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INVESTIGATION INTO WASTE HEAT TO WORK IN THERMAL SYSTEMS IN ORDER TO GAIN MORE EFFICIENCY AND LESS ENVIRONMENTAL DEFECT

By

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Submitted in partial fulfilment of the requirements for the degree

MASTER OF SCIENCE (Mechanical Engineering)

in the

Faculty of Engineering, Built Environment and Information Technology

University of Pretoria

2016

Abstract

Title: Investigation into waste heat to work in thermal systems in order to gain more efficiency and less environmental defect

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Degree: Master of Science (Mechanical Engineering)

In most previous studies that have been conducted on converting waste heat energy from exhaust gases into useful energy, the engine waste heat recovery system has been placed along the exhaust flow pipe where the temperature differs from the temperature just behind the exhaust valves. This means that an important fraction of the energy from the exhaust gases is still lost to the environment. The present work investigates the potential thermodynamic analysis of an integrated exhaust waste heat recovery (EWHR) system based on a Rankine cycle on an engine's exhaust manifold. The amount of lost energy contained in the exhaust gases at the exhaust manifold level, at average temperatures of 500 °C and 350 °C (for petrol and diesel), and the thermodynamic composition of these gases were determined. For heat to occur, a temperature difference (between the exhaust gas and the working fluid) at the pinch point of 20°C was considered. A thermodynamic analysis was performed on different configurations of EWHR thermal efficiencies and the selected suitable working fluids. The environmental and economic aspects of the integrated EWHR system just behind the exhaust valves of an internal combustion engine (ICE) were analysed. Among all working fluids that were used when the thermodynamic analysis was performed, water was selected as the best working fluid due to its higher thermal efficiency, availability, low cost and environmentally friendly characteristics. Using the typical engine data, results showed that almost 29.54% of exhaust waste heat can be converted. This results in better engine efficiency and fuel consumption on a global scale by gaining an average of 1 114.98 Mb and 1 126.63 Mb of petrol and diesel respectively from 2020 to 2040. It can combat global warming by recovering 56.78 1 011 MJ and 64.65 1 011 MJ of heat rejected from petrol and diesel engines, respectively. A case study of a Volkswagen Citi Golf 1.3i is considered, as it is a popular vehicle in South Africa. This idea can be applied to new-design hybrid vehicles that can use the waste heat to charge the batteries when the engine operates on fossil fuel.

***Keywords**: Waste heat recovery, thermal efficiency, Rankine cycle, fuel consumption, global warming*

Acknowledgement

First and foremost, praises and thanks to God, the Almighty, for His showers of blessings throughout my research work to complete the research successfully.

I would like to express my sincere gratitude and appreciation to my supervisor, Dr Mohsen Sharifpur, and my co-supervisor, Prof Josua P Meyer, for their valuable guidance, encouragement and help.

I would also like to thank the National Research Foundation (NRF) for its financial support and the Clean Energy Research Group of the Department of Mechanical and Aeronautical Engineering at the University of Pretoria.

I am extending my exceptional thanks to my family and many friends for their love, prayers and supports throughout the process.

A special feeling of gratitude to my lovely wife, Flavienne Zelie Katamba whose words of encouragement and push for tenacity ring in my ears.

“Trust in the Lord with all your heart, and lean not on your own understanding, in all your ways acknowledge Him, and He shall direct your paths.” Proverbs 3:5-6

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Nomenclature

Alphabetic symbols

h – Enthalpy (kJ)

C_p – Specific heat (kJ/kgK)

$\dot{m}_{wf}$ – Working fluid mass flow rate (kg/s)

Q – Energy, heat (kW)

E_f – Energy contained in fuel (kJ)

$E_{f/d}$ – Total available fuel energy produced per day (kJ/d)

$E_{f/y}$ – Total available fuel energy produced per year (kJ/y)

m_f – Fuel mass admitted (kg)

$m_{f/d}$ – Fuel mass produced per day (kJ/kg)

H_u – Net fuel calorific value (kJ/kg)

ρ – Density (kg/m^3)

η_{th} – Thermal efficiency (%)

P – Gas pressure in cylinder (kPa)

V – Volume in cylinder (m^3)

v – Gas specific volume (m^3/kg)

R – Gas constant of air (kJ/kmol-K)

T – Temperature (°C)

u – Specific internal energy (kJ/kg)

C_p, C_v – Specific heats (kJ/kg-K)

K – Specific heat ratio C_p/C_v

Abbreviations

BC	Boiling condenser
BDC	Bottom-dead-centre
b_e	Effective specific fuel consumption
BMEP	Brake mean effective pressure
bsfc	Brake specific fuel consumption
Btu	British thermal units
C	Carbon
CFH	Closed feedwater heater
CI	Compression ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
ECE	External combustion engine
EER	Exhaust energy recovery
EES	Engineering Equation Solver
EGEC	Exhaust gas energy converter
EIA	Energy Information Association
EWEC	Engine waste energy converter
EWHR	Exhaust waste heat recovery
FWD	Front wheel drive
HC	Hydrocarbons
GHG	Greenhouse gases
H_2O	Water
HEV	Hybrid electrical vehicles
HLEC	Hot liquid energy converter
IAPWS	International Association for the Properties of Water and Steam

ICE	Internal combustion engine
isfc	Indicated specific fuel consumption
LPG	Liquefied Petroleum Gas
Mb	Mega barrels
MJ	Mega joules
mb/d	Million barrels a day
N_2	Nitrogen
NAAMSA	National Association of Automobile Manufacturers of South Africa
NEDC	New European Driving Cycle
NOx	Nitrogen oxide
NRF	National Research Foundation
OECD	Organisation for Economic Cooperation and Development
OFH	Open feedwater heater
ORC	Organic Rankine cycles
PM	Particulate matter
qkJ/y	Quadrillion kilojoules per year
sfc	Specific fuel consumption
SI	Spark ignition
SO_2	Sulphur dioxide
SOHC	Single overhead camshaft
TDC	Top- dead-centre
WHR	Waste heat recovery
WOT	Wide-open throttle

Chapter 1: Introduction

1.1. Historical background

The rapid economic and industrial development and unexpected global demographic growth of countries that do not belong to the Organisation for Economic Cooperation and Development (OECD), such as China and India, has resulted in a projected 56% increase in worldwide energy consumption between 2010 and 2040. According to the Energy Information Association (EIA) (2013), the total world energy use is expected to increase from 524 quadrillion British thermal units (Btu) in 2010 to 820 quadrillion Btu in 2040, as illustrated in Figure 1.1.

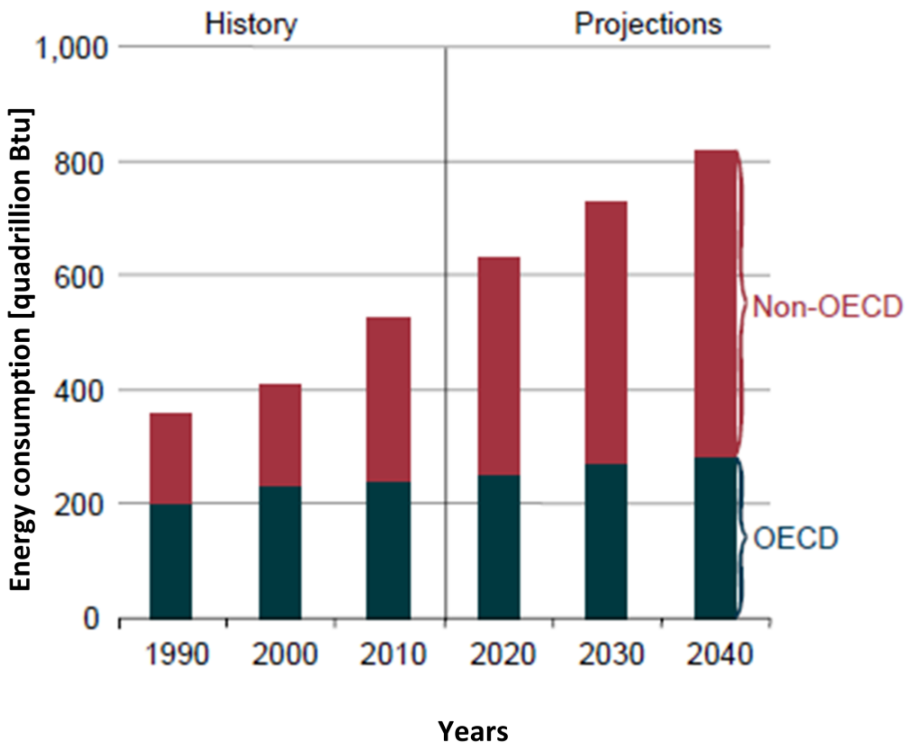


Figure 1.1: World energy consumption projection up to 2040 (EIA, 2013).

Although there are other means of energy production, such as coal and natural gas, renewable resources and nuclear, petroleum and other liquid fuels will remain the fuels that are consumed the most in the world. The use of these fuels will increase by more than one-third (33 MMbbl/d), from 87 MMbbl/d in 2010 to 119 MMbbl/d in 2040 (EIA, 2014).

Between 2010 and 2014, most of the growth in liquid fuel consumption came from the transport and industrial sectors. Despite an increase in and the anticipation of non-liquid-based transport technologies, these are not enough to compensate for the rising demand in the global transport sector. Transport services still account for 63% of the overall share growth in liquid fuel consumption worldwide from 2010 to 2040 (EIA, 2013) and from 2011 to 2040 (see Figure 1.2) (OPEC, 2014).

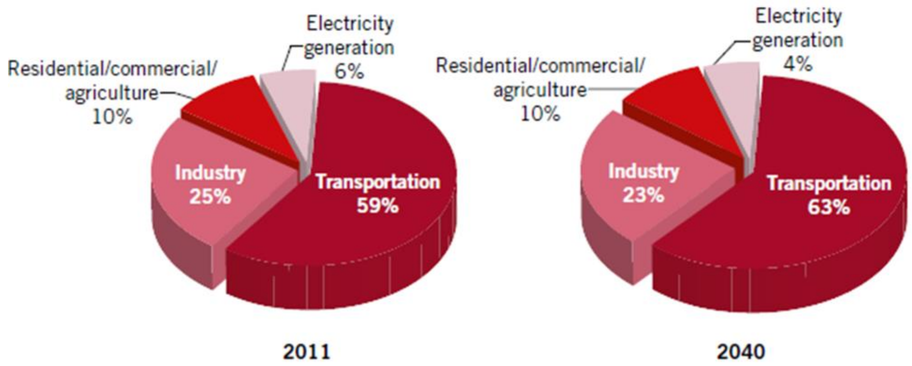


Figure 1.2: Oil percentage shares demand by sector in 2011 and the projected world demand by 2040 (OPEC, 2014).

This energy demand projection in the transport sector has been the leading key for many scientific researchers, due the fact that the biggest source of energy used in the transport sector is the thermal engine. Of these engines, the internal combustion engine (ICE) is the most popular. These engines convert thermal energy or heat into mechanical work on the one hand; on the other hand, the release of greenhouse gases (nitrogen oxide (NO_x), sulphur dioxide (SO_2) and carbon dioxide (CO_2)) and heat into the atmosphere from thermal engines can, directly or indirectly, impact negatively on both human lives and the environment. This release of greenhouse gases causes global warming such as the melting of glacier regions, changes in rainfall patterns, a rise in sea levels and ocean warming, heat waves, as well as unusual periods of warm weather (Marguerite Richardson and Agnes Phahlane 2009). Many guideline policies, such as the Kyoto Protocol, and considerable investigations into and solutions for more energy-efficient and environmentally friendly technologies have been implemented to reduce the emission of greenhouse gases and the effects of global warming.

Various types of thermal systems are classified according to the process these systems undergo to produce the work. This case study will focus on thermal engines, which can be internal or external combustion engines. Thermal engines are heat engines that convert thermal energy or heat into mechanical work. In an ICE, the fuel is burnt in a cylinder or vessel. In an external combustion engine (ECE), the working fuel is not burnt because it is heated from an external source. The fuel is heated and expanded through the internal mechanism of the engine, which results in work. Examples of such engines include steam turbines and steam trains.

Estimates showed that 20% to 50% of the energy input in a thermal system is lost as waste heat in the form of hot exhaust gases and cooling water, as well as heat lost from the hot surfaces of equipment and heated products (Breu, Guggenbichler and Wollmann, 2008). This waste heat does not only affect the thermal system's efficiency, but can, directly or indirectly, have a negative impact on the environment.

Nowadays, a call for more energy efficient and environmentally friendly technologies, such as energy recovery, can be heard all over the world to ensure a reduction in greenhouse gas emissions and global warming.

1.2. Problem statement

The world's population is expected to increase from 7 billion in 2010 to nearly 9 billion in 2040 (The Outlook, 2014). This global population increase will lead to considerable growth in the world's automotive vehicle use, which implies a significant expansion in energy demand in the transport sector on the one hand and a great risk for more global warming and negative environmental impacts on the other hand. This is because ICEs are still the main source of power used in transporting people and goods in our daily lives. As a result, researchers have taken a noticeable interest in cleaner and more efficient technologies for ICEs. The public has also become more aware of the world's environmental and energy consumption issues.

It has been shown that the two predominant available waste heat sources for an automotive vehicle's engine are the radiator and exhaust gas systems. Hatazawa, Hiroshi, Takahiro and Yoshitoki (2004) presume that more than 35% of the combustion energy produced is released as heat lost to the environment as exhaust gas or in the form of other losses, which are a function of the engine load.

A major amount of energy in an ICE is wasted through the exhaust gas in the form of heat, which could conceivably be recovered and re-used to improve the engine's output work. Conklin and Szybist (2010) show that only 10.4% of fuel energy is converted into useful work, while a considerable amount of thermal energy (27.7%) is wasted through the exhaust gas. In the same perspective, Yu and Chau (2009) stated that, from the thermal combustion process generated in an ICE (petrol), 40% of the heat generated is wasted to the environment through the exhaust manifold, which is higher than the energy dedicated to the engine's operation (25%).

1.3. Purpose of the study

Generally, due to the thermal limitations of the engines of automotive vehicle, it is inevitable that a fraction of the energy produced in ICEs is wasted in the form of heat discharged to the environment. According to a number of studies, the use of waste heat recovery technologies gives us a large potential ability to save some part of the energy lost in ICEs, which in return can be used to generate electrical or mechanical work or for heating the cabin. Previously, in most cases, the recovery of waste heat has been used for turbochargers, supercharging for high-performance engines or for warming the passengers' cabin.

Although some waste heat is used for turbochargers or supercharging, some heat is still lost through different streams in the automotive vehicle's engine system. A study by Mei, Chaturvedi, and Lavan (1979) has shown that almost half of the heat is wasted in an engine's exhaust manifold at low Reynolds numbers for a standard exhaust system.

Talom and Beyene (2009) conducted an experiment on the exhaust gas of a 2.8-litre V6 ICE that was used to operate a modified three-ton (10.55 kW) absorption chiller by utilising the hot exhaust gas intake from the engine. This proved to be applicable for refrigeration and vehicle transport. It could also considerably improve the system's performance in terms of the engine load.

Saidur, Rezaei, Muzammil, Hassan, Paria and Hasanuzzaman (2012) studied waste heat recovery technologies and their newest improvements on the exhaust gas from automotive combustion engines involving thermoelectric generators, an organic Rankine cycle, a six-stroke cycle ICE and new developments on a turbocharger. Their study shows that there is a latent opportunity for thermal efficiency to be maximised when incorporating these technologies with other devices. They also pointed out that the heat recovery technologies would not only increase the efficiency of the ICE, but would also have a positive impact on global warming as long as the fossil fuel reserves could be reduced.

In most of the previous studies that were conducted on converting waste heat energy from exhaust gases into useful energy, the waste heat recovery system is placed along the exhaust flow pipe where the temperature differs from the temperature behind the exhaust valves. This means that an important fraction of the energy from the exhaust gases is still lost to the environment. In this study, an investigation was conducted into the potential of integrating a waste heat recovery system based on a Rankine cycle just behind the exhaust valves of an ICE. The heat exchanger was fitted on the exhaust manifold, with the temperature of the exhaust gases at the exhaust valves ranging from 400 °C to 600 °C and from 200 °C to 500 °C for spark ignition (SI) and compression ignition (CI) engines respectively. In order to recover the large amount of waste energy from exhaust gases, an analysis of a suitable working fluid with great potential for recovering heat at high temperature was also conducted. The total energy consumed and wasted in transport sector throughout the outlook period from 2013 to 2040 was analysed. The environmental benefit of an EWHR was shown by comparing the amount of CO_2 released from an ICE with and without an EWHR system.

1.4. Dissertation layout

This thesis consists of a literature survey in Chapter 2, in which the energy consumption in the transport sector is highlighted. Heat recovery technologies and their advantages and limitations are also investigated. From the literature review, the problem is formulated and solved in Chapter 3. The results and discussions are elaborated on in Chapter 4. Conclusions, recommendations and possibilities for further work are described in Chapter 5.

Chapter 2: Literature survey

2.1. Introduction

A literature review is conducted in this chapter. An overview is given of the world energy demand and projection in the transport sector. The theory, energy balance, and fuel and engine efficiencies of ICEs are analysed. Different waste heat recovery (WHR) systems, as well as their limitations and choice of working fluid, are considered and compared. A WHR system based on a Rankine cycle is chosen for further investigation. Engine exhaust gas emissions and their environmental effects are also taken into consideration. In the end, concluding observations are made.

2.2. Energy consumption in the transport sector

2.2.1 Wold transport fuel demand and projection

In the recent decade, the use of energy in an efficient way in the transport sector is one of the major keys to reducing energy consumption and global warming. According to the 2013 Outlook for Energy (ExxonMobil, 2013), over 40% of energy increases from 2010 to 2040 will come from the transport sector, especially from commercial sources such as planes, trucks, trains and ships. Figure 2.1 gives an overview of the worldwide trend in liquid fuel consumption in the transport and other sectors.

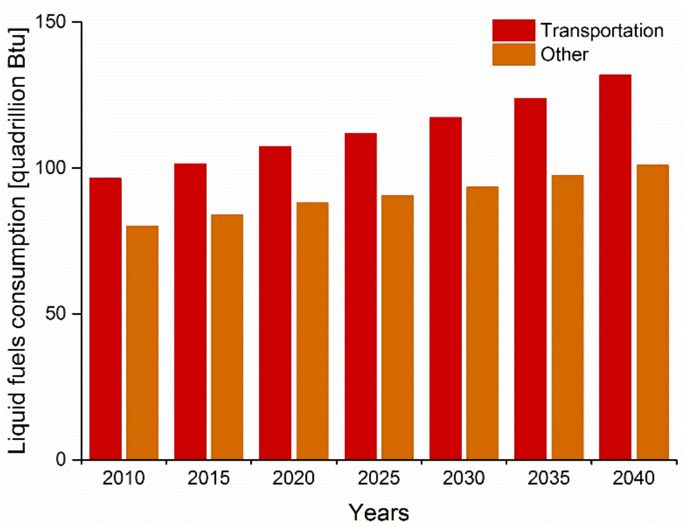


Figure 2.1: World liquid fuel consumption by end-use sector, 2010–2040 (quadrillion Btu), adapted from (EIA 2013).

According to the (EIA, 2013), worldwide energy consumption in the transport sector increases by an average of 1.1% per annum and it shares 63% of the total growth in worldwide petroleum consumption. A growth in petroleum consumption in the transport sector means an increase in the number of automotive vehicles used worldwide. In August 2011, Ward's Auto reported that the worldwide number of motor vehicles has increased from 980 million in 2009 to 1.015 billion in 2010 (Sousanis, 2011).

Despite the fact that the number of light motor vehicles will nearly double (from 800 million to more than 1.6 billion by 2040), the overall energy demands in the personal transport sector will remain almost stable due to advances in automotive technology, such as energy recovery, hybrid motor vehicles, as well as smaller and lighter motor vehicles.

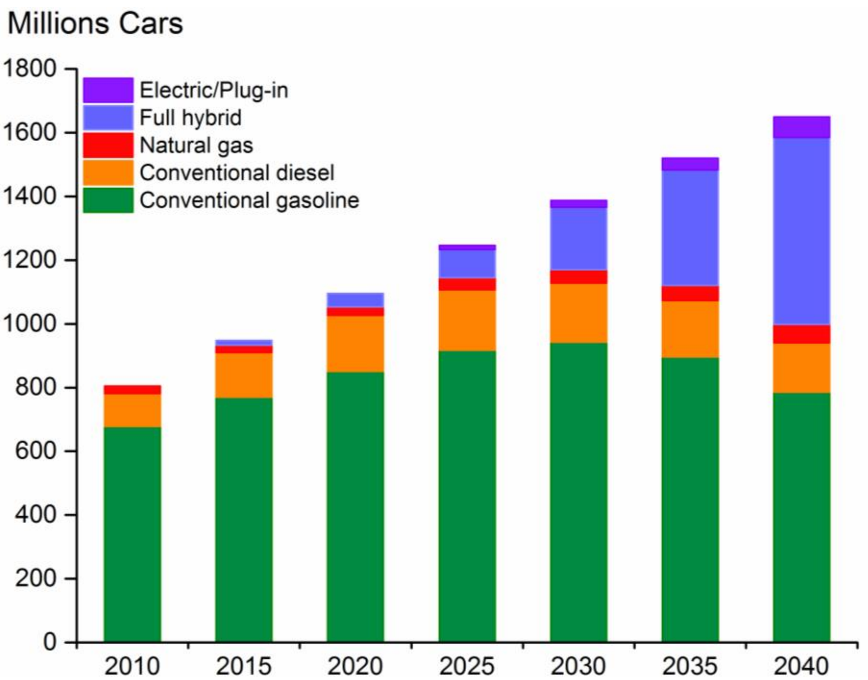


Figure 2.2: Vehicle fleet by type adapted from (ExxonMobil, 2013).

The trend in Figure 2.2. of vehicle type per fleet shows us, on one hand, that the growth of conventional petrol and diesel motor vehicles will decrease by about 50%, with more energy-efficient motor vehicles being available by 2040. On the other hand, full hybrid motor vehicles will increase by around 40% by 2040 (or 50% of the new motor vehicle market), while electric and plug-in hybrid motor vehicles will increase by 10% in the same year.

Although the number of conventional petrol and diesel motor vehicles will decrease by 2040, oil will remain the most important supply point of fuel in the transport sector. Despite the higher cost and functional limitations of alternative technologies such as plug-in hybrids or electric motor vehicles, it has also been shown that a motor vehicle can travel 563.27 km with 45.36 litres of petrol, while an electric motor vehicle's battery, which takes hours of charging, can only accomplish about 24.14 km of travel (ExxonMobil, 2013). One can see that plug-in hybrids or electric motor vehicles still need to improve substantially before the technology can make an important impact on the market. Thus, improving the technology in conventional diesel and petrol vehicles will still be one of researchers' main focuses as long as it not only helps make vehicles more efficient (fuel economy), but also plays a major role in considering the environmental effects (such as greenhouse gas emission).

2.2.2 Oil-refined products' demand and projection from 2013 to 2040

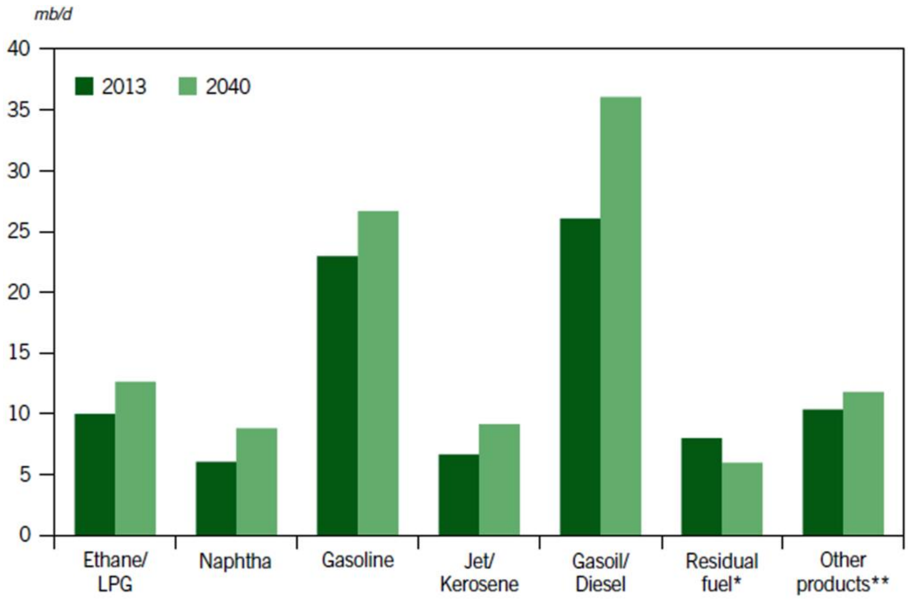
Table 2.1 and Figure 2.3 give a clear view of the global trend and shares of fuel products by type during the forecast period from 2013 to 2040. The increase in the demand of the middle distillates in the transport sector, which mainly focuses on diesel/gasoil and petrol, is mostly emphasised (OPEC, 2014).

Table 2.1: Global demand and shares of fuel products according to type from 2013 to 2040

	Global demand millions of barrels per day (mb/d)							**Shares %**	
	2013	**2015**	**2020**	**2025**	**2030**	**2035**	**2040**	**2013**	**2040**
Light products									
Ethane/liquefied petroleum gas (LPG)	10.0	10.3	11.0	11.6	12.1	12.4	12.6	11.1	11.4
Naphtha	6.0	6.2	6.6	7.1	7.6	8.1	8.8	6.7	7.9
Petrol	23.0	23.6	24.6	25.4	25.9	26.3	26.7	25.5	24.0
Middle distillates									
Jet/kerosene	6.6	6.8	7.3	7.8	8.2	8.7	9.2	25.5	24.0
Diesel/gasoil	26.1	27.1	29.7	31.7	33.3	34.7	36.1	29.0	32.5
Heavy products									

	Global demand millions of barrels per day (mb/d)							Shares %	
	2013	2015	2020	2025	2030	2035	2040	2013	2040
Residual fuel*	8.0	7.8	7.1	6.9	6.6	6.3	6.0	8.9	5.4
Other**	10.4	10.4	10.6	10.9	11.2	11.5	11.8	11.5	10.6
Total	**90.0**	**92.3**	**96.9**	**101.3**	**104.8**	**108.0**	**111.1**	**100.0**	**100.0**

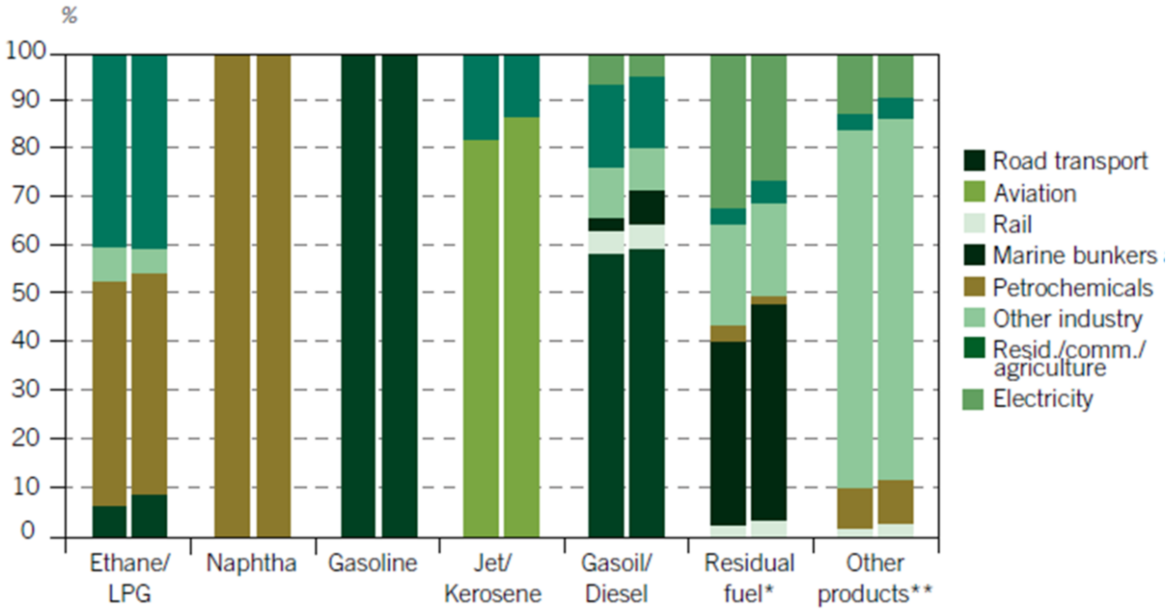
This data allows the trend in worldwide energy demand throughout the forecast period (2013 to 2040) to be evaluated later in this study.



* *Includes refinery fuel oil.*
** *Includes bitumen, lubricants, petroleum coke, waxes, still gas, sulphur, direct use of crude oil, etc.*

Figure 2.3: Global refined product demand from 2013 to 2040 (mb/d).

2.2.3 Fuel share demand by type and sector



* *Includes refinery fuel oil.*

** *Includes bitumen, lubricants, petroleum coke, waxes, still gas, sulphur, direct use of crude oil, etc.*

Figure 2.4: The refined share products' demand according to sector for 2013 and 2040.

The rate of increase in the demand for gasoil/diesel and petrol in other sectors, such as commercial, agriculture, residential and industry, will be lower than in the transport sector. Almost 60% of the total gasoil/diesel oil demand will come from the road transport sector, as illustrated in Figure 2.4.

2.3. Internal combustion engines

2.3.1 General theory of internal combustion engines

ICEs are heat engines that convert the chemical energy contained in the fuel into mechanical energy, which is then made available on a rotating crankshaft. Firstly, the fuel's chemical energy is transformed into thermal energy through combustion or oxidation with air in the engine. Secondly, this thermal energy raises the pressure and temperature of the gas inside the engine and the high-pressure gas is converted into a rotation motion of the shaft (mechanical energy) by means of the engine's mechanical linkages. Finally, the mechanical energy of the shaft is transmitted to the desired end use via a power train. This can be visualised via the first law of thermodynamics which stipulates; the change in internal energy of system in a given state, equals to heat added unto the system minus the work done by the system Borgnakke and Stonntag (2009).

$$\Delta Engery = +in - out \quad or\ \Delta U = Q - W \tag{2.1}$$

2.3.2 Engine classifications

Various types of engines are classified according to the process they undergo to produce the mechanical work.

This study will pay attention to thermal engines. These can be ICEs or ECEs. In an ICE, the fuel is burnt in a cylinder or vessel. In an ECE, the working fuel is not burnt, but is heated from an external source. The fuel is heated and expanded through the internal mechanism of the engine, which results in work such as in steam turbines and steam trains.

The best-known examples of internal combustion are internal combustion and gas turbines (Heywood, 1988).

There are different ways of classifying ICEs. Pulkrabek (2004) provides the following overview of their specifications:

1. **Types of ignition**

- In an SI engine, the cyclical combustion process is initiated by a high-voltage electrical discharge between the two electrodes of a spark plug, which ignites the air-fuel mixture within the combustion chamber surrounding the spark plug.
- In a CI engine, the combustion process starts when the highly compressed air-fuel mixture auto-ignites because of the high temperature in the combustion chamber.

2. **The engine cycle**

- In a four-stroke cycle, four pistons accomplish up and down movements over two engine revolutions per cycle.
- In a two-stroke cycle, each cycle corresponds to the movement of two pistons over one revolution.

3. The location of the valves

- In a head (overhead valve), also called an I-head engine.
- In a block (flat head), also called an L-head engine.
- In a block with valves on one side of the cylinder and the exhaust on the other side, also called a T-head engine.

4. The basic design

- A reciprocating engine is characterised by one or more cylinders in which the pistons move up and down. A combustion chamber is located at the top of each cylinder and the output power is usually delivered to the crankshaft by means of a mechanical linkage with the pistons.
- A rotary engine consists of a non-rotating block (stator), which houses a non-concentric rotor and crankshaft; combustion chambers are built into the stator.

5. Position and number of cylinders of reciprocating engines (Figure 2.5)

- A single-cylinder engine has one cylinder and a piston connected to the crankshaft.
- In the case of in-line cylinders, there can be two to 11 cylinders or more. The cylinders are in a straight line along the length of the crankshaft. Four-cylinder in-line engines are commonly used in most applications, especially in the engines of motor vehicles.
- In a V-engine, two banks of cylinders are placed at an angle to each other (varying from 15° to 120°, but mostly between 60° and 90°) along a single crankshaft. They have even numbers from 2 to 20 or more. V6 and V8 engines are commonly used in motor vehicles.
- An opposed cylinder engine has two banks of cylinders mounted on a crankshaft opposite each other (180° V). It is mostly used on small motor vehicle and aircraft engines.
- A W-engine is the same as a V-engine, except that it has three cylinder banks on the same crankshaft. It is mostly used for racing cars.
- An opposed piston engine is characterised by two pistons placed within the same cylinder with a common combustion chamber at the centre of the two pistons that are mounted on two different crankshafts, which means that two power strokes are produced at the same time.
- A radial engine has pistons placed in a circular plane position around a central crankshaft. It usually consists of an odd number of pistons and operates on a four-stroke cycle.

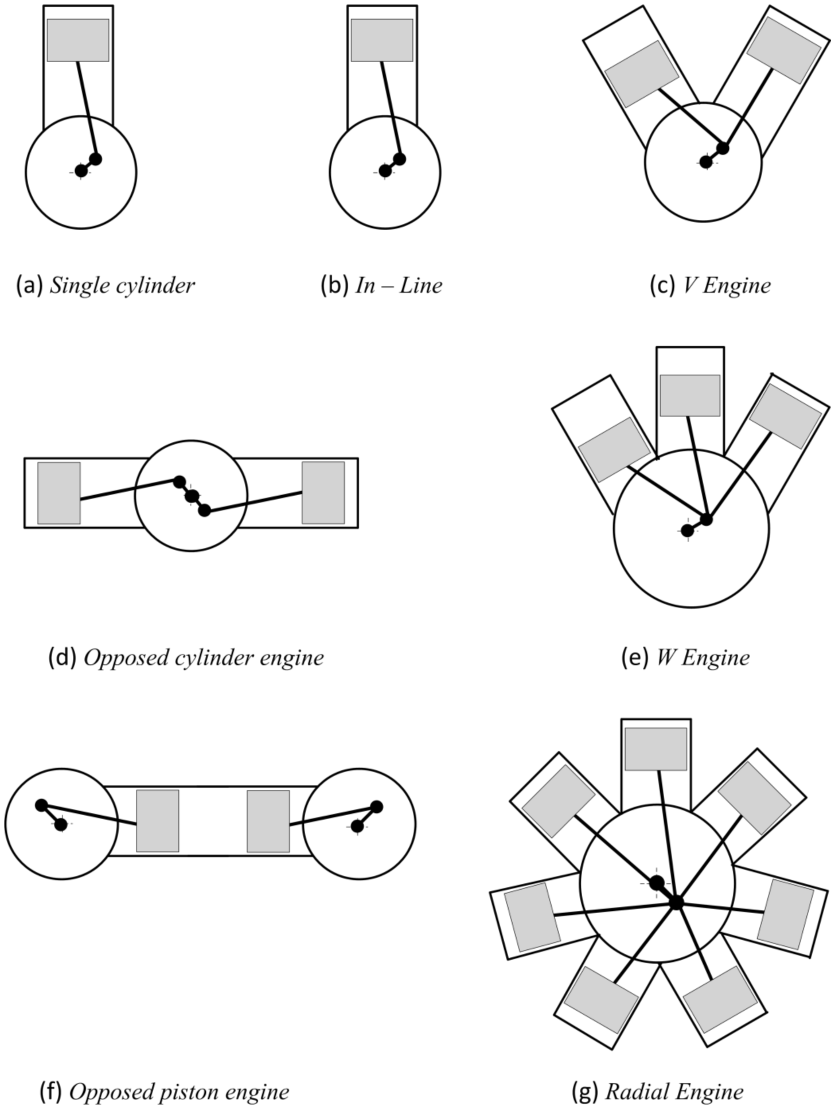


Figure 2.5: The classification of engines according to the cylinder arrangement, adapted from Pulkrabek (2004).

6. Air intake process

- During the naturally aspirated process, the intake air pressure is not boosted by an external system.
- The supercharged intake air pressure is increased by an auxiliary compressor driven off the engine crankshaft.
- The turbocharged intake air pressure is increased with the turbine compressor driven by the exhaust gases.
- The two-stroke cycle engine uses the crankcase as the intake air is compressed.

7. Method of fuel input for SI engines

- Some engines use a carburetted fuel input.
- A multipoint port fuel injection has one or more injectors placed at each cylinder intake.
- A throttle body fuel injection has injectors placed up front on the intake manifold.

8. Fuel used

- Petrol
- Diesel oil or fuel oil
- Gas, natural gas or methane
- Liquefied petroleum gas (LPG)
- Alcohol – ethyl or methyl
- Dual fuel: a combination of two or more fuels such as methane and diesel fuel, which is usually used in large CI engines; a combination of petrol and alcohol fuels commonly used as a straight substitute fuel for petrol automotive engines
- Gasohol: a common fuel containing 90% petrol and 10% alcohol

9. Application

- Motor vehicles, trucks and buses
- Locomotives
- Stationary
- Marine
- Aircraft
- Small portable devices, chainsaws, model airplanes

10. Type of cooling

- Air cooled
- Liquid and water cooled

One can combine some or all of the above classifications into the following engine characteristics: turbocharged, reciprocating, SI, four-stroke cycle, overhead valve, water-cooled, petrol, multipoint fuel-injected and V8 automotive vehicle engine.

All these distinctions are important, as they illustrate the variety of the engine designs that are available. Table 2.2 gives a description of the most usual applications, predominant type and approximate power range of ICEs.

Table 2.2: Classification of ICEs by application

Class	Service	Approximate engine power range (kW)	Predominant type		
			D or SI	Cycle	Cooling
Road vehicles	Motorcycles, scooters	0.75 – 70	SI	2, 4	A
	Small passenger vehicles	15 – 75	SI	4	A, W
	Large passenger vehicles	75 – 200	SI	4	W
	Light commercial vehicles	35 – 150	SI, D	4	W
	Heavy (long-distance commercial vehicles	120 – 400	D	4	W
Off-road vehicles	Light vehicles (factory, airport, etc.)	1.5 – 15	SI	2, 4	A, W
	Agricultural	3 – 150	SI, D	2, 4	A, W
	Earth moving	40 – 150	D	2, 4	W
	Military	40 – 2 000	D	2, 4	A, W
Railroad	Rail cars	150 – 400	D	2, 4	W
	Locomotives	400 – 3 000	D	2, 4	W
Marine	Outboard motors	0.4 – 75	SI	2	W
	Inboard motor crafts	4 – 750	SI, D	4	W
	Light naval crafts	30 – 2 200	D	2, 4	W
	Ships		D	4	W
	Ships' auxiliaries		D	4	A
Airborne vehicles	Airplanes	45 – 2 700	SI	4	A
	Helicopters	45 – 1 500	SI	4	A
Home use	Lawn mowers	0.7 – 3	SI	2, 4	A
	Snow blowers	2 – 5	SI	2, 4	A
	Light tractors	2 – 8	SI	4	A
Stationary	Building service	7 – 400	D	2, 4	W
	Electric power	35 – 22 000	D	2, 4	W
	Gas pipeline	750 – 5 000	SI	2, 4	W

Legend:

SI: Spark ignition D: Diesel A: Air W: Water cooled

2.3.3 The energy balance of internal combustion engines

The amount of energy provided to an engine is the heat value of the fuel (chemical energy) consumed. However, as we know, only a portion of the total chemical energy that enters an engine is converted to useful crankshaft work. Most of time, the remaining energy is lost from the engine in several ways. The energy flow distribution in the engine can be categorised as follows:

- Energy to useful work
- Energy to exhaust gas
- Energy to coolant
- Direct energy loss through walls
- Energy to the sump

From the abovementioned engine flow energies, only two principal parts of the heat do not contribute to the crankshaft work: the heat lost through the cooling medium and the exhaust gases.

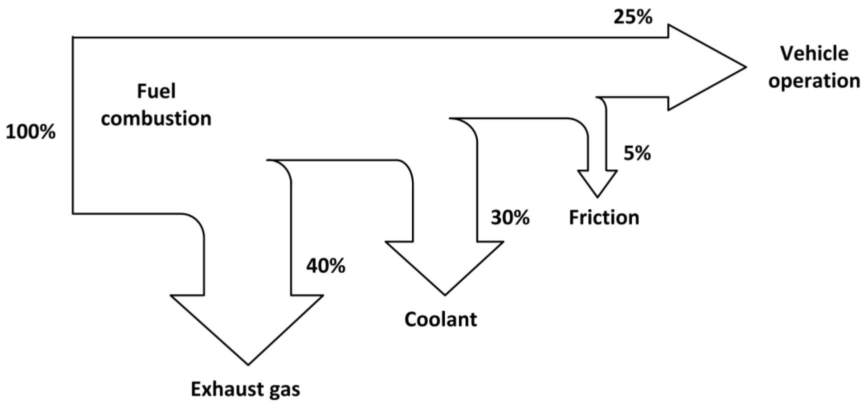


Figure 2.6: Energy distribution in an ICE, adapted from Arlotto and Millikin (2005) and Yu and Chau (2009).

Figure 2.6 depicts the energy distribution in an ICE. From the 100% of available combustion energy, only about 25% is actually dedicated to the motion of the car and its accessories. The rest of the energy is wasted (+5% to parasitic and friction losses, 30% to cooling and 40% to the exhaust). On the other hand, diesel engines and lean combustion petrol engines fare somewhat better, as 35% of the energy flows to mobility and accessories.

2.3.4 Distribution of energy in internal combustion engines

In an ICE, the available amount of useful energy or power is given by:

$$\dot{W} = \dot{m}_f Q_{HV}, \tag{2.2}$$

where $\dot{m}_f$ is the rate of fuel into the engine and Q_{HV} is the fuel-heating value.

However, the fuel flow rate is limited by the air mass flow rate that needs to react with the fuel. Therefore, the brake thermal efficiency is the ratio of the converted useful energy (power) at the crankshaft to the total available energy (Pulkrabek, 2004), given in Equation 2.3.

$$(\eta_t)_{brake} = {\dot{W}_b}/{\dot{m}_f Q_{HV} \eta_C}, \tag{2.3}$$

where η_t *is* the thermal efficiency, η_c is the combustion efficiency and $\dot{W}_b$ is the brake power.

It can be seen that a considerable part of energy is wasted as heat loss, parasitic load and waste exhaust gas. Thus, a typical ICE energy distribution as a percentage of the total fuel energy is shown in Figure 2.8, where the friction losses have been counted twice: once as the original loss and once as the resulting heat losses (Pulkrabek, 2004).

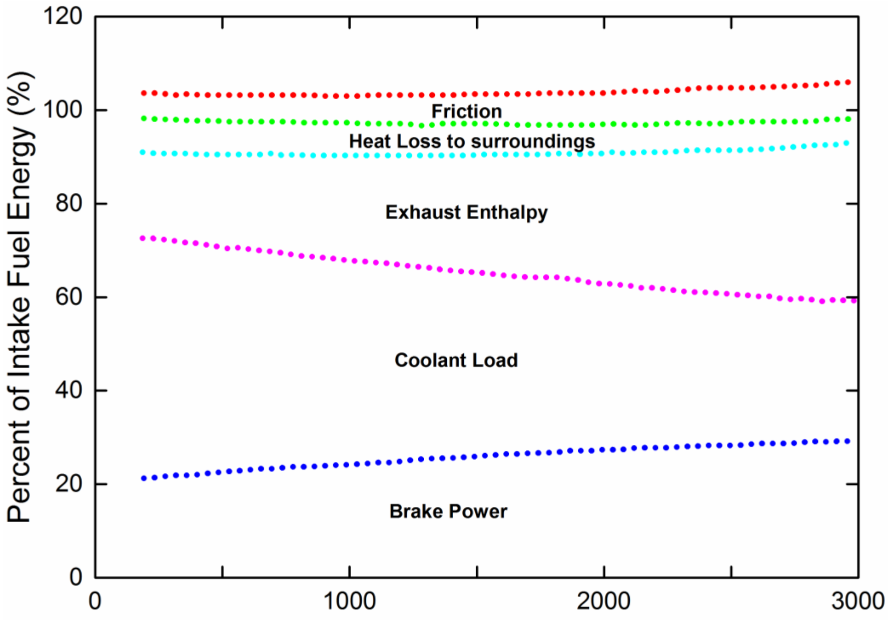


Figure 2.7: Energy distribution in a typical SI engine as a function of engine speed, adapted from Pulkrabek (2004).

For any engine, the distribution of energy is given as follows:

- Total engine energy:

$$Power\ generated = \dot{W}_{shaft} + \dot{Q}_{exhaust} + \dot{Q}_{loss} + \dot{W}_{acc} \tag{2.4}$$

with $\dot{W}_{shaft}$ being the brake output power, $\dot{Q}_{exhaust}$ the energy lost in the exhaust flow, $\dot{Q}_{loss}$ all the other energy lost to the surroundings by heat transfer, and $\dot{W}_{acc}$ the power to run the engine's accessories.

As a function of the engine size and geometry, and how the engine is operated, the shaft output power is represented by the following equation:

- Shaft output energy:

$$\dot{W}_{shaft} \approx 25 - 45\%$$

Generally, a CI engine, which refers to a diesel engine, has a higher shaft power output than an SI engine, which refers to a petrol engine.

- Energy lost in exhaust gas:

$$\dot{Q}_{exhaust} \approx 20 - 45\%$$

Due to their higher exhaust temperatures, a major part of the energy in SI engines is wasted through the exhaust flow in the form of enthalpy (heat) and chemical energy. At full load, half of the exhaust loss is constituted by the chemical energy, while it is usually higher than the brake power output of the engine under many operating conditions

- Other heat losses:

$$\dot{Q}_{loss} \approx 10 - 35\%$$

Generally, in many engines, the heat losses can be subdivided:

$$\dot{Q}_{loss} = \dot{Q}_{coolant} + \dot{Q}_{oil} + \dot{Q}_{ambient} \tag{2.5}$$

With CI engines on the high end, energy dissipated in the coolant is in order of:

$$\dot{Q}_{coolant} \approx 10 - 30\%$$

This can account for half of the output brake energy at a high load and increases for about twice the brake output power at a low load.

Oil loss energy depends on its type and on the engine speed.

$$\dot{Q}_{oil} \approx 5 - 15\%$$

Energy directly transferring to the surroundings is shown by the following equation:

$$\dot{Q}_{ambient} \approx 20 - 10\%$$

Friction losses are in the following range:

$$\dot{W}_{friction} \approx 10\%$$

In this regard, considerable interest in the advanced technologies of highly efficient ICEs has been shown by researchers due to public awareness of environmental and energy issues.

In ICEs, vast amounts of energy are wasted through the exhaust gas in the form of heat, which could conceivably be recovered and re-used to improve the engine's work output. Conklin and Szybist (2010) show that only 10.4% of the fuel energy is converted into useful work, while a considerable amount of the fuel energy (27.7% of the thermal energy) is wasted through the exhaust gas. In the same perspective, Yu and Chau (2009) stated that, from the thermal combustion process generated by an ICE (petrol), 40% of the heat generated is wasted to the environment through the manifold, compared to 25% of the energy dedicated to the engine's operation.

The two predominant available waste heat sources for the engine of a motor vehicle are the radiator and the exhaust gas systems. Hatazawa et al. (2004) presume that more than 35% of the combustion energy produced during the engine's combustion is released as heat lost to the environment as exhaust gas or in the form of other losses. It can be seen that recovering this kind of loss must be a function of the engine load.

2.4. Fuel and engine efficiencies

The engine efficiency is known as the ratio of the output energy (mechanical work done by the engine) and the input energy (heat or heat contained in the fuel), as shown in Figure 2.8. It can also be defined as the rate at which the heat is converted to the mechanical work.

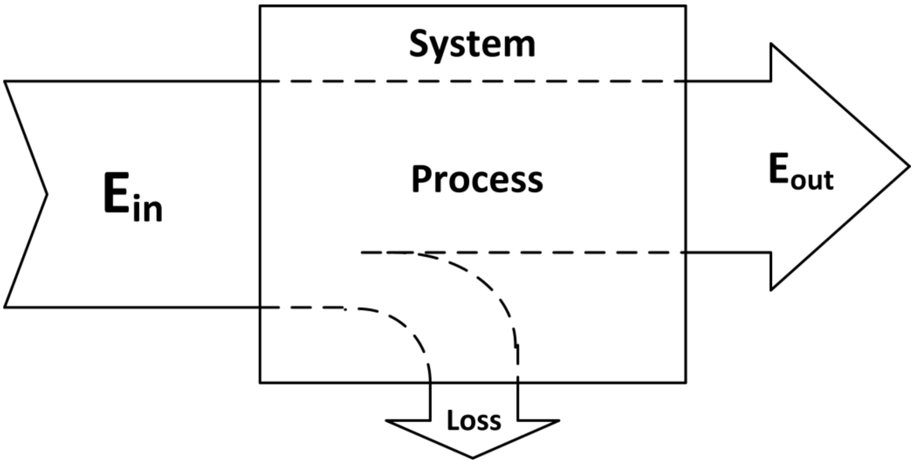


Figure 2.8: An efficiency diagram, adapted from (Jordan Hanania, Braden Heffernan, James Jenden, Kailyn Stenhouse 2015) (Sankey diagram).

Fuel efficiency is a form of thermal efficiency, which means the efficiency of the process that converts the chemical potential energy contained in a carrier fuel into kinetic energy or work. Overall fuel efficiency may vary per device, which in turn may vary per application. This spectrum of variance is often illustrated as an ongoing energy profile.

The admitted energy of the fuel is determined as follows:

$$E_f = m_f \times H_u, \tag{2.6}$$

where m_f is the weight of the fuel admitted, and H_u is the net calorific value of the fuel.

The fuel consumption is measured as a volumetric flow or as a mass flow per unit of time.

$$\dot{m}_f = \frac{m_f}{t} = \rho_f \times \dot{V}_f, \tag{2.7}$$

with ρ_f as the density of the fuel.

2.4.1 Specific fuel consumption (sfc)

The specific fuel consumption is a useful engine parameter, which is used to determine how efficiently fuel has been used in the engine to produce the work (Pulkrabek, 2004).

$$\text{sfc} = \frac{\dot{m}_f}{\dot{W}}, \tag{2.8}$$

where $\dot{m}_f$ is the fuel mass flow rate into the engine and $\dot{W}$ is the power delivered by the engine.

For better comparability, the indicated or effective power can also be used as a reference for fuel consumption.

2.4.2 Indicated specific fuel consumption (isfc or bi)

$$\text{isfc} = \frac{\dot{m}_f}{\dot{\text{W}}_i}, \tag{2.9}$$

where $\dot{m}_f$ is the mass flow rate, and $\dot{\text{W}}_i$ or $\dot{P}_i$ is the indicated power.

2.4.3 Effective specific fuel consumption (b_e)

$$b_e = \frac{\dot{m}_k}{P_e} = \frac{1}{\eta_e H_u}, \tag{2.10}$$

where η_e is the effective efficiency.

2.4.4 Brake specific fuel consumption (bsfc)

$$\text{bsfc} = \frac{\dot{m}_f}{\dot{\text{W}}_b}, \tag{2.11}$$

where $\dot{\text{W}}_b$ is the brake power.

There is also an important parameter, the “fuel conversion efficiency, η_f”, which is the measure of engine efficiency given by:

$$\eta_f = \frac{W}{m_f Q_{HV}} = \frac{\dot{W}}{\dot{m}_f Q_{HV} \eta_c} \tag{2.12}$$

2.4.5 Combustion efficiency

Combustion efficiency is a measure of how efficiently a device consumes fuel. Ideally, combustion efficiency would be measured at 100%, which means that the fuel was completely consumed. In practice, this level of combustion efficiency is impossible to achieve, but it is possible to come close to that. The lower the combustion efficiency, the less efficient the device is, which makes it expensive to run, wasteful of fuel and harmful to the environment. The time available for the combustion process of an engine cycle is very brief, and not all fuel molecules may find an oxygen molecule with which to combine, or the local temperature may not favour a reaction. Consequently, a small fraction of fuel does not react and exits with the exhaust flow. A combustion efficiency (TJc) is defined to account for the fraction of fuel that burns. TJc typically has values in the range of 0.95 to 0.98 when an engine operates properly (Pulkrabek, 2004).

2.4.6 Mechanical efficiency

The major part of the indicated engine power is dedicated to evacuate the exhaust gases and let in a fresh charge. However, an additional part is drained by the friction forces of the pistons, bearings and other mechanical components of the engine (Heywood, 1988). This leads to the notion of mechanical efficiency, which is defined as the ratio of the brake work at the crankshaft to indicate work in the combustion chamber (Pulkrabek, 2004).

$$\eta_m = \frac{\dot{W}_b}{\dot{W}_i}, \tag{2.13}$$

where $\dot{W}_b$is the brake work and $\dot{W}_i$ is the indicated work.

In modern automotive vehicle engines that operate at wide-open throttle, the mechanical efficiency is in the range of 75% to 95% at high speed, which then decreases as the engine speed decreases to zero at idling conditions when no work is taken by the crankshaft (Pulkrabek, 2004).

2.4.7 Thermal efficiency

Thermal efficiency is the ratio of the output and input energy of a system. It normally has to be between 0% and 100%. A thermal efficiency of 100% means that all the input energy that enters the system is converted into output energy, despite its form. Practically, thermal efficiencies are usually under 100% for different reasons.

$$\eta_{th} = \frac{W}{Q_{in}} = \frac{\dot{W}}{\dot{Q}_{in}} = \frac{\dot{W}}{\dot{m}_f Q_{HV} \eta_c} = \frac{\eta_f}{\eta_c} \tag{2.14}$$

$$\eta_{th} = \frac{\dot{W}}{(sfc) Q_{HV}} \tag{2.15}$$

The input heat transfer and the input heat transfer rate are given by the following equations:

$$Q_{in} = m_f Q_{HV} \eta_c \tag{2.16}$$

$$\dot{Q}_{in} = \dot{m}_f Q_{HV} \eta_c \tag{2.17}$$

where: W = work of one cycle

- $\dot{W}$ = power mf = mass of fuel for one cycle
- $\dot{m}_f$ = mass flow rate of fuel into the engine
- Q_{HV} = heating value of the fuel
- η_f = fuel conversion efficiency

2.5. Heat recovery technologies

Generally, the two main sources of energy waste in an engine are the engine cooling system (radiator) and the exhaust gas, Hendricks and Lustbader (2002b). This is due to the higher temperature that is carried in the exhaust gases and engine coolant fluids, as illustrated in Figure 2.9 (adapted from Pulkrabek, 2004).

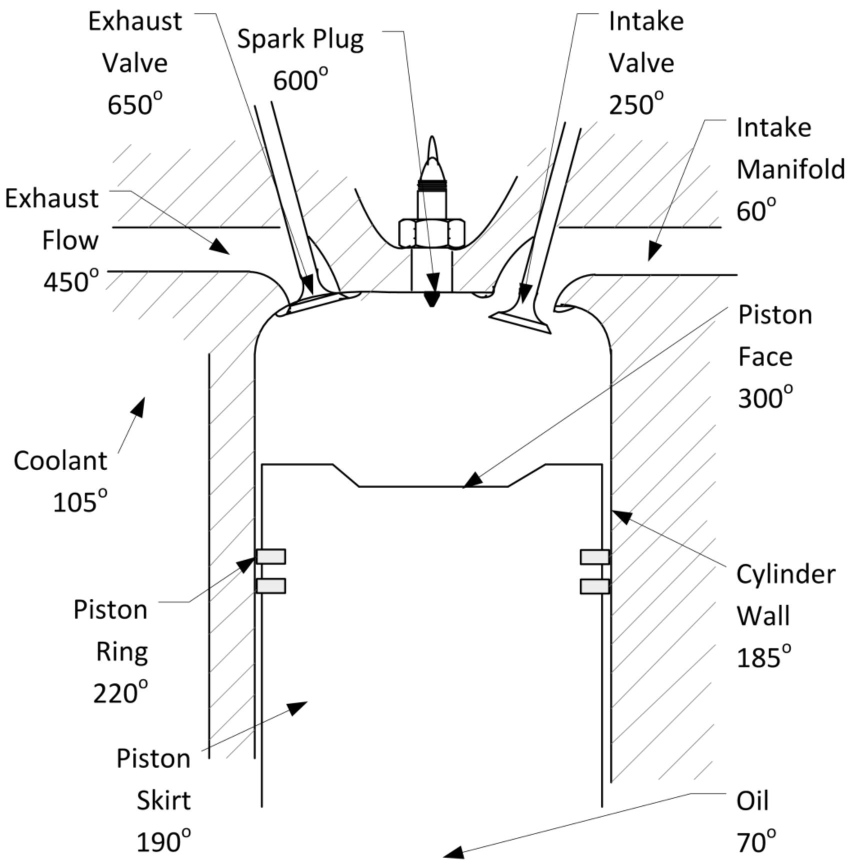


Figure 2.9: The temperature distribution (˚C) of an SI engine that operates in steady-state conditions, adapted from Pulkrabek (2004).

Therefore, in the recent decade, several studies on WHR improvement to enhance the thermal efficiency of ICEs have been undertaken using different technologies based on different principles. The two most promising ways to recover thermal energy from exhaust gases are thermoelectricity and Rankine cycles. Thermoelectricity is based on the electrical principle of converting thermal energy into electrical energy by means of temperature gradients Hendricks and Lustbader (2002), while Rankine cycles are based on a smart thermodynamic cycle parallel to the main cycle. This cycle works on the temperature difference between the hot working fluid and the environment, which will be the focus of this case study.

A smart thermodynamic cycle usually incorporates a heat exchanger (or evaporator), turbine, condenser or pump. In this study, water was used as the working fluid. To understand smart thermodynamic cycles, a brief overview of the theory of the different components used in the thermodynamic cycle is presented.

1. Heat exchanger

A heat exchanger is a heat transfer device that transfers the available heat (or thermal energy) to or from fluids at different temperatures. Generally, the fluids do not mix in heat exchangers, as they are separated by a heat transfer surface. Heat exchangers are used in the processing, power, petroleum, transport, air conditioning, refrigeration, cryogenic, heat recovery, alternate fuels and other industries.

For a steady-state heat exchanger, some aspects have to be considered. There is no shaft work or electrical work, and pressure drop may or may not be taken into consideration as the process occurs at an almost constant pressure (a small pressure drop due to the wall friction).

2. Turbine

A turbine is a rotary machine that produces shaft work (power on a rate basis) by converting the working fluid's kinetic energy, which can be water, steam or gas, into mechanical energy.

Generally, changes in the potential and inlet kinetic energy are negligible. The kinetic energy can mostly be neglected too, as can any heat rejection from the turbine.

3. Condenser

A condenser (typically a heat exchanger) consists of an arrangement of tubes or pipes in which a fluid condensates from its gaseous to its liquid state. In a steady-state condition, the condenser does no shaft or electrical work and the pressure drop can also be neglected because it is usually very small. Changes in potential and kinetic energy are negligible, as they are generally small.

4. Pump

The principal role of a pump is to increase the fluid pressure by supplying shaft work.

In terms of the first law of thermodynamics, changes in inlet kinetic and potential energy are negligible. The exit kinetic energy and the heat rejection from the compressor, which are relatively small, can often also be neglected. The following sections provide a brief overview of the different smart thermodynamic cycles.

2.5.1 Engine waste energy converter

Engine waste energy converters (EWECs) can recover some part of waste heat from the engine jacket, as well as the exhaust gas, and convert it into useful work for air conditioning or any hybrid application purpose, for example (Sharifpur, 2008). Based on a smart thermodynamic cycle, the EWEC has two different sections of heat recovery: the cylinder jacket and the exhaust gas jacket, in which a working fluid circulates at a given pressure and temperature if its type is known. An EWEC also has the following components:

- A condenser (radiator)
- A thermostat that allows the working fluid to flow from the cylinder jacket to the exhaust jacket once its temperature reaches that of the engine operating condition
- A recirculation pump that circulates the working fluid in a controlled and precise manner inside the housing so that the quantity of exchanged heat can be controlled by adjusting the recirculation pump's performance
- A small turbine that produces shaft work by means of vapour from the exhaust gas jacket

2.5.2 Exhaust gas energy converter

In the exhaust gas energy converter (EGEC) thermodynamic cycle, the only source of energy recovery is the exhaust gas. To recover a certain amount of heat from the exhaust gas at a high temperature, heat is exchanged from the hot exhaust gas while crossing the tube bundle to an appropriate subcooled working fluid in the outside of the tube bundle. Once vapour is produced in the boiling heat exchanger, it can enter a specific turbine, which results in shaft work. A condenser, a pump with a smart control system and a recirculation pump with the same function as the one from the EWEC have been used.

2.5.3 Hot liquid energy converter

The hot liquid energy converter (HLEC) is quite similar to the boiling condenser and can be used in a large number of ignition engines with less modification in order to transform some part of the heat lost in the engine.

2.5.4 Bottom Rankine cycle

An evaluation of the potential WHR using a Rankine cycle has been analysed by Domingues, Santos and Costa (2013). Their analyses on a 2.8-litre VR6 SI engine for the different operating conditions of the vehicles and various working fluids have proved that a Rankine cycle has tremendous potential for ICE exhaust WHR. They also added that one should improve the condenser (evaporator) design and choose well-suited expender (turbine) devices that will provide a high evaporating pressure to get the maximum available WHR energy from the Rankine cycle.

2.5.5 Advantage of heat recovery methods compared to other technologies

A theoretical analysis, which is supported by the simulation results of a study by Weerasinghe, Stobart and Hounsham (2010) that compares the turbo-compounding method to the Rankine method, proved that the heat recovery and expansion method with water as the working fluid presented significant advantages. It saved up to 20% in fuel consumption compared to turbo-compounding. Arias, Shedd and Jester (2006) mentioned that achieving this kind of efficiency depends on both the effectiveness of heat recovery technology and the expansion used in the Rankine cycle.

The experiment done by Peng, Wang, He, Yang and Lu (2013) on a 1.4-litre light-duty petrol engine found that the maximum efficiency of an exhaust energy recovery (EER) system can reach up to 14%. However, for their research, the efficiency was considered to be between 1% and 10% as it is under general operating conditions for personal vehicle engines. According to the Peng, Wang He, Yang and Lu (2013), around 3.9% of fuel energy can be recovered using an EER thermal cycle, which is an estimated fuel saving of 17.5% for the engine tested and a 22.3% saving of the total engine efficiency according to the New European Driving Cycle (NEDC).

2.6. Limitations of waste heat recovery methods

Each technology has its positive and negative attributes. Therefore, one should think about the limitations that can be encountered in applying Rankine WHR methods, either on the equipment or on a specific cycle (Tian, Shu,Wei, Liang and Liu, 2012):

- Constraint on the available heat from the heat source
- Production of backpressures while recovering heat
- Safety and environmental effects of the working fluid
- Heat rejection from the engine

2.6.1 Bad effect of backpressure

The backpressure occurred due to the presence of the heat exchanger on the exhaust manifold, which is undesirable in ICEs. Therefore, the heat exchanger size and design will play a key role in backpressure limitation on the one hand, while, on the other hand, it should remain efficient so that it can extract as much heat from the hot exhaust gas as possible.

2.6.2 Choice of working fluids

Previous studies have shown that the working fluid is the major key in determining the efficiency of the thermodynamic cycle (Saidur et al., 2012). This means that the choice of working fluid strongly affects the efficiency of a cycle. Therefore, working fluids should be selected judiciously according to their specific properties.

Tian et al. (2012) mention some important physical and chemical characteristics that should be considered when choosing suitable working fluids. These characteristics include stability, non-fouling, non-corrosiveness, non-toxicity and non-flammability. According to Vaja and Gambarotta (2010) and Gu, Weng, Wang and Zheng (2009), organic fluids are better than water as a working fluid for their limited power and low temperature heat source. Organic fluids are also better because of their lower vaporisation temperature, which results in reduced evaporator irreversibility. Domingues et al. (2013) presented water as a better working fluid compared to an organic one because of its higher net output power, despite its high condensation temperature. Thus, it can be seen that not all the desirable requirements can be met in a particular cycle by using a specific working fluid (Chen, Goswami and Stefanakos, 2010). However, their two main criteria suggested for selecting which working fluid to operate in which cycle are based on the critical temperature and ξ value (slope of the saturation vapour curve on the T-s diagram).

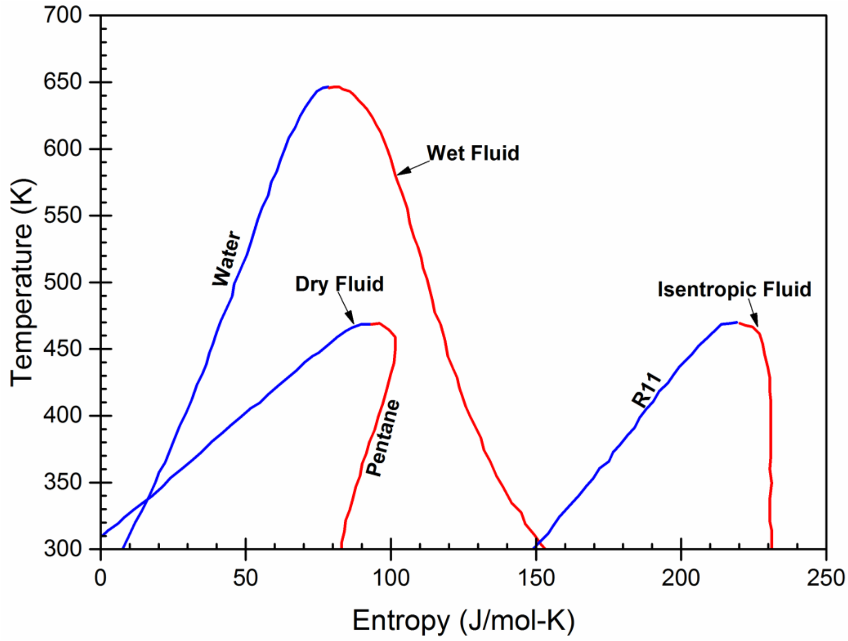


Figure 2.10: The three types of working fluid, adapted from Chen et al. (2010).

2.7. Engine exhaust gases' emissions and their environmental effects

Air pollution from the exhaust gases of automotive vehicles' engines has been one of the major concerns in scientific studies in recent years. This is not only due to the negative impact air pollution has on the environment, but also due to the fact that it has many adverse side effects on our daily lives. It has caused the deaths of thousands of people in developing countries, has necessitated billions to be injected into the medical sector, and has resulted in vast amounts of money to be lost in production each year (Arapatsakos, Karkanis and Strofylla, 2012).

In the current situation, as ICEs are still the fundamental sources of energy in the automotive industry. Legislation regulating emissions from vehicles has been implemented to control and limit the emission of regulated components such as carbon monoxide (CO), NO_x, hydrocarbons (HC) and particulate matter (PM) from the exhausts of motor vehicles (Czerwinski, Winkelmann, and Morin 2012). According to Boloy, Silveira, Tuna, Coronado and Antunes (2011), PM_{10} can easily be inhaled and settle in the lungs, which endangers human health. CO_2 is considered to be one of the main agents that cause greenhouse gas effects, while SO_2 causes acid rain. NO_x is considered to be the main cause of acidification in ecosystems.

That is one of the raisons the European Union (EU) commission was put into place. Their main objective is to periodically implement new emission standards regulations for new passenger cars and light commercial vehicles. Table 2.3 reflects the Euro 1, 2, 3, 4, 5 and 6 emission standards applied to vehicles of category M_1 with a reference mass less or equal to 2610 kg (DieselNet 2012). The standards for all categories (M_1, M_2, N_1 and N_2) of vehicles are shown in an extended table in Appendix M (European comission 2012).

Table 2.3: European Union emission standards for passenger cars (category M1[a])

Stage	Date	CO (g/km)	HC (g/km)	$HC+NO_x$ (g/km)	NO_x (g/km)	PM (g/km)	PN (#/km)
Compression ignition (diesel)							
Euro 1[b]	1992.07	2.72 (3.16)	–	0.97(1.13)	–	0.14(0.18)	–
Euro 2, IDI	1996.01	1	–	0.7	–	0.08	–
Euro 2, IDI	1996.01[c]	1	–	0.9	–	0.1	–
Euro 3	2000.01	0.64	–	0.56	0.5	0.05	–
Euro 4	2005.01	0.5	–	0.3	0.25	0.025	–
Euro 5a	2009.09[d]	0.5	–	0.23	0.18	0.005[e]	–
Euro 5b	2011.09[f]	0.5	–	0.23	0.18	0.005[e]	$6.0x10^{11}$
Euro 6	2014.09	0.5	–	0.17	0.8	0.005[e]	$6.0x10^{11}$
Positive ignition (gasoline)							
Euro 1[b]	1992.07	2.72 (3.16)	–	0.97(1.13)	–	–	–
Euro 2	1996.01	2.2	–	0.5	–	–	–
Euro 3	2000.01	2.3	0.2	–	0.15	–	–
Euro 4	2005.01	1	0.1	–	0.08	–	–
Euro 5	2009.09[d]	1	0.1[f]	–	0.06	0.005[g,h]	
Euro 6	2014.09	1	0.1[f]	–	0.06	0.005[g.h]	$6.0x10^{11}$[g,1]

[a] At the Euro 1-4 stages, passenger vehicles>2500kg were type approved as category N1, vehicles.
[b] Values in brackets are conformity of the production limits
[c] Until 30 September; after that date, DI engines must meet the IDI limits
[d] 2011.01 for all models
[e] 0.0045 g/km using the particulate Measurement Programme Procedure
[f] 2013.01 for all models
[g] And NMHC = 0.068 g/km[h] Applicable only to vehicles using DI engines
[i] 6.0×10^{12} L/km within first 3 years from Euro 6 effective dates
CO, carbon monoxide; DI, direct injection; HC, hydrocarbons; IDI, injection; NMHC, non-methane hydrocarbons; NO_x, nitrogen oxide ; PM, particulate matter ; PN, particle number.

Figures 2.10 and 2.11 provide an approximation of the exhaust emission composition of petrol and diesel engines. Petrol engines might also discharge SO_2 in addition to the regulated components mentioned above Volkswagen Group (2000).

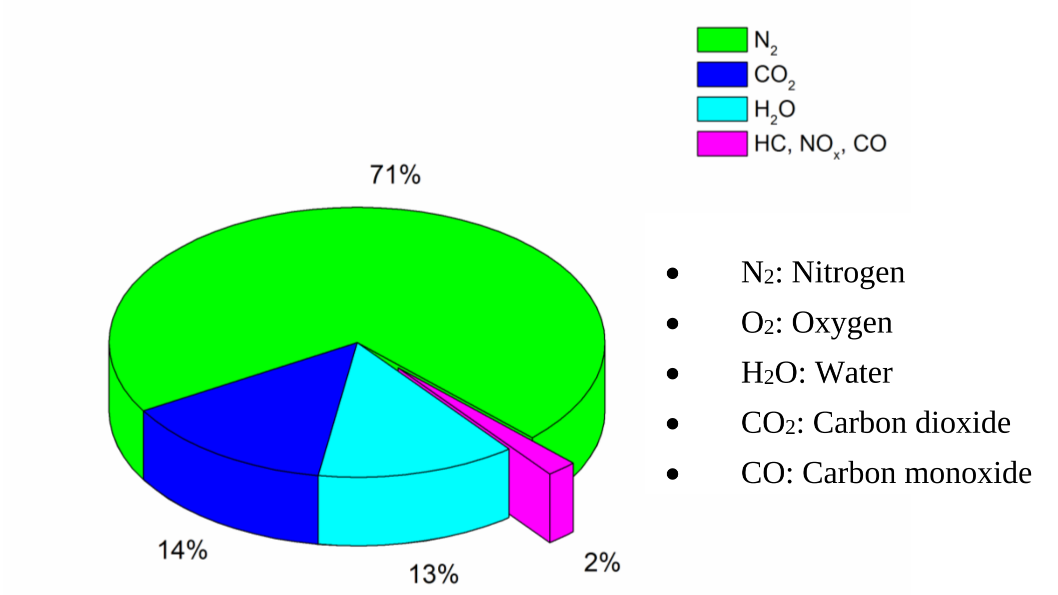


Figure 2.11: The composition of the exhaust emissions of petrol engines, adapted from Volkswagen Group (2000).

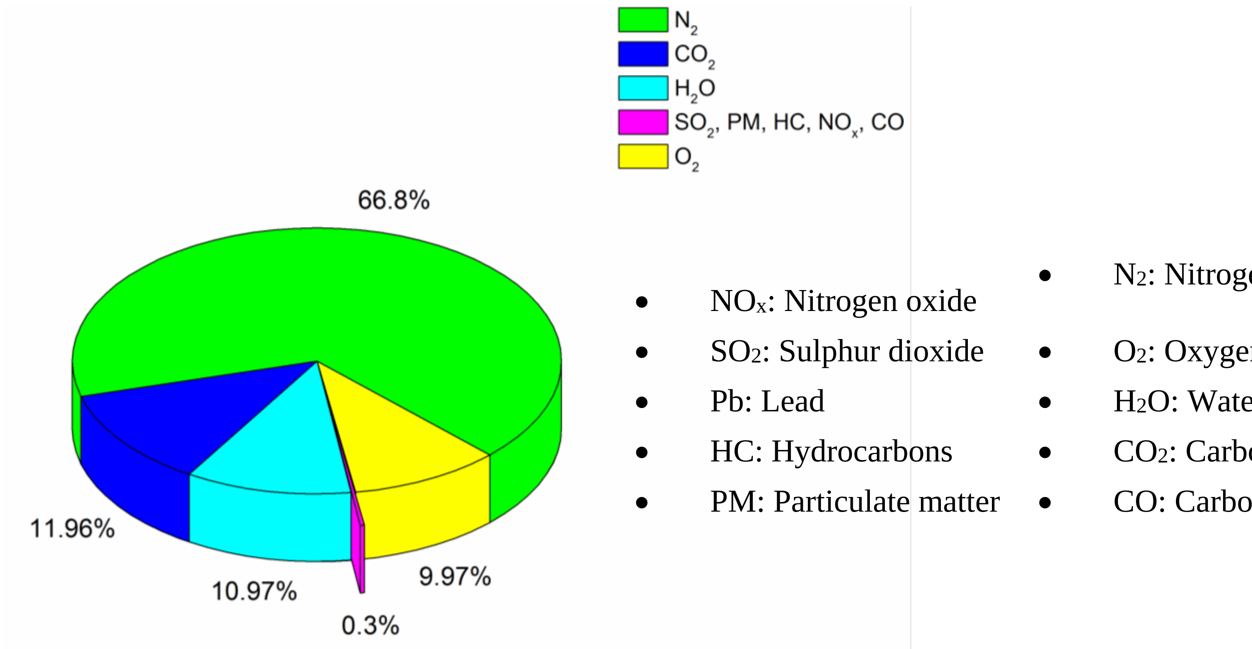


Figure 2.12: The composition of the exhaust emissions of diesel engines, adapted from Volkswagen Group (2000).

In this regard, the experiments conducted by Arapatsakos et al. (2012) on a four-stroke petrol engine have proven that CO_2, CO and HC can be reduced by reducing the exhaust gas temperature of the engine.

2.8. The economics of waste heat recovery

The EER method based on the Rankine cycle applied on ICEs does not only aim to improve the engine's thermal efficiency by reducing its fuel consumption. It also has a considerable impact on the environment by decreasing the emission of the various components of exhaust gas (CO_2, SO_2, NO_x and MP) that have a negative effect on the environment.

Peng et al. (2013) conducted a study on the cost and carbon emission payback times on vehicles that incorporate an EER system, and compared them to hybrid electrical vehicles (HEVs). The comparison was based on fuel price, variable fuel price and possible high mileage. The study showed that EER takes 10.1 years of cost payback and 1.9 years of carbon emission payback compared to 11.9 and 1.4 years for HEVs. This was evaluated in the UK market for light-duty motor vehicles by using the following equations:

$$CO_{2-cumulative} = CO_{2-embedded} + S_{cumulative} \times FC \times \rho_{fuel} \times \chi_c \times 44/12, \tag{2.18}$$

where:

$S_{cumulative}$ is the cumulative mileage
FC is the fuel consumption (litre per mile or L/mile)
ρ_{fuel} is the fuel density (kilogram per litre or kg/L)
$\chi_c$is the rate of carbon in the fuel
44 and 12 are the molecular weights of CO_2 and carbon (C)

$$Cost_{cumulative} = Cost_{embedded} + S_{cumulative} \times FC \times P_{fuel} + \sum_0^n (Ins + Tax + MOT + Service), \tag{2.19}$$

where:

P_{fuel} is the fuel price (pound per litre or pound/L)

Ins is insurance

MOT is the annual authority inspection of the Ministry of Roads and Transport

Jadhao, Thombare, Post-Graduate Student and Sangali (2013) categorised the benefit of WHR as follows:

Direct benefits are applied to fuel consumption and the process of cost reduction because it has a straight impact on the efficiency of the combustion process.

Indirect benefits are characterised by the following factors:

- Pollution reduction is possible by recovering waste heat from exhaust gas as this reduces the level of pollution in the environment.
- The downsizing of equipment is possible due to reduced fuel consumption, which means reduced flue gas production.
- A reduction in auxiliary energy consumption and downsizing of some equipment leads to a lower fuel consumption.

2.9. Automobile market investigation

According to the October 2015 New Vehicle Sales Statistics of the National Association of Automobile Manufacturers of South Africa (NAAMSA), Volkswagen (VW) South Africa still maintains the best-seller position in the new passenger vehicle market (NAAMSA, 2015).

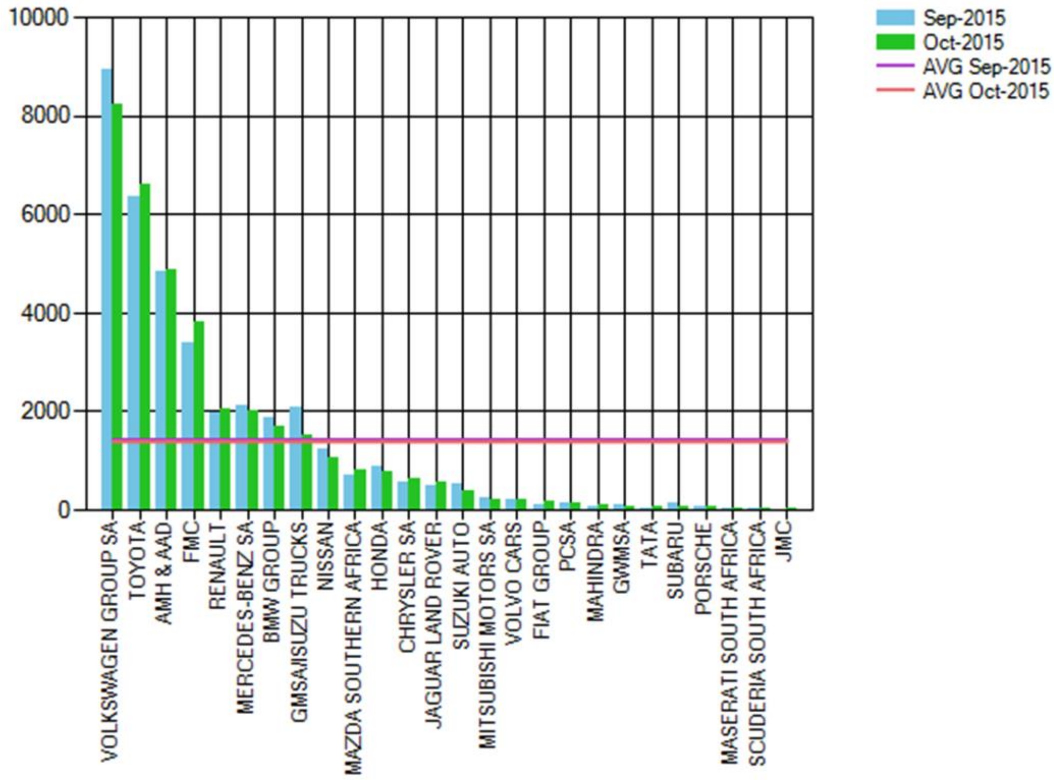


Figure 2.13: New vehicle sales statistics for September and October (NAAMSA, 2015).

It has been reported that VW South Africa has sold 8 246 units, with its highest monthly sales in 2012 and highest ever sales in August 2007. The record figure was bolstered by the Polo Vivo (3 714 units), which achieved its best-ever monthly sales since its launch in March 2010. The Polo Vivo was also the best-selling model in the total new vehicle market in August. The Polo was the second best-selling model in the new passenger vehicle market with 2 355 units (NAAMSA, 2015).

2.10. Previous research

Vaja and Gambarotta (2010) studied three different thermodynamic WHR configurations. In the first case, they studied exhaust gases as the only thermal source; in the second case, they studied exhaust gases and coolant. The third case entailed a regenerated cycle that was incorporated on a stationary ICE. To conduct their theoretical analysis, three different organic working fluids (R11, benzene and R134a) were used in the three configurations to convert waste heat to useful work. In conclusion, their work has proven that the total efficiency of an ICE can be increased by 12%.

A mathematical model and experimental setup conducted by Talom and Beyene (2009) showed that exhaust gas heat recovered from a 2.8-litre V6 ICE was used instead of a burner to run a modified 10.55 kW, three-ton absorption chiller matched to it. They concluded that their system was thermodynamically feasible and could be applied to the air conditioning and refrigeration of transport vehicles.

A steady state experiment on exhaust gas energy balance and exergy analysis conducted by He, Zhang, Zeng, Gao (2011), which has shown that waste heat recovery of ICEs can be improved by implementing a combined thermodynamic cycle consisting of two different cycles to recover waste heat from the exhaust gas and lubricant.

Endo, Kawajiri, Kojima, Takahashi, Baba, Ibaraki, Takahashi and Shinohara (2007) studied a practical approach to exergy maximisation from two sources (exhaust gas and cooling loss) in automotive engines applied on a 2.0-litre Honda Stream engine. In their heat recovery management system, the exhaust port was replaced by an innovative evaporation device to recover a great amount of high-quality energy. A three-layer water jacket configuration was built around the combustion chamber to allow both cooling and heat recovery at the same time. In the end, their experimental results showed an increase of 3.8% in thermal efficiency from 28.9% to 32.7% (a 13.2% relative increase in the control vehicle) at a constant speed of 100 km/h.

An experimental analysis conducted by Wang, Zhang, Zhang, Shu and Peng (2013) on exhaust waste energy from a 1.3-litre CA4GA1 light-duty petrol engine connected to a multi-coil helical heat exchanger, stated that the fuel conversion efficiency of the engine could be improved by up to 14%, but only in the range of 3% to 8% under the vehicle’s accustomed operating conditions. They also highlighted the importance of controlling the working fluid flow rate if the engine’s operating conditions varied in order to achieve the desired overheat and steam pressure, and better heat transfer efficiency.

Domingues, Santos, and Costa (2013) evaluated two different methodologies (Rankine cycle thermodynamic and heat exchanger models) to recover the potential waste energy contained in the exhaust gas of a 2.8-litre VR6 SI vehicle engine. The mass flow rate and temperature of the exhaust gases’ measurements were recorded by means of a dynamometer chassis coupled to the engine running at different steady-state working conditions. The results proved that a Rankine cycle as a potential WHR method to be used after the simulations on ideal and shell-and-tube heat exchangers has shown an increase in the thermal and mechanical efficiency of 1.4% to 3.52% and 10.16% to 15.95% for the first case, and 0.85% to 1.2% and 2.64% to 6.96% for the second case. In these cases, water, R123 and R245fa were used as working fluids at 2MPa as the evaporating pressure.

Ringler, Seifert, Guyotot and Hübner (2009) analysed different working fluids and selected the most suitable one. They then developed a simulation model to evaluate different WHR systems from ICEs using a Rankine cycle and they ran test bench experiments on a four-cylinder ICE at various speeds. Water was chosen as the suitable heat recovery working fluid due to its high evaporation enthalpy. From their reviewed WHR method, Ringler et al. (2009) highlighted only two basic single-loop systems: exhaust gas only and exhaust gas plus coolant. They found that an additional output power of 0.7 kW to 2 kW represents 10% increases in engine performance.

2.11. Conclusions

Some 40% of the energy increase from 2010 to 2040 will mostly come from the transport sector, predominantly in the road transport sector, where the number of vehicles is expected to almost double from 800 million to more than 1.6 billion by 2040. Although the number of conventional ICE vehicles will decrease by 2040 due to the growth of more energy-efficient vehicles, hybrid and electric motor vehicles will still present some challenges because of their high cost and functional technology limitations, such as charging time and distance travelled in the case of electric vehicles. Therefore, improving diesel and petrol vehicles is still a focus of thermodynamics research.

Despite the type of ICE used, studies have shown that only 25% of fuel energy input in a motor vehicle is converted into useful shaft work and the other 75% is wasted through different mechanisms. Some 40% of energy is lost through exhaust gas, 30% through cooling and 5% through friction, which makes the ICEs inefficient. Thus, technology to transform some part of waste heat energy into useful energy is implemented. From the two most promising thermal energy recovery methods (thermoelectricity and Rankine cycles), the

Rankine system that incorporates a turbine, heat exchanger, condenser and pump with an appropriate working fluid was chosen in this study.

In previous studies of WHR from exhaust gases, the WHR system has been placed along the exhaust flow pipe where the temperature differs from the temperature behind the exhaust valves. This means that an important fraction of energy is lost to the environment from exhaust gases. In this study, the heat exchanger will be fitted on the exhaust manifold just behind the exhaust valves to recover a large amount of waste energy from exhaust gases, and water will be used as the working fluid due to its potential to recover heat at a high temperature.

Chapter 3: Methodology

3.1. The operating cycles of internal combustion engines

There are several different types of engine thermodynamic operating cycles depending on the type of engine. This study will focus more on the analysis of an air standard Otto cycle than on the CI and combined (CI dual) cycle.

3.1.1 Basic engine cycles

Generally, reciprocating ICEs (SI or CI) operate on either a four-stroke or a two-stroke cycle. These are quite large standard basic cycles of all engines, but differ slightly according to the engine's design.

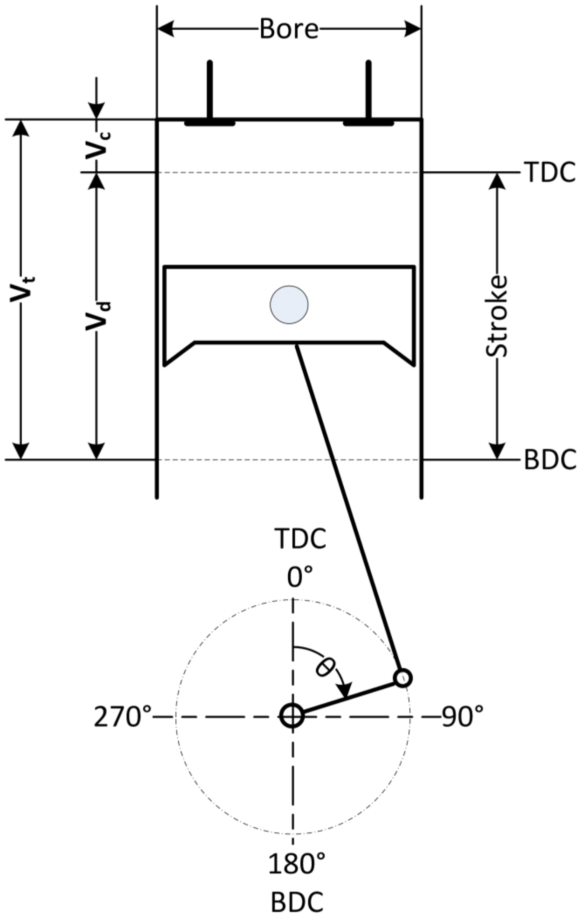


Figure 3.1: Reciprocating IC engines' basic cycle, adapted from Heywood (1988).

The alternative motion of the piston within the cylinder results in the rotary motion of the crankshaft, which produces mechanical work. During this process, the piston rests at the top-dead-centre (TDC) crankshaft position, where the volume corresponds to its minimum value (Vc), and the bottom-dead-centre (BDC) crankshaft position corresponds to the maximum volume (V_t). The piston's swept-out volume (V_d) is the difference between the maximum or total volume and the minimum or clearance volume.

A piston within a cylinder in a four-stroke ICE accomplishes the following four processes:

1. **First stroke:** An intake stroke is characterised by the displacement of the piston from the TDC to the BDC, resulting in the inlet of the gas mixture into the cylinder with the exhaust valves closed and the intake valve opening shortly before the stroke starts to induct a large quantity of the mixture, then closing at the end of its cycle.

2. **Second stroke:** In a compression stroke, the piston travels from the BDC to the TDC with both valves closed. This results in the compression of the trapped mixture to the minimum volume, which increases the pressure and temperature of the gas within the cylinder. At the end of compression stroke, combustion is initiated over a very short finite period, increasing the cylinder pressure more rapidly.

3. **Third stroke:** A power stroke or expansion stroke is characterised by a sudden push down of the piston from its TDC to its BDC due to the high pressure and temperature produced during the combustion of the compressed mixture, which results in crankshaft work. When the piston reaches its BDC, the exhaust valve is opened to start the exhaust process, but the intake valve still closes. Therefore, the pressure and temperature within the cylinder drops near that of the exhaust pressure and temperature.

4. **Fourth stroke:** An exhaust stroke forces the remaining burnt gases into the cylinder due to the displacement of the piston as it travels from the BDC to the TDC point. The remaining burnt gases are also forced into the cylinder because the pressure within the cylinder might be substantially higher than that of the exhaust valve. As the piston approaches the top centre, the inlet valve opens. Just after the top centre, the exhaust valve closes and the cycle starts again.

3.1.2 The air standard Otto cycle

The processes undergone during a cycle in the cylinder of an ICE are very complex due to the fact that the intake of the air mixture and fuel is mixed into the remaining exhaust gas again from the previous cycle. It is then compressed and ignited to produce work. The composition of the mixture changes, which makes the analysis of a cycle more complex than before. Therefore, a real cycle will approximately be assimilated to an ideal air standard cycle that makes the engine cycle analysis much more controllable. The following assumptions indicate the difference between them (Pulkrabek, 2003):

- The gas mixture in the cylinder is treated as air for the entire cycle and the property values of air are used in the analysis. This is a good approximation during the first half of the cycle when most of the gas in the cylinder is air with only up to about 7% fuel vapour. Even in the second half of the cycle, when the gas composition is mostly CO_2, water (H_20) and nitrogen (N_2), using the air properties does not create large errors in the analysis. Air will be treated as an ideal gas with constant specific heats.

- The real open cycle is changed into a closed cycle by assuming that the exhaust gases are fed back into the intake system. This works with ideal air standard cycles, as air is the intake and exhaust gas. Closing the cycle simplifies the analysis.

- The combustion process is replaced with a heat addition term Q_{in} of equal energy value. Air alone cannot combust.

- The open exhaust process, which carries a large amount of enthalpy out of the system, is replaced with a closed-system heat rejection process Q_{out} of equal energy.

- The actual engine processes are approximated with ideal processes:

  - The almost-constant pressure intake and exhaust strokes are assumed to be constant pressure. At wide-open throttle (WOT), the intake stroke is assumed to be at a pressure P_0 of 1 atmosphere. At partially closed throttle or when supercharged, the inlet pressure will be some constant value other than 1 atmosphere. The exhaust stroke pressure is assumed to be constant at 1 atmosphere.

- Compression and expansion strokes are approximated by isentropic processes. To be truly isentropic would require these strokes to be reversible and adiabatic. There is some friction between the piston and cylinder walls, but because the surfaces are highly polished and lubricated, this friction is kept to a minimum and the processes are close to frictionless and reversible. If this were not true, automotive vehicle engines would wear out long before the 241 400 to 321 860 km they now last if they are properly maintained. There is also fluid friction because of the gas motion within the cylinders during these strokes. This too is minimal. Heat transfer for any one stroke will be negligible due to the very short time involved for that single process. Thus, an almost reversible and almost adiabatic process can quite accurately be approximated with an isentropic process.

- The combustion process is idealised by a constant-volume process (SI cycle), a constant-pressure process (CI cycle) or a combination of both (CI dual cycle).

- Exhaust blowdown is approximated by a constant-volume process.

- All processes are considered reversible.

Taking air as an ideal gas, the following equations can be applicable for an air standard cycle:

$$Pv = RT \tag{3.1}$$

$$PV = mRT \tag{3.2}$$

$$P = \rho RT \tag{3.3}$$

$$dh = C_p dT \tag{3.4}$$

$$du = C_v dT \tag{3.5}$$

$$Pv^k = constant \text{ (isentropic process)} \tag{3.6}$$

$$Tv^{k-1} = constant \text{ (isentropic process)} \tag{3.7}$$

$$TP^{k-1/k} = constant \text{ (isentropic process)} \tag{3.8}$$

3.1.3 Thermodynamic analysis of the air standard Otto cycle

From a thermodynamic point of view, the stated points of each process can be determined as follows:

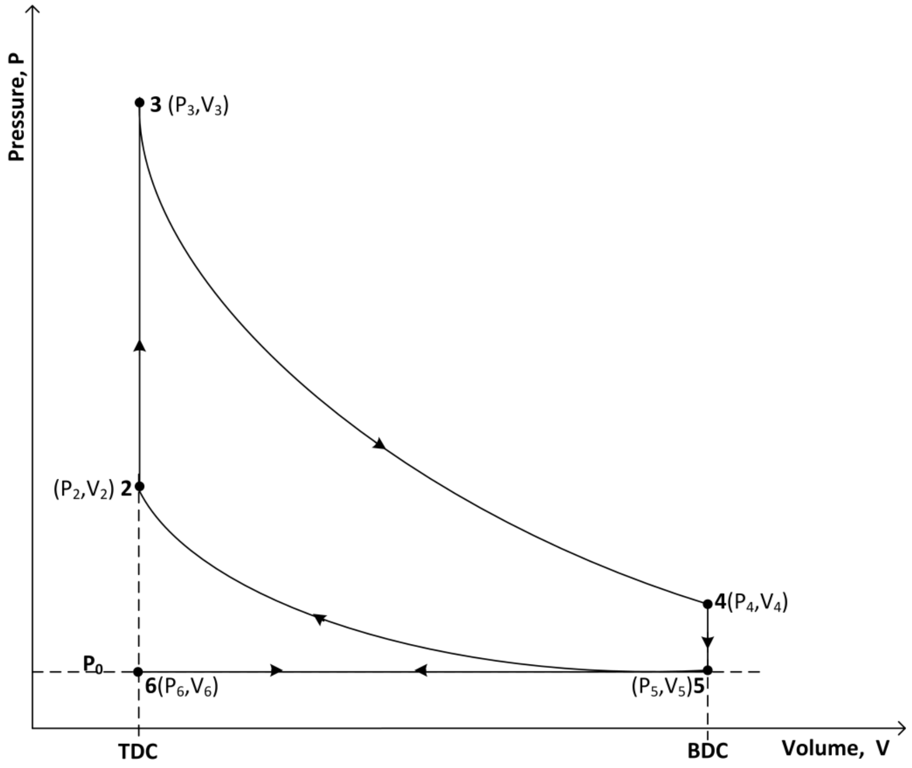


Figure 3.2: The pressure volume of an ideal standard Otto cycle, adapted from Heywood (1988).

- **Process 6 to 1** is an isobaric process with the intake of air at atmospheric pressure. The intake valve is open and the exhaust valve is closed.

$$P_6 = P_1 = P_o \tag{3.9}$$

$$w_{6-1} = \int_6^1 P dv = P_0(v_1 - v_6) \tag{3.10}$$

- **Process 1 to 2** is an isentropic and adiabatic process. The mixture is compressed at $P = constant$, with $P_2 > P_1$, and both valves are closed.

$$T_2 = T_1 \left(\frac{v_1}{v_2}\right)^{k-1} = T_1 \left(\frac{V_1}{V_2}\right)^{k-1} = T_1 (r_c)^{k-1} \tag{3.11}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k = P_1\left(\frac{V_1}{V_2}\right)^k = P_1(r_c)^k \tag{3.12}$$

$$q_{1-2} = 0 \tag{3.13}$$

$$w_{1-2} = \left(\frac{P_2v_2 - P_1v_1}{1-k}\right) = \frac{R(T_2 - T_1)}{1-k} = (u_1 - u_1) = c_v(T_1 - T_2) \tag{3.14}$$

- **Process 2 to 3** is an isochoric heat input process (combustion of the mixture) with both valves closed.

$$v_3 = v_2 = v_{TDC} \tag{3.15}$$

$$\mathrm{w}_{2-3} = 0 \tag{3.16}$$

$$Q_{2-3} = Q_{in} = m_f Q_{HV}\eta_c = m_m c_v(T_3 - T_2) = \left(m_a + m_f\right)c_v(T_3 - T_2) \tag{3.17}$$

$$q_{2-3} = q_{in} = c_v(T_3 - T_2) = (u_3 - u_2) \tag{3.18}$$

$$T_3 = T_{max} \tag{3.19}$$

$$P_3 = P_{max} \tag{3.20}$$

- **Process 3 to 4** is an isentropic process or power expansion stroke (expansion of the burnt mixture) and all the valves are closed.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1}{r_c}\right)^{k-1} \tag{3.21}$$

$$P_4 = P_3\left(\frac{v_3}{v_4}\right)^{k} = P_3\left(\frac{V_3}{V_4}\right)^{k} = P_3\left(\frac{1}{r_c}\right)^{k} \tag{3.22}$$

$$q_{3-4} = 0 \tag{3.23}$$

$$w_{3-4} = \left(\frac{P_4v_4 - P_3v_3}{1-k}\right) = \frac{R(T_4 - T_3)}{1-k} = (u_4 - u_3) = c_v(T_4 - T_3) \tag{3.24}$$

- **Process 4 to 5** is an isochoric process with heat rejection (exhaust blowdown) and the exhaust valve open and the intake valve closed.

$$v_5 = v_4 = v_1 = v_{BDC} \tag{3.25}$$

$$w_{4-5} = 0 \tag{3.26}$$

$$Q_{4-5} = Q_{out} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \tag{3.27}$$

$$q_{4-5} = q_{out} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4) \tag{3.28}$$

- **Process 5 to 6** is an isobaric process at P_0 with the exhaust valve open and the intake valve closed.

$$P_6 = P_5 = P_o \tag{3.29}$$

$$w_{5-6} = \int_5^6 Pdv = P_0(v_6 - v_5) = P_0(v_6 - v_1) \tag{3.30}$$

The thermal efficiency is given by the following equation:

$$\eta_{OTTO} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} \tag{3.31}$$

After substitution and rearranging, the Otto cycle's efficiency expressed in terms of the compression ratio, $v_1/v_2 = r_c$ is given by the following equation:

$$\eta_{OTTO} = 1 - \left(\frac{1}{r_c}\right)^{k-1} \tag{3.32}$$

3.2. Exhaust flow process analysis

After the production of crankshaft work towards the high pressure of the gases produced during combustion, the vacuum of the burnt gases occurs in two steps: the exhaust blowdown and the exhaust stroke. The exiting flow throughout the exhaust pipe is a non-steady-state, pulsing flow, which is usually assumed to be a pseudo steady-state flow. A brief analysis of the two steps of the exhaust flow is provided below.

3.2.1 The blowdown step

This operation occurs when the exhaust valve begins to open as the power stroke ends, which is around 60° to 40° before the BDC. At this moment, the pressure within the cylinder is around 4 to 5 atmospheres (or 405.3 kPa to 506.625 kPa), while the temperature is upwards of 1 000 K or 726.85 °C, while the pressure in the exhaust system is around 1 atmosphere or 101.325 kPa (Pulkrabek, 2003).

The usual equation used to determine the temperature in the exhaust system is the ideal gas isentropic expansion equation that relates the temperature to the pressure.

$$T_{ex} = T_{EVO} \left(\frac{P_{ex}}{P_{EVO}}\right)^{\frac{k-1}{k}}, \tag{3.33}$$

where T_{ex} and P_{ex} are respectively the exhaust temperature and pressure, and T_{EVO} and P_{EVO} are the temperature and pressure in the cylinder at the valve openings.

3.2.2 The exhaust stroke step and gas temperature

During this process, the piston travels from the BDC to the TDC and the exhaust process of the burnt gases continues as the pressure inside the cylinder is still slightly higher than the pressure of 1 atmosphere in the exhaust system. This process can best be assumed to be an isobaric process with the properties of the gas remaining constant at the conditions of point 5 in Figure 3.2, which is not really an ideal Otto cycle.

During the exhaust process, the mass flow rate and the temperature of the burnt mixture vary considerably due to different factors like the variation of the volume during the blowdown process, a restriction of the exhaust gas flow throughout the exhaust valve and the lift of the exhaust gas, which varies with time. The temperature of the exhaust gas decreases during the whole exhaust process because of the heat gas that is lost during this operation.

The exhaust gas temperature in an SI engine has been estimated to vary from 300 °C to 400 °C during idling and to about 900 °C at high-power operation. The most common range is 400 °C to 600 °C (Heywood, 1988).

According to Van Basshuysen (2004), the exhaust gas temperatures were found to be 850 °C in petrol engines and 650 °C in diesel engines. In the same year, Pulkrabek (2004) noted that the temperature of the exhaust gases in typical SI engines ranges from 400 °C to 600 °C, decreasing to about 300 °C to 400 °C in idling conditions and increasing again to 900 °C at maximum power load. Temperatures range from 200 °C to 500 °C in typical CI engines.

3.3. Type of engine chosen

For the purpose of this research, the chosen thermal source of waste heat energy is the engine of a VW Citi Golf 1.3i model. An overview of the technical specifications of all VW Citi Golf vehicles used in South Africa is given in Table 3.1. Refer to the Appendix for more information.

3.3.1 Volkswagen Citi Golf 1.3i (2000 model)

This vehicle is a five-seat hatchback with an engine of four in-line arranged cylinders, a carburettor and a single overhead camshaft (SOHC), all placed in the front of the vehicle. The mechanical and other specifications are mentioned in Table 3.1.

Table 3.1: Volkswagen Citi Golf 1.3i specifications

Make	**Volkswagen**
Model	Citi Golf 1.3i
Year	2000
Car category	Small or economy cars
Car engine	1.3 litres, 1 272 cubic centimetres (cc), four cylinders
Car valves per cylinder	Two
Car maximum power	65.00 PS (48 kW or 64 HP) at 5 600 revolutions per minute
Car maximum torque	100 Nm (10 kgf-m or 73 ft.lbs) at 3 000 revolutions per minute
Car brake mean effective pressure (BMEP)	143.26 psi (987.74 kPa or 9.88 bar)
Car compression ratio	9.00:1
Car bore stroke	75 mm x 72 mm (2.9528 x 2.8346 inches)
Car top speed	152 km/h (94.45 mph)
Car fuel	Petrol
Car transmission	Manual, five-speed
Car power per weight	0.0980 PS/kg
0 to 100 km/h 0 to 62 mph	13.72 seconds
Car drive	Front wheel drive (FWD)
Car seats	Five
Car doors	Five
Car weight	794 kg (1750.47 pounds)
Car total length	3815.00 mm (150.2362 inches)
Car total width	1610.00 mm (63.4252 inches)
Car total height	1410.00 mm (55.4724 inches)
Car wheelbase	2399.00 mm (94.4488 inches)
Car front brakes type	Disks
Car rear brakes type	Drums
Car cargo space	260 litres (68.66 gallons)
Car fuel tank capacity	50.00 litres (13.21 gallons)

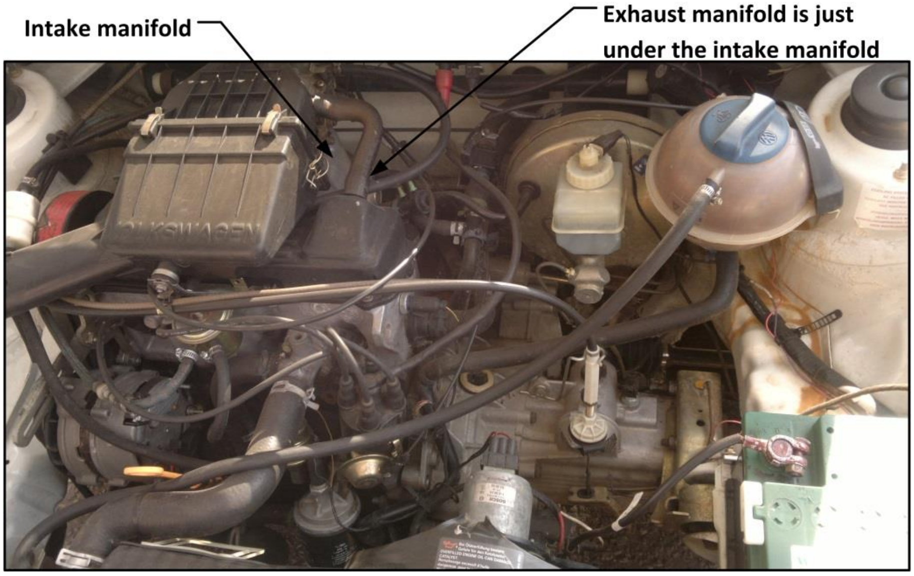


Figure 3.3: Engine of a VW Citi Golf 1.3i.

Figure 3.3 above shows a VW Citi Golf 1.3i with the exhaust manifold just under the intake manifold. There is enough space around the engine where the EWHR can be fitted. More details will be given below.

3.4. Formulation of energy contained in fuel

In order to determine the amount of thermal energy that is wasted through the exhaust pipe by the fuel type, the amount of chemical potential energy carried in each fuel should first be determined. In the study of Pulkrabek (2003), the chemical potential energy that is available in a fuel is expressed as the product of the fuel mass admitted (m_f) (kg) (with one barrel = 117 kg for petrol and 134.1 kg for diesel) (Hofstrand, 2008)) and the net fuel calorific value (H_u) (kJ/kg), which is 43 448 kJ/kg for petrol and 42 791 kJ/kg for diesel (GREET, 2011). This is expressed as follows:

$$E_f = m_f \times H_u \tag{3.34}$$

Hence, from Equation 3.34, with n in Equation 3.36 and 0.4 in Equation 3.37 representing the number of days per year and 40% of the total admitted energy wasted in an ICE through the exhaust gas, the total available fuel energy consumed per day and per year and the total exhaust waste energy per year can be derived by the following equations:

$$E_{f/d} = m_{f/d} \times H_u \quad (3.35)$$

$$E_{f/y} = n \times E_{f/d} \quad (3.36)$$

$$E_{f/w} = 0.4 \times E_{f/y} \quad (3.37)$$

3.5. The thermodynamic properties of the exhaust gas composition

To evaluate the amount of energy available in the exhaust mixture, it is important to know the stoichiometric composition of the exhaust gas mixture, and to determine the specific characteristics of each component contained in it, as well as the total exhaust mass flow rate of the mixture after the engine valves. According to the Volkswagen Group (2000), the exhaust gas of petrol engines is composed of approximately 71% N_2, 14% CO_2, 13% H_2O and 1% to 2% HC, NO_x and CO.

For the simplicity of calculations, the following assumptions have been considered:

- The percentage of HC, NO_x and CO would be neglected
- The exhaust gas would behave like an ideal gas mixture
- The exhaust gas pressure would vary between 300 kPa and 400 kPa
- The exhaust gas temperature would range from 400 °C to 600 °C

According to the Gibbs-Dalton law, the following assumptions can be made:

A gas mixture can be considered as the ideal gas, thus it should obey the equation of the state of ideal gas, which is the following:

$$pV = mRT \quad (3.38)$$

with:

P = pressure (N/m^2)

V = volume (m^3)

m = mass (kg)

R = specific gas constant (kJ/kgK)

T = absolute or thermodynamic temperature (k)

The total gas mixture pressure is equal to the sum of the partial pressures exerted by each element individually as each of them occupies the total volume of the mixture at the same temperature.

The internal energy, enthalpy and entropy of the mixture are equal to the sums of the internal energy, enthalpy and entropy of the various components respectively, as each occupies the whole volume at the mixture temperature.

3.5.1 Molar fraction

$$x_i = \frac{n_i}{n}, \tag{3.39}$$

where n_i is the mole number of each ideal gas contained in the mixture, and n the total mole number of the mixture. Thus, the molar fractions of each component are as follows:

$$x_{N_2} = \frac{n_{N_2}}{n} \tag{3.40}$$

$$x_{CO_2} = \frac{n_{CO_2}}{n} \tag{3.41}$$

$$x_{H_2O} = \frac{n_{H_2O}}{n} \tag{3.42}$$

Table 3.2: Values of molar fractions

Molar fraction	N_2	H_2O	CO_2
x_i	0.71	0.13	0.14

3.5.2 Constant pressure-specific heat of the exhaust mixture

The constant pressure-specific heat of a mixture, $C_{p,mix}$, is a function of temperature and is defined as the sum of the individual ideal gas constant pressure-specific heat, $C_{p,m}$, multiplied by the molar fraction, x_i, of each ideal gas contained in the mixture. Therefore, the following expression can be used to determine the constant pressure-specific heat:

$$C_{p0} = C_0 + C_1\theta + C_2\theta^2 + C_3\theta^3 \tag{3.43}$$

with: $\theta = \frac{T_{exh}}{1\,000}$, where the constants C_0, C_1,C_2 and C_3 can respectively be read in Table 3.3 (Borgnakke and Sonntag, 2013).

Table 3.3: Constant pressure-specific heat of various ideal gases, values of constants C_0, C_1, C_2 and C_3

Gas	Formula	C_0	C_1	C_2	C_3
Nitrogen	N_2	1.11	-0.48	0.96	-0.42
Steam	H_2O	1.79	0.107	0.586	-0.20
Carbon dioxide	CO_2	0.45	1.67	-1.27	0.39

- C_p of N_2

$$C_{pN_2} = C_{0N_2} + C_{1N_2}\theta + C_{2N_2}\theta^2 + C_{3N_2}\theta^3 \tag{3.44}$$

- C_p of CO_2

$$C_{pCO_2} = C_{0CO_2} + C_{1CO_2}\theta + C_{2CO_2}\theta^2 + C_{3CO_2}\theta^3 \tag{3.44}$$

- C_p of H_2O

$$C_{pH_2O} = C_{0H_2O} + C_{1H_2O}\theta + C_{2H_2O}\theta^2 + C_{3H_2O}\theta^3 \tag{3.45}$$

- C_p of the exhaust gas mixture

$$C_{p,mix} = \sum x_i \left(C_{p,m}\right)_i \tag{3.46}$$

$$C_{p,mix} = x_{N_2} \times C_{pN_2} + x_{CO_2} \times C_{pCO_2} + x_{H_2O} \times C_{pH_2O} \tag{3.47}$$

3.5.3 Partial pressure of the mixture gas component

The partial pressure of each exhaust gas component can be defined as the multiplication of each component's mole fraction x_i and the total pressure of the mixture gas P_{exh}.

$$P_{N_2} = x_{N_2} \times P_{exh} \tag{3.48}$$

$$P_{CO_2} = x_{CO_2} \times P_{exh} \tag{3.49}$$

$$P_{H_2O} = x_{H_2O} \times P_{exh} \tag{3.50}$$

Therefore, the general form of the partial pressure is as follows:

$$P_i = x_i \times P_{exh} \tag{3.51}$$

3.5.4 Density of the exhaust mixture

The relationship between the density ρ and the ideal gas equation is given by the following:

$$p = \rho RT, \tag{3.52}$$

where the density is proportional to the mass (m) and inversely proportional to the volume (V):

$$\rho = \frac{m}{V} \tag{3.53}$$

From the law of ideal gas, the following expression can be written:

$$pV = mRT = \frac{m}{m_w} m_w RT \tag{3.54}$$

Therefore, the following equations apply to the separate gas mixture components:

$$pV_{N_2} = m_{N_2} R_{N_2} T \tag{3.55}$$

$$pV_{CO_2} = m_{CO_2} R_{CO_2} T \tag{3.56}$$

$$pV_{H_2O} = m_{H_2O} R_{H_2O} T \tag{3.57}$$

with:

$$m_{mix} = m_{N_2} + m_{CO_2} + m_{H_2O} + \cdots = \sum_{i=1}^{n} m_i \tag{3.58}$$

$$V_{mix} = V_{N_2} + V_{CO_2} + V_{H_2O} + \cdots = \sum_{i=1}^{n} V_i \tag{3.59}$$

Thus, the density of the gas mixture is determined with the following equation:

$$\rho_{mix} = \frac{\rho_1 V_1 + \rho_2 V_2 + \rho_3 V_3 + \cdots + \rho_n V_n}{V_1 + V_2 + V_3 + \cdots V_n} \tag{3.60}$$

where $x_i = \frac{V_i}{\sum_{i=1}^{n} V_i}$, the mixture density, is given by the following equation:

$$\rho_{mix} = x_1\rho_1 + x_2\rho_2 + x_3\rho_3 + \cdots + x_n\rho_n \tag{3.61}$$

3.5.5 Volumetric and mass flow rate of the gas mixture

Knowing the total volume, V_{Tot}, and the flywheel rotation speed, R_{pm}, at maximum power, which are given in Table 3.1, and the gas mixture density, the volumetric and mass flow rates can be calculated as follows:

$$\dot{V}_{mix} = \frac{V_{Tot} \times Rpm}{60}, \tag{3.62}$$

where $\dot{V}_{mix}$ is the volume flow rate in m^3/s.

Hence,

$$\dot{m}_{mix} = \dot{V}_{mix} \times \rho_{mix}, \tag{3.63}$$

with $\dot{m}_{mix}$ being the mass flow rate in kg/s.

With all these variables known, one can determine the volume and mass flow rate of the exhaust gas per cylinder.

3.6. Thermal energy characteristics in exhaust gases

3.6.1 Determination of the exhaust mixture's available energy

The amount of energy contained in the exhaust gas mixture behind the engine valves that can be reutilised as a source of energy intake in the bottom Rankine cycle to enhance the thermal efficiency can now be determined.

Since the constant pressure-specific heat, exhaust mass flow rate and average temperature of the exhaust gas mixture, $T_{exh} = 500\ °C$, are known, and the ambient temperature is assumed to be $T_{amb} = 25°C$, the available heat in the exhaust gas can be calculated as follows:

$$\dot{Q}_{av} = \dot{m}_{mix} \times C_{p,mix}(T_{exh} - T_{amb}), \tag{3.64}$$

where $\dot{Q}_{av}$ is in kW, and T_{exh} and T_{amb} in K.

3.6.2 Calculation of the exhaust gas properties

The unknown thermodynamic properties of the exhaust mixture have been calculated using the known properties given in Table 3.4.

Table 3.4: Known inputs

Engine total volume (m^3)	0.0013
Flywheel speed rotation (rpm)	From 3 300 to 5 600
Exhaust gas pressure (kPa)	From 300 to 400
Exhaust gas temperature (°C)	From 400 to 600

All equations in Section 3.6 were solved through a code written in Engineering Equation Solver (EES) V9.486-3D.

3.7. Rankine thermal recovery system

3.7.1 Exhaust waste heat recovery model based on a Rankine cycle model

To recover heat from the hot exhaust gas and transform it into useful energy, a smart thermodynamic cycle incorporates a helical coil heat exchanger, turbine, condenser and pump. Water was used as the working fluid in this study.

Figure 3.4 shows a schematic design setup of this study’s thermodynamic cycle, which is normally a Rankine cycle (heat engine with a vapour power cycle).

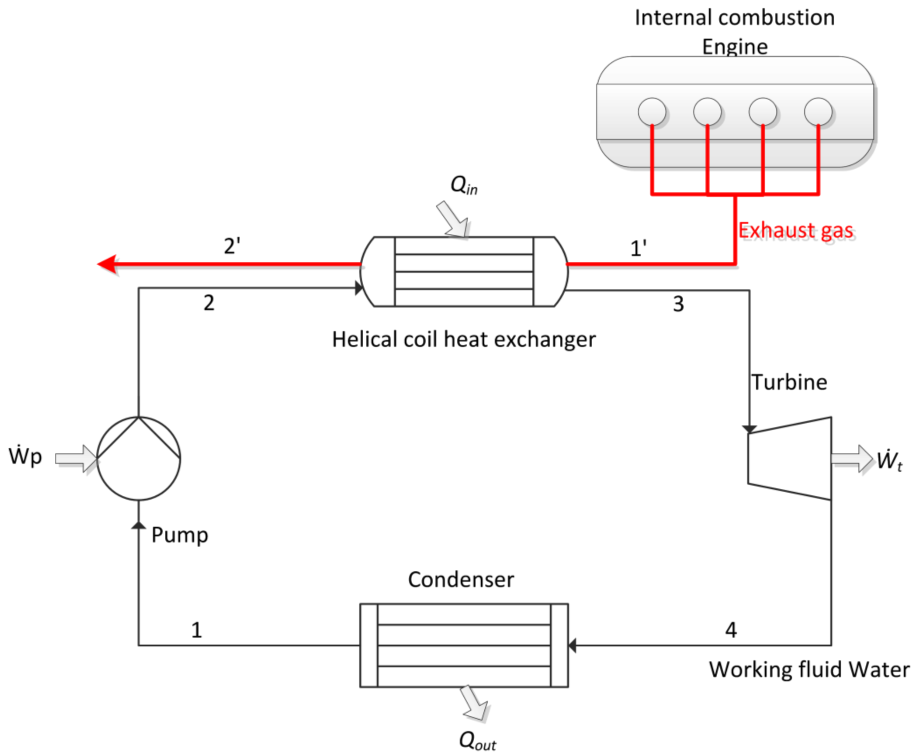


Figure 3.4: The WHR cycle.

Taking each device separately as a control volume and water as the working fluid, the following thermodynamic analysis of the cycle's four processes has been conducted:

- State 1 to 2: The pump has a control volume and the compression process is isentropic, while the external work (power) is applied to the pump.
- State 2 to 3: The heat exchanger has a control volume and an isobaric heat supply process. Heat from the hot exhaust gas is transferred to the cool working fluid to convert it into superheated steam.
- State 3 to 4: The steam turbine has a control volume and isentropic process (constant entropy of working fluid). Work is done through the working fluid's expansion in the turbine.
- State 4 to 1: The condenser has a control volume and isobaric process. Heat is rejected at a constant pressure.

1) **Pump**

The following assumptions are made:

- Steady-state condition
- Isentropic compression $(s_1 = s_2)$
- Adiabatic compression $\left(q_{1\_2} = 0\right)$
- Neglecting the potential and kinetic energy

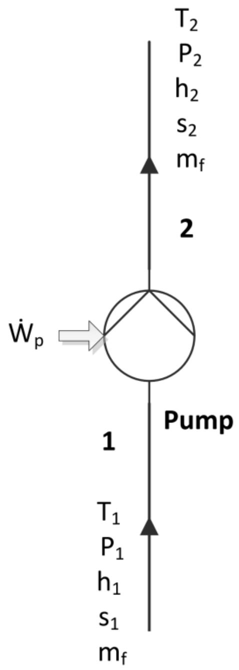


Figure 3.5: An alimenting pump.

Thus, from the continuity and energy balance equations, the following is found:

Continuity:

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{3.65}$$

Energy equation:

$$\dot{Q}_{CV} + \dot{m}_i\left(h_i + \frac{V_i^2}{2} + gZ_i\right) = \dot{m}_e\left(h_e + \frac{V_e^2}{2} + gZ_e\right) + \dot{W}_{CV}, \tag{3.66}$$

where i and e indicate the inlet and outlet flow and $(c.v)$ the control volume indices.

The continuity and energy equations can now be applied to the pump as a control volume.

The following assumptions can also be considered:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{f,} \tag{3.67}$$

with f indicating the cold working fluid (water)

$$\dot{W}_{1\_2} = \dot{m}_f(h_2 - h_1), \tag{3.68}$$

where $\dot{W}_{1\_2}$ or $\dot{W}_p$ is the pump power in kW.

Then, when both sides of the equations are divided by $\dot{m}_f$, the following is found:

$$w_{1\_2} = (h_2 - h_1), \tag{3.69}$$

with w_{1_2} or w_p being the pump work in (KJ/kg)

2) Heat exchanger

The following assumptions are made:

- Steady-state condition
- Isobaric heat transfer ($P_2 = P_3$)
- No shaft work $\left(w_{2\_3} = 0\right)$
- Neglecting the potential and kinetic energy

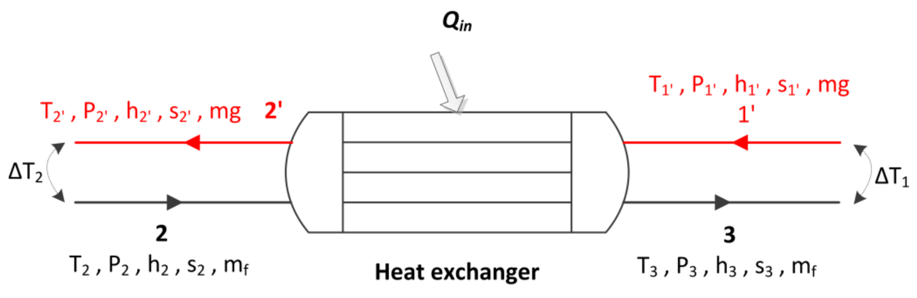


Figure 3.6: The heat exchanger.

Firstly, the amount of heat available in the exhaust gas that will be transferred to the working fluid of the smart thermocycle should be determined.

From the continuity and balance energy equations, the energy available in the exhaust gas is given by the following:

$$\dot{Q}_{1'\_2'} = \dot{m}_g c_{pg}\ (T_{2'} - T_{1'}) \tag{3.70}$$

Assuming that the all the heat from the exhaust gas has been transferred to the working fluid, it means the following:

$$\dot{Q}_{1'\_2'} = \dot{Q}_{2\_3} = \ \dot{m}_f (h_3 - h_2), \tag{3.71}$$

with $\dot{m}_2 = \dot{m}_3 = \dot{m}_f$

Dividing this by the mass flow rate of the working fluid:

$$q_{2\_3} = (h_3 - h_2) \tag{3.72}$$

3) **Turbine**

The following assumptions are made:

- Steady-state condition
- Isentropic expansion $(s_2 = s_3)$
- No rejection of heat from the turbine $\left(q_{2\_3} = 0\right)$
- Neglecting the potential and kinetic energy

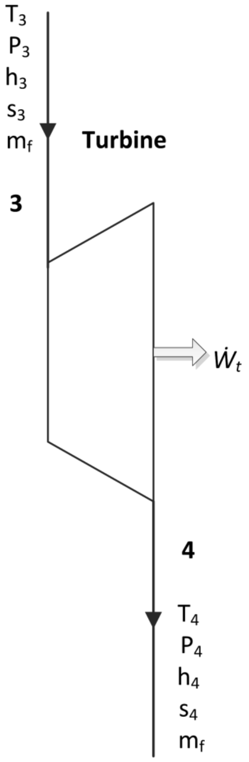


Figure 3.7: The turbine.

Applying the above assumptions, the conservation of mass and energy equations give the following:

with $\dot{m}_3 = \dot{m}_4 = \dot{m}_f$,

$$\dot{W}_{3\_4} = \dot{m}_f(h_3 - h_4) \tag{3.73}$$

Thus, the work done by the turbine is the power over the working fluid mass flow rate:

$$w_{3\_4} = (h_3 - h_4) \tag{3.74}$$

4) **Condenser**

The following assumptions are made:

- Steady-state condition
- Isobaric heat transfer in the condenser $(\mathrm{P}_2 = \mathrm{P}_3)$

- No shaft work $\left(w_{4\_1} = 0\right)$
- Neglecting the potential and kinetic energy

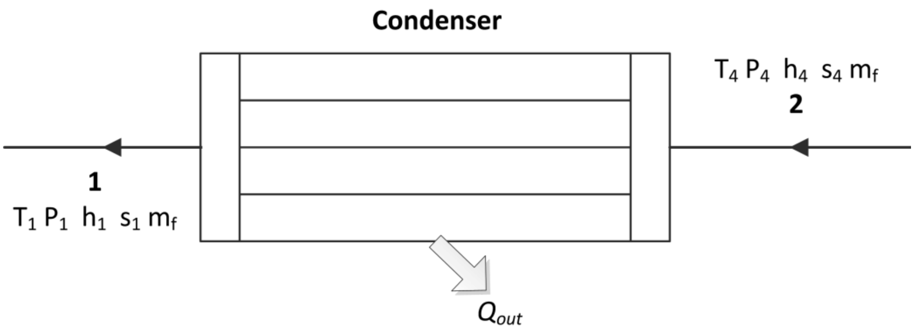


Figure 3.8: The condenser.

When the first law of thermodynamics for a steady-state flow is applied, the following is found:

$$\dot{Q}_{4\_3} = \dot{m}_f(h_4 - h_1) \tag{3.75}$$

with $\dot{m}_4 = \dot{m}_1 = \dot{m}_f$

Divided by the mass flow rate of the working fluid:

$$q_{4\_1} = (h_4 - h_1) \tag{3.76}$$

5) **Thermal efficiency**

The thermal efficiency of the cycle is given by the following equations:

$$\eta_{thermal} = \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\dot{Q}_{in}} \tag{3.77}$$

$$\eta_{th} = \frac{\dot{W}_{3\_1} - \dot{W}_{1\_2}}{\dot{Q}_{2\_3}} \tag{3.78}$$

where $w_{net} = w_T - w_p$

From the above equations, the characteristic thermodynamic points were determined using the EES V9.486-3D. The Rankine cycle was also plotted.

3.7.2 The different configurations of WHR based on a Rankine cycle

The mechanical or electrical power can be produced by using a Rankine cycle when there is a heat source available, $\dot{Q}_{av}$ (waste heat from exhaust gases). For the same exhaust gas specification applied on different WHR configurations, as shown in Table 3.5. Different thermal efficiencies have also been recorded.

Table 3.5: Exhaust gas specifications

$T_{g,in}$	$T_{g,out}$	P_{exh}	$C_{p,mix}$	ρ_{mix}	$\dot{m}_{Tot,mix}$	$\dot{Q}_{av}$
773.2	313.2	400	1.258	0.9513	0.1154	66.78

Configuration I

In terms of Figure 3.5 and the exhaust gas and working fluid specifications given in Tables 3.5 and 3.6, the thermal efficiency was found to be 38.68%.

Table 3.6: Working fluid specifications in ideal conditions (case I)

$T_{w,in}$	$T_{g,out}$	P_{HEX}	P_{COND}	$\dot{m}_{w}$	$\dot{Q}_{av}$	η_{th}
319.2	723.2	8 000	10	0.02065	63.44	38.68

It is evident that this high thermal efficiency can only be achieved theoretically because the turbine outlet is a two-phase mixture, which means that there is moisture in the steam that leaves the turbine. This can cause the erosion of turbine blades in the turbine’s low-pressure stages if it exceeds 10% of the steam and leads to a decrease in turbine efficiency.

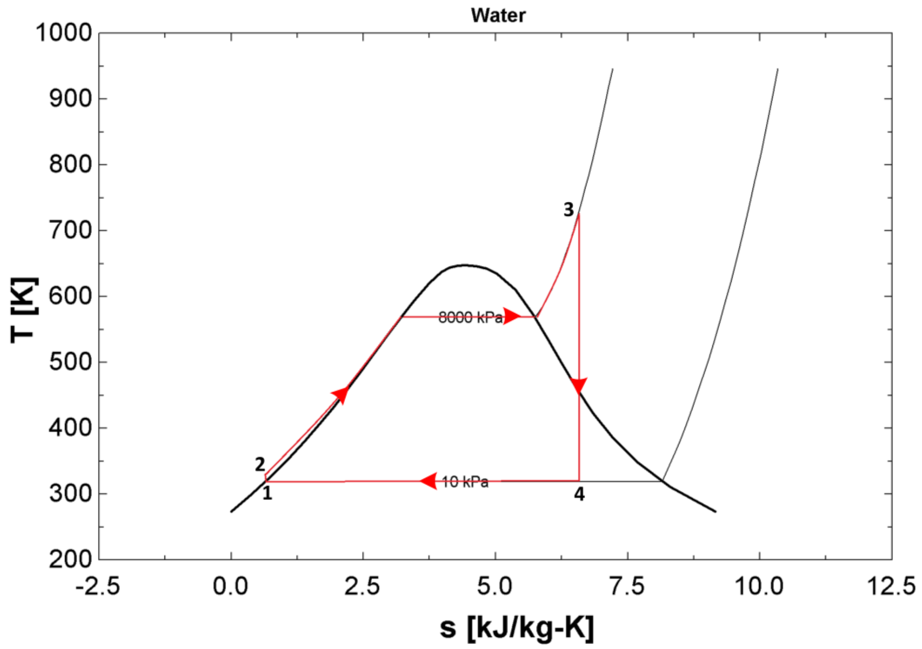


Figure 3.9: A temperature entropy (T-s) diagram of an ordinary Rankine cycle.

Configuration II

Considering the previous configuration, but taking the turbine outlet as a saturated vapour of water ($x = 1$), with the same exhaust gas specifications as stated in Table 3.5, the achieved thermal efficiency was 19.46%.

Table 3.7: Working fluid specifications in practical conditions (case II)

$T_{w,in}$	$T_{g,out}$	P_{HEX}	P_{COND}	$\dot{m}_w$	$\dot{Q}_{av}$	η_{th}
457.9	723.2	8 000	1 092	0.02554	63.44	19.46

In this case, the turbine lifetime is lengthened, but the steam leaving the turbine still has a high temperature and pressure. This means that heat is lost and the turbine performs less work. Less thermal efficiency is therefore achieved.

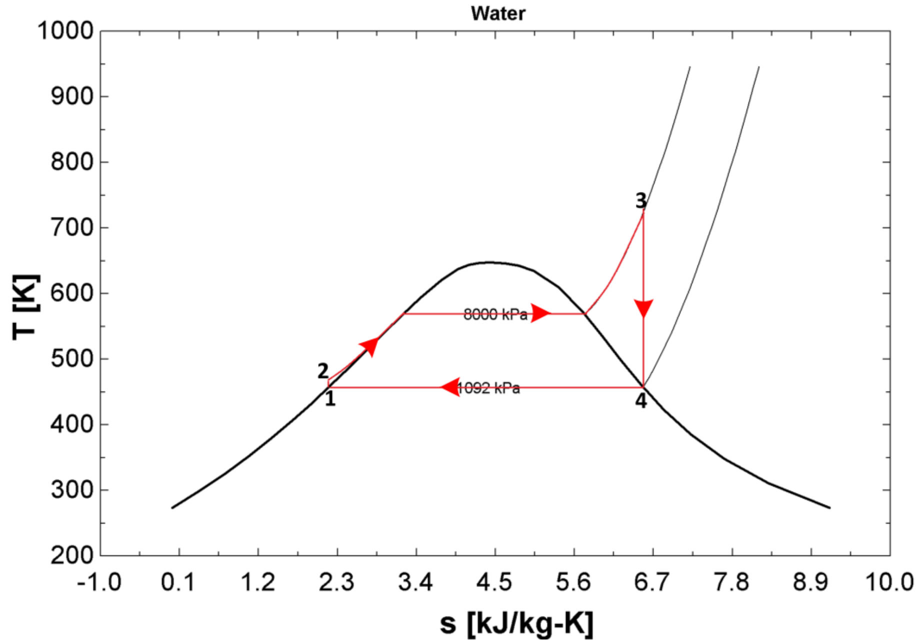


Figure 3.10: T-s diagram of an ideal ordinary Rankine cycle.

Configuration III

In this configuration, the working fluid is expanded at some intermediate pressure within a high-pressure turbine. It is then reheated in the boiler, which is then expanded through a low-pressure turbine to the exhaust pressure (see Figure 3.11). The advantage of the reheat configuration is to enhance the efficiency at a higher pressure by avoiding excessive moisture at the outlet of the low-pressure turbine, as shown in the T-s diagram (see Figure 3.12). With this exhaust gas specification, which is the same as in previous cases, the thermal efficiency was found to be 39.65% due to an increase in work and heat transferred in the boiler.

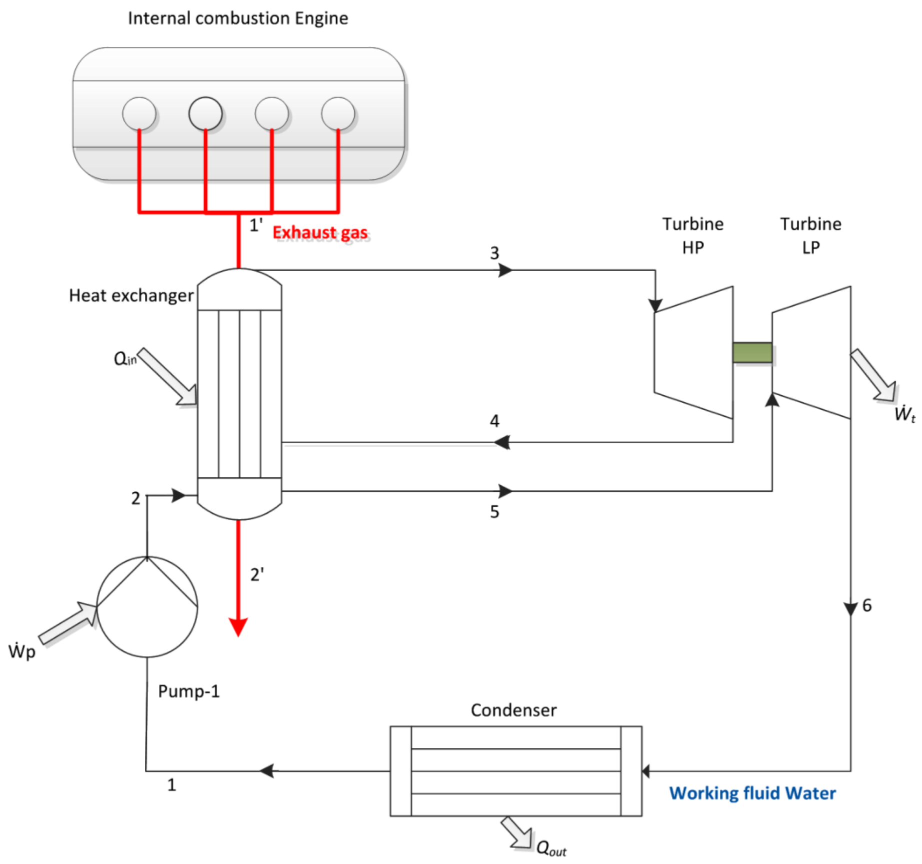


Figure 3.11: The reheating cycle.

Table 3.8: The working fluid specifications in ideal conditions (case III)

$T_{w,in}$	$T_{g,out}$	P_{HEX}	P_{INT}	P_{COND}	$\dot{m}_w$	$\dot{Q}_{av}$	η_{th}
319.2	723.2	8 000	1 092	10	0.01733	63.44	39.65

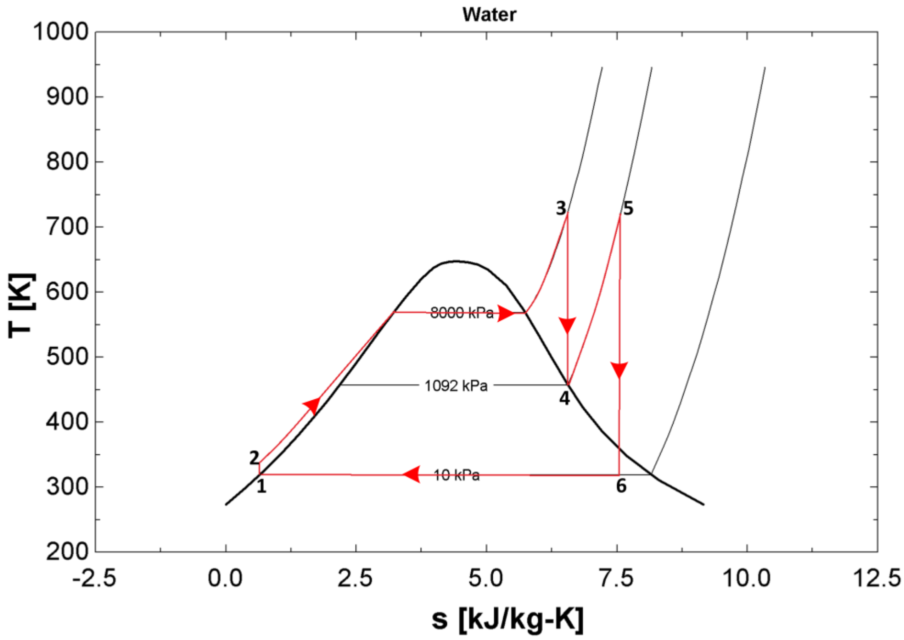


Figure 3.12: The T-s diagram of a reheating cycle.

Configuration IV

In this case, the steam temperature and exhaust pressure are both kept constant and the high- and the low-pressure turbine outlets operate as saturated water vapour.

Table 3.9: Working fluid specifications in practical conditions (case IV)

$T_{w,in}$	$T_{g,out}$	P_{HEX}	P_{INT}	P_{COND}	$\dot{m}_w$	$\dot{Q}_{av}$	η_{th}
356.2	723.2	8 000	1 092	52.53	0.01733	63.44	33.37

In the case of avoiding moisture in the low stages of the turbine, the working fluid in the low-pressure turbine exits in a saturated state (x = 1). This has the advantage of increasing the lifetime of the turbine by eliminating moisture in the low-pressure stage of the turbine. On the other hand, there is a decrease in thermal efficiency.

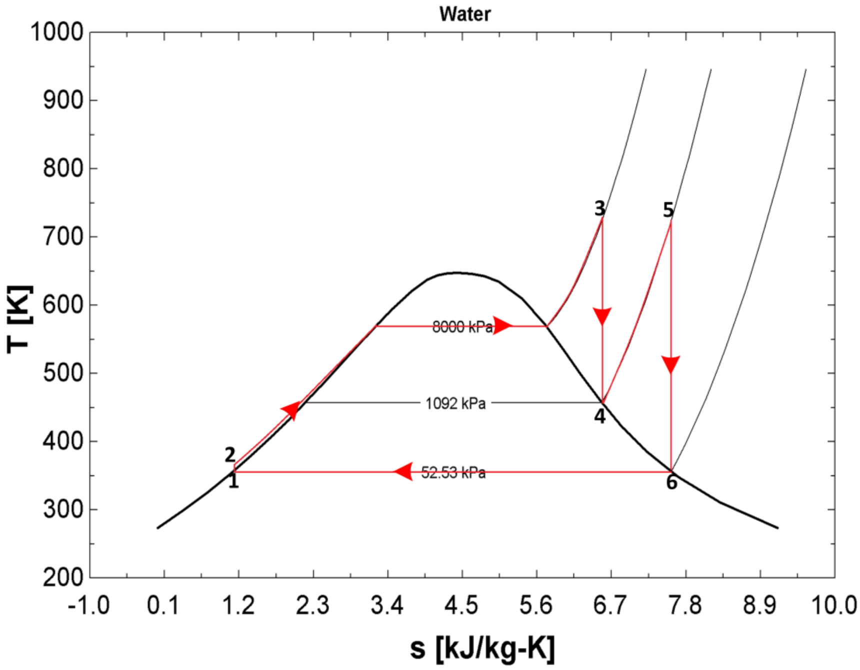


Figure 3.13: T-s diagram of an ideal reheating cycle.

Configuration V

The approach behind this configuration is to increase the steam power cycle's thermal efficiency by extracting a portion of steam from the turbine (at Stage 6) and to use it to preheat the compressed liquid from the compressor (at Stage 2) before it enters the boiler. This operation is accomplished by directly mixing the steam extracted from the turbine and compressed liquid from the condenser in the open feedwater heater (OFH).

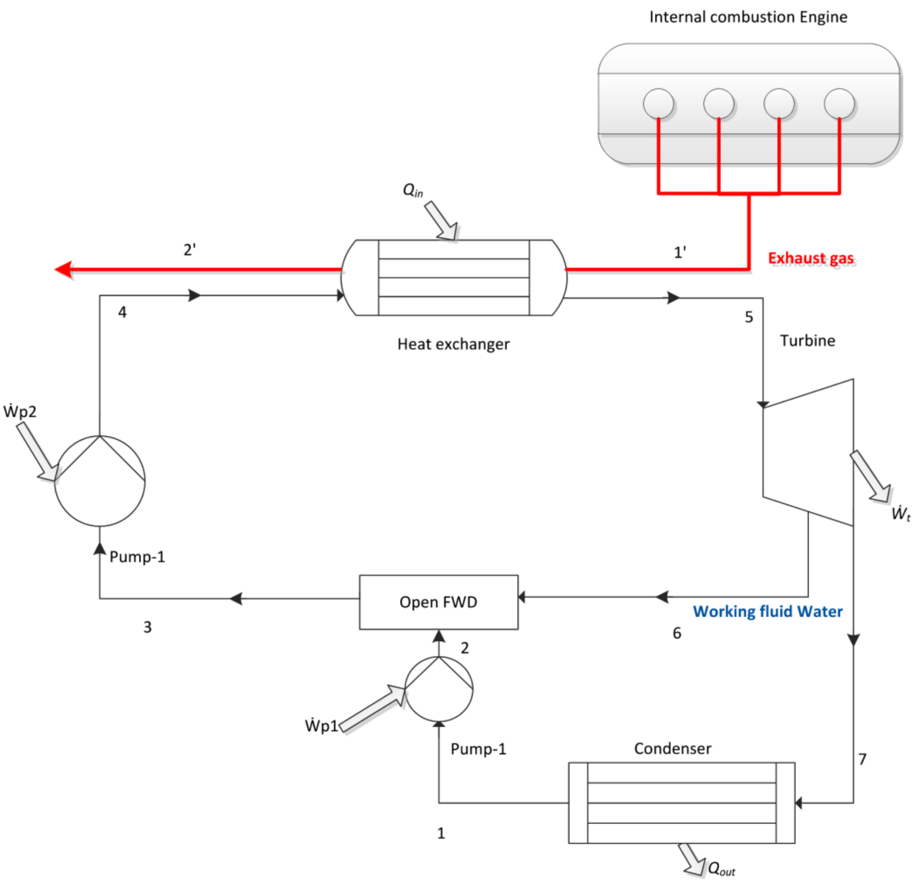


Figure 3.14: A regenerative cycle with an OFH.

Table 3.10: Working fluid specifications

$T_{w,in}$	$T_{g,out}$	P_{HEX}	P_{INT}	P_{COND}	$\dot{m}_w$	$\dot{Q}_{av}$	η_{th}
444.5	723.2	8 000	800	10	0.02495	63.44	41.4

It is noticed that the thermal efficiency of the power cycle that incorporates an OFH is slightly higher than that of a reheat cycle. The OFH has the advantage of being less expensive and having a better heat transfer rate compared to that of the closed feedwater heater (CFH). The only advantage is that it requires an auxiliary pump to handle the pressure between the feedwater and the boiler.

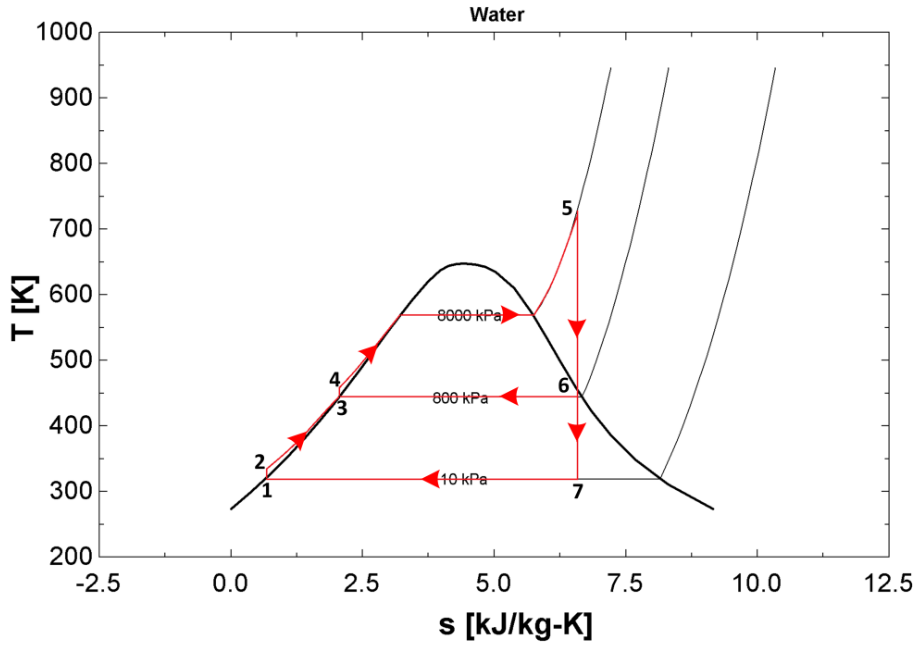


Figure 3.15: T-s Diagram of a regenerative cycle with OFH.

3.8. Conclusions

Among the different types of thermodynamic operating cycles, such as air standard Otto, CI and the combined CI dual cycle, only the exhaust flow process of an air standard Otto or SI cycle was analysed. For simplicity of analysis, the exhaust gas flow process was assumed to be a pseudo steady-state flow, even though it is a non-steady-state pulsing flow.

It was stated that the exhaust gas temperatures commonly range from 400 °C to 600 °C in SI engines, and from 200 °C to 500 °C in typical CI engines. In this study, which was conducted on a VW Citi Golf 1.3i, an average temperature of 500 °C and a pressure between 300 kPa and 400 kPa for the exhaust gas mixture were considered to establish the set of equations to determine the exhaust mass flow rate ($\dot{m}_{mix}$) by applying the Gibbs-Dalton law. The exhaust gas mixture consisted of 71% N_2, 14% CO_2, 13% H_2O and 1% to 2% HC, NO_x and CO. The laws of thermodynamics were solved to compute the thermal energy available ($\dot{Q}_{av}$) in the exhaust mixture by means of a code written in EES V9.486-3D. For the same exhaust gas characteristics and working fluid, different thermal efficiencies were found using different waste recovery configurations.

Chapter 4: Results and discussions

The results of the analysis are shown in this chapter. The thermal efficiency of different working fluids for a system operating under the same condition (x = 0.9 and x = 1) is discussed. Fuel demand and consumption, as well as the environmental impact of an EWHR system, are analysed.

4.1. Thermodynamic performance of working fluids

The thermal efficiency of five different working fluids (water, R-718, Steam_NBS, steam and Steam_IAPWS) under the same conditions have been determined by the means of a Rankine cycle.

If the temperature of the exhaust gas just behind the valves is 500 °C (773.2K), as the temperature of the exhaust gas must always be higher than the working fluid, a minimum difference in temperature at the pinch point of 20 °C (293.2K) is necessary for heat transfer to occur.

$$\mathrm{T_{g,out}} = \mathrm{T_{c,in}} + \Delta\mathrm{T_{PP}} \tag{4.1}$$

Therefore, $\mathrm{T_{c,in}} = 480°C\ (573.2K)$

with

$T_{g,out}$: Exhaust gas temperature behind the valves

$T_{c,in}$: Inlet working fluid temperature

ΔT_{PP}: Minimum temperature difference at pinch point

4.2. Design configuration

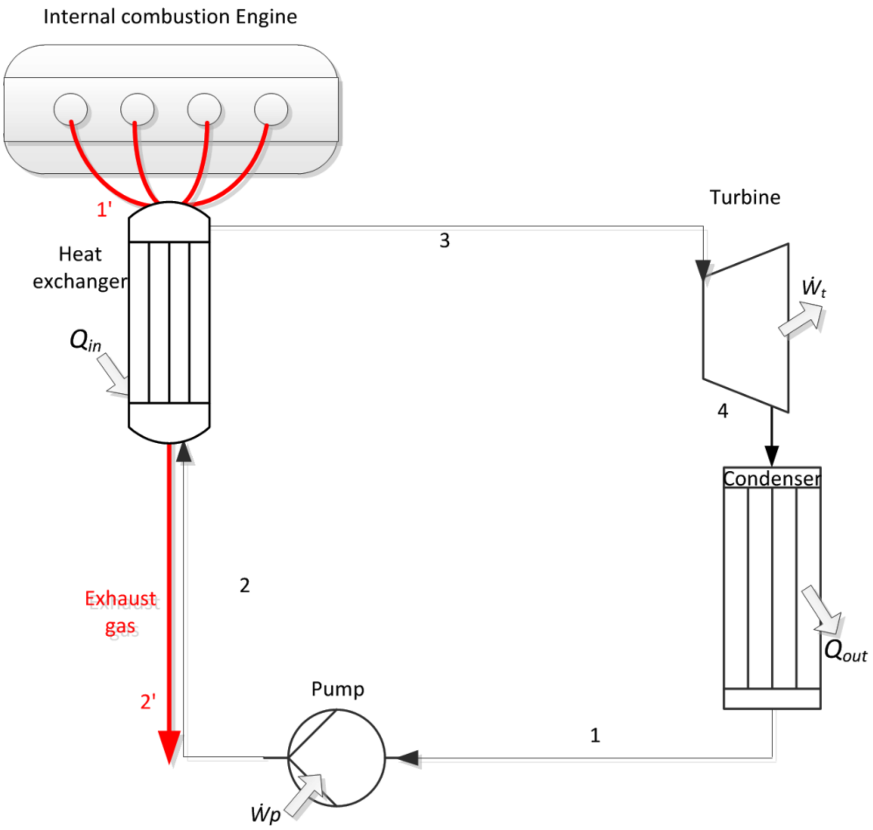


Figure 4.1: The energy recovery configuration.

For this energy recovery configuration, ten different working fluids have been used under the same conditions. Their thermal efficiency was compared. Table 4.1 shows the thermal properties of the exhaust gas, especially the amount of energy it carries, which is the major data in the determination of the Rankine cycle's thermal efficiency for all five selected working fluids.

Table 4.1: Exhaust gas specifications

$T_{g,in}$	$T_{g,out}$	P_{exh}	$C_{p,mix}$	ρ_{mix}	$\dot{m}_{Tot,mix}$	$\dot{Q}_{av}$
500	40	400	1.258	0.9513	0.1154	66.78

The cycle's highest and lowest temperature is fixed at 480 °C (outlet temperature of the working fluid at the heat exchanger) and 60 °C (temperature of working fluid at the condenser). The thermodynamic properties that characterise each point of the cycle for all five working fluids were determined as follows:

4.2.1 Configuration I: water as the working fluid

Table 4.2: Thermodynamic properties of the cycle

Point	T	P	H	S	x	v	$\dot{m}_f$
1	60	19.93	251.2	0.8312	0.0	0.001017	0.02008
2	60.12	2 851	254	0.8312	-	0.001016	0.02008
3	480	2 851	3 413	7.2000	-	0.118900	0.02008
4	60	19.93	2 373	7.2000	0.9	6.907000	0.02008

Table 4.3: Thermal efficiency of the cycle

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.879	1 040	1 038	3 159	2 122	**32.84**

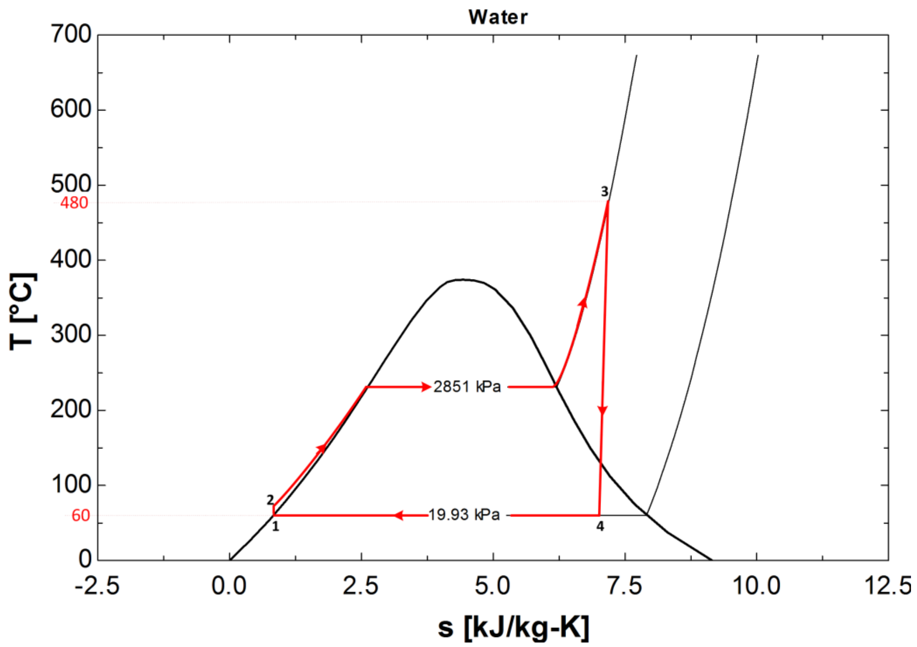


Figure 4.2: T-s diagram for water, x = 0.9.

4.2.2 Configuration II: R-718 as the working fluid

Table 4.4: Thermodynamic properties of the cycle

Point	T	P	H	S	x	v	$\dot{m}_f$
1	60	19.93	251.2	0.8312	0.0	0.001017	0.02008
2	60.12	2 851	254	0.8312	-	0.001016	0.02008
3	480	2 851	3 413	7.2000	-	0.118900	0.02008
4	60	19.93	2 373	7.2000	0.9	6.907000	0.02008

Table 4.5: Thermal efficiency of the cycle

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.879	1 040	1 038	3 159	2 122	**32.84**

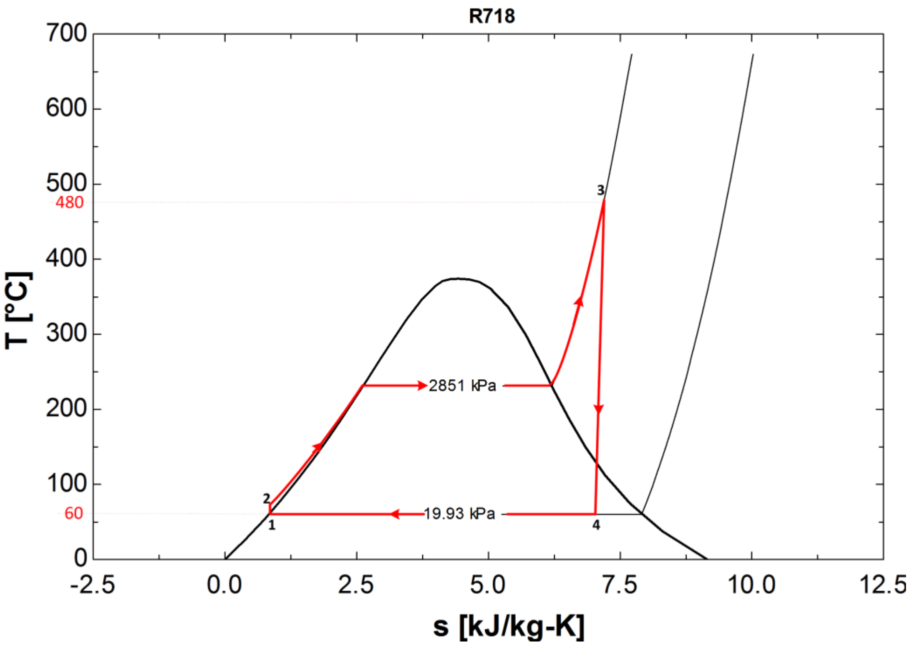


Figure 4.3: T-s diagram for R718, x = 0.9.

4.2.3 Configuration III: Steam$_{NBS}$ as the working fluid

Table 4.6: Thermodynamic properties of the cycle

Point	T	P	H	S	x	v	$\dot{m}_f$
1	60	19.93	251.2	0.8312	0.0	0.001017	0.02008
2	60.12	2 851	254	0.8312	-	0.001016	0.02008
3	480	2 851	3 413	7.2000	-	0.118900	0.02008
4	60	19.93	2 373	7.2000	0.9	6.907000	0.02008

Table 4.7: Thermal efficiency of the cycle

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.879	1 040	1 038	3 159	2 122	**32.84**

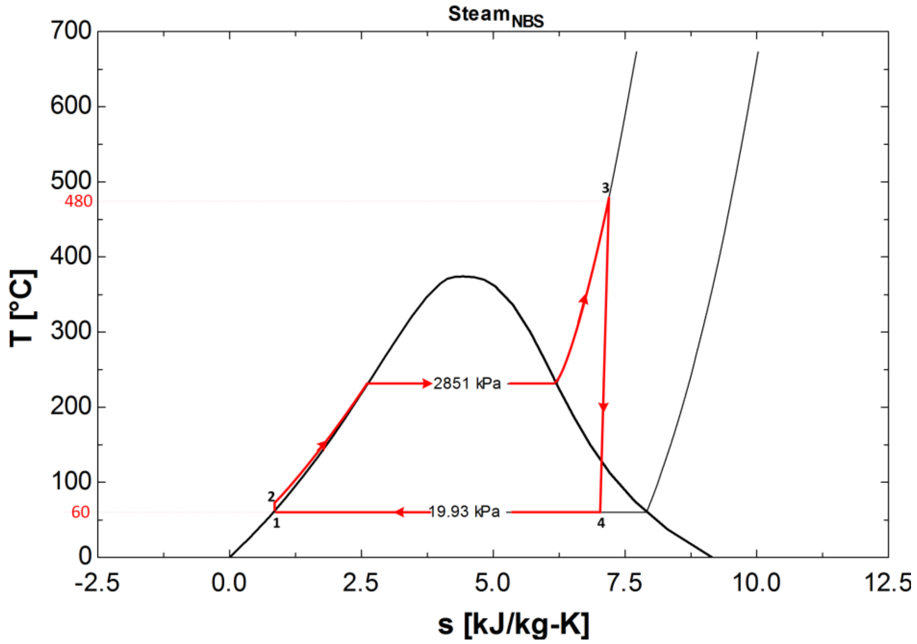


Figure 4.4: T-s diagram for steam NBS, x = 0.9.

4.2.4 Configuration IV: steam as the working fluid

Table 4.8: Thermodynamic properties of the cycle

Point	T	P	H	S	x	v	$\dot{m}_f$
1	60	19.93	251.2	0.8312	0.0	0.001017	0.02008
2	60.12	2 851	254	0.8312	-	0.001016	0.02008
3	480	2 851	3 413	7.2000	-	0.118900	0.02008
4	60	19.93	2 373	7.2000	0.9	6.907000	0.02008

Table 4.9: Thermal efficiency of the cycle

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.879	1 040	1 038	3 159	2 122	**32.84**

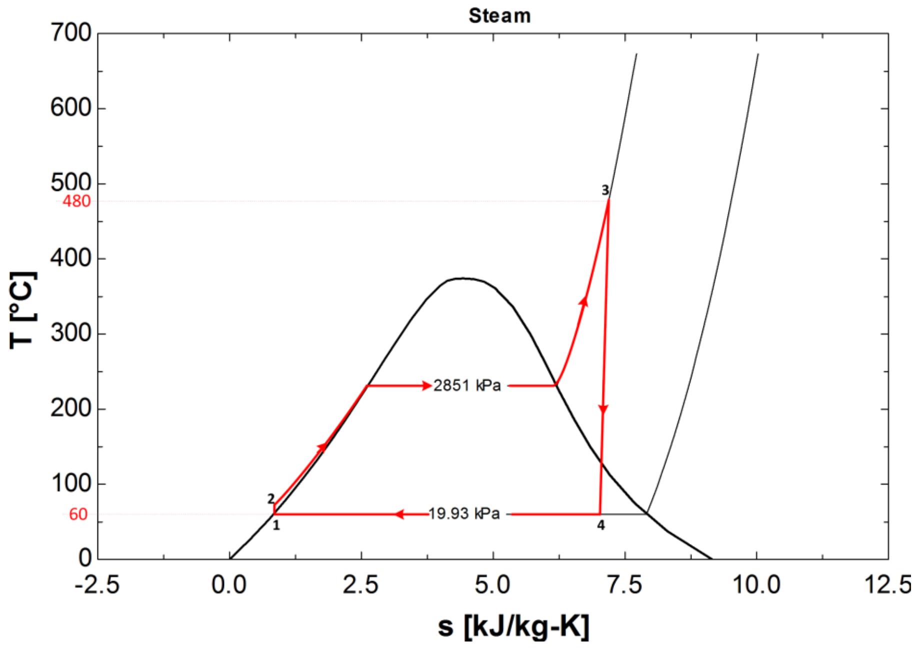


Figure 4.5: T-s diagram for steam, x = 0.9.

4.2.5 Configuration V: Steam_IAPWS as the working fluid

Table 4.10: Thermodynamic properties of the cycle

Point	T	P	H	S	x	v	$\dot{m}_f$
1	60	19.93	251.2	0.8312	0.0	0.001017	0.02008
2	60.12	2 851	254	0.8312	-	0.001016	0.02008
3	480	2 851	3 413	7.2000	-	0.118900	0.02008
4	60	19.93	2 373	7.2000	0.9	6.907000	0.02008

Table 4.11: Thermal efficiency of the cycle

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.879	1 040	1 038	3 159	2 122	**32.84**

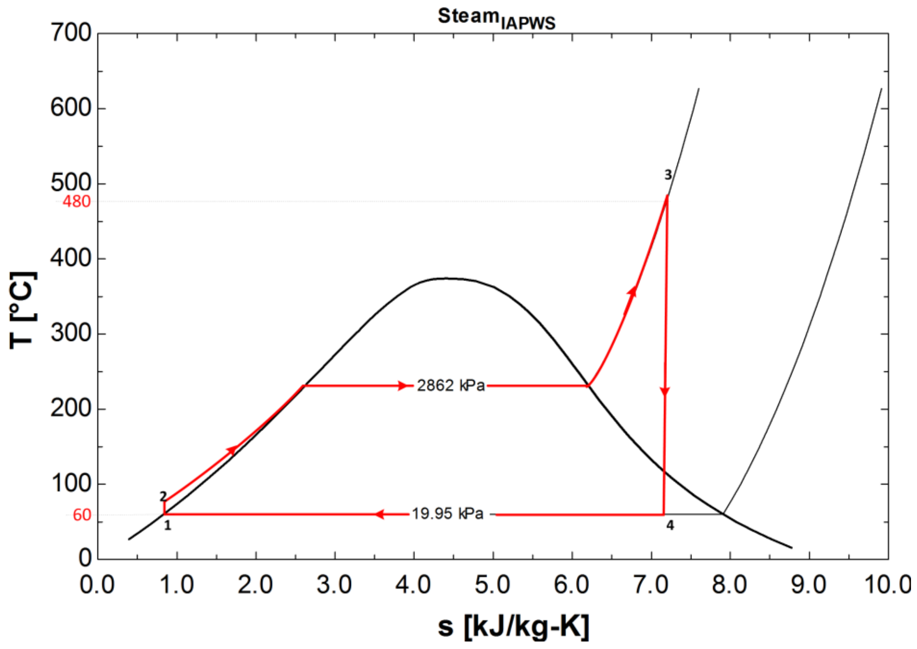


Figure 4.6: T-s diagram for Steam_IAPWS, x = 0.9.

4.2.6 Configuration V: ammonia as the working fluid

Table 4.12: Thermodynamic properties of the cycle [x4 = 0.9]

Point	*T*	*P*	*H*	*S*	*x*	*v*	$\dot{m}_f$
1	15.73	746.6	273.5	1.260	0.0	0.001622	0.04391
2	18.17	11 333	290.6	1.260	-	0.001609	0.04391
3	199.7	11 333	1 735	5.009	-	0.015590	0.04391
4	15.73	746.6	1 357	5.009	0.9	0.153600	0.04391

Table 4.13: Thermal efficiency of the cycle [x4 = 0.9]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
17.17	378.8	361.6	1 445	1 083	**25.03**

Table 4.14: Thermodynamic properties of the cycle [x4 = 1]

Point	*T*	*P*	*H*	*S*	*x*	*v*	$\dot{m}_f$
1	15.73	746.6	273.5	1.260	0.0	0.001622	0.04391
2	18.17	11 333	290.6	1.260	-	0.001609	0.04391
3	258	11 333	1944	5.426	-	0.015590	0.04391
4	15.73	746.6	1 477	5.426	1.0	0.153600	0.04391

Table 4.15: Thermal efficiency of the cycle [x4 = 1]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
17.17	466.9	449.8	1 653	1 203	**27.21**

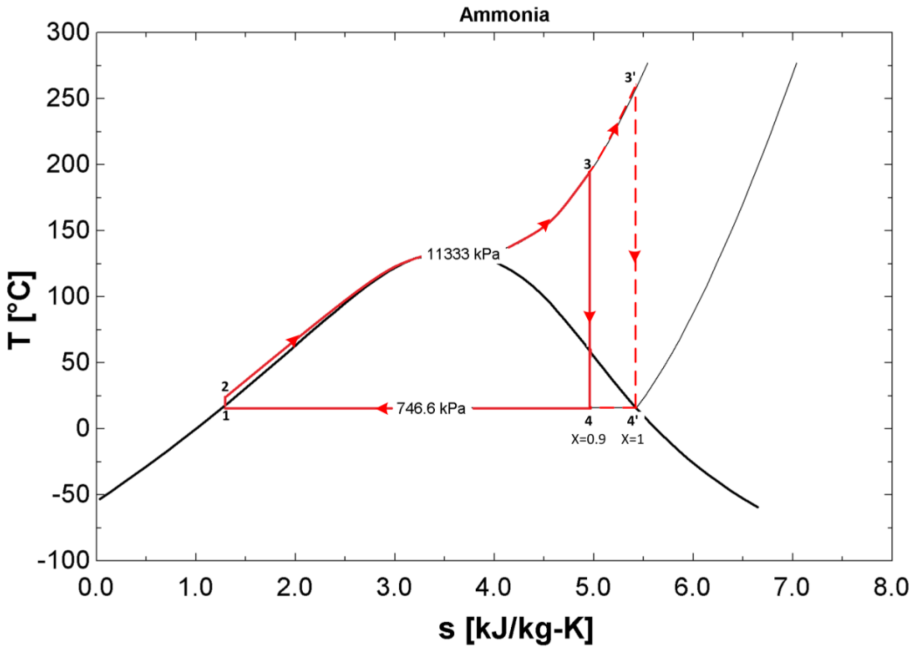


Figure 4.7: T-s diagram for ammonia, x = 0.9 and x = 1.

4.2.7 Configuration V: R11 as the working fluid

Table 4.16: Thermodynamic properties of the cycle [x4 = 0.9]

Point	*T*	*P*	*H*	*S*	*x*	*v*	$\dot{m}_f$
1	20.51	90	51.7	0.1968	0.0	0.0006726	0.2931
2	29.45	4 408	59.47	0.1968	-	0.0006824	0.2931
3	198.4	4 408	275.9	0.7531	-	0.0023750	0.2931
4	20.51	90	215.1	0.7531	0.9	0.1714000	0.2931

Table 4.17: Thermal efficiency of the cycle [x4 = 0.9]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.904	60.83	57.93	216.4	163.4	**26.77**

Table 4.18: Thermodynamic properties of the cycle [x4 = 1]

Point	T	P	H	S	x	v	$\dot{m}_f$
1	20.51	90	51.7	0.1968	0.0	0.0006726	0.2581
2	29.45	4 408	59.47	0.1968	-	0.0006824	0.2581
3	210.1	4 408	305.3	0.8150	-	0.0036280	0.2581
4	20.51	90	233.2	0.8150	1.0	0.1904000	0.2581

Table 4.19: Thermal efficiency of the cycle [x4 = 1]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.904	72.1	69.19	245.8	181.5	**28.15**

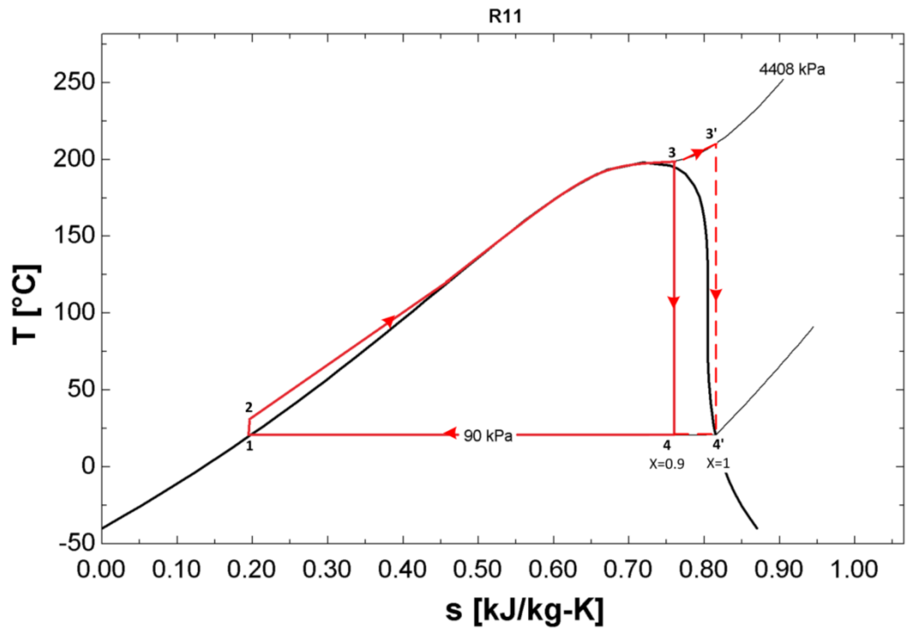


Figure 4.8: T-s diagram for R11, x = 0.9 and x = 1.

4.2.8 Configuration V: R134a as the working fluid

Table 4.20: Thermodynamic properties of the cycle [x4 = 0.9]

Point	T	P	H	S	x	v	$\dot{m}_f$
1	20	572.1	79.32	0.3006	0.0	0.0008161	0.3263
2	21.76	4 059	82.15	0.3006	-	0.0008083	0.3263
3	103.5	4 059	276.6	0.8602	-	0.0033080	0.3263
4	20	572.1	243.4	0.8602	0.9	0.0324500	0.3263

Table 4.21: Thermal efficiency of the cycle [x4 = 0.9]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.846	33.24	30.39	194.4	164	**15.63**

Table 4.22: Thermodynamic properties of the cycle [x4 = 1]

Point	T	P	H	S	x	v	$\dot{m}_f$
1	20	572.1	79.32	0.3006	0.0	0.0008161	0.3597
2	21.76	4 059	82.15	0.3006	-	0.0008083	0.3597
3	111.8	4 059	300.2	0.9223	-	0.0042720	0.3597
4	20	572.1	261.6	0.9223	1.0	0.0359700	0.3597

Table 4.23: Thermal efficiency of the cycle [x4 = 1]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.846	38.65	35.81	218.1	182.3	**16.42**

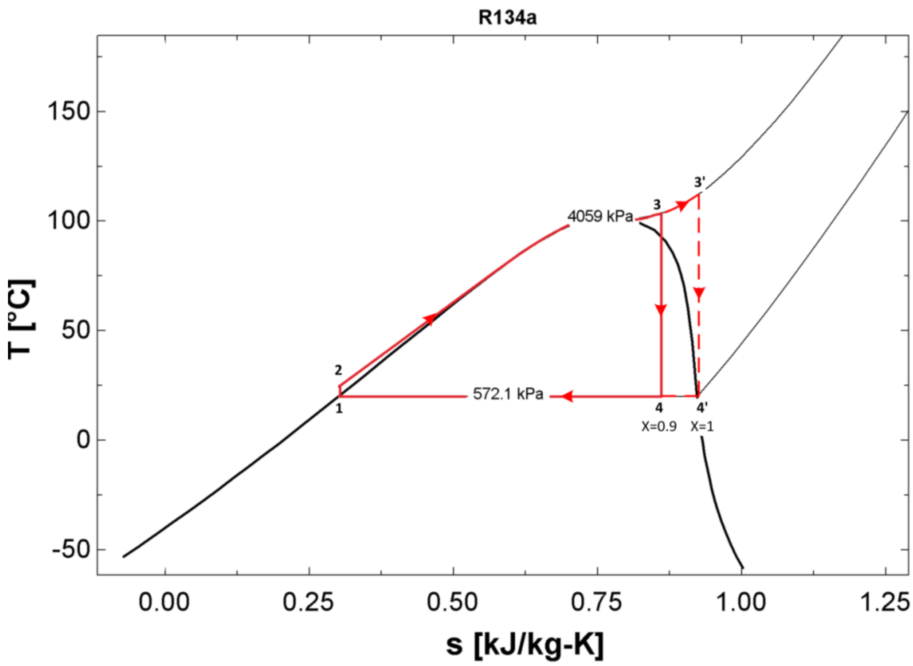


Figure 4.9: T-s diagram for R134a, x = 0.9 and x = 1.

4.2.9 Configuration V: R12 as the working fluid

Table 4.24: Thermodynamic properties of the cycle [x4 = 0.9]

$Point$	T	P	H	S	x	v	$\dot{m}_f$
1	20	566.8	54.86	0.2077	0	0.0007524	0.4249
2	27.06	4 114	61.69	0.2077	-	0.0007673	0.4249
3	115	4 114	211	0.6403	-	0.0029510	0.4249
4	20	566.8	181	0.6403	0.9	0.0277900	0.4249

Table 4.25: Thermal efficiency of the cycle [x4 = 0.9]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.669	29.31	26.64	149.3	126.8	**17.84**

Table 4.26: Thermodynamic properties of the cycle [x4 = 1]

$Point$	T	P	H	S	x	v	$\dot{m}_f$
1	20	566.8	54.86	0.2077	0.0	0.0007524	0.3772
2	27.06	4 114	61.69	0.2077	-	0.0007673	0.3772
3	124.9	4 114	229.9	0.6884	-	0.0038000	0.3772
4	20	566.8	195.8	0.6884	1.0	0.0307900	0.3772

Table 4.27: Thermal efficiency of the cycle [x4 = 1]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
2.669	34.08	31.41	168.2	140.9	**18.68**

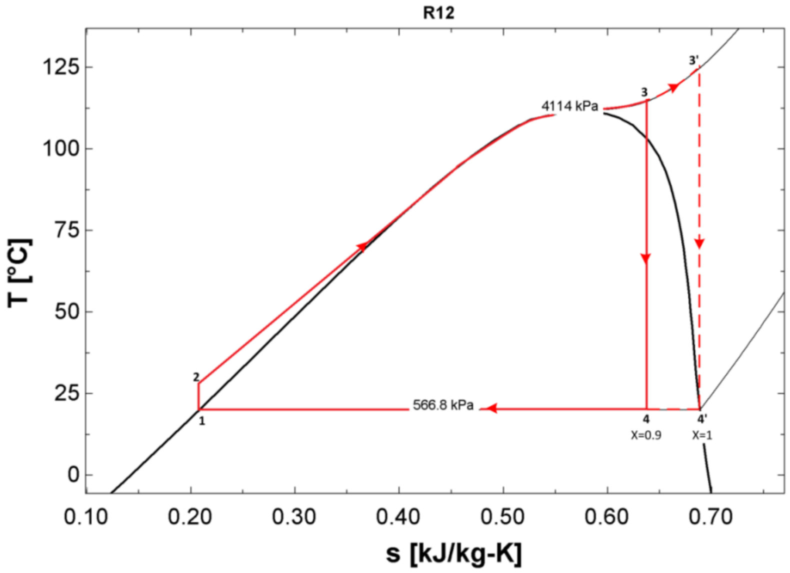


Figure 4.10: T-s diagram for R12, x = 0.9 and x = 1

4.2.10 Configuration V: R717 as the working fluid

Table 4.28: Thermodynamic properties of the cycle [x4 = 0.9]

Point	T	P	H	S	x	v	$\dot{m}_f$
1	16.27	760	276.1	1.269	0.0	0.001624	0.04406
2	18.72	11 333	293.2	1.269	-	0.001611	0.04406
3	199.2	11 333	1 733	5.004	-	0.015550	0.04406
4	16.27	760	1 357	5.004	0.9	0.151000	0.04406

Table 4.29: Thermal efficiency of the cycle [x4 = 0.9]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
17.17	375.8	358.6	1 440	1 081	**24.91**

Table 4.30: Thermodynamic properties of the cycle [x4 = 1]

Point	T	P	H	S	x	v	$\dot{m}_f$
1	16.27	760	276.1	1.269	0.0	0.001624	0.03851
2	18.72	11 333	293.2	1.269	-	0.001611	0.03851
3	257	11 333	1941	5.419	-	0.019590	0.03851
4	16.27	760	1 477	5.419	1.0	0.167600	0.03851

Table 4.31: Thermal efficiency of the cycle [x4 = 1]

w_p	w_t	w_{net}	q_{HEX}	q_{cond}	η_{th}
17.17	463.1	446	1 647	1 201	**27.07**

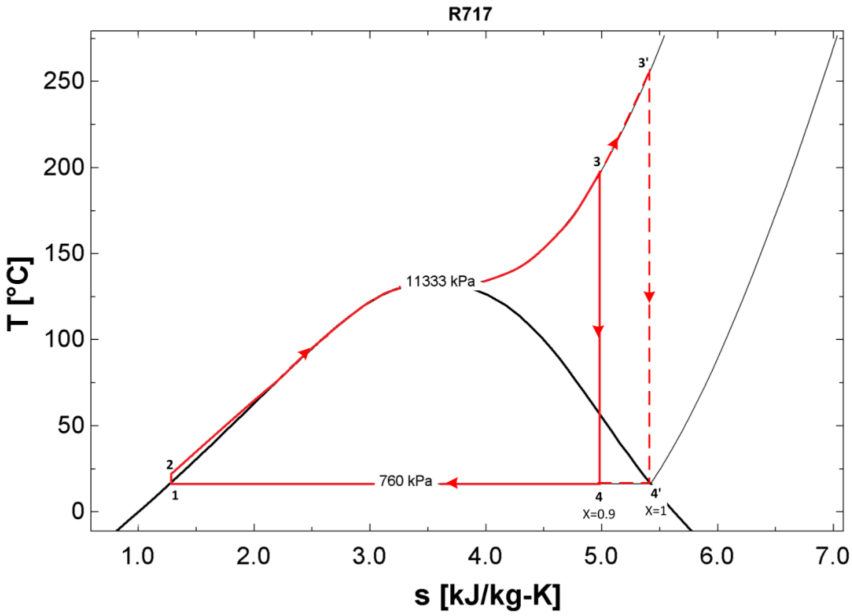


Figure 4.11: T-s diagram for R717, x = 0.9 and x = 1.

It is evident from the 10 working fluids used to determine the thermal efficiency that water and its three variants give a thermal efficiency of 32.84%, while the rest recorded a thermal efficiency under 28%. For this study, water was chosen as the working fluid due to its availability, low cost and environmentally friendliness.

4.3. Fuel demand and consumption analysis

In terms of Table 2.1, only the petrol and diesel or gasoil fuel projection demand is analysed. Therefore, from the abovementioned table, one can determine the total amount of energy available in each fuel, the total amount of energy subjected to produce net shaft work and the total amount of thermal energy wasted through the exhaust pipe per year.

Using the data from Table 2.1 in equations 3.34, 3.35, 3.36 and 3.37, the amount of energy consumed annually and wasted through an exhaust pipe for both petrol and diesel from 2013 to 2040 can be determined (see Figure 4.12).

The graph below gives a clear view of the growth in the world energy demand throughout the coming years. The global trend in energy demand seems to increase by almost 30.03 quadrillion kJ/y during the forecast period, reaching 126.513 quadrillion kJ/y by 2040, compared to 96.491 in 2013, which is also proportional to the growth in waste energy. This increase in energy demand in the transport sector is mostly due to the fact that the number of light motor vehicles will nearly double from 800 million in 2010 to more than 1.6 billion by 2040. This increase is led by the growth in conventional petrol and diesel vehicles on the road, although electrical, full hybrid and natural gas vehicles are available on the market.

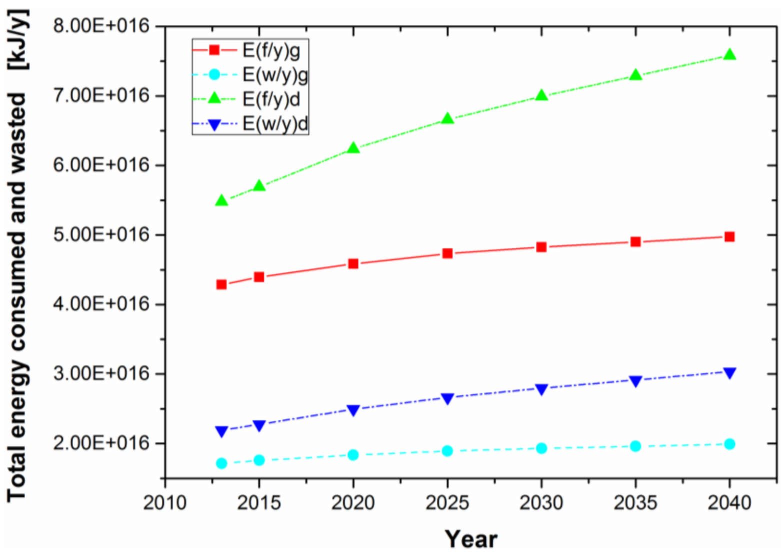


Figure 4.12: Total energy consumed and wasted, 2013-2040 (kJ per year)

4.4. The environmental impact of EWHR

The release of greenhouse gases like NO_x, SO_2 and CO_2 from ICEs into the atmosphere acts like a blanket by absorbing energy and slowing or preventing the loss of heat into the atmosphere. Therefore, in this study, only the reduction of CO_2 has been analysed in terms of its environmental benefit by comparing an ICE with and without an EWHR system. The estimated amount of CO_2 accumulated from a vehicle until it reaches five active years was determined using the formula of Peng, Wang, He, Yang and Lu (2013), where $CO_{2-embedded}$ is the average of embedded CO_2 emitted during the production of a standard-sized petrol ICE (tonnes), $S_{cumulative}$ is the cumulative mileage (km), FC is the fuel consumption (L/km), ρ_{fuel} is the fuel density (kg/m3), χ_c is the rate of carbon content in the fuel, and 44 and 12 represent the CO_2 and C molecular weights respectively.

From Figure 4.13 below, it can be seen that 2 750 tonnes of CO_2 could approximately be reduced using an EWHR system within five active years.

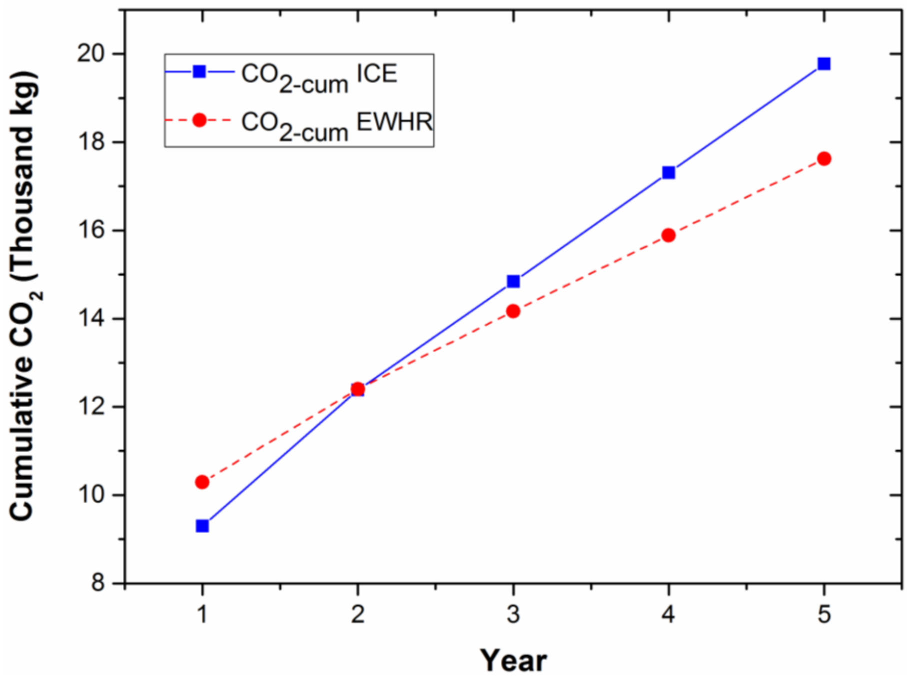


Figure 4.13: Carbon dioxide emission for ICE and ICE+EWHR (Thousand kg)

In the case of petrol, by knowing the average amount of waste heat that can be recovered from 2020 to 2040 using an EWHR system, one can assume that this amount of heat has been released in the North Pole region and one can determine the amount of ice that will be melted in that region during the specific period.

$$\mathrm{m_{ice}} = \frac{\Delta Q_{phase\ change}}{H_{if}}, \tag{4.2}$$

with $\Delta Q_{phase\ change}$ representing the thermal energy absorbed by the ice.

In this case, it is the average amount of heat released into the environment (petrol and diesel combined), H_{if} water is the latent heat of fusion (333.7 kJ/kg), and $\mathrm{m_{ice}}$ is the mass of ice melted, which equals 3 638.742283 1 010 kg or 3 638.742283 107 m^3 of water from 2020 to 2040.

4.5. Conclusions and recommendation

The results of the analysis performed on an EWHR system with 10 different working fluids have shown that water and its three variants (Steam_NBS, steam and Steam_IAPWS) have produced a thermal efficiency of 32.84%, while the rest offered a thermal efficiency of less than 28% with the quality of 0.9 (x=90%). The second analysis performed with quality of 1 (x=100%), the thermal efficiency was increased by an average of 1.29% with water and its variants still achieved the highest thermal efficiency.

The global average of energy wasted through the exhaust pipes of both petrol and diesel vehicles in the transport sector were recorded to be 1.92 10^{10} kJ and 2.78 10^{10} kJ per year, from 2013 to 2040. A reduction of 2.75 thousand tonnes of CO_2 could be achieved within a five-year forecast period when an EWHR system based on a Rankine cycle implemented on an ICE. The same research for some more actual cases and newly designed vehicles is recommended.

Chapter 5: Conclusions and recommendations

5.1. Conclusions

It can be seen that 40% of the energy increase from 2010 to 2040 will mostly come from the transport sector, predominantly the road transport sector where the number of vehicles is expected to almost double from 800 million to more than 1.6 billion by 2040. Although the number of conventional ICE vehicles will decrease by 2040 due to the growth of more energy-efficient vehicles, hybrid and electric vehicles will still present some challenges because of the limitations posed by their high cost and functional technology. For example, in the case of electric vehicles, charging time and distance travelled could present challenges. Therefore, improving diesel and petrol vehicles is still the main objective.

Despite the type of ICE used, studies have shown that, from the 100% of fuel energy put in a vehicle, about 25% is converted into useful shaft work and the rest is wasted throughout different mechanisms (about 40% through exhaust gas, 30% through cooling and 5% through friction). This makes ICEs inefficient engines. Thus, technology to transform some part of waste heat energy into useful energy is implemented. From the two most promising thermal energy recovery methods (thermoelectricity and Rankine cycles), the Rankine system, which incorporates a turbine, heat exchanger, condenser and pump with an appropriate working fluid, was chosen.

In most previous WHR studies on exhaust gases, the WHR system has been placed along the exhaust flow pipe where the temperature differs from the temperature behind the exhaust valves. This means that some important fraction of energy from exhaust gases is still lost to the environment. In this study, in order to recover a large amount of waste energy from exhaust gases, the main heat exchanger will be fitted on the exhaust manifold just behind the exhaust valves, and water will be used as the working fluid due to its potential to recover heat at a high temperature.

Among different types of thermodynamic operating cycles (air standard Otto, CI and combined CI dual cycles), only the exhaust flow process of an air standard Otto or SI cycle was analysed. For simplicity of analysis, the exhaust gas flow process was assumed to be pseudo steady-state flow, even though it is a non-steady-state pulsing flow.

Exhaust gas temperatures commonly range from 400 °C to 600 °C in SI engines and from 200 °C to 500 °C in typical CI engines. In this study, which was conducted on a VW Citi Golf 1.3i, an average temperature of 500 °C and a pressure between 300 kPa and 400 kPa for the exhaust gas mixture were considered to establish the set of equations to determine the exhaust mass flow rate ($\dot{m}_{mix}$) using the Gibbs-Dalton law. The exhaust gas mixture consisted of 71% N_2, 14% CO_2, 13% H_2O and 1% to 2% HC, NO_x and CO. The laws of thermodynamics were solved to compute the available thermal energy ($\dot{Q}_{av}$) in the exhaust mixture by means of a code written in EES V9.486-3D. For the same exhaust gas characteristics and the same working fluid, different thermal efficiencies were found in different waste recovery configurations.

The results of the analysis performed on an EWHR system using 10 different working fluids have shown that water and its three variants (Steam_NBS, steam and Steam_IAPWS) have produced a thermal efficiency of 32.84%, while the rest offered a thermal efficiency of less than 28%.

The global average energy wasted through the exhaust pipes of both petrol and diesel vehicles in the transport sector was recorded to be 1.92 10^{10} kJ and 2.78 10^{10} kJ per year from 2013 to 2040. A reduction of 2.75 thousand tonnes of CO_2 could be achieved within a five-year forecast period when an EWHR system based on the Rankine cycle is implemented on an ICE.

Ultimately, although the development of waste energy recovery systems has been explored for some time, an important fraction of energy is still lost to the environment. This study has highlighted that recovering energy by incorporating a WHR system on the exhaust manifold of ICEs shows significant potential. On one hand, it has been shown that an increase in thermal efficiency up to 7.78% can be achieved when this idea is applied to a VW Citi Golf 1.3i. On the other hand, it can significantly improve fuel consumption in the transport sector on a global scale by gaining an average of 1 114.98 and 1 126.63 Mb of petrol and diesel, respectively, from 2020 to 2040. It can also positively affect global warming by recovering 56.78 10^{11} and 64.65 10^{11} mJ of heat released into the environment, which prevented 3.648 10^{10} m^3 of ice to be melted in the ocean. This would have resulted in the sea level rising, which could have led to natural disasters.

In fact, waste energy recovery also benefits the environment by reducing the amount of CO_2 emissions by 2.75 thousand tonnes during an active period of five years compared to traditional ICEs.

5.2. Recommendations

There are a lot of possibilities for future work. This research only focused on a theoretical approach of an integrated waste heat recovery system based on a Rankine cycle just behind the exhaust valves of an ICE. Similar work can be done by combining the exhaust waste heat recovery system and the cooling recovery system, to improve the engine efficiency and environment effects further more.

The most interesting possibility would be to apply the EWHR system on a practical case and on a newly designed engine vehicle.

REFERENCES

Arapatsakos, Charalampos, Anastasios Karkanis, Stella Maria Strofylla, Management Engineering, and V Sofias Street. 2012. "The Effect of Temperature on Gas Emissions 1,2,3" 11 (April): 89–100.

Arias, D., Shedd, T., and Jester, R. 2006. "Theoretical Analysis of Waste Heat Recovery from an Internal Combustion Engine in a Hybrid Vehicle." *SAE Technical Paper*, 01–1605.

Boloy, Ronney Arismel Mancebo, Jose Luz Silveira, Celso Eduardo Tuna, Christian R Coronado, and Julio Santana Antunes. 2011. "Ecological Impacts from Syngas Burning in Internal Combustion Engine: Technical and Economic Aspects." *Renewable and Sustainable Energy Reviews* 15 (9): 5194–5201. doi:http://dx.doi.org/10.1016/j.rser.2011.04.009.

Breu, FX, S Guggenbichler, and JC Wollmann. 2008. "Waste Heat Recovery: Technology and Opportunities in U.S." *Vasa*.

Chen, Huijuan, D. Yogi Goswami, and Elias K. Stefanakos. 2010. "A Review of Thermodynamic Cycles and Working Fluids for the Conversion of Low-Grade Heat." *Renewable and Sustainable Energy Reviews* 14 (9). Elsevier Ltd: 3059–67. doi:10.1016/j.rser.2010.07.006.

Claus Borgnakke, Richard E. Sonntag. 2013. *Fundamentals of Thermodynamics, 8th Edition SI Version -. John Wiley & Sons, Inc.*

Conklin, James C., and James P. Szybist. 2010. "A Highly Efficient Six-Stroke Internal Combustion Engine Cycle with Water Injection for in-Cylinder Exhaust Heat Recovery." *Energy* 35 (4). Elsevier Ltd: 1658–64. doi:10.1016/j.energy.2009.12.012.

Czerwinski, Jan, Ronny Winkelmann, and Jean-Paul Morin. 2012. "Toxicity of Exhaust Gases and Particles from IC-Engines – International Activities Survey (EngToxIn)*)." www.iea-amf.org/app/webroot/files/.../Annex Reports/AMF_Annex_42-2012.pdf.

DieselNet. 2012. "Emission Standards: Summary of Worldwide Emission Standards," 469–86. www.dieselnet.com/standards/.

Domingues, António, Helder Santos, and Mário Costa. 2013. "Analysis of Vehicle Exhaust Waste Heat Recovery Potential Using a Rankine Cycle." *Energy* 49: 71–85.

Don Hofstrand. 2008. "Liquid Fuel Measurements and Conversions." *Iowa State University*. http://www.extension.iastate.edu/agdm.

EIA. 2013. *International Energy Outlook 2013*. Report No.:DOE/EIA-0484(2013), U.S. Energy Information Administration, Washington, pp1. doi:U.S. Energy Information Administration.

———. 2014. *International Energy Outlook 2014*. Report No.:DOE/EIA-0484(2014), U.S. Energy Information Administration, Washington, pp4.

Endo, T., S. Kawajiri, Y. Kojima, K. Takahashi, T. Baba, S. Ibaraki, T. Takahashi, and M. Shinohara. 2007. "Study on Maximizing Exergy in Automotive Engines." *SAE*

Technical Paper, April, 01–0257.

European comission. 2012. “Amendment No 459/2012 as Regards Emissions from Light Passenger and Commercial Vehicles (Euro 6).” *Official Journal of the European Union* 2012 (459/2012): 16–24. doi:10.3000/19770677.L_2012.142.eng.

ExxonMobil. 2013. “The Outlook for Energy: A View to 2040.” Exxon Mobil Corporation, Irving, Texas 75039-2298. exxonmobil.com/energyoutlook.

GREET. 2011. “Argonne GREET Model.” In *Biomass Energy Data Book*, Appendix.

Gu, W, Y Weng, Y Wang, and B Zheng. 2009. “Theoretical and Experimental Investigation of an Organic Rankine Cycle for a Waste Heat Recovery System.” *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy* 223 (5): 523–33.

Hatazawa, Masayasu, Hiroshi Sugita, Takahiro Ogawa, and Yoshitoki Seo. 2004. “Performance of a Thermoacoustic Sound Wave Generator Driven with Waste Heat of Automobile Gasoline Engine.” *Nippon Kikai Gakkai Ronbunshu, B Hen/Transactions of the Japan Society of Mechanical Engineers, Part B* 70 (689): 292–99.

He, Maogang, Xinxin Zhang, Ke Zeng, and Ke Gao. 2011. “A Combined Thermodynamic Cycle Used for Waste Heat Recovery of Internal Combustion Engine.” *Energy* 36 (12). Elsevier Ltd: 6821–29.

Hendricks, Terry J, and Jason A Lustbader. 2002a. “Advanced Thermoelectric Power System Investigations for Light-Duty / Heavy-Duty Vehicle Applications - Presentation,” TBD. z:%5CArticles%5CFossil.Fuel.Applications%5CHendricks-ICT-2001c.pdf.

Hendricks and Lustbader. 2002b. “Advanced Thermoelectric Power System Investigations for Light-Duty and Heavy Duty Applications: Part I.” *21st International Conference on Thermoelectronics,* 381–86.

Heywood, John B. 1988. *Internal Combustion Engine Fundamentals*. McGraw-Hil. New York : McGraw-Hill.

Jadhao, J S, D G Thombare, Post-Graduate Student, and Dist Sangali. 2013. “Review on Exhaust Gas Heat Recovery for I . C . Engine.” *Ternational Journal of Engineering and Innovative Technology (IJEIT)* 2 (12): 93–100.

Jordan Hanania, Braden Heffernan, James Jenden, Kailyn Stenhouse, Jason Donev. 2015. “Thermal Efficiency - Energy Education.” *Energy Education*. http://energyeducation.ca/encyclopedia/Thermal_efficiency.

Marguerite Richardson and Agnes Phahlane, Department of Environmental Affairs. 2009. “State of Air Report 2005 A Report on the State of the Air in South Africa.” Pretoria, 0001, Republic of South Africa. https://www.environment.gov.za/sites/default/.../stateofair_executive_introduction.pdf.

Mary, Arlotto Mike, Millikin. 2005. “Green Car Congress: DOE Co-Funds 12 Projects to Increase Engine Efficiency.” *Green Car Congress*. http://www.greencarcongress.com/2005/02/doe_cofunds_12_.html.

Mei, Vincent, Sushil K Chaturvedi, and Zalman Lavan. 1979. “Truck Exhaust Gas-Operated

Absorption Refrigeration System." *ASHRAE Transactions* 85 (Pt 2): 66–76.

OPEC 2014. n.d. *World Oil Outlook 2014*. Organization of the Petroleum Exporting Countries, Vienna, Austria, pp89.

Outlook, The. 2014. "The Outlook for Energy: A View to 2040 2014," no. May: 1–32.

Peng, Zhijun, Tianyou Wang, Yongling He, Xiaoyi Yang, and Lipeng Lu. 2013. "Analysis of Environmental and Economic Benefits of Integrated Exhaust Energy Recovery (EER) for Vehicles." *Applied Energy* 105 (0): 238–43.

Plaza, Nedbank, and Beatrix Streets. 2015. "National Association of Automobile Manufacturers of South Africa." *Passenger Market*. http://www.naamsa.co.za/flash/pas.html.

Pulkrabek, Willard W. 2003. *Engineering Fundamentals of the Internal Combustion Engine* -. Prentice Hall PTR.

Pulkrabek, Willard W. 2004. *Engineering Fundamentals of the Internal Combustion Engine, 2nd Ed. Journal of Engineering for Gas Turbines and Power*. Vol. 126.

Ringler, J., M. Seifert, V. Guyotot, and W. Hübner. 2009. "Rankine Cycle for Waste Heat Recovery of IC Engines." doi:10.4271/2009-01-0174.

Saidur, R., M. Rezaei, W.K. Muzammil, M.H. Hassan, S. Paria, and M. Hasanuzzaman. 2012. "Technologies to Recover Exhaust Heat from Internal Combustion Engines." *Renewable and Sustainable Energy Reviews* 16 (8). Elsevier: 5649–59.

Sharifpur, Mohsen. 2008. "Designing New Cooling System for Automobiles to Get More Fuel Efficiency and Less Environment Defects." In *Volume 17: Transportation Systems*, 355–59. ASME. doi:10.1115/IMECE2008-68413.

Sousanis, John. 2011. "World Vehicle Population Tops 1 Billion Units." *WardsAuto*. http://wardsauto.com/ar/world_vehicle_population_110815.

Stonntag, Claus Borgnakke; Richard. 2009. *Fundamentals of Thermodynamics*. 7th ed. John Willey & Sons, Inc.

Talom, Hugues L, and Asfaw Beyene. 2009. "Heat Recovery from Automotive Engine." *Applied Thermal Engineering* 29 (2–3): 439–44.

Tian, Hua, Gequn Shu, Haiqiao Wei, Xingyu Liang, and Lina Liu. 2012. "Fluids and Parameters Optimization for the Organic Rankine Cycles (ORCs) Used in Exhaust Heat Recovery of Internal Combustion Engine (ICE)." *Energy* 47 (1): 125–36.

Vaja, Iacopo, and Agostino Gambarotta. 2010. "Internal Combustion Engine (ICE) Bottoming with Organic Rankine Cycles (ORCs)." *Energy* 35 (2). Elsevier Ltd: 1084–93.

Van Basshuysen, Richard. 2004. *Internal Combustion Engine Handbook: Basics, Components, Systems, and Perspectives*. Edited by Richard van Basshuysen and Fred Schäfer. SAE International,.

VOLKSWAGEN AG. 2000. "Motor Vehicle Exhaust Emissions, Programme, Self-Study." Wolfsburg.

Wang, Tianyou, Yajun Zhang, Jie Zhang, Gequn Shu, and Zhijun Peng. 2013. "Analysis of Recoverable Exhaust Energy from a Light-Duty Gasoline Engine." *Applied Thermal Engineering* 53 (2). Elsevier Ltd: 414–19.

Weerasinghe, W.M.S.R., R K Stobart, and S M Hounsham. 2010. "Thermal Efficiency Improvement in High Output Diesel Engines a Comparison of a Rankine Cycle with Turbo-Compounding." *Applied Thermal Engineering* 30 (14–15): 2253–56.

Yu, Chuang, and K.T. Chau. 2009. "Thermoelectric Automotive Waste Heat Energy Recovery Using Maximum Power Point Tracking." *Energy Conversion and Management* 50 (6). Elsevier Ltd: 1506–12.

Appendix: Fluid property information

Steam, Steam_NBS, water, R718, and ice

The thermodynamic properties of water have been implemented using the thermodynamic property correlation of Harr, Gallagher, and Kell, 1984 (*NBS/NRC steam tables*, Hemisphere Publishing Co.). The correlations are valid up to a pressure of 815 bar. This formulation is now slightly outdated, since a more recent formulation for the properties of water that extends to pressures of 1 000 MPa is available in the substance Steam_IAPWS.

EES will return the properties of ice at temperatures below 0 °C and pressures above the saturation vapour pressure of ice based on the property information of Hyland and Wexler, 1983 (*Formulations for the thermodynamic properties of the saturated phases of H_2O from 173.15 K to 473.15 K*, ASHRAE Trans., Part 2A, Paper 2 793). The isothermal compressibility of ice is from Equation 22 of Hyland and Wexler, 1983 (*Formulations for the thermodynamic properties of dry air from 173.15 to 373.15 and saturated moist air from 173.15 to 372.15 K at pressure to MPa*, ASHRAE Trans., Part 2A, Paper 2 794).

Enthalpy and entropy values refer to 0 for saturated liquid at 0 °C.

Note that the substances steam, Steam_NBS, water, R718 and ice all use the same correlations for water in the solid, liquid and vapour regions and provide exactly the same properties. Earlier versions of EES used the less accurate correlations for the substances water, steam and R718. The substance ice has been implemented to allow preference to ice properties in the calculations when a specific volume is provided as one of the inputs. In this case, there may be two solutions.

The transport property correlations are from Electrical Research Association, 1967 (*Steam tables, thermodynamic properties of water and Steam; viscosity of water and steam, thermal conductivity of water and steam*, Edward Arnold Publishers, London). The transport functions are valid for temperatures between 273.15 K and 815 K at pressures up to 830 bars.

The thermal conductivity of solid ice is provided by a correlation from Willard Coles, 1954 (Experimental determination of the thermal conductivity of low-density ice, National Advisory Committee for Aeronautics, Technical Note, 3 143, Lewis Flight Propulsion Laboratory, Cleveland, OH).

Steam_IAPWS

Steam_IAPWS implements the high-accuracy thermodynamic properties of water with the 1995 *Formulations for the thermodynamic properties of ordinary water substance for general and scientific use* issued by The International Association for the Properties of Water and Steam (IAPWS). This correlation replaced the 1984 formulation of Haar, Gallagher and Kell (*NBS/NRC steam tables*, Hemisphere Publishing Co.), which is implemented in the substance steam. The new formulation is based on the correlations of Saul and Wagner, 1087 (*Journal of Physical and Chemical Reference Data*, 16, 893) with modifications to adjust to the International Temperature Scale of 1990. The modifications are described by Wagner and Pruss, 1993 (*Journal of Physical and Chemical Reference Data*, 22, 783).

This correlation provides accurate results for temperatures between 273.15 K and 1273.15 K at pressures up to 1 000 MPa. The formulation allows extrapolation of properties to 5 000 K. Steam_IAPWS is only available in the professional version.

Water also provides steam properties, but it uses less accurate correlations that require significantly less computational effort. Use steam or water for the properties of ice at temperatures below 0 ˚C based on the ice property information of Hyland and Wexler, 1983 (*Formulations for the thermodynamic properties of the saturated phases of H_2O from 173.15 K to 473.15 K*, ASHRAE Trans., Part 2A, Paper 2 793).

Enthalpy and entropy values refer to 0 for saturated liquid at 0 ˚C.

The thermal conductivity, viscosity and surface tension correlations for Steam_IAPWS are from Kestin, Sengers, Kangmar-Parsi and Levelt Sengers, 1984 (“Thermophysical properties of fluid H_2O”, *Journal of Physical and Chemical Reference Data*, 13, 175. The range of applicability for these transport correlations is the same as for the thermodynamic properties.

5.3. Appendix B: EES code – Thermodynamic analysis of exhaust gas composition

```
{$DS.}" EXHAUST GAS SPECIFICATIONS"

"Knowns"

V_Tot=0.0013 " Total volume of the 4 cylinders"
N= 5600 "Flyweel rotation"
"P_exh= 500 Exhaust gas pressure varies between 400 and 500kPa"
"T_exh=873.2 Exhaust gas temperature ranges between 400 to 600 oC"
T_exh=300 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=20  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
"m_dot_Tot_mix=0.1154  Exhaust gas mass flow rate"
"C_p_mix=1.258  Exhaust gas constant-pressure specific heat"

" Computations"
" CONSTANT SPECIFIC  HEAT OF THE MIXTURE"
" C_p_0=C_0 + C_1*tTETA + C_2*TETA^2 + C_3*TETA^3"
theta=(273.2 +T_exh)/1000
```

```
" For N2"

C_p_N2=C_0_N2 + C_1_N2*theta + C_2_N2*theta^2 + C_3_N2*theta^3
C_0_N2=1.11
C_1_N2=-0.48
C_2_N2=0.96
C_3_N2=-0.42

" For H2O"

C_p_H2O=C_0_H2O + C_1_H2O*theta + C_2_H2O*theta^2 + C_3_H2O*theta^3

C_0_H2O= 1.79
C_1_H2O=0.107
C_2_H2O=0.586
C_3_H2O=0.20

" For CO2"

C_p_CO2=C_0_CO2 + C_1_CO2*theta + C_2_CO2*theta^2 + C_3_CO2*theta^3

C_0_CO2=0.45
C_1_CO2=1.67
C_2_CO2=-1.27
C_3_CO2=0.39

" C_p_mix"

Alpha_N2=0.71
Alpha_H2O=0.13
Alpha_CO2=0.14

C_p_mix= Alpha_N2*C_p_N2 + Alpha_H2O*C_p_H2O + Alpha_CO2*C_p_CO2
```

```
" DENSITY OF THE EXHAUST MIXTURE"

"1 : Determination of pressures"

P_N2=Alpha_N2*P_exh
P_H2O=Alpha_H2O*P_exh
P_CO2=Alpha_CO2*P_exh

"2: Density"

R_N2=0.2968
R_H2O=0.4615
R_CO2=0.1889

Rho_N2=P_N2/(R_N2*T_exh)
Rho_H2O=P_H2O/(R_H2O*T_exh)
Rho_CO2=P_CO2/(R_CO2*T_exh)
Rho_mix= Alpha_N2*Rho_N2 + Alpha_H2O*Rho_H2O + Alpha_CO2*Rho_CO2

 "EXHAUST MASS FLOW RATE DETERMINATION"

V_cyl=V_Tot/4
V_dot_mix_cyl=V_cyl * N/60
m_dot_mix_cyl=V_dot_mix_cyl*Rho_mix
m_dot_Tot_mix=m_dot_mix_cyl*4

"HEAT AVAILABLE IN THE EXHAUST MISTURE"

T_amb=20
Q_dot_av=m_dot_Tot_mix*C_p_mix*(T_exh - T_amb) "Rate of heat transfer"
Q_dot_in_w=Q_dot_av*0.95

"WORKING FLUID MASS FLOW RATE DETERMINATION"
```

```
T_c_out= 280
T_c_in=30
C_p_c_w=4.178

C_c_w=Q_dot_in_w/(T_c_out - T_c_in)
m_dot_c_w=C_c_w/C_p_c_w

"Rankine Cycle - Config-Water"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"

" Control volume :Turbine"

" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480

"Exist state: T4 and x4 known; state fixed"

T_4=60
x_4=0.9
P_4=Pressure(Water,T=T_4,x=x_4)
h_4=Enthalpy(Water,T=T_4,x=x_4)
s_4=Entropy(Water,T=T_4,x=x_4)
v_4=Volume(Water,T=T_4,x=x_4)
```

```
" Analysis"

s_3=s_4
x_3=1
P_33=Pressure(Water,x=x_3,s=s_3)
P_3=Pressure(Water,T=T_3,s=s_3)
h_3=Enthalpy(Water,T=T_3,s=s_3)
v_3=Volume(Water,T=T_3,s=s_3)
w_t=h_3 - h_4

" Control volume :Pump"

" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(Water,P=P_1,x=x_1)
s_1=Entropy(Water,P=P_1,x=x_1)
T_1=Temperature(Water,P=P_1,x=x_1)
v_1=Volume(Water,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Water,P=P_2,s=s_2)
h_2=Enthalpy(Water,P=P_2,s=s_2)
v_2=Volume(Water,T=T_2,s=s_2)

" Analyse"

w_p=v_1 * (P_2 - P_1)
"h_22=w_p + h_1"
```

```
" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
W_dot_t=m_dot_w*w_t
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
W_dot_net=w_net*m_dot_c_w
ETA_1=(W_dot_net/Q_dot_in_w)*100
```

5.4. Appendix C: EES code – EWHR with water as working fluid

```
{$DS.}"Rankine Cycle - Config-Water"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
```

(3) Neglecting also the heat lost from each component and pipe system
(4) Isentropic turbine efficiency = 0.7
(5) Insentropic pump efficiency = 0.8
(6) Evaporating pressure varying between condensation and critical pressure
(7) water as working fluid"

```
"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"
" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480
ETA_t=0.90

"Exist state: T4 and x4 known; state fixed"

T_4=60
x_4=0.9
P_4=Pressure(Water,T=T_4,x=x_4)
h_4=Enthalpy(Water,T=T_4,x=x_4)
s_4=Entropy(Water,T=T_4,x=x_4)
v_4=Volume(Water,T=T_4,x=x_4)
```

```
" Analysis"

s_3=s_4
x_3=1
P_33=Pressure(Water,x=x_3,s=s_3)
P_3=Pressure(Water,T=T_3,s=s_3)
h_3=Enthalpy(Water,T=T_3,s=s_3)
v_3=Volume(Water,T=T_3,s=s_3)

"h_4=h_3 - (h_3 - h_4s)*ETA_t"
w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"

ETA_p=0.95

" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(Water,P=P_1,x=x_1)
s_1=Entropy(Water,P=P_1,x=x_1)
T_1=Temperature(Water,P=P_1,x=x_1)
v_1=Volume(Water,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Water,P=P_2,s=s_2)
h_2=Enthalpy(Water,P=P_2,s=s_2)
v_2=Volume(Water,T=T_2,s=s_2)
```

" Analyse"

w_p=(v_1 * (P_2 - P_1))/ETA_p

"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"

"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"

"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2

m_dot_w=Q_dot_in_w/q_in_w

W_dot_t=m_dot_w*w_t

w_net=w_t - w_p

ETA_th=(w_net/q_in_w)*100

5.5. Appendix D: EWHR with R-718 as working fluid

{$DS.}"Rankine Cycle - Config-R-718"

" General Assumptions"

```
" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480
PC=P_crit(R718)
TC=T_crit(R718)
P_3=PC

h_3=Enthalpy(R718,T=T_3,P=P_3)
s_3=Entropy(R718,T=T_3,P=P_3)
v_3=Volume(R718,T=T_3,P=P_3)

"Exist state: T4 and s4 known; state fixed"
```

```
T_4=60 "case 1"
"x_4=1"

" Analyse"

s_3=s_4
P_4=Pressure(R718,T=T_4,s=s_4)
h_4=Enthalpy(R718,T=T_4,s=s_4)
x_4=Quality(R718,T=T_4,s=s_4) "case 1"
v_4=Volume(R718,T=T_4,s=s_4)

{P_4=Pressure(R718,s=s_4,x=x_4)
T=Temperature(R718,s=s_4,x=x_4)
h_4=Enthalpy(R718,s=s_4,x=x_4)
v_4=Volume(R718,s=s_4,x=x_4)}

w_t=h_3 - h_4

" Control volume :Pump"

" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(R718,P=P_1,x=x_1)
s_1=Entropy(R718,P=P_1,x=x_1)
T_1=Temperature(R718,P=P_1,x=x_1)
v_1=Volume(R718,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
```

```
h_2=Enthalpy(R718,P=P_2,s=s_2)
T_2=Temperature(R718,P=P_2,s=s_2)
v_2=Volume(R718,T=T_2,s=s_2)

" Analyse"

w_p=v_1 * (P_2 - P_1)

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.6. Appendix E: EWHR with Steam_NBS as working fluid

```
{$DS.}"Rankine Cycle - Steam_NBS"
```

```
" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480

"Exist state: T4 and x4 known; state fixed"

T_4=60
x_4=0.9
```

```
P_4=Pressure(Steam_NBS,T=T_4,x=x_4)
h_4=Enthalpy(Steam_NBS,T=T_4,x=x_4)
s_4=Entropy(Steam_NBS,T=T_4,x=x_4)

" Analyse"

s_3=s_4
P_3=Pressure(Steam_NBS,T=T_3,s=s_3)
h_3=Enthalpy(Steam_NBS,T=T_3,s=s_3)
w_t=h_3 - h_4

" Control volume :Pump"

" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(Steam_NBS,P=P_1,x=x_1)
s_1=Entropy(Steam_NBS,P=P_1,x=x_1)
T_1=Temperature(Steam_NBS,P=P_1,x=x_1)
v_1=Volume(Steam_NBS,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Steam_NBS,P=P_2,s=s_2)
h_2=Enthalpy(Steam_NBS,P=P_2,s=s_2)

" Analyse"

w_p=v_1 * (P_2 - P_1)
"h_22=w_p + h_1"
```

```
" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.7. Appendix F: EWHR with Steam as working fluid

```
{$DS.}"Rankine Cycle - Config-Steam"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
```

(7) water as working fluid"

"Given Exhaust gas"

```
T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480

"Exist state: T4 and x4 known; state fixed"

T_4=60
x_4=0.9
P_4=Pressure(Steam,T=T_4,x=x_4)
h_4=Enthalpy(Steam,T=T_4,x=x_4)
s_4=Entropy(Steam,T=T_4,x=x_4)

" Analyse"

s_3=s_4
P_3=Pressure(Steam,T=T_3,s=s_3)
```

```
h_3=Enthalpy(Steam,T=T_3,s=s_3)
w_t=h_3 - h_4

" Control volume :Pump"

" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(Steam,P=P_1,x=x_1)
s_1=Entropy(Steam,P=P_1,x=x_1)
T_1=Temperature(Steam,P=P_1,x=x_1)
v_1=Volume(Steam,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Steam,P=P_2,s=s_2)
h_2=Enthalpy(Steam,P=P_2,s=s_2)

" Analyse"

w_p=v_1 * (P_2 - P_1)
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1
```

```
" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.8. Appendix G: EWHR with Steam_IAPWS as working fluid

```
{$DS.}"Rankine Cycle - Steam_IAPWS"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
```

```
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

" Inlet sate: T3 know, and S3 as calculated; state fixed"

T_3=480

"Exist state: T4 and x4 known; state fixed"

T_4=60
x_4=0.9
P_4=Pressure(Steam_IAPWS,T=T_4,x=x_4)
h_4=Enthalpy(Steam_IAPWS,T=T_4,x=x_4)
s_4=Entropy(Steam_IAPWS,T=T_4,x=x_4)
v_4=Volume(Steam_IAPWS,T=T_4,x=x_4)

" Analyse"

s_3=s_4
P_3=Pressure(Steam_IAPWS,T=T_3,s=s_3)
h_3=Enthalpy(Steam_IAPWS,T=T_3,s=s_3)
v_3=Volume(Steam_IAPWS,T=T_3,s=s_3)
w_t=h_3 - h_4

" Control volume :Pump"

" Inlet sate: P1 know, saturated water; state fixed"
```

```
P_1=P_4
x_1=0
h_1=Enthalpy(Steam_IAPWS,P=P_1,x=x_1)
s_1=Entropy(Steam_IAPWS,P=P_1,x=x_1)
T_1=Temperature(Steam_IAPWS,P=P_1,x=x_1)
v_1=Volume(Steam_IAPWS,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Steam_IAPWS,P=P_2,s=s_2)
h_2=Enthalpy(Steam_IAPWS,P=P_2,s=s_2)
v_2=Volume(Steam_IAPWS,T=T_2,s=s_2)

" Analyse"

w_p=v_1 * (P_2 - P_1)
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"
```

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w)*100

5.9. Appendix H: EWHR with R-11 as working fluid

{$DS.}"Rankine Cycle - Config-R11"

" General Assumptions"

" (1) Let consider all devices as steeady - state steady - flow system
 (2) Neglecting kinectic and potential engeries
 (3) Neglecting also the heat lost from each component and pipe system
 (4) Isentropic turbine efficiency = 0.7
 (5) Insentropic pump efficiency = 0.8
 (6) Evaporating pressure varying between condensation and critical pressure
 (7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40 " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258 "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

```
Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"

Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

"Inlet: P3  known"
"P_3=4405"
PC=P_crit(R11)
TC=T_crit(R11)
P_3=PC
ETA_t=0.90
"Outlet : P4 and X4 known"

P_4=90
x_4=0.9
h_4=Enthalpy(R11,P=P_4,x=x_4)
s_4=Entropy(R11,P=P_4,x=x_4)
T_4=Temperature(R11,P=P_4,x=x_4)
v_4=Volume(R11,P=P_4,x=x_4)

"Analysis"

s_3=s_4
T_3=Temperature(R11,P=P_3,h=h_3)
h_3=Enthalpy(R11,P=P_3,s=s_3)
v_3=Volume(R11,P=P_3,s=s_3)

w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"
ETA_p=0.95
" Inlet sate: P1 know, saturated water; state fixed"
```

```
P_1=P_4
x_1=0
h_1=Enthalpy(R11,P=P_1,x=x_1)
s_1=Entropy(R11,P=P_1,x=x_1)
T_1=Temperature(R11,P=P_1,x=x_1)
v_1=Volume(R11,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(R11,P=P_2,h=h_2)
h_2=Enthalpy(R11,P=P_2,s=s_2) "Properties for R11 in the subcooled region have been estimated assuming the fluid to be incompressible."

v_2=Volume(R11,T=T_2,s=s_2)

" Analysis"

w_p=(v_1 * (P_2 - P_1))/ETA_p
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"
```

```
"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.10. Appendix I: EWHR with R-134a as working fluid

```
{$DS.}"Rankine Cycle - Config-R134a"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"
C_p_mix=1.258  "Exhaust gas constant-pressure specific heat"
```

```
"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

"Inlet: P3  known"
"P_3=3984"
PC=P_crit(R134a)
P_3=PC
ETA_t=0.90
"Outlet : T4 and X4 known"

T_4=20
x_4=0.9
h_4=Enthalpy(R134a,T=T_4,x=x_4)
s_4=Entropy(R134a,T=T_4,x=x_4)
P_4=Pressure(R134a,T=T_4,x=x_4)
v_4=Volume(R134a,T=T_4,x=x_4)

"Analysis"

s_3=s_4
T_3=Temperature(R134a,P=P_3,v=v_3)
h_3=Enthalpy(R134a,P=P_3,s=s_3)
v_3=Volume(R134a,P=P_3,s=s_3)

w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"
ETA_p=0.95

" Inlet sate: P1 know, saturated water; state fixed"
```

```
P_1=P_4
x_1=0
h_1=Enthalpy(R134a,P=P_1,x=x_1)
s_1=Entropy(R134a,P=P_1,x=x_1)
T_1=Temperature(R134a,P=P_1,x=x_1)
v_1=Volume(R134a,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(R134a,P=P_2,h=h_2)
h_2=Enthalpy(R134a,P=P_2,s=s_2)
v_2=Volume(R134a,T=T_2,s=s_2)

" Analysis"

w_p=(v_1 * (P_2 - P_1))/ETA_p
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"
```

```
"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.11. Appendix J: EWHR with R-12 as working fluid

```
{$DS.}"Rankine Cycle - Config-R12"

" General Assumptions"

" (1) Let consider  all devices as steeady - state steady - flow system
  (2) Neglecting kinectic and potential engeries
  (3) Neglecting also the heat lost from each component and pipe system
  (4) Isentropic turbine efficiency = 0.7
  (5) Insentropic pump efficiency = 0.8
  (6) Evaporating pressure varying between condensation and critical pressure
  (7) water as working fluid"


"Given Exhaust gas"

T_exh=350 "Exhaust gas temperature ranges between 400 to 600 oC"
T_g_in=T_exh  " Inlet exhaust gas Temperature at the heat exchanger"
T_g_out=40  " Outlet exhaust gas Temperature at the heat exchanger 40 oC"
P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"
m_dot_Tot_mix=0.1258 " Exhaust gas mass flow rate"
C_p_mix=1.150  "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"
```

```
Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"
Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

"Inlet: P3  known"
"P_3=4060"
PC=P_crit(R12)
P_3=PC
ETA_t=0.90
"Outlet : T4 and X4 known"

T_4=20
x_4=0.9
h_4=Enthalpy(R12,T=T_4,x=x_4)
s_4=Entropy(R12,T=T_4,x=x_4)
P_4=Pressure(R12,T=T_4,x=x_4)
v_4=Volume(R12,T=T_4,x=x_4)

"Analysis"

s_3=s_4
T_3=Temperature(R12,P=P_3,s=s_3)
h_3=Enthalpy(R12,P=P_3,s=s_3)
v_3=Volume(R12,P=P_3,s=s_3)

w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"
ETA_p=0.95
" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
```

```
x_1=0
h_1=Enthalpy(R12,P=P_1,x=x_1)
s_1=Entropy(R12,P=P_1,x=x_1)
T_1=Temperature(R12,P=P_1,x=x_1)
v_1=Volume(R12,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(R12,P=P_2,h=h_2)
h_2=Enthalpy(R12,P=P_2,s=s_2) "Properties for R12 in the subcooled region have been estimated assuming the fluid to be incompressible."
v_2=Volume(R12,T=T_2,s=s_2)

" Analysis"
w_p=(v_1 * (P_2 - P_1))/ETA_p
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"
```

q_in_w = h_3 - h_2

m_dot_w=Q_dot_in_w/q_in_w

w_net=w_t - w_p

ETA=(w_net/q_in_w)*100

5.12. Appendix K: EWHR with R-717 as working fluid

{$DS.}"Rankine Cycle - Config-R717"

" General Assumptions"

" (1) Let consider all devices as steeady - state steady - flow system

(2) Neglecting kinectic and potential engeries

(3) Neglecting also the heat lost from each component and pipe system

(4) Isentropic turbine efficiency = 0.7

(5) Insentropic pump efficiency = 0.8

(6) Evaporating pressure varying between condensation and critical pressure

(7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"

T_g_in=T_exh " Inlet exhaust gas Temperature at the heat exchanger"

T_g_out=40 " Outlet exhaust gas Temperature at the heat exchanger 40 oC"

P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"

m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"

C_p_mix=1.258 "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"

Q_dot_in_w=Q_dot_HEX*0.95

```
" Control volume :Turbine"

"Inlet: P3  known"
"P_3=11166"
PC=P_crit(R717)
P_3=PC
ETA_t=0.90
"Outlet : P4 and X4 known"

P_4=760
x_4=0.9
h_4=Enthalpy(R717,P=P_4,x=x_4)
s_4=Entropy(R717,P=P_4,x=x_4)
T_4=Temperature(R717,P=P_4,x=x_4)
v_4=Volume(R717,P=P_4,x=x_4)

"Analysis"

s_3=s_4
T_3=Temperature(R717,P=P_3,h=h_3)
h_3=Enthalpy(R717,P=P_3,s=s_3)
v_3=Volume(R717,P=P_3,s=s_3)

w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"
ETA_p=0.95
" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(R717,P=P_1,x=x_1)
s_1=Entropy(R717,P=P_1,x=x_1)
```

```
T_1=Temperature(R717,P=P_1,x=x_1)
v_1=Volume(R717,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(R717,P=P_2,h=h_2)
h_2=Enthalpy(R717,P=P_2,s=s_2)
v_2=Volume(R717,T=T_2,s=s_2)

" Analysis"

w_p=(v_1 * (P_2 - P_1))/ETA_p
"h_22=w_p + h_1"

" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
m_dot_w=Q_dot_in_w/q_in_w
```

w_net=w_t - w_p

ETA=(w_net/q_in_w)*100

5.13. Appendix L: EWHR with Ammonia as working fluid

{$DS.}"Rankine Cycle - Config-Ammonia"

" General Assumptions"

" (1) Let consider all devices as steeady - state steady - flow system

(2) Neglecting kinectic and potential engeries

(3) Neglecting also the heat lost from each component and pipe system

(4) Isentropic turbine efficiency = 0.7

(5) Insentropic pump efficiency = 0.8

(6) Evaporating pressure varying between condensation and critical pressure

(7) water as working fluid"

"Given Exhaust gas"

T_exh=500 "Exhaust gas temperature ranges between 400 to 600 oC"

T_g_in=T_exh " Inlet exhaust gas Temperature at the heat exchanger"

T_g_out=40 " Outlet exhaust gas Temperature at the heat exchanger 40 oC"

P_exh= 400 "Exhaust gas pressure varies between 400 and 500kPa"

m_dot_Tot_mix=0.1154 " Exhaust gas mass flow rate"

C_p_mix=1.258 "Exhaust gas constant-pressure specific heat"

"Heat Exchanger"

Q_dot_HEX=m_dot_Tot_mix*C_p_mix*(T_g_in - T_g_out) "Rate of heat transfer"

Q_dot_in_w=Q_dot_HEX*0.95

" Control volume :Turbine"

```
"Inlet: T3  known"
PC=P_crit(Ammonia)
P_3=PC
ETA_t=0.90
"P_3=11238"
"T_3=200"

"Outlet : P4 and X4 known"

P_4=746.6
x_4=0.9
h_4=Enthalpy(Ammonia,P=P_4,x=x_4)
s_4=Entropy(Ammonia,P=P_4,x=x_4)
T_4=Temperature(Ammonia,P=P_4,x=x_4)
v_4=Volume(Ammonia,P=P_4,x=x_4)



"Analysis"

s_3=s_4
T_3=Temperature(Ammonia,P=P_3,v=v_3)
h_3=Enthalpy(Ammonia,P=P_3,s=s_3)
v_3=Volume(Ammonia,P=P_3,s=s_3)

w_t=(h_3 - h_4)*ETA_t

" Control volume :Pump"
ETA_p=0.95
" Inlet sate: P1 know, saturated water; state fixed"

P_1=P_4
x_1=0
h_1=Enthalpy(Ammonia,P=P_1,x=x_1)
s_1=Entropy(Ammonia,P=P_1,x=x_1)
```

```
T_1=Temperature(Ammonia,P=P_1,x=x_1)
v_1=Volume(Ammonia,P=P_1,x=x_1)

"Exist state: P2 kown and S2; state fixed"

P_2=P_3
s_2=s_1
T_2=Temperature(Ammonia,P=P_2,h=h_2)
h_2=Enthalpy(Ammonia,P=P_2,s=s_2)
v_2=Volume(Ammonia,T=T_2,s=s_2)

" Analyse"

w_p=(v_1 * (P_2 - P_1))/ETA_p
"h_22=w_p + h_1"


" Control Volume: CONDENSER"

"Inlet sate: P4, h4 known; state fixed"
"Exist sate: P1, h1 known; state fixed"

"Analysis"

q_cond=h_4 - h_1

" Control Volume: HEAT EXCHANGER"

"Inlet sate: P2, h2 known; state fixed"
"Exit state: state 3 fixed(as given)"

"Analysis"

q_in_w = h_3 - h_2
```

```
m_dot_w=Q_dot_in_w/q_in_w
w_net=w_t - w_p
ETA=(w_net/q_in_w )*100
```

5.14. Appendix M: Commission regulation (EU) No 459/2012

ANNEX I

Amendments to Regulation (EC) No 715/2007

Annex I to Regulation (EC) No 715/2007 is amended as follows:

(1) the text in the second row of the last column of Table 1 (Euro 5 emission limits) is replaced by the following:

'Number of particles (PN)';

(2) Table 2 is replaced by the following table:

'*Table 2*

Euro 6 Emission Limits

			Limit values													
		Reference mass (RM) (kg)	Mass of carbon monoxide (CO)		Mass of total hydrocarbons (THC)		Mass of non-methane hydrocarbons (NMHC)		Mass of oxides of nitrogen (NO_x)		Combined mass of hydrocarbons and oxides of nitrogen (THC + NO_x)		Mass of particulate matter (PM) (1)		Number of particles (PN)	
			L_1 (mg/km)		L_2 (mg/km)		L_3 (mg/km)		L_4 (mg/km)		$L_2 + L_4$ (mg/km)		L_5 (mg/km)		L_6 (#/km)	
Category	Class		PI	CI	PI	CI	PI	CI	PI	CI	PI	CI	PI (2)	CI	PI (2) (3)	CI
M	—	All	1 000	500	100	—	68	—	60	80	—	170	4,5	4,5	$6{,}0 \times 10^{11}$	$6{,}0 \times 10^{11}$
N_1	I	RM ≤ 1 305	1 000	500	100	—	68	—	60	80	—	170	4,5	4,5	$6{,}0 \times 10^{11}$	$6{,}0 \times 10^{11}$
	II	1 305 < RM ≤ 1 760	1 810	630	130	—	90	—	75	105	—	195	4,5	4,5	$6{,}0 \times 10^{11}$	$6{,}0 \times 10^{11}$
	III	1 760 < RM	2 270	740	160	—	108	—	82	125	—	215	4,5	4,5	$6{,}0 \times 10^{11}$	$6{,}0 \times 10^{11}$
N_2	—	All	2 270	740	160	—	108	—	82	125	—	215	4,5	4,5	$6{,}0 \times 10^{11}$	$6{,}0 \times 10^{11}$

Key: PI = Positive Ignition, CI = Compression Ignition

(1) A limit of 5,0 mg/km for the mass of particulate emissions applies to vehicles type approved to the emission limits of this table with the previous particulate mass measurement protocol, before 1.9.2011.

(2) Positive ignition particulate mass and number limits shall apply only to vehicles with direct injection engines.

(3) Until three years after the dates specified in Article 10(4) and (5) for new type approvals and new vehicles respectively, a particle number emission limit of $6{,}0 \times 10^{12}$ #/km shall apply to Euro 6 PI direct injection vehicles upon the choice of the manufacturer. Until those dates at the latest a type approval test method ensuring the effective limitation of the number of particles emitted by vehicles under real driving conditions shall be implemented.'

5.15. Appendix N: Graphs' use permission

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