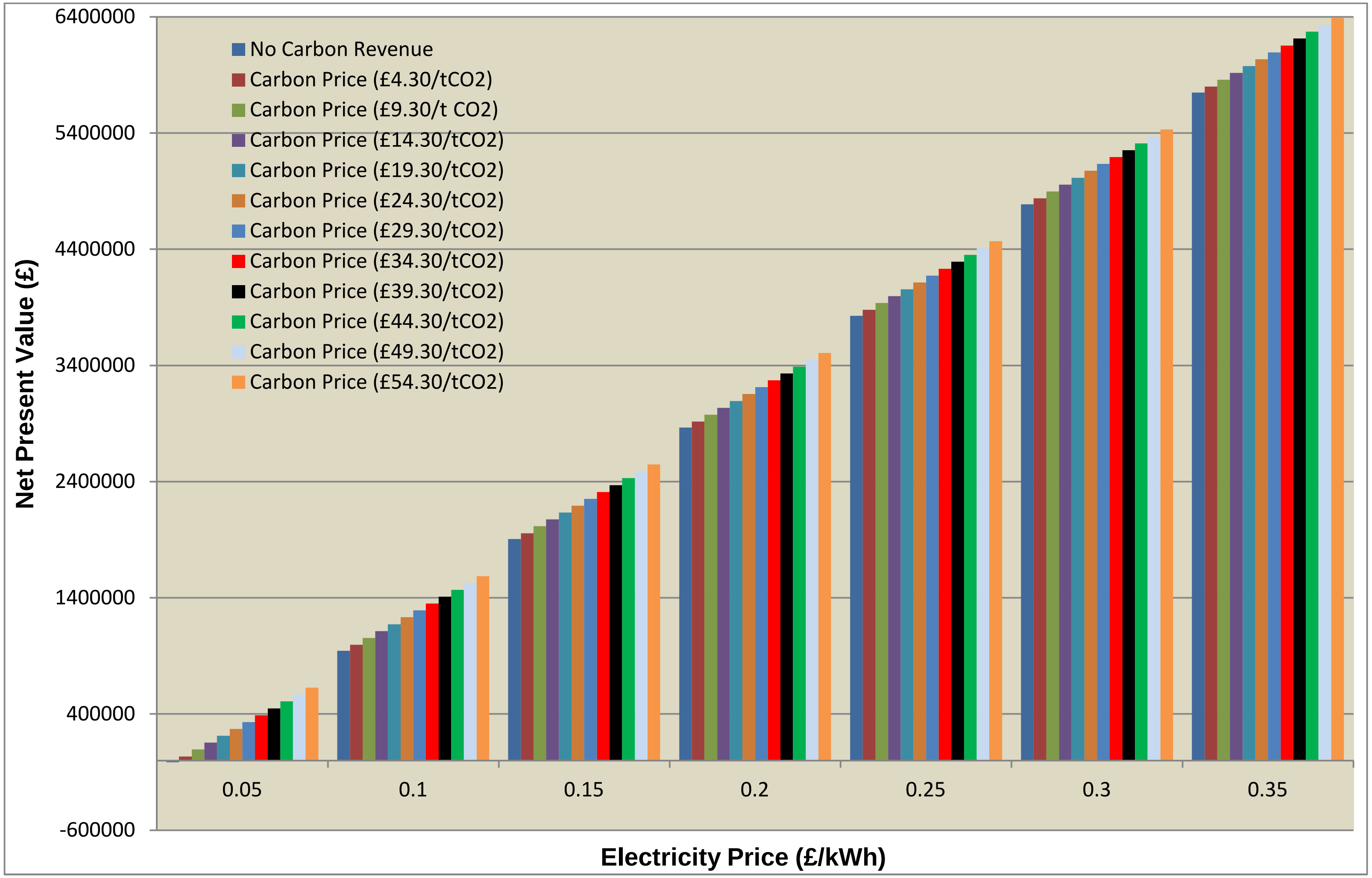


Figure 7–17: Net Present Value at Different Carbon and Electricity Prices

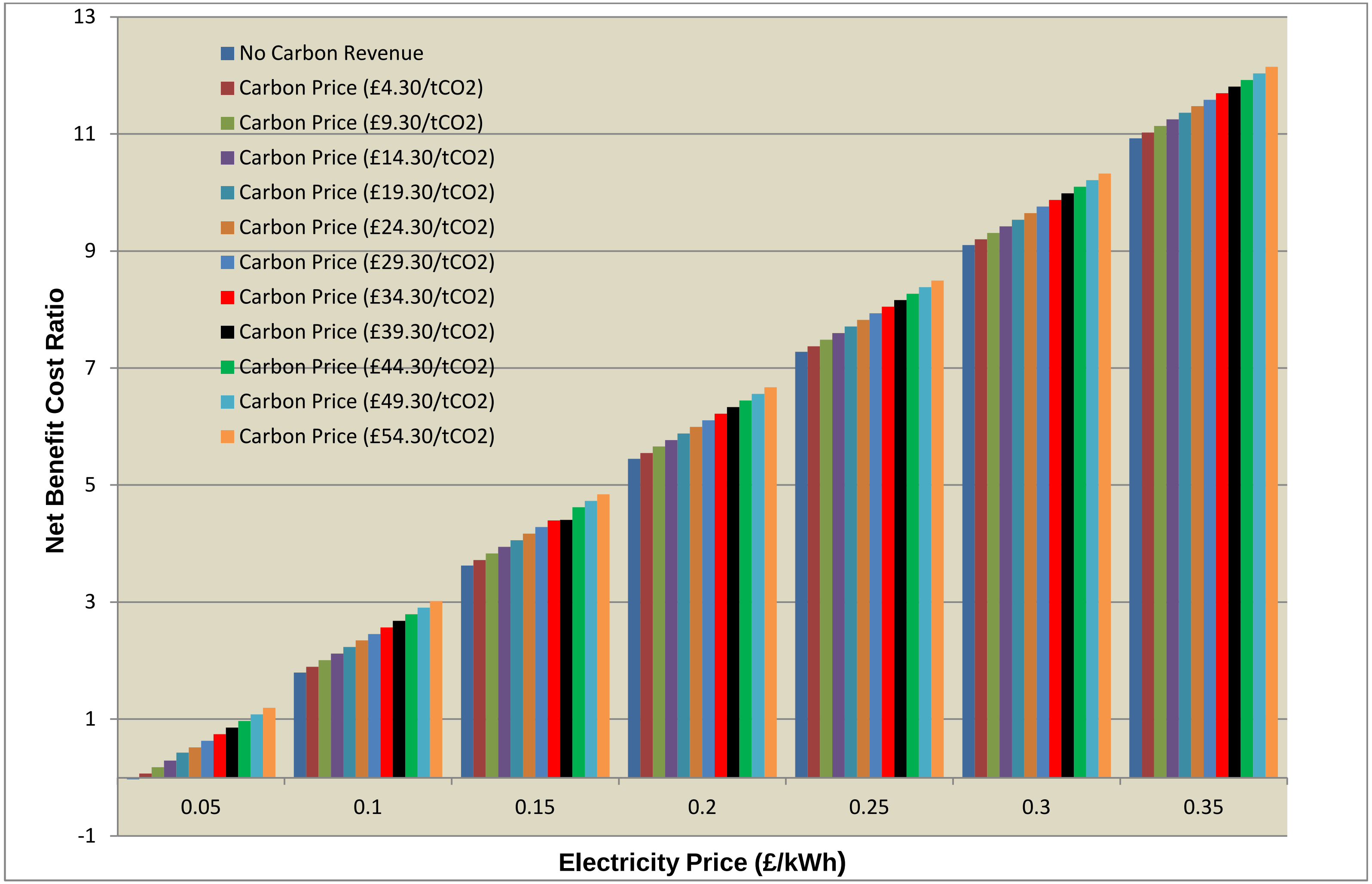


Figure 7–18: Net Benefit Cost Ratio at Different Carbon and Electricity Prices

From trends so far, the price of energy will increase with time and hence this would bring in more revenue, thus improving the economic viability of the proposed project.

Chapter Eight

This is the concluding chapter of this thesis; it highlights all the major findings of this project and provides recommendations for future work in this field.

8 Conclusions and Recommendations

8.1 Conclusions

In this work, the use of low-grade waste heat energy from a crisp manufacturing plant (which is currently being emitted into the environment, thus causing energy wastage and heat pollution) to generate electricity using the dual heat source ORC system proposed in this work has been established to be an economically viable and environmentally friendly project.

Although the use of exhaust waste heat to generate power using the Organic Rankine Cycle has been in existence for a long time, its application to the crisp manufacturing industry, especially in the UK, has never previously occurred. Therefore, this project, through its theoretical study, has been able to establish that this same technology can be adequately transferred to the food processing industries, as well as to other processes where waste heat is in existence.

The proposed project has been established not only to offset the carbon footprint of the crisp manufacturing process but also to improve resource savings, as well as reducing the over-dependence on fossil fuels to generate power, which will lead to a reduction in CO_2 emissions and ultimately contribute to the reduction of global warming.

The modelling and simulation carried out in this work using the IPSEpro Process Simulation tool has also established that the waste heat emitted from the fryer section and exhaust stack can be used to drive a dual heat source ORC system to generate about 199.4 kW_e (net) which is about 92% of the electricity needed in the crisp manufacturing process.

Furthermore, from the ‘gate to grave’ life cycle assessment carried out in this work, it was established that the use of the waste heat to generate power using the dual heat source ORC system proposed in this work will also reduce the quantity of carbon emission by 1395.8 tonnes/year when the generated electricity is used to displace electricity which would have been generated from a conventional coal-fired power plant.

The economic analysis shows that at an average electricity and carbon price of £0.16/kWh and £13.77/tCO_2 the proposed project will give an NPV of about £2,260,593.99 with a positive NBCR and payback period of 2.868 years making it an economically viable venture.

8.2 Recommendations for Future Research

One major challenge encountered during the course of this study was to obtain reliable cost and performance data from major ORC manufacturers, especially for the specialized model (dual heat source ORC model) proposed in this work. This constraint can be overcome by developing an experimental rig (which is outside the scope of the objective of this project) in order to verify some of the data obtained from the published literature which are used as the basis of this modelling and simulation work.

Therefore, the author suggests that further research should be carried out to develop an experimental dual heat source ORC system rig which could be used to validate the theoretical model already developed in this work. This will not only generate more insight into the operation of the system but will also reposition this university as a centre for the study of the waste heat to electricity process, since none is currently in existence in UK. It will also serve as a teaching facility to undergraduates and decision-makers from industry some of who currently are sceptics about the capability of converting "ordinary" heat to electricity using the ORC technology.

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Appendices

Appendix A

Screenshots showing the implementation of the developed models in IPSEpro MDK

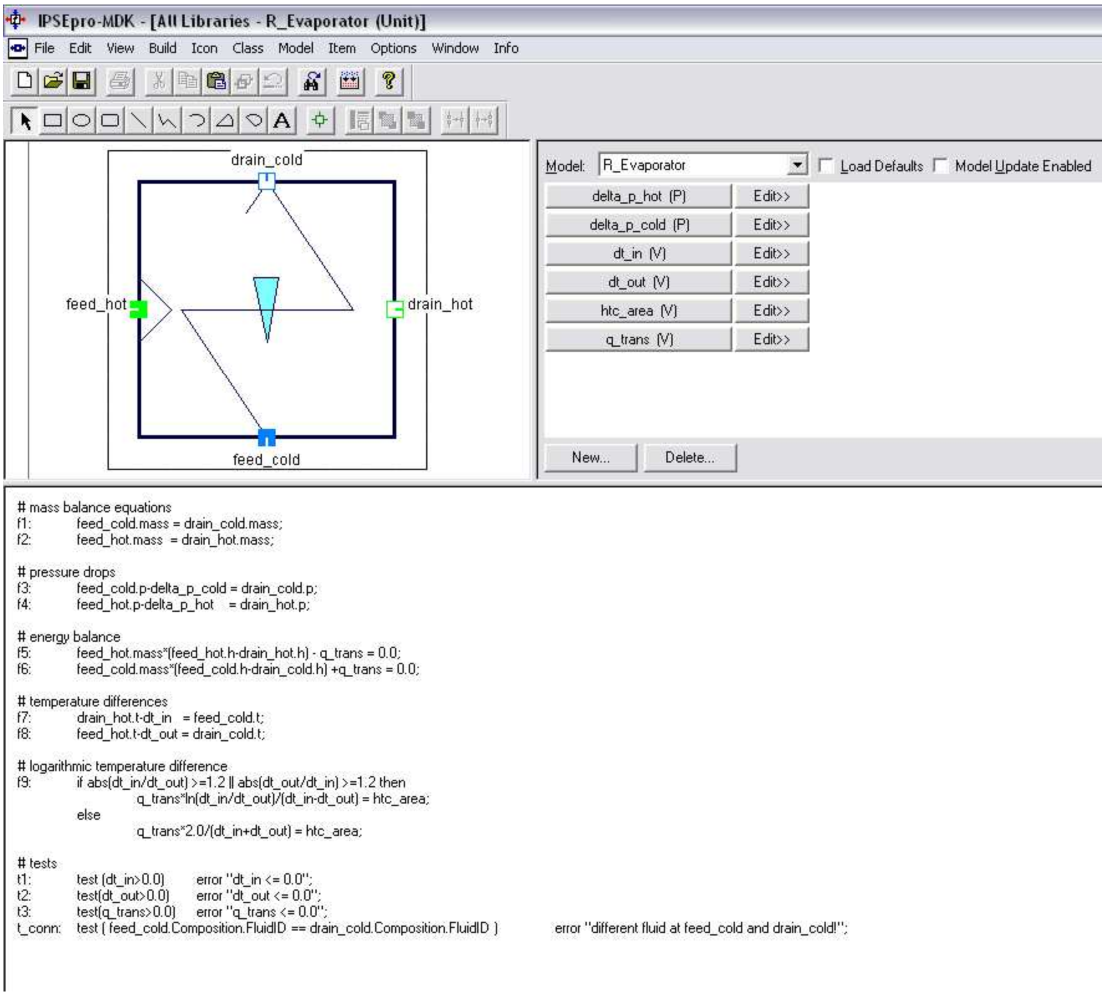

Figure A–1: Evaporator Model in IPDEpro MDK

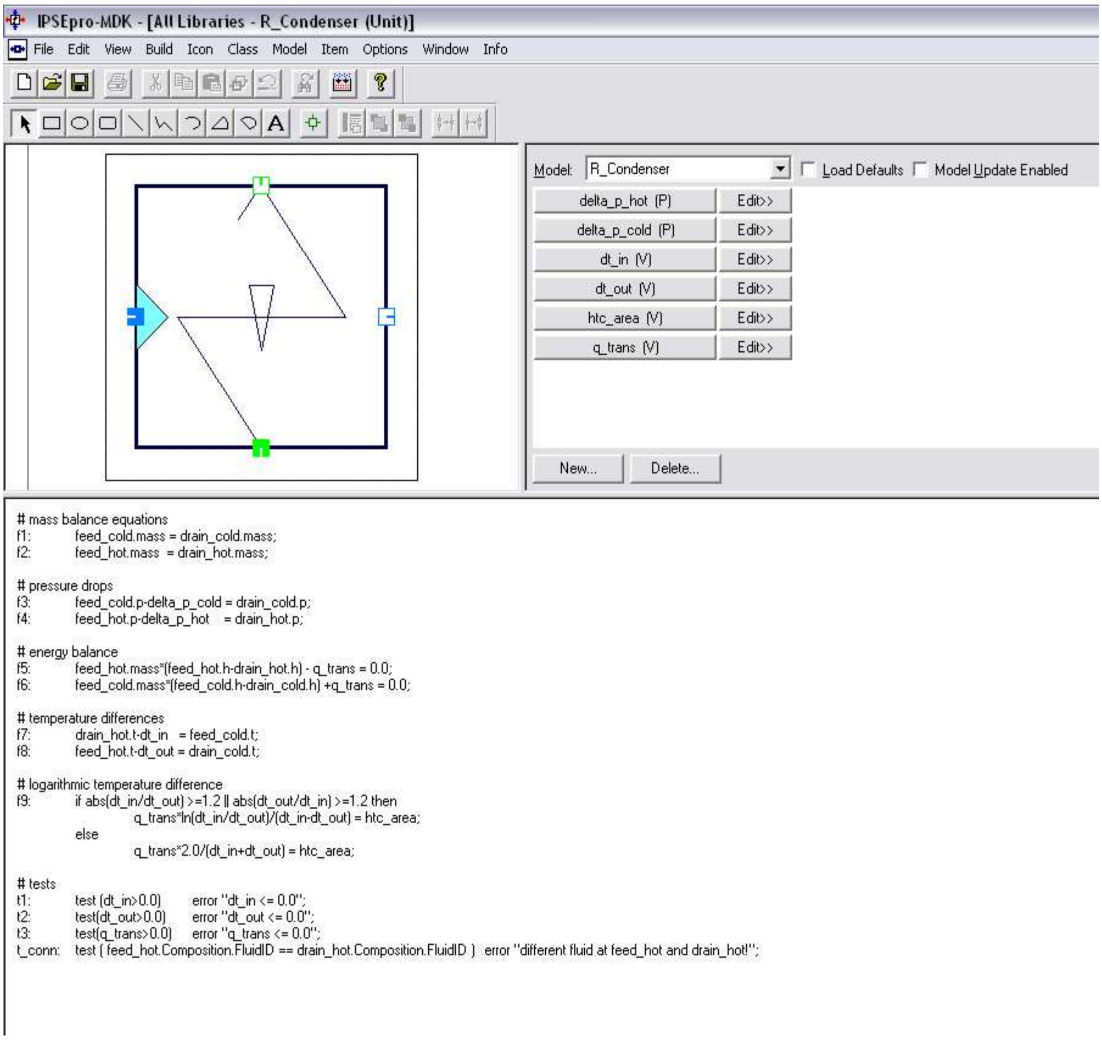

Figure A–2: Condenser Model in IPSEpro MDK

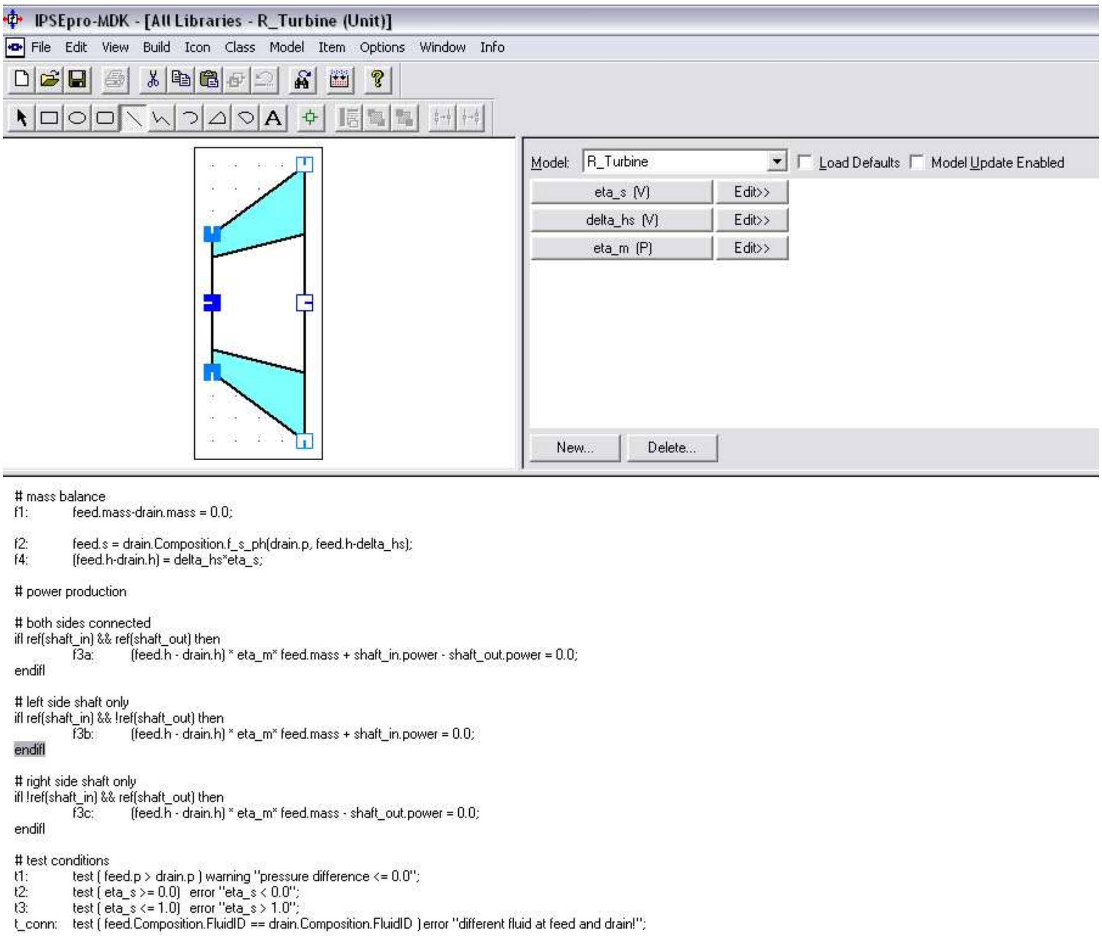

Figure A–3: Turbine Model in IPSEpro MDK

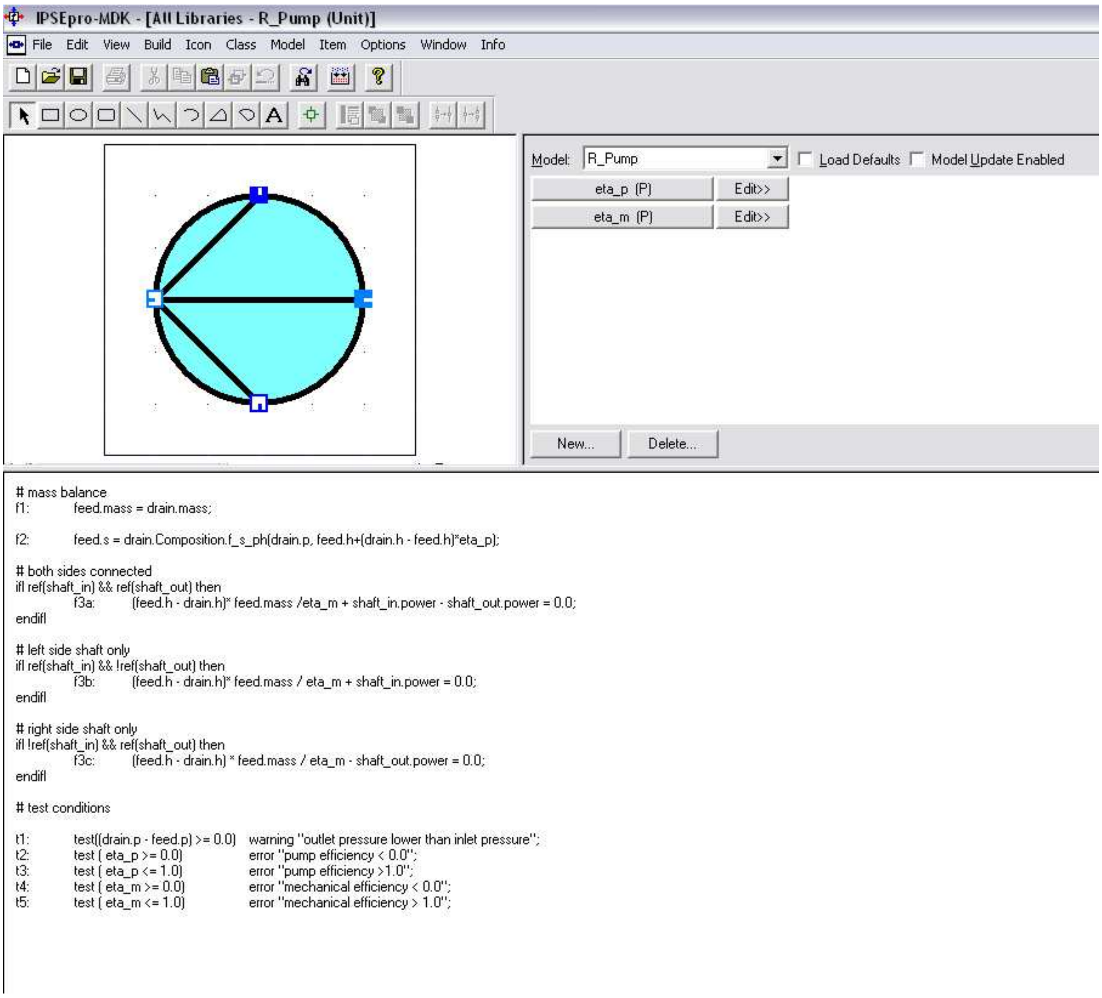

Figure A–4: Pump Model in IPSEpro MDK

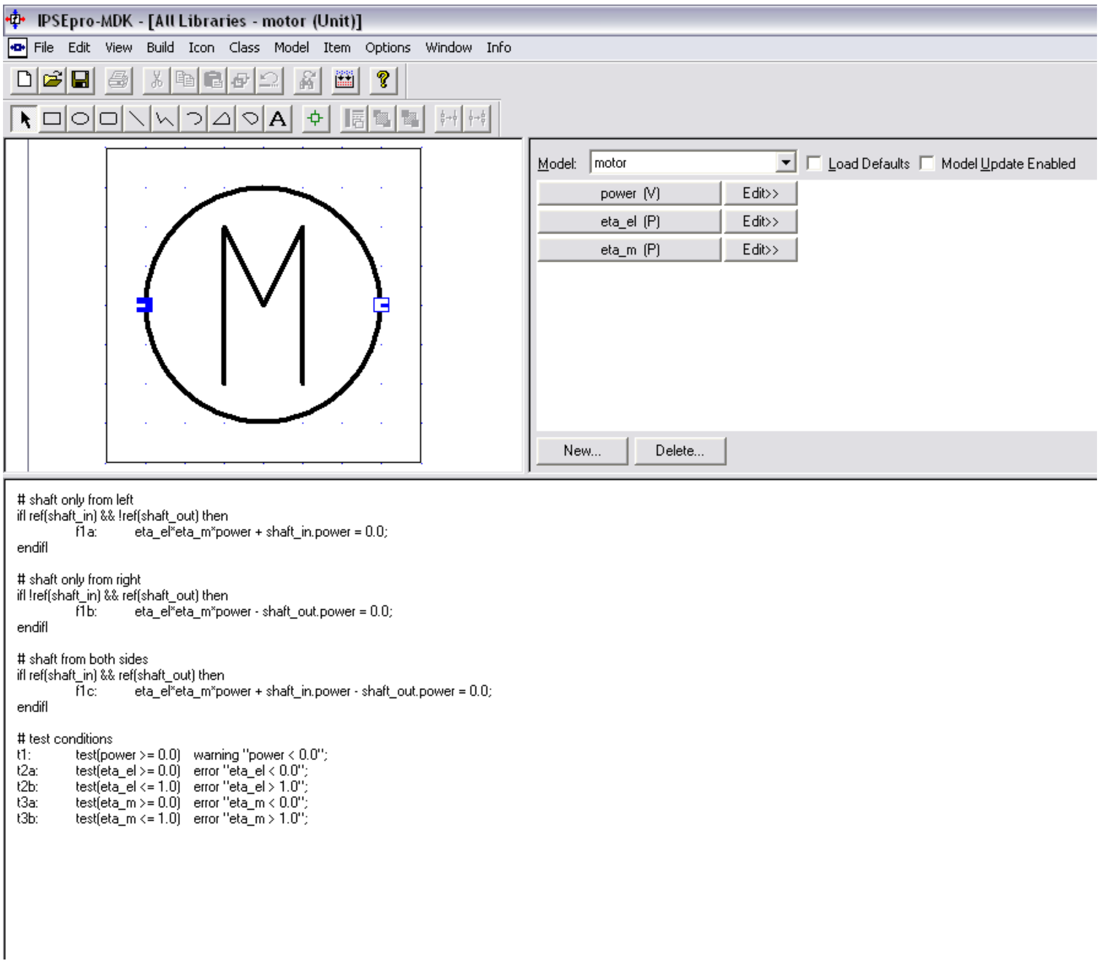

Figure A–5: Motor Model in IPSEpro MDK

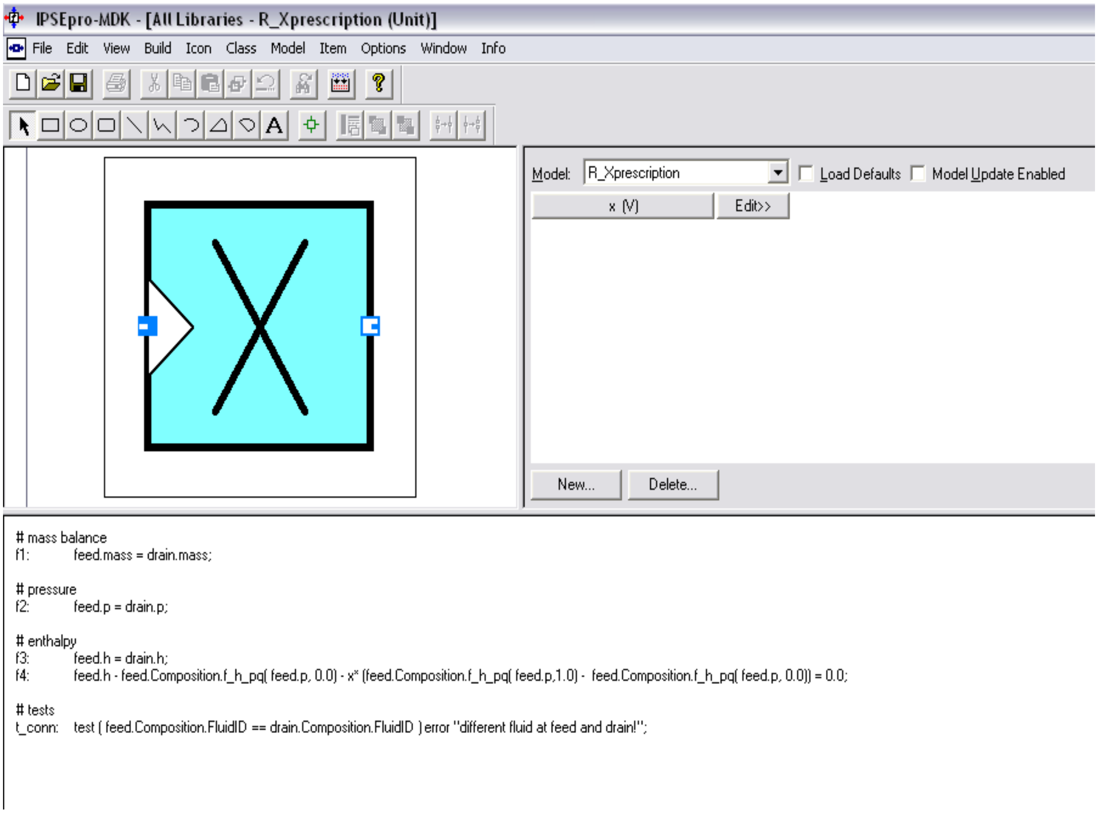

Figure A–6: Enthalpy Parameter Model in IPSEpro MDK

Appendix B

Price List from ORC Manufacturer: Infinity Turbine®

Infinity Turbine LLC · Price List Prices Valid Until December 20, 2011 20111118

All Prices U.S. Dollars unless otherwise noted. Shipping and crating is not included.

ORC Turbine-Generator Only · Working pressure up to 300 psi (20bar, 2 Mpa, 2,000 kpa) · Magnets rated to 150 C

Model	Power Output (DC)	Ports (NPT) Inlet/Outlet	Dimensions inches(metric)	Weight Lbs(kg)	Turbine Kit (no mag rotor/coil)	Turbine-Gen Assembled
ITmini	300 watts (max 1,000 watts)	3/8" In 3/4" Out	3x3x3" (76x76x76 mm)	7lb (3 kg)	$2,000	$3,000
ITxr	Up to 10 kW	1" Inlet 4" Out	10x10x10" (254x254x254 mm)	70lb (32 kg)	$10,000	$15,000
ITxr	Up to 30 kW	3 x 1" Inlet 4" Out	10x10x14" (254x254x356 mm)	110lb (50 kg)		$35,000
ITxr	Up to 60 kW	6 x 1" Inlet 2 x 4" Out	10x10x26" (254x254x660 mm)	220lb (100 kg)		$55,000

ORC Compete Systems · Uses R134a or R245fa not included · Working pressure up to 300 psi (20bar) 80-140 C Temp

Model	Power Output (turbine-generator)		Heat Exchanger	BTU input (kWt)	Flow GPM(lpm)	Price
IT01	Using ITmini	DC 300-900 watts	Flat Plate	80,000 (24 kWt)	2 gpm (8 lpm)	$20,000
IT10xr	Using ITxr	DC up to 10 kW)	Flat Plate	500,000 (147 kWt)	30 gpm (114)	$50,000
IT50xr - FP	Using ITxr60	AC Inverter 50 kW Net	Flat Plate	2.5 mmbtu (733 kWt)	140 gpm (530)	$130,000
IT50xr - ST	Using ITxr60	AC Inverter 50 kW Net	Shell / Tube	2.5 mmbtu (733 kWt)	140 gpm (530)	$130,000
IT100xa-ST	Using ITxa	AC Inverter 100 kW Net	Shell / Tube	5 mmbtu (1465 kWt)	280 gpm (1060)	$230,000

ORC Turbine Only · Licensing · Blueprints (blue-copy) · Buy these blueprints to machine your own turbine

Model	(shaft)	One-Time License Fee	Consulting (per month)	Yearly Design Update Fee	Webpage Fee (per year)
ITmini	(10mm)	30,000 Euro	5 ,000 Euro	10,000 Euro	10,000 Euro
ITxr	(17mm)	40,000 Euro	5 ,000 Euro	10,000 Euro	10,000 Euro
Note: the ITxr is rail mounted and expandable to 100 kW. Turbine-Generator uses permanent magnet alternator.					
ITxa	(60mm)	50,000 Euro	5 ,000 Euro	10,000 Euro	10,000 Euro
Note: the ITxa is the turbine only, and for large applications to direct-drive AC generator up to 400 kW.					

ORC Consulting · Plans for ORC System only (no turbine blueprints) · One Month Continuous Consulting (up to 140 hours)

Model	Plans for ORC System	Consulting	Email/Skype Support	Additional Drawing	Price
IT01-IT10	IT01/ITmini/IT10/IT10xr	One Month Continuous	Yes	$60 /hr	$5,000
IT50	IT50 ST/IT50xr/IT50	One Month Continuous	Yes	$60 /hr	$30,000

Turbine Parts · To be eligible to purchase parts, you must have purchased same model in kit or fully assembled version

Model	Front-Mod	Inlet-Mod	Stator-Mod	Shaft	O-Rings	Turbine	Mag-Rotor	Coil	Back-Mod	Bearings	Cavitation Disc
ITmini	$600	$600	--	$150	$40	$800	$150	$1,000	--	$ 80	$150
ITxr	$1,500	$2,000	$2,500	$250	$100	$2,500	$600	$600	$1,500	$ 200	$400

Computer Software

Infinity Turbine Applications Design and Engineering Program

We are now offering a simple program on Filemaker, which allows you to input data to get flow rates, btu heat flows, and a host of other features. This program is designed for sales and initial ORC system data. The Filemaker solution requires that you purchase(separately) and install Filemaker on your computer (Windows or Mac).

Price: $10,000 USD. Support is not included, but will be available from our support staff at $5,000 per year. Updates are included for the first year.

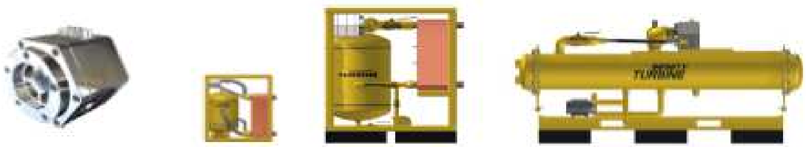

Figure B–1: ORC Price List from Infinity Turbine®

ORC Price Estimation Using Regression Analysis

The price of the ORC is estimated in this project using the price data from the manufacturer (shown in Figure B–1). The relationship between the price and the ORC plant size is estimated using regression analysis and the plot is as shown below (see Figure B–2).

ORC Size (Power Output) kW	Price ($)
0.9	20000
10	50000
50	130000
100	230000

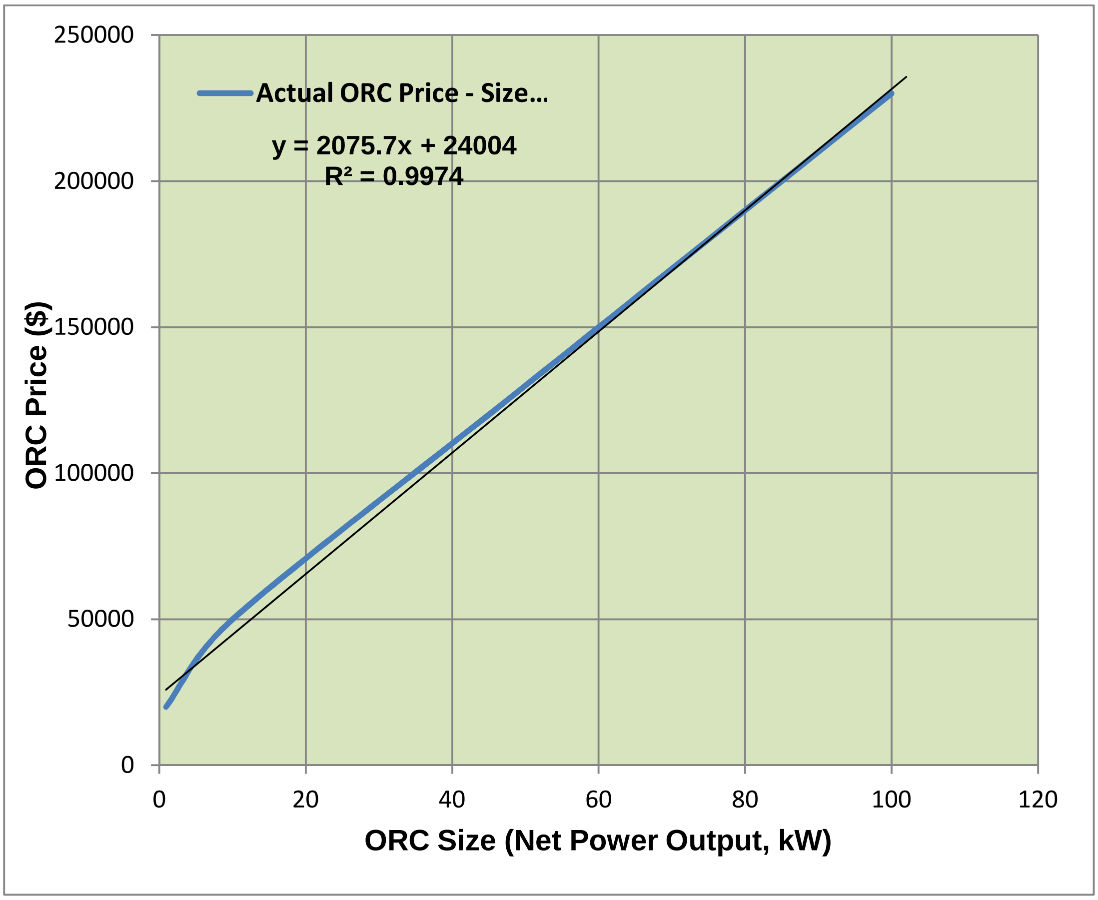

Figure B–2: ORC Price Estimation Plot

Appendix C

Chat with R245fa Vendor

Message History with Ting Yu (by Web-based TradeManager)

chat with:Ting Yu date:2011-12-19 11:01:24

Mathew Aneke(2011-12-19 10:13:44)
hello
how are you
Ting Yu(2011-12-19 02:14:00)
fine
Ting Yu(2011-12-19 02:14:05)
how about u?
Mathew Aneke(2011-12-19 10:14:14)
am okay, thanks
Mathew Aneke(2011-12-19 10:14:32)
Please do you deal on R245fa refrigerant?
Ting Yu(2011-12-19 02:15:08)
yes, we do
Ting Yu(2011-12-19 02:15:15)
are u in need of it?
Mathew Aneke(2011-12-19 10:15:31)
yes, but i will like to know the price first
Ting Yu(2011-12-19 02:15:52)
what the quantity do u need?
Mathew Aneke(2011-12-19 10:16:24)
i will determine that if i know the unit price
Ting Yu(2011-12-19 02:16:25)
I should check these with u in order to give the best price to u
Ting Yu(2011-12-19 02:16:36)
what the package?
Mathew Aneke(2011-12-19 10:16:58)
do you have the price per kg?
Ting Yu(2011-12-19 02:17:33)
ok, let me check it with recyclable cylinder 926L for u
Ting Yu(2011-12-19 02:17:44)
also what is the port of destination？
Mathew Aneke(2011-12-19 10:17:47)
ok
Mathew Aneke(2011-12-19 10:17:56)
Am in UK currently
Ting Yu(2011-12-19 02:18:21)
does the good should be to UK?
Mathew Aneke(2011-12-19 10:18:55)
Let me know the price first please
Ting Yu(2011-12-19 02:19:11)
ok
Ting Yu(2011-12-19 02:19:30)
pls have a wait
Ting Yu(2011-12-19 02:19:37)

let me check it
Mathew Aneke(2011-12-19 10:19:42)
ok, am waiting
Ting Yu(2011-12-19 02:23:49)
FOB NINGBO $9.85/kg with package
Ting Yu(2011-12-19 02:24:40)
hello~
Mathew Aneke(2011-12-19 10:24:45)
hi
Ting Yu(2011-12-19 02:25:00)
did u see the price?
Mathew Aneke(2011-12-19 10:25:7)
ya
Ting Yu(2011-12-19 02:25:29)
this is the latest price
Mathew Aneke(2011-12-19 10:25:38)
pls i will have to discuss with my client and get back to you soon
Ting Yu(2011-12-19 02:25:53)
ok
Mathew Aneke(2011-12-19 10:26:1)
i hope is in US DOLLARS
Ting Yu(2011-12-19 02:26:08)
yes
Ting Yu(2011-12-19 02:26:11)
it is USD
Ting Yu(2011-12-19 02:26:22)
USD 9.85/KG
Ting Yu(2011-12-19 02:28:10)
also the validity is 2 working days
Ting Yu(2011-12-19 02:28:24)
the gases market in this week is unstable
Ting Yu(2011-12-19 02:28:33)
the price is going up now
Mathew Aneke(2011-12-19 10:30:4)
Please whats the standard package size
Ting Yu(2011-12-19 02:30:22)
recyclable cylinder 926L
Ting Yu(2011-12-19 02:30:34)
it could pack 1000kg per cylinder
Mathew Aneke(2011-12-19 10:30:43)
ok
Ting Yu(2011-12-19 02:32:23)
waiting for your good news

Appendix D

Publications

Aneke, M., Agnew, B. and Underwood, C. (2011) Performance analysis of Chena binary geothermal power plant, *Applied Thermal Engineering*, 31, 1825–1832.

Aliyu, B., Agnew, B., **Aneke, M.**, Walker, S. and Atan, R. (2012) Prospects of Croton Megalocarpous (Musine) as a Source of Bio-Diesel (currently under review for NAE Conference).

Aneke, M., Agnew, B. and Underwood, C. (2011) Power generation through the use of waste heat energy from process industries: a greener approach to reducing CO2 emission and global warming in Nigeria, Proceeding of the FUTO 2011 Renewable & Alternative Energy Conference.

Aneke, M., Agnew, B. and Underwood, C. (2011) Approximate analysis of the economic advantage of a dual source ORC system over two single ORC systems in the conversion of dual low and mid grade heat energy to electricity, *EUEC Journal*.

Aneke, M., Agnew, B., Underwood, C., Wu, H. and Masheiti, S. (2012) Power generation from waste heat in a food processing application, *Applied Thermal Engineering*, 36, 171–180.

Aneke, M. C. and Menkiti, M. C. (2011) Geothermal: History, Classification and Utilization for Power Generation. *Geothermal Energy.* John Wiley & Sons (under review at John Wiley and Sons' Encyclopaedia of Geothermal Energy).

Aneke, M. C. (2011) Introduction to Process Modeling, Simulation and Optimization, in Nwoha, C., Holloway, M. and Onyewuenyi, O. A. (eds) *Process Plant Equipment Operation, Reliability and Control.* John Wiley & Sons (forthcoming).

Menkiti M. C, **Aneke, M. C.**, Onukwuli, O. D, and Ekumankama, E. O (2011) Response surface methodology as optimization tool for alum driven coag-flocculation of brewery effluent, *Proceedings of the Nigeria Society of Chemical Engineers Conference*, Lagos, Nigeria

Aneke, M., Agnew, B. and Underwood, C. (2012) Thermodynamic Analysis of Alternative Refrigeration Cycles Driven from Waste Heat in a Food Processing Application, *Internal Journal of Refrigeration*, 35, 1349 - 1358

Menkiti, M. C, **Aneke, M. C** and Onukwuli, O. D (2012) Optimization and kinetics of coag-flocculation of coal effluent by crab extract via response surface methodological analysis, Journal of Nigeria Society of Chemical Engineers (article in press)

Menkiti, M. C, **Aneke, M. C**, Ogbuene, E. B, Onukwuli, O. D, and Ekumankama, E. O (2012) Optimal evaluation of coag-flocculation factors for alum-brewery effluent system by response surface methodology, Journal of Minerals and Material Characterization and Engineering (JMMCE), USA

Work Experience

1. Process Engineer Consultant for Operation PTC UK

I am responsible for modelling energy-saving processes in order to improve energy usage and reduce greenhouse gas emissions.

2. Journal Reviewer, *International Journal of Refrigeration*

I am responsible for reviewing journal articles for the *International Journal of Refrigeration*, which is published by Elsevier.

3. Journal Reviewer, *Journal of Renewable and Sustainable Energy*