Experimental Investigation of Scroll Based Organic Rankine Systems

by

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Abstract

In this thesis, an experimental research is conducted on scroll-based Organic Rankine Cycle (ORC) focusing on the expansion process. An important feature of the ORC is the ability to utilize low or moderate temperature heat sources derived from renewable energy such as concentrated solar radiation, biomass/biofuels combustion streams, geothermal heat and waste heat recovery. The ORC is more appropriate than steam Rankine cycle to generate power from low capacity heat sources (5-500 kW thermal). For example, expansion of superheated steam from 280°C/1000 kPa to a pressure corresponding to 35°C saturation requires a volume ratio as high as 86, whereas for the same operating conditions toluene shows an expansion ratio of 6 which can be achieved in a single stage turbine or expander.

The objective of this work is to experimentally study the performance of a selected refrigeration scroll compressor operating in reverse as expander in an ORC. To this purpose, three experimental systems are designed, built and used for conducting a comprehensive experimental programme aimed at determining the features of the expansion process. In preliminary tests the working fluid utilized is dry air while the main experiments are done with the organic fluid R134a.

Experimental data of the scroll expander are collected under different operating conditions. Power generation in various conditions is analyzed in order to determine the optimum performance parameters for the scroll expander. In addition, thermodynamic analysis of the system is conducted through energy and exergy efficiencies to study the system performance.

Based on the experimental measurements, the optimum parameters for an ORC cycle operating with the Bitzer-based expander-generator unit are determined. The cycle energy and exergy efficiencies are found 5% and 30% respectively from a heat source of 120°C.

**Keywords:** Scroll compressor, expander, organic Rankine cycle, heat recovery, heat engine, energy, exergy, efficiency.
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Nomenclature

A  area, m$^2$

$c$  specific heat, kJ/kg.K

$c_p$  specific heat at constant pressure, J/kg.K

$c_v$  specific heat at constant volume, J/kg.K

$ex$  specific exergy, kJ/kg

$\dot{Ex}$  exergy rate, kW

$h$  specific enthalpy, kJ/kg

$I$  current, Amp

$m$  mass, kg

$m$  mass flow rate, kg/s

$\dot{Q}$  heat flow rate, J/s

$r_b$  radius of basic circle of the scroll, m

$r_o$  orbiting radius of rotating scroll, m

$s$  specific entropy, J/kg.K

$T$  temperature, °C

$TV$  throttle valve,

$U$  overall heat transfer coefficient, W/m$^2$.K

$V$  volume, m$^3$

$v$  specific volume, m$^3$/kg

$W$  work, kJ

$\dot{W}$  work rate, kW

Greek letters

$\varepsilon$  uncertainty

$\eta$  efficiency

$\Pi$  isochoric pressure building work ratio

$\phi$  angle, rad

$\psi$  exergy efficiency
ρ  density, Kg/m$^3$
θ  rotation angle, rad
ζ  leakage coefficient
ω  angular velocity, rad/s
μ  coefficient of friction
σ  Stefan-Boltzmann constant, W/m$^2$K$^4$

**Subscripts**

0  reference state
acc  acceleration
b  basic
bp  boiling point
bi  boiler in
bo  boiler out
c  condenser or compressor
co  compressor out
cog  cogeneration
cl  compressor low
crt  critical
e  expander
eb  expander body
ei  expander in
eo  expander out
eq  equivalent
h  high
I  current
l  low
liq  liquid
mix  mixture
mot  motor
<table>
<thead>
<tr>
<th>p</th>
<th>constant pressure or pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>sc</td>
<td>subcooled</td>
</tr>
<tr>
<td>vop</td>
<td>vapour</td>
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CHAPTER 1: INTRODUCTION

1.1 Importance of Energy
Energy plays a vital role in the development of any country. It has been clearly shown that the energy consumption of a country is proportional to its economic status. By extension, the development of a country can be quantified as a function of its energy consumption. Therefore, sufficient and reliable sources of energy are necessary for both developed countries and developing countries seeking to further their economic growth. Developing countries have the highest growth rate and the highest energy consumption. The EIA has projected that the global energy consumption will increase by 1.4 percent per year with a total growth of 49% from 2007 to 2035 (US-EIA, 2010). Economic growth is considered the most important factor in assessing and projecting changes in global energy consumption (US-EIA, 2010). This projection includes the supply of marketed energy from all fuel sources (see Figure 1). Energy is used for everything including heating, cooking, agriculture, transportation, manufacturing and even telecommunication.

Concisely put, energy can be classified as non-renewable or renewable based on its source. Over 85% of the energy used in the world is from non-renewable supplies (UOC-UCCP, 2009). Non-renewable energy sources can be sub-divided into fossil fuels (coal and oil) and nuclear power. The global reserve of non-renewable energy sources is finite. They cannot be reproduced or regenerated quickly enough to meet the growing demand. The result of this energy imbalance is the depletion of energy reserves. Other energy sources that are completely natural and plentiful are renewable energies. Some examples of renewable energy sources are: solar, geothermal, hydroelectric, biomass, and wind. While renewable energy sources are abundantly available in nature, industrialized nations heavily depend on non-renewable energy sources. Primarily, fossil fuels are the most widely used non-renewable energy source. Their application as a main source of thermal energy is due to the increase use of heat engines since their development in 18th century. Today the majority of heat engines use fossil fuels as an energy source to produce mechanical or electrical power. Some examples of these heat engines are coal power plants, gas turbines, and internal combustion engines. Figure 1.1 depicts consumption history and the future projection of energy consumption until 2030 based on the type of energy source.
Liquid fuels are the most widely used source of energy in the world. Their consumption remains the largest and it is predicted as the fastest consumed source of energy (US-EIA, 2010). The liquid fuel consumption increase predicted to grow steadily per year by 0.9% from 2007 to 2035. The use of liquid fuels in power generation is declining whereas in the transportation sector, consumption continues to increase despite rising oil prices (US-EIA, 2010).

The projected growth of natural gas consumption is 1.3% per year for 2007 to 2035. This is due to growing use of natural gas-fuelled power plants. Also, the increase of oil prices and the environmental impact of the burning of coal broaden the use of natural gas in worldwide power generation.

Despite the fact that the burning of coal is a major contributor to greenhouse gas emissions, global coal consumption is predicted to increase at an average rate of 1.6% per year. This is primarily due to the steady economic development in China which requires cheap electricity to operate competitively. China is the one of the largest user of coal (UOC-UCCP, 2009).

However, the use of renewable energy source in the power generation industry is projected to grow faster than nuclear power. Due to government support throughout the world to
construct renewable energy-based power plants it is predicted that the power generation industry will increase its share from 18% in 2007 to 23% in 2035 (US-EIA, 2010).

With respect to nuclear power, several factors such as plant safety, radioactive waste disposal, and nuclear material proliferation may hinder plans for new installations in some countries. The annual growth projection of nuclear-based electrical power generation from 2007-2035 is 2%. This is a slightly slower rate than the predicted 5% increase in the use of renewable energy sources in electrical power generation (US-EIA, 2010).

1.2 Motivation and Objectives

With respect to heat engines there are many sustainable energy sources available for heat supply. Typical examples of these are solar, geothermal, biomass combustion, nuclear heat, and waste heat from human activities. Coupled with a suitable external heat driven engine, the thermal energy from these renewable sources can be harvested and converted into useful work. Another possibility is cogeneration to produce heat and power simultaneously. Therefore, from both thermodynamic and economic points of view, the application of externally supplied heat engines to generate an acceptable amount of power has solid motivation. The purpose of this work is to investigate the performance of a scroll expander in an organic Rankine cycle and to develop efficient design criteria for low capacity heat engines supplied by low grade heat sources to generate electricity. Scroll machines have some unique and attractive design features as a positive displacement machine. They are compact, reliable and can be used as a compressor or, in reverse, as an expander. Furthermore, they are widely available as refrigeration compressors. Off the shelf, refrigeration scroll compressors are easily modifiable and can be converted to operate as an expander in an ORC.

The goal of this work is to contribute to the development of low-capacity heat engine for sustainable power production or cogeneration. Thus, the specific objectives of this thesis are the following:

- To evaluate the operation of a modified scroll compressor as an expander;
- To model the thermodynamic process in the scroll expander;
- To design, build and conduct experiments in scroll with an expander-dynamometer and expander-generator test units;
• To perform an analysis of the experimental results from the scroll expander in ORC test unit;
• To validate the theoretical model with experimental result.
CHAPTER 2: LITERATURE REVIEW

2.1 Introduction

The industrial revolution began some 200 years ago with the invention of the steam engine. The main component of the steam engine is the steam piston/cylinder, a device capable of producing mechanical work from the expansion of high pressure steam. With current terminology, the steam piston is categorized as “positive displacement expander”. The steam engine was invented by James Watt, but the thermodynamic cycle bears the name of Rankine, the scientist who developed the theoretical background of the thermodynamic cycle. With respect to the working fluid, Rankine cycle can be categorized in three ways: (i) steam Rankine cycle (the most common for large scale power plants), (ii) Rankine cycle with inorganic working fluids, and (iii) Rankine cycle with organic fluids – or, so-called Organic Rankine Cycle (ORC). There is an abundant quantity of published literature on the development of Rankine cycles operating with working fluids other than water. Due to lower boiling temperatures and pressures these fluids are capable of operating with low-capacity heat sources. The number of applications of such heat engines is remarkably large. These heat engines are quite relevant to the paradigm of efficient power production from renewable sources. In this chapter, the literature pertaining to ORC is reviewed with respect to the applications, design, prime movers and experimental systems.

2.2 ORC Applications

There is no standard classification system of heat sources based on their operating temperature range. However, Peterson et al. (2008) designates heat sources as “low-temperature” for a range of 80°C -150°C; “medium-temperature” for a range of 150°C -500°C, and “high-temperature” for temperatures above 500°C. Borsukiewicz-Gozdur and Nowak (2007) consider a low-temperature range between 25°C and 150°C. Saleh et al. (2007) assumes that the low-temperature is around 100°C, while medium-temperature heat sources are between 100°C-350°C. Latour et al. (1982) categorizes temperature levels of 0°C-121°C as low, 121°C-649°C and medium, 649-1093°C as high. The nature of the heat source is also important. For example, one may consider OTEC (Ocean Thermal Energy Conversion) applications as the design constraint for the heat engine. According to Uehara and Ikekani (1993), the temperature is 4°C for the heat sink and 28°C for
the heat source. These figures represent the deep water temperature and the ocean surface water temperature, respectively. Most of the geothermal heat sources are available at temperatures of 80°C-150°C. Solar radiation on flat panels generates heat at a temperature of approximately 80°C, while at high concentration the temperature goes up to 200°C.

The nature of the application is directly related to the temperature range of the heat source. Schuster et al. (2009) identified the following applications of low-capacity, externally supplied heat engines as follows: biomass combustion, solar desalination and waste heat recovery from biogas digestion. Zamfirescu et al. (2010) illustrate the application of heat engines for small-capacity concentrated solar power and heat cogeneration. He had shown that with a grid connected system with possibility of sell-back electricity, the return on investment time was approximately 7 years. For all these cases, the heat source temperature can be considered below 200°C. We will restrict the scope of our review to the range of source temperature between the standard 25°C and 250°C, and denote this range as “low-temperature”. The justification of choosing this range relates to two aspects: the vast majority of renewable and sustainable energy sources fall within these temperature ranges and the current technology allows the use of positive displacement expanders or expander-generator units operating at temperatures below 250°C, respectively.

2.3 ORC Design Aspects
When developing externally supplied heat engines one has to start the design with two thermodynamic constraints in mind. They are the heat source and the heat sink. They determine the magnitudes of the heat transfer that occurs between the heat source and the working fluid followed by the fluid and the heat sink. Rankine cycle and its variations are excellent choices for the previously mentioned temperature ranges as mentioned by Zamfirescu and Dincer (2008). This is due to their flexible nature and the possibility to precisely adapt to the temperature difference between the heat source and heat sink by proper selection of working fluid.

Among inorganic working fluids, interesting options are ammonia, ammonia-water and carbon dioxide. Zamfirescu and Dincer (2008) proposed a trilateral flash Rankine cycle with a zeotropic ammonia-water as the working fluid. In this cycle, liquid ammonia-water at high pressure is heated to its saturation temperature and then flashed into the two-phase region to
generate work with the help of a positive displacement expander. The cycle shows an exergy efficiency of 30% when supplied with a 150°C geothermal heat source. This compares favourably to the 13% exergy efficiency achieved with a Kalina cycle operating under the same conditions. The Kalina (1984) cycle is a variation of the Rankine cycle that uses ammonia-water as the working fluid. As opposed to the trilateral flash Rankine cycle, the Kalina cycle heats the working fluid to superheated vapour. During the boiling process, the ammonia-water varies its temperature. Due to this, the Kalina cycle offers the opportunity to better match the temperature profiles in the heat exchangers. A Rankine cycle operating on pure ammonia has also been investigated for low power applications. It was found to be attractive for heat recovery applications (Koji, 2004). Carbon dioxide can be used as working fluid in a transcritical Rankine cycle configurations. Though carbon dioxide shows excellent heat transfer properties, the main drawback of this cycle is the high pressure at which the system operates. The condensation pressure is typically around 80 bar and the expander inlet pressure is typically around 160 bar; this makes a pressure ratio of 2, a pressure difference across the expander of 80 bar. This poses a problem for the reliability/life time of the expander. Due to such challenges, the development of commercial transcritical carbon dioxide Rankine cycles for low capacities may be judged as difficult. Furthermore, Chen et al. (2006) had indicated that for a 140°C heat source a transcritical carbon dioxide Rankine showed an efficiency of 9.2%. When compared to a R123 organic Rankine cycle operating at the same conditions, only a minor increase in efficiency was observed (1.4%).

The R123 ORC mentioned by Chen et al. (2006) operated at a boiling pressure of 5.87 and a condensation pressure of 0.85. Thus the pressure ratio was 6.9 for a pressure difference of 5.02 bar. This is much lower than the 80 bar pressure difference associated with the transcritical carbon dioxide Rankine. Therefore the organic Rankine cycle poses less technical problems than the transcritical carbon dioxide cycle. This point emphasizes the advantage of using organic fluids as working fluid in Rankine cycles. There are many configurations of organic Rankine cycle derived all from the basic one. In the basic configuration four elements are included: the pump, the boiler, the prime mover (turbine or expander) and the condenser component. Dai et al. (2009) conducted a comprehensive parametric study of multiple configurations of ORC. Apart from the basic Rankine configuration they identified the following:
- ORC with internal heat recovery via a heat exchanger. This uses the heat of the expanded gases to preheat the liquid before saturation;

- Supercritical ORC (also named transcritical by some authors viz. transcritical CO$_2$ cycle discussed above) where the liquid is pressurized to supercritical pressure and then heated to temperature above the critical one before expansion;

- Subcritical ORC operating with retrograde working fluids with expansion of saturated vapour.

The nature of the working fluid is a very important criterion for selecting the appropriate cycle configuration. For example, if a retrograde fluid is used then the expansion of saturated vapour is an expansion into the superheated region. This is due to the fact that the slope of the saturation curve on a T-s diagram for retrograde fluids is always positive. Conversely, if the fluid is regular then the expansion of saturated vapour is an expansion into the two-phase region. An example of a retrograde organic fluid would be Siloxane as mentioned by Colonna et al. (2008). Other Rankine cycle configurations are those with expansion into the two-phase region. The benefit of expanding into the two phase region has been emphasized in the work by Wagar et al. (2010). He had shown that expanding into the two-phase region is a means of adjusting the cycle to the exterior conditions. Smith et al. (1996) proposed an interesting cycle of combined flash expansion and saturated vapour expansion working with R134a as the working fluid. In the first stage, saturated liquid is flashed into the two-phase region. In the second stage, the two phase mixture is separated gravitationally into saturated liquid and saturated vapour components. The liquid is flashed in an expander by reducing its pressure while the saturated vapour is expanded into the two-phase region. The proper selection of expansion device is absolutely crucial due to the pressure sensitivity of two-phase flows. Smith et al. (1994) selected a positive displacement screw expander to flash the liquid and a turbine for expanding the two-phase mixture. The reason for selecting the screw expander is its ability to flash the liquid and expand the saturated vapour while retaining an R134a stream of high quality.

Many works can be found in the open literature regarding the appropriate selection of the best working fluid for an organic Rankine cycle. Available options can include pure working fluids or a mixture of working fluids, where the mixtures can be zeotropic or azeotropic. Harada (2010) emphasized the importance of the retrograde or regular behaviour of the working fluid for

The working fluid selections should be the result of technical, economical, and ecological analyses. Many criteria can be inferred as identified by Papdopoulos et al. (2010). According to them, the most important fluid properties are: density, boiling enthalpy, liquid heat capacity, viscosity, thermal conductivity, melting point temperature, critical temperature, critical pressure, ozone depletion potential (ODP), global warming potential (GWP), toxicity, flammability and the zeotropic or azeotropic characteristic. The life time of the working fluid is also important. An example of this would be a comparison of R11 and R152. Harada (2010) reported that the life time of R11 is 45 years while R152 is only 0.6 years. However, R11 has a GWP of 6370 and ODP of 11, while R152 has a GWP of 187 and ODP of 0.

2.4 Prime Movers for ORC
According to Harada (2010), the prime mover is the component that affects the overall efficiency of the heat engine. As the most previously mentioned, when operating with an organic working fluid the pressure ratio across the prime mover is high. Turbo-expanders operate at a high pressure ratio and need to have a special construction as developed by Larioala (1995). Remarkable progress in the area of computational simulation of ORC turbines is attributed to Harinck et al. (2010). They had shown that real gas effects are crucial due to typical operation in the dense gas regions. Yamamoto (2001) designed a turbine for a small scale ORC and obtained 15-46% isentropic efficiency when the heat input varied from 13 kW to 9 kW. In general, the ORC turbine is characterized by very high rotational speeds as also supported by Yogoub et al. (2006). According to them, the rotational speeds of the ORC turbines reach 60,000 RPM or
more. Arguably, designing and manufacturing ORC turbines can be considered expensive due to the fact that they are sensitive to operating conditions and therefore the design must be customized for each specific application. Furthermore, due to the tip leakages that are relatively constant with turbine size, operation at low capacities can be very inefficient. The fact is that at low capacity, the magnitude of the bypass flow is at a higher percentage of the main stream. Another option for ORC prime mover is a positive displacement expander. Several kinds of those can be found suitable for ORC: axial piston, rolling piston, rotary vane, screw, scroll and others. Maurer et al. (1999) performed an investigation on axial piston expanders and had shown that its volumetric and isentropic efficiencies are approximately 30% and 41% respectively. Wang et al. (2010) used a rolling piston type expander in an organic cycle ORC with R245fa using low grade solar heat source. It had achieved a maximum isentropic efficiency of 45.2%. Mustafah and Yamada (2010) analyzed a rotary vane expander with R245fa for a heat source temperature range of 60°C to 120°C. Furthermore, they had projected isentropic efficiencies upwards of 80%. For a range of capacities between ten and a few hundred kW, twin screw expanders were found suitable (Smith et al., 1996). Screw expanders had shown good adaptability to the operating conditions as they are able to handle two-phase flows (Infante Ferreira et al., 2004). For low capacities, the best choice of positive displacement expanders is the scroll type expander. The geometry of scroll machines is simpler than that of screw expanders because the scroll has 2D geometry while the screw has a 3D geometry. Scroll units are quieter than the screw types and easier to manufacture for low capacity applications. Nagatomo et al. (1999) investigated the performance characteristics of a scroll expander modified from a refrigeration compressor in an ORC. They had found a maximum expander isentropic efficiency of 74%. Also, Yanagisawa et al. (1988) had modified a refrigeration scroll compressor to operate as an expander and tested it in an experimental loop operating with compressed air. They had found a maximum isentropic efficiency of approximately 75%. Husband and Beyene (2008) presented a theoretical model of low-grade heat-driven Rankine cycle with scroll expander and showed a thermal efficiency of 11% for a fixed 10 kW work output. Kim et al. (2001) used a scroll expander in a low temperature heat recovery system, and had found its volumetric efficiency to be 42.3%. Zanelli and Favrat (1994) converted a hermetic scroll compressor to operate as an expander and tested it in a Rankine cycle. Overall the
maximum isentropic efficiencies were about 63% at a rotating speed of 2400 RPM. Harada (2010) tested a scroll expander modified from a refrigeration scroll compressor with R134a and R245fa. They had found its isentropic efficiency to be over 70% for 1 kW power output. It was found that the rotational speed slightly influences the performance, but the optimal expansion pressure ratio was determined to be approximately 3.3. The geometrical modeling of scroll expanders have been investigated by Bush and Beagle (1992) and Chen et al. (2002) based on involute curve theory. Expression for fluid pocket volumes were developed by Wang et al. (2005) and are very important in thermodynamic modeling. Thermodynamic models for scroll machines were presented by Lemort et al. (2009), Oralli et al. (2010) and Harada (2010). Leakage or by-pass flows in scroll machines were studied by Tojo et al. (1986), Sufueji et al. (1992) and Puff and Mogolis (1992). From these studies the leakage flow regime is between two limits: Fanno (isenthalpic flow) and isentropic flow, respectively.

2.5 Experimental Systems

Three kinds of testing loops for scroll expanders are mentioned in the literature. The simplest of which is the open-loop air test bench. Air at a fixed pressure and temperature is provided at the expander inlet and the resulting torque and rational speed are measured with dynamometer and tachometer, respectively. An example of this approach was that of Yanagisawa et al. (1988). A second possibility is the integration of the tested expander in an ORC loop in place of a turbine. This approach has been adopted by a large number of authors some of which are Peterson et al. (2008), Lemort et al. (2009), and Mathias et al. (2009). The third possibility is to use a compressor for working fluid pressurization. Afterwards, the expanded working fluid is delivered back to the compressor inlet to close the loop (Harada, 2010).

In summary, the above literature review reveals the following:

- An analysis of the four discussed aspects (ORC applications, design, prime movers and experimental systems) leads us to believe that a thorough analysis has yet to be fully completed;
- Leakage was identified by most of the researchers as an important irreversibility factor within the expander;
• Selection of the expander for the test unit seems to be random due to the lack of selections based on detailed analysis;
• Wide flexibility of the operating parameters of the test bench was not adopted by designing a modified Rankine cycle;
• Research in the literature was based on the performance with a single scroll machine.
CHAPTER 3: BACKGROUND

3.1 Introduction
There are many sustainable sources of thermal energy that can be used for various purposes, either directly or converted into mechanical work (viz. electric power). Conversion of sustainable thermal energy sources into mechanical work represents a very important issue with regards to the achievement of a clean, non-polluting, un-exhaustible energy supply system for future generations. In this chapter, we will identify and categorize sustainable thermal energy sources and discuss the available heat engines that can be used to convert thermal energy into useful work. The importance and scope of applications of ORC-based heat engines are explained and the main issues regarding their development are discussed. As the main organ of this heat engine is the expansion device, the principal categories of expanders and their required design parameters for optimal operation in ORC are reviewed. This work focuses on scroll expanders in ORC. Thus attention is paid to the identification of the machines’ main problems for low-capacity applications.

3.2 Sustainable Energy Sources for Heat Supply
Our modern world is supplied with energy that is mainly drawn from coal, petroleum and natural gas. The combustion gases are used to generate high pressure steam in large scale power plants to produce electricity. Thermal energy sources and heat engine systems supply the energy needed for our society. As a by-product, combustion of fossil fuels to generate high temperature heat is accompanied by emissions of large amounts of GHG (Green House Gas) and other pollutants. Fossil fuels are combusted not only to generate electricity, but also for space heating, water heating and many industrial processes.

Other thermal energy sources have great potential to replace fossil fuel combustion. These heat sources are mainly derived from renewable energy, nuclear energy and waste recovery. Figure 3.1 illustrates available thermal energy sources that can be called as “sustainable”. Minimal GHG emissions can be associated with these thermal energy sources. Figure 3.1 also suggests that heat engines can be used for sustainable power generation and other
valuable usage such as heating, cooling and so forth. Thermodynamically, it makes more sense to use a heat engine to generate power and heat than to generate power or heating individually.

Figure 3.1 Sustainable thermal energy sources and their applications in heat engines.

A simple example can clarify the importance of heat engines for better resource utilization from the point of view of thermodynamics. Assume that the temperature of air needs to be heated from 18°C to 23°C for space heating in a household application. Two possibilities
are considered: (i) heating with a heat pump, (ii) heating with a gas furnace. In the first case (i), the working fluid of the heat pump is a refrigerant that condenses at 30°C. Mechanical power is consumed to run the heat pump. On the expense of the mechanical power, the working fluid temperature is increased at a suitable level to heat the air with minimal exergy destruction as indicated schematically in Figure 3.2(a). Of course, this manner of heating is justified even more if renewable electricity is available to run the heat pump.

![Diagram](image)

**Figure 3.2 Example of justifying heat-engine applications for better resource utilization.**

In the second case (ii), the primary energy used to generate heat is natural gas (or, possibly biomass-derived fuel), which is combusted to produce hot exhaust gases. Typically, the combustion gas in high-efficiency furnaces evolves from about 150°C to 175°C. A heat exchanger is interposed between the air to be heated and the hot exhaust gases. The temperature profile of the heat exchanger looks similar to that illustrated in Figure 3.2(b). As this figure suggests, the exergy destruction is very important in this case due to the large temperature differences. Thus, from a purely thermodynamic point of view, heating air with exhaust gases does not make sense because it destroys too much exergy. A better justifiable solution is indicated in Figure 3.2(c) where a heat engine is interposed between the hot stream and the cold stream. The heat engine transfers heat between the two streams and at the same time it generates mechanical work. The exergy destroyed is thus kept to acceptable value. The generated
mechanical work can be converted into power and used to drive the air blower or for other purposes.

There are many applications of heat engines driven with external sources of heat. The heat engines are coupled with the heat source. A heat exchanger is used to transfer heat to a working fluid. Thus the working fluid reaches high enthalpy and is able to expand for generating useful work. Further, the working fluid is cooled with the help of the heat sink. For this purpose, a heat exchanger is used. These kinds of heat engines coupled with the heat source and heat sink through heat exchangers are referred as “externally heat driven heat engine”. This term differentiates from another heat engine, which is the internal combustion engine (ICE).

Table 3.1 Characteristics of sustainable thermal energy sources (except waste heat).

<table>
<thead>
<tr>
<th>Thermal source</th>
<th>Temperature, °C</th>
<th>Carnot factor, ((1 - T_o/T))</th>
<th>Heat transfer mode</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar</td>
<td>80-1500</td>
<td>0.16-0.83</td>
<td>Radiation</td>
<td>With or without concentration</td>
</tr>
<tr>
<td>Geothermal</td>
<td>60-350</td>
<td>0.10-0.52</td>
<td>Convection</td>
<td>Sensible heat (brine) heat transfer</td>
</tr>
<tr>
<td>Ocean heat</td>
<td>25-30</td>
<td>0.07-0.09</td>
<td>Convection</td>
<td>Constant sink/source temperature</td>
</tr>
<tr>
<td>Biomass/biofuel/biogas/landfill gas</td>
<td>200-1000</td>
<td>0.37-0.76</td>
<td>Convection</td>
<td>Combustion process; sensible heat (flue gas) heat transfer</td>
</tr>
<tr>
<td>Nuclear</td>
<td>250-900</td>
<td>0.43-0.75</td>
<td>Radiation</td>
<td>Gamma and nuclear radiation</td>
</tr>
<tr>
<td>Moderator heat</td>
<td>60-85</td>
<td>0.10-0.17</td>
<td>Convection</td>
<td>Constant source temperature</td>
</tr>
<tr>
<td>Waste incineration</td>
<td>1000-1500</td>
<td>0.76-0.83</td>
<td>Convection</td>
<td>Combustion process; sensible heat (flue gas) heat transfer</td>
</tr>
</tbody>
</table>

Note: the reference temperature assumed: \(T_o = 25^\circ\text{C}\); (*) for ocean heat \(T_o = 4^\circ\text{C}\)

Details of the types and temperature levels of various thermal energy sources are indicated in Figure 3.1. The most important parameter from a thermodynamic point of view is the temperature associated with the heat sources. Also important is the nature of the fluid that carries the heat. Exception is the case when the heat is transferred by radiation. If the specific heat of the hot fluid is high, then the associated exergy is also high.

Table 3.1 lists the main characteristics (including the exergy content) of the sustainable energy sources included in Figure 1, except the waste heat that is treated separately. The exergy
of the source listed in Table 3.1 is estimated with the help of the Carnot factor calculated based on the heat source temperature \((1 - \frac{T_o}{T})\). The higher the temperature of the heat source the higher is the value of the Carnot factor.

Table 3.2 Characteristics of various sources of waste heat.

<table>
<thead>
<tr>
<th>Thermal source</th>
<th>Temperature(°C)</th>
<th>Carnot factor, ((1 - \frac{T_o}{T_{eq}}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam power plants</td>
<td>230-250</td>
<td>0.40-0.43</td>
</tr>
<tr>
<td>Gas turbine power plants</td>
<td>600-800</td>
<td>0.65-0.72</td>
</tr>
<tr>
<td>Heating furnace exhaust</td>
<td>175-230</td>
<td>0.33-0.41</td>
</tr>
<tr>
<td>Automobile engine exhaust</td>
<td>400-700</td>
<td>0.55-0.69</td>
</tr>
<tr>
<td>Nitrogenous fertilizers plants</td>
<td>195-230</td>
<td>0.36-0.40</td>
</tr>
<tr>
<td>Pulp mills</td>
<td>140-200</td>
<td>0.27-0.37</td>
</tr>
<tr>
<td>Paper mills</td>
<td>140-200</td>
<td>0.27-0.37</td>
</tr>
<tr>
<td>Paperboard mills</td>
<td>140-200</td>
<td>0.27-0.37</td>
</tr>
<tr>
<td>Alkalines and chlorines</td>
<td>170-220</td>
<td>0.32-0.40</td>
</tr>
<tr>
<td>production</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Industrial inorganic</td>
<td>120-200</td>
<td>0.24-0.37</td>
</tr>
<tr>
<td>chemical production</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Industrial organic</td>
<td>120-300</td>
<td>0.24-0.47</td>
</tr>
<tr>
<td>chemicals</td>
<td></td>
<td></td>
</tr>
<tr>
<td>petroleum crudes and</td>
<td>150-200</td>
<td>0.29-0.37</td>
</tr>
<tr>
<td>intermediate refining</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Glass factory</td>
<td>400-500</td>
<td>0.55-0.61</td>
</tr>
<tr>
<td>Cement production</td>
<td>200-300</td>
<td>0.37-0.47</td>
</tr>
<tr>
<td>Blast furnace</td>
<td>500-700</td>
<td>0.58-0.67</td>
</tr>
<tr>
<td>Iron foundries</td>
<td>425-650</td>
<td>0.57-0.67</td>
</tr>
<tr>
<td>Drying and baking oven</td>
<td>93-230</td>
<td>0.18-0.40</td>
</tr>
<tr>
<td>Aluminum production</td>
<td>650-760</td>
<td>0.67-0.71</td>
</tr>
<tr>
<td>Copper production</td>
<td>760-815</td>
<td>0.71-0.72</td>
</tr>
</tbody>
</table>

There are various sources of waste heat due to human activity. For example, common steam cycle power plants reject heat at the condenser in the environment at temperature level in
the range of 30°C-60°C. Gas turbine power plants reject exhaust gases at 600°C-800°C. Usual fuel combustion processes for generation of low temperature heat (e.g., gas furnaces) generate flue gas at 150°C-200°C. Automobile engines produce exhaust gases at 800°C-1000°C. Table 3.2 lists temperature levels and Carnot factors associated with various sources of waste heat encountered in commercial, residential, transportation and industrial sectors.

Based on the report by Latour et al. (1982) more than 65% of waste heat rejected by the industries in the United States is at a low temperature level (below 200°C). From this information, it can be inferred that low temperature heat is available in abundance to recover from industrial thermal waste. Table 3.1 suggests that most of the sustainable thermal energy sources are within this low temperature range. It is of prime importance now to evolve ways for harnessing thermal energy from such prevailing sources to meet a substantial portion of energy demand.

Another point of discussion is the capacity of the thermal energy sources. No precise heat engines classification is found in the literature based on their capacity (or generated power). Thus, for the purpose of the present thesis we will introduce the following ranges of electric power generation with heat engines:

- Low-power, 1-5 kW electric or 5-25 kW thermal energy at heat source.
- Medium-power, 5-50 kW electric or 25-250 kW thermal energy at heat source.
- Intermediate-power, 50-500 kW electric or 250-2,500 kW thermal energy at heat source.
- High-power, over 500 kW electric or 2,500 kW thermal energy at heat source.

There are many sustainable energy sources in the low-power range. Recalling the classification from Figure 3.1, the following sources fit the range for low power generation: solar energy from small scale concentrators (of 10-50 m²), biomass or biofuel combustion, low-scale waste incineration and waste heat recovery. Waste heat sources at low capacity can be found in processes such as air conditioning, non-efficient heating and so forth. These potentially allow heat recovery for enhanced efficiency, cooking, drying, various food processing techniques done in farms and households, activities at small workshops, restaurants, etc. Many processes can be made more efficient by including heat engines. For example, a furnace can be replaced with a co-generation unit that incorporates a low-power heat engine for power and heat production simultaneously.
3.3 Heat Engines

3.3.1 General Aspects

Utilization of thermal energy to produce useful work has been started with the invention of heat engine. In 1712, the practical steam engine was built as designed by Thomas Newman. In the early stage, its use was limited to mining and industrial equipment. Later in 1803, steam locomotive started its journey which continued for over a century. In the 19th century, the internal combustion engine replaced the steam engine as an improvement of heat engine for wide applications.

A heat engine is a device that converts thermal energy to mechanical energy, using temperature gradients between the heat source and the heat sink. A heat engine typically uses the heat energy to do the work and then exhausts the remaining heat to a sink or environment which does not take part to do useful work. The first law and second law of thermodynamics constrain the operation of a heat engine. The first law is the application of conservation of energy to the system, and the second law sets limits on the possible efficiency of the machine and determines the direction of energy flow (Zamfirescu and Dincer, 2008).

![Figure 3.3 P-V diagram of Carnot cycle.](image)

The term “heat engine” is generally used in a broader sense to include power producing devices that operate based on a thermodynamic cycle. Any heat engine includes a prime mover.
that uses thermal energy to do useful mechanical work. The selection and design of the thermodynamic cycle depends on the temperature range of source and sink, the availability of resources, safety, and environmental impact.

The most efficient, theoretical, heat engine cycle is the Carnot cycle which consists of two isothermal and two adiabatic processes. A system undergoing a Carnot cycle is called a Carnot heat engine, although in real life this type of engine is impossible to build. The processes involved in this cycle are considered reversible (Rajput, 2007).

Figure 3.3 shows the Carnot thermodynamic cycle where the working fluid undergoes a series of four internally reversible processes. Those are as follows:

- 1-2, Reversible adiabatic compression from low temperature $T_L$ to high temperature $T_H$ due to work input on the fluid;
- 2-3, Reversible isothermal expansion at temperature $T_H$ as heat added to the fluid;
- 3-4, Reversible adiabatic expansion as the fluid performs work and temperature drops from $T_H$ to $T_L$;
- 4-1, Reversible isothermal compression at temperature $T_L$.

Figure 3.4 Representation of Carnot heat engine in $T$-$s$ diagram.
Figure 3.4 represents the Carnot cycle in the $T$-$s$ diagram and also illustrates a heat engine schematically by indicating the heat source, heat sink, heat fluxes and work generated. The characteristic of the Carnot cycle is that it establishes the maximum possible efficiency for an engine cycle operating between $T_H$ and $T_L$.

The amount of heat input ($Q_H$) in the Carnot cycle from the source is graphically represented on Figure 3.4 as the area under line 2-3. The heat rejected ($Q_L$) is depicted by the area underline 1-4. The net work is thus represented by the area of the rectangle 1-2-3-4. The efficiency ($\eta$) of the cycle is the ratio of the network of the cycle to the heat input ($Q_H$) to the cycle. This ratio can be expressed as

$$\eta = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H}$$

(3.1)

where $\eta$ is cycle efficiency, $T_L$ is sink temperature in K, and $T_H$ is source temperature in K.

From the above equation it can be seen that the maximum possible efficiency exists when $T_H$ is the largest possible value or $T_L$ is at the smallest value. The assumptions made in a Carnot engine are:

- The piston moving in the cylinder does not develop any friction during movement.
- The walls of the piston and cylinder are considered perfect insulator of heat.
- The cylinder head is arranged in such a way that it can be a perfect heat conductor or perfect heat insulator.
- Transfer of heat does not affect the heat source or sink.
- Working medium is a perfect gas.
- Compression and expansion are reversible.

In practice, all processes are irreversible. That is, for example:

- It is impossible to perform a frictionless process.
- It is impossible to transfer the heat without temperature potential.
- Isothermal process can be achieved only if the piston moves very slowly and adiabatic process can be achieved if the piston moves very fast.

The above facts depicts that it is impossible to build a Carnot heat engine. However, Carnot's cycle is important since it describes the best possible use of heat energy and identifies an optimal system. It also describes theoretical limits of the transformation of heat into physical
work. Hence, the Carnot cycle is very useful for the explanation of an ideal heat engine and represents the upper limit of efficiency for any given systems operating between the same two temperatures.

In what it follows, we will categorize and discuss the available heat engines cycles with practical implementations and quantify their efficiency and their applicability to sustainable heat sources. The implementation of a thermodynamic cycle in an actual device (or plant) is called “power cycle” in a broad way. Obviously, any power cycle requires a working fluid. These working fluids are moving through the equipments and act as media to convey the energy while flowing. The power cycles are categorized in two kinds based on the nature of the working fluid; these are “gas power cycles” and “vapour power cycle”. The difference between the two categories is that in the first case the fluid is gaseous and does not encounter any phase change process; for the second kind, there is a liquid-vapour phase change process of the working fluid within the cycle. Figure 3.5 presents a non-exhaustive classification of heat engines.

Figure 3.5 Classification of heat engines.
3.3.2 Gas Power Cycles

The gas power cycles mainly consist of compressions and expansions of working fluid and transform the heat supplied to the system into mechanical work through the process. Typically, gas power-cycles consist of four different processes. Those are:

- Isothermal process: This process takes place at constant temperature;
- Isobaric process: During this process, pressure is constant;
- Constant volumetric process: A constant volume process is isochoric;
- Adiabatic process: In this process, no heat is removed from the system and no heat is added to the system.

Some important gas-powered cycles are discussed in the following sections to explain their characteristics, applicability, and scope of low-capacity power generation.

3.3.2.1 Otto Cycle

The standard Otto cycle is used in the spark-ignition internal combustion engines. In this cycle, four processes occur as shown in Figure 3.6: 1-2 isentropic compression; 2-3 constant volume heat addition; 3-4 isentropic expansion; 4-1 constant volume heat rejection.

Despite its wide use in the automotive industry, the Otto cycle is popular in reciprocating engines of small capacity for power generation. It uses gasoline or propane or compressed
natural gas as the working fluid and involves high temperature combustion. Otto-based engines covering a broad spectrum of powers -from low to intermediate- are available on the market.

3.3.2.2 Diesel Cycle

The air-standard Diesel cycle was introduced by Rudolf Diesel in 1897 and is widely used in Diesel combustion engines. It comprises the following processes as indicated in Figure 3.7: 1-2 isentropic compression; 2-3 constant pressure heat addition, 3-4 isentropic expansion and 4-1 constant volume heat rejection. The practical Diesel engine works at combustion temperature limit of 2100 K.

![Figure 3.7 P-V and T-s diagrams of an ideal Diesel cycle.](image)

Because of its relatively low cost, high reliability and high efficiency, the Diesel engine is a very mature and widespread technology. The Diesel cycle is suitable in internal combustion engine to transform heat energy into mechanical power at a much higher source temperature. Diesel engines are available from the medium power (road vehicles) to high power range (for ocean ships propulsion).
3.3.2.3 Brayton Cycle

The Brayton cycle is widely used in gas-turbine heat engines. The following are the processes that occur in the ideal Brayton cycle (see Figures 3.8 and 3.9):

- Process 1-2: The air is compressed isentropically from the lower pressure $P_1$ to higher pressure $P_2$ with a temperature rise from $T_1$ to $T_2$. No heat addition or rejection occurs at this stage.
- Process 2-3: Heat addition occurs here at constant pressure. Volume increases from $V_1$ to $V_2$ and temperature rises from $T_1$ to $T_2$.
- Process 3-4: In this process air is expanded isentropically from $P_2$ to $P_1$ and the temperature drops from $T_2$ to $T_1$. No heat flow occurs here.
- Process 4-1: Heat is rejected in this process with a decrease of volume from $V_4$ to $V_1$ and temperature drop from $T_4$ to $T_1$. Pressure remains constant here.

The thermal efficiency of the cycle may be represented as:

$$
\eta = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{(T_3 - T_4) - m \cdot c_p (T_2 - T_1)}{m \cdot c_p (T_3 - T_2)}
$$

(3.2)

where $\eta$ is the cycle efficiency, $m$ is the mass flow rate, $C_p$ is specific heat at constant pressure.

![Figure 3.8 Brayton heat engine cycle.](image-url)
A number of methods are applied in improving the thermal efficiency of a gas turbine: 1) using two-stage compressions, inter-cooling between them; 2) two-stage expansion in turbine and reheating between them; 3) passing the exhaust gas through a regenerator to extract the waste heat and preheat the compressed air before combustion. Its requirement for robust expansion device (turbine), high source temperature and complex control mechanism, declines the applicability of its use in low capacity, low grade heat engine. The combustion gases normally rejected by the Brayton heat engine come at a high temperature; thus, Brayton cycle can be used as a topping cycle in a hybrid system, where the bottoming cycle can be an ORC. Brayton heat engine can be also applied without combustion: this is called air-Brayton cycle. Jaffe (1989) mentions the application of an air-Brayton cycle for concentrated solar power that converts 30 kW incident solar radiation into mechanical work. For obtaining reasonable high efficiency with air-Brayton cycle, the hot end temperature must be of the order of 1000°C (Jaffe, 1989).

3.3.2.4 Ericsson Cycle
The Ericsson cycle comprises of two isothermal and two isobaric processes. The processes are
described with the P-V diagram as shown in Figure 3.10. The efficiency of the Ericsson cycle is similar to that of Carnot, namely $\eta = \frac{T_1 - T_2}{T_1}$.

The cycle processes are:

- **Isothermal expansion 1-2**: In this process the working fluid, usually air, is heated by the external source and expansion occurs at constant temperature with the addition of heat. During this process the work is obtained from the engine.

- **Isobaric Heat removal 2-3**: The air is then passed through the regenerator, where its temperature reduces at constant pressure. The regenerator absorbs heat which is then used for heating the gas in the next cycle. The air is then released as the exhaust gas.

- Isothermal compression 3-4: In this process the air is compressed in the cylinder at constant temperature maintained by an intercooler. The pressurized air then flows into the storage tank at constant pressure.

- **Isobaric heat absorption 4-1**: The compressed air at high pressure then flows through the regenerator and absorbs the previously stored heat on the way to heated power cylinder.
3.3.2.5 Stirling Cycle

Stirling cycle in the ideal form which consists of four thermodynamic processes acting on the working fluid. As shown in P-V diagram in Figure 3.11 the processes are:

- **Constant –volume (isochoric) heat-addition 1-2**: The compressed gas flows through the regenerator and receives heat on the way to the next process;
- **Isothermal Expansion 2-3**: The expansion-space is heated externally, and the gas undergoes near-isothermal expansion;
- **Constant-Volume or isochoric heat-removal 3-4**: The gas is passed through the regenerator thus cooling the gas, and transferring heat to the regenerator for use in the next cycle;
- **Isothermal Compression 4-1**: The compression space is intercooled, so the gas undergoes near-isothermal compression.

The Stirling engine principle is such that the transfer of heat occurs from external sources and that has an advantage of using any type of heat source to run the engine. A disadvantage for this external heat addition is that the effective heat transfer rate into the system is low (Simon and Seume, 1990).

Kongtragool et al. (2003) performed a research study on solar-powered Stirling engines and low temperature differential Stirling engine technology. They reported that Stirling engine efficiency however is low, but reliability is high and costs are low. On the other hand, they also pointed out that though Stirling engines have some advantages to use low-grade heat sources the operating technology in practical engine has several disadvantages. The cylinders of a Stirling engine are heated and cooled by external sources which needs some respond time and also starting warm up time. So it has pulsation type power generation which is not suitable for synchronization of electrical generator. In a theoretical analysis, Thombare and Verma (2008) explained that Stirling engines can be used with low temperature heat source using helium or air as the working fluid. However, Stirling engine design is a most complicated task because the dynamic behaviour of the engine mechanism and performance of the heat exchangers highly influence the efficient operation of the engine.
3.3.3 Vapour Power Cycles

The vapour power cycle take advantage of the relatively low power needed to pressurize the working fluid in liquid phase, of the rather high enthalpy gained by the working fluid during boiling and of the relatively large amount of work generated at expansion in either gaseous phase or two-phase. Vapour power cycle bear the name of Rankine from its inventor. There are many variations of the Rankine cycle with regards to configuration and the working fluid; the most common kind is the steam power cycle. Other working fluids include ammonia-water, ammonia, carbon dioxide, and organic fluids. Known cycle configurations are as follows: basic, multi-pressure, regenerative, Kalina, and supercritical.

3.3.3.1 Basic Rankine Cycle

The diagram of a basic Rankine power cycle which involves four components - the pump, the heater-boiler-superheater, the turbine and the condenser- is depicted in Figure 3.12. In the ideal Rankine cycle the working fluid experiences the following thermodynamic processes, with reference to the T-s diagram in Figure 3.13:

- Process 1-2: The working fluid in the liquid state is pressurized from low pressure to high pressure with the help of a pump;
- Process 2-3-4-5: The high pressure liquid is heated in a boiler at constant pressure to convert the liquid into dry saturated vapour;
- Process 5-6: The dry saturated vapour is passed through a turbine where it expands and generates power. This fluid exits the turbine at lower temperature and pressure as a wet or dry vapour;
- Process 6-1: The vapour then enters a condenser where it is condensed at a constant pressure and temperature to become again saturated liquid.

![Basic Rankine cycle configuration](image)

Figure 3.12 Basic Rankine cycle configuration.

In an ideal Rankine cycle isentropic- expansion and isentropic pressurization occur in the turbine and pump respectively, whereas in the boiler and condenser the process is isobaric. Rajput (2007) described that the Rankine cycle efficiency can be improved by appropriate selection of operating conditions as follows: 1) increasing the boiler pressure, 2) superheating the fluid before expansion and 3) reducing the condenser pressure. Zamfirescu and Dincer (2008) mentioned that steam power cycle is used in 80% of all electric power generation systems throughout the world, including virtually all solar thermal, biomass, coal and nuclear power plants. The basic Rankine cycle also operates with organic fluids for specific application as it will be detailed subsequently. However, when used with steam for large-scale power generation (power plants) the basic configuration is not technically and economically justifiable because it
implies too much exergy losses at the boiler/heater. The losses are mainly due to the pinch point problem which occurs in the boiler at the point where the working fluid reaches saturation. The exergy losses are illustrated in the shaded area of the $T$-$s$ diagram in Figure 3.13. For better utilisation of the heat source temperature, the basic Rankine cycle requires a number of configuration improvements, as discussed next.

![Figure 3.13 T-s diagram of a basic steam Rankine cycle.](image)

### 3.3.3.2 Reheat Rankine Cycle

The reheat Rankine cycle aims to better match the temperature profile of the heat source and working fluid by applying reheating of the superheated vapour after a first stage of expansion. In the reheat cycle, two staging or two turbines known as high pressure turbine and low pressure turbine are used. After expansion in the high pressure turbine, the steam is reheated and expanded in the low pressure turbine. The addition of heat to the high pressure steam after exhaust from the high pressure turbine increases the cycle efficiency. This cycle is suitable where very high pressure and high temperature superheated steam is used. Basically, the turbine
stages are divided in two or more pressures. Assuming that $P_H$ is the highest pressure and $P_L$ is the condensation pressure, then the inter-stage pressure would be $\sqrt{P_H \times P_L}$.

3.3.3.3 Multi-Pressure Steam Rankine Cycle

The multi-pressure steam Rankine cycle is a variation of the reheat Rankine cycle capable of obtaining an even better match between the temperature profiles at source side. In the multi-pressure steam Rankine cycle, the expansion and reheating are applied multiple times and several direct contact heat exchangers (mixing steam with liquid water) and pre-heaters are designed for internal heat regeneration purpose and for minimizing the pinch point problem. The multi-pressure steam Rankine cycle is the base for coal, natural gas and nuclear power plants. Basically, a heat exchanger network is designed to optimally preheat, boil and superheat the water with minimal exergy losses. For a typical coal-fired power plant, the energy efficiency goes to 38% and exergy efficiency to 37% (Dincer and Al-Muslim, 2001, Dincer and Rosen, 2007). The characteristic exergy destruction of the main components of the coal power plant is presented in Figure 3.14.

Also some modifications of the basic configurations can improve the cycle efficiency, for example: 1) reheat cycle; 2) regenerative cycle; 3) combined reheat and regenerative cycle. In the regenerative cycle steam after partial expansion in the turbine bled from different staging well above the region of phase change. This bled steam is mixed with the saturated liquid of the condenser in a mixing chamber and ultimately fed to the boiler. The cycle efficiency thus increases by adding heat at higher temperature. Rankine cycle with the combination of reheat and regenerative process has the higher efficiency than any other steam power Rankine cycle operating between similar temperature and pressure regions.
Figure 3.14 Typical exergy destruction of a coal-fired steam power plant.

### 3.3.3.4 Ammonia-Water Rankine Cycle

When the source temperature is in the low range, pure steam is not satisfactory from thermodynamics point of view for power generation. This fact is due to the low normal boiling point of water (100°C). At temperatures below 100°C, the steam engine must operate completely under vacuum; this is inconvenient. To increase the boiling point, water can be combined with ammonia to form a zeotropic mixture. The ammonia-water mixture, being zeotropic, has the property to vary its temperature during liquid-vapour phase change. This brings the opportunity to obtain an excellent match of temperature profiles at sink and source. In fact, matching the temperature profiles at sink and source is stringently important when the heat engine operates with low temperature differential. OTEC applications are one example where ammonia-water Rankine cycles are best suited. In these applications, as mentioned above, the source temperature is in the range of 25-30°C, while the sink is at 4°C. Some recent papers are noted in the literature regarding ammonia-water Rankine cycle development, Desideri and Bidini (1997), Roy et al. (2009), Pouraghaie et al. (2010) and Wagaret al. (2010). By energy and exergy analyses, and calculation of the heat exchangers' surfaces, their results show that evaporation pressure increases with the source inlet temperature and concentration of ammonia in the heat exchanger.
The higher evaporation pressure in turn maximises the thermal and exergetic efficiencies. The ammonia-water Rankine cycle can have a basic configuration or regenerative configuration (discussed latter within ORC section below) and it can be equipped either with a turbine or with a positive displacement expander.

The advantage of the expander is that it can be used to expand in two-phase region, a fact that confers more flexibility of the cycle. One remarkable feature of the ammonia-water cycle is that the ammonia concentration can be adjusted during the heat engine operation as a mechanism to adapt the cycle to possible fluctuation at the heat source or heat sink temperatures. The concentration can be varied from nil (zero) when the cycle is a pure steam one or to 100% when the cycle becomes pure ammonia Rankine. Thus, ammonia-water Rankine cycle appears suitable to heat sources that are variable in nature, such as solar radiation. According to Wagar et al. (2010) the energy efficiency of this cycle can reach 30% with a heat source as low as 250°C. Some other configurations of the ammonia-water Rankine cycle are discussed below.

### 3.3.3.5 Kalina Cycle

Kalina cycle is a thermodynamic cycle used to convert thermal energy to mechanical power with a relatively lower temperature heat source. The Kalina cycle was developed by Aleksandr Kalina in the late 1970 and early 1980 (Zamfirescu and Dincer, 2008) and then several modifications have been proposed depending on the various applications. The Kalina cycle can be depicted as a modified Rankine cycle- using a mixture of two different components (ammonia and water) as working fluid rather than a single pure component like water in Rankine cycle. Kalina et al. (1986) described that this cycle may show up to 10% to 20% higher exergy efficiency than the conventional Rankine cycle. The ammonia concentration in the fluid imposes a property to increases the thermodynamic reversibility which achieves a higher thermodynamic efficiency. A basic configuration of the Kalina cycle is shown in Figure 3.15.

As depicted in the Figure 3.15 ammonia-water mixture is pumped to the generator through the heat recovery unit. In the generator, high temperature and high pressure concentrated vapour develops and flows through the super heater to the turbine. In the turbine this superheated vapour mixture transfers thermal energy to mechanical work. Finally, the low temperature low
pressure vapour mixture flows back to the absorber. Once in the absorber, at a low temperature and mixed with a weak solution it becomes a concentrated low temperature liquid mixture.

The heat recovery potential of Kalina cycles is rather high as compared to usual multi-pressure steam power plants, as indicated by Corman et al. (1995) and Martson (1909). Park and Sontag (1990) show as a case study that the exergy efficiency of Kalina cycle was 15% superior to steam power cycle. Hettiarachchi (2007) had shown the suitability of applying a Kalina cycle to low temperature sources.

### 3.3.3.6 Ammonia-Water Trilateral Flash-Rankine Cycle

Another configuration of the ammonia-water cycle is called “ammonia-water flash Rankine cycle”. This type of cycle is applicable to geothermal power generation or to any other case when low heat is provided by a heat transfer fluid which only exchanges sensible heat. This cycle allows for the best match of temperature profiles at both sink and source. The cycle operates without boiling in a boiler. Rather, after preheating to saturation, the working fluid is expanded by flashing into the two-phase region using a positive displacement expander to
generate power. Scroll or screw expanders are preferred for this type of expansion since they are capable of handling a two-phase flow.

The trilateral ammonia-water flash Rankine cycle is indicated in Figure 3.16 in \( T - \dot{Q} \) coordinates; here \( \dot{Q} \) represents the heat transferred in the heat exchangers. Observe the excellent match between the temperature profiles at source and sink. It is due to the fact that the cycle has high exergy efficiency. For a brine temperature of 150°C, the exergy efficiency of this cycle, as calculated by Zamfirescu and Dincer (2008) has been 30% while that of Kalina was 13% in the same operating conditions.

![Figure 3.16 Cycle diagram of the trilateral ammonia-water flash Rankine cycle [modified from Zamfirescu and Dincer (2008)].](image)

3.3.3.7 Supercritical Rankine Cycle with Inorganic Fluids

Another method to avoid the pinch point problem is by pressurizing the working fluid till above the critical point. During a heating process in the constant-temperature region, boiling is avoided due to the supercritical pressures. The heat exchanger operates in this case with fluids that exchange sensible heat. Due to the high pressures associated with supercritical cycles, one must exercise care when selecting a working fluid. This is due to mechanical limitations on materials. Supercritical water Rankine cycles are gaining popularity as an emerging technology. An example of this application is supercritical water heater as illustrated by Tsiklauri et al. (2005) and Boehm et al. (2005). Water is pressurized to 22 MPa or over and then heated over 400°C.
(supercritical). The high pressure and temperature allow this cycle to be applied to large scale power plants. However, this cycle is not suitable for low power applications.

Two possible working fluids are considered for low power supercritical Rankine cycles: carbon dioxide and organic fluids. Supercritical Rankine cycle must operate at pressures above 75 bar. Though this high pressure may cause safety concerns, this cycle is considered due to the enhanced heat transfer characteristics of the supercritical carbon dioxide. Yamaguchi et al. (2006) demonstrated a solar-powered Rankine cycle with supercritical carbon dioxide and had calculated an efficiency of 25%. Many organic fluids were tried for supercritical and transcritical Rankine cycles as depicted by Schuster et al. (2010).

3.3.3.8 Organic Rankine Cycle
Organic Rankine cycle works on the same principle as that of Rankine cycle. The working fluid in the form of liquid is pumped to a boiler where it changes its state to vapour then passes through an expansion device and finally condenses to liquid state. In the ORC however instead of using water, organic fluid is used as working substance. The advantage of using organic fluid over water is that it has a low boiling temperature which is a prime criterion for the use with any low grade heat source. Due to the low liquid to vapour-volume ratios associated with organic working fluids, a single stage expansion device can be used to convert thermal energy to mechanical work. In water fed Rankine cycle, a robust turbine is used which requires the working fluid to be superheated to avoid any condensation droplets forming during expansion; removing chances for possible damage to the blades. In an ORC, a compact low-speed expansion device may be used which does not require the mandatory superheating of the fluid. This has an advantage on using low grade heat sources as in some cases superheating has adverse effect on the overall cycle efficiency.

In an ORC, a large variety of working fluids may be used depending on the operating condition and the temperature difference between source and sink. However, the purpose of an ORC focuses on use of low grade heat energy. That being said, the organic fluid should have the following characteristics:

- Low boiling and freezing point.
- High stability temperature.
• High heat of vaporisation.
• High density.
• Low environmental impact.
• Safety.

A typical example of externally supplied heat engine application – like an ORC – is a bottoming cycle coupled with an ICE. Bombarda et al. (2010) studied the application of both Kalina and ORC type heat engines as a bottoming cycle coupled to a diesel ICE and had shown efficiency improvements. Miller et al. (2009) coupled an ORC as bottoming cycle to a thermoelectric power converter. Bradz et al. (2005) developed a heat engine coupled to a moderate temperature geothermal source. Kane et al. (2003) developed a heat engine driven by solar radiation. Another example of sola- driven heat engine is that of Saitoh et al. (2007) which uses a scroll expander. Manolakos et al. (2007, 2009) had shown the benefits of heat engines used in systems that generate other products in addition to power. They had integrated an ORC with a desalination plant. Gu et al. (2008) evaluated an ORC theoretically and experimentally with low to moderate heat sources (60-200°C). Their findings were that the cycle is relatively insensitive to source temperature but highly sensitive to evaporating pressure. Mago et al. (2008) performed an analysis of a regenerative ORC and found increased performance over a simple ORC.

### 3.3.4 Comparison of Heat Engines

From the above discussion, it can be concluded that thermal energy can be efficiently converted to mechanical work with a suitable heat engine based on a properly chosen appropriate thermodynamic cycle that is matched with the heat source temperature and capacity. Figure 3.17 had been assembled based on heat engine manufacturers’ catalogues and general literature data regarding the temperature and power generation range for various heat engines. Otto cycle-based heat engines are found to operate at the highest temperature and are available from a “low” to “medium” power range. Furthermore, the pressures are found to be the highest in Diesel cycle-based engines due to high compression ratios. Otto cycle engines are ranked second in terms of maximum pressures.
Gas turbines are typically used for large scale power generation (typically over 100 kW) and feature a relatively low maximum pressure (typically 15-20 bars). Stirling and Ericsson engines are applicable for low power range due to their construction being based on reciprocating concept; normally the hot end temperature in these engines is roughly below 900°C. As supercritical CO₂ cycle must employ on compact heat exchangers and prime movers. Due to the high pressure and temperature the capacity of these machines are limited to applications requiring less than 100 kW. The Kalina cycle has been determined to be ideal for low power applications in the intermediate capacity range. Furthermore, a second limitation on these machines is due to the tendency of ammonia to decompose at temperatures higher than 250°C. Organic Rankine cycles also have high temperatures. Due to the nature of their molecular structure, organic working fluids have a tendency to decompose at high temperatures. Organic working fluids have limits to a maximum temperature of 400°C. Until recently, the ORC technology was limited from low to intermediate power range covering 1.0 to approximately 800 kW.

![Comparative chart of heat engines with applicable temperature range.](image)

Figure 3.17 Comparative chart of heat engines with applicable temperature range.
3.4 ORC Configurations

The use of an organic Rankine cycle in small scale power generation system is of a growing interest since about twenty years. Its advantage of working with low grade heat source had convinced researchers to focus their effort in developing a feasible and practical application of small scale power generation. It has already been mentioned in Section 3.1 that low and moderate temperature thermal energy sources are available as sustainable and renewable energy, and waste heat sources can be used for ORC applications. Engin et al. (2005) conducted a case study on industrial waste heat and pointed out that 40% of the heat used in the cement industry was lost in flue gases. These flue gases were at a temperature between 215°C and 315°C. Hung (2001) pointed out that due to economical constraints this heat energy is not suitable to be recovered with traditional steam cycles. Delgado-Torres et al (2010) had mentioned that application of ORC generally deals with the selection of the working fluid, the optimization of the ORC unit and analysis of further modification in order to achieve higher thermodynamic and mechanical efficiency. Analysis of all factors is required to determine the optimal configuration for maximum efficiency.

Saleh et al. (2007) performed a thermodynamic analysis of a number of pure working fluids with and without superheating. There results had shown that the highest efficiencies were obtained with fluids in a subcritical cycle with regeneration. A pinch point analysis was performed in their study and it had shown that the heat transfer between the source and the working fluid was the largest in the super critical region for some fluids and the least for subcritical fluids with high boiling points. They had recommended to do an analysis on mixture of pure fluids which they believe might give interesting results.

Khennich et al. (2010) analyzed the performance of a Rankine power cycle using R134a as the working fluid and a finite low temperature (100°C) heat source. They had shown that the net power output and the total conductance of the heat exchangers are functions of the evaporation pressure and the pinch points of the heat exchangers. Hettiarachchi et al. (2007) in their research project considered the ratio of the total heat exchanger area to net power output as one of the criterion for ORC design. They had determined the optimum cycle efficiency by varying the evaporation and condensation temperatures. Among the four working fluids studied, they had found that ammonia meets the optimum design criterion. Its first Law and second Law
efficiencies were found lower than those obtained with the organic fluid HCFC 123 and n-pentane. In an analytical model of an ORC with R113, Badr et al. (1984) had explained that the evaporator temperature and the expander efficiency were both important parameters in optimizing the performance of an ORC. They had varied the isentropic expander efficiency, evaporator temperature, and condenser temperature. They had concluded that expander efficiency was the most sensitive parameter for cycle efficiency in a low temperature ORC system.

3.4.1 Basic ORC
The basic ORC configuration is identical to that depicted in Figure 3.18 and is typically used when the working fluid expands into the two-phase region. This is the case when the fluid behaviour is “regular” as described in the above paragraphs. An example of basic ORC cycle is represented in the T-s diagram for the working fluid R134a.

![Basic ORC configuration with R134a](image)

Figure 3.18 Basic ORC configurations with R134a.

Basic ORC configurations could also operate with expansion of superheated vapour into the superheated region. This case would require the vapours to be cooled in a condenser. In co-
generation applications this rejected heat has some use. If the degree of superheat of the expanded working fluid is high, then a regenerative ORC configuration should be employed for greater efficiency.

### 3.4.2 Regenerative ORC

The configuration of a regenerative ORC is presented in Figure 3.19. An additional heat exchanger is placed into the basic configuration and it allows for a transfer of heat between the hotter working fluid at the turbine exhaust and the colder liquid at the pump discharge. The term “regenerative” suggests that the heat is “regenerated” internally within the cycle. Due to this, less thermal energy is required from a source to drive the cycle, since the preheating of the working fluid is provided through “regeneration”.

![Figure 3.19 Regenerative configuration of ORC.](image)

Figure 3.20 illustrates the T-s diagram of a regenerative ORC, operating with R134a to be compared with the basic cycle operating with the same fluid as shown in Figure 3.18. The energy
balance on the regenerative heat exchanger using the enthalpy of the state points (Figure 3.20) can be written as: \( h_3 - h_2 = h_7 - h_8 \).

From a thermodynamic point of view, the regenerative ORC configuration makes a lot of sense, even more so when the working fluid is retrograde. An example of T-s diagram of ORC with a retrograde working fluid toluene is presented in Figure 3.21. It was assumed that a superheated vapour is expanded. Establishing the level of vapour superheating is a matter of cycle optimization from thermodynamic, fluid mechanics, heat transfer and economic points of view. The parameters that must be accounted for the design are the source and sink temperatures, the pressure ratio across the expander and the pinch point of the heat exchanger. In the design shown in Figure 3.21, the source temperature is 320°C and it is suggested that cycle operates with 40°C of superheating.

![Figure 3.20](image)

Figure 3.20 the T-s diagram of a regenerative ORC with regular working fluid: R134a.

Another ORC design is shown in Figure 3.22 with toluene as the working fluid and identical pressure limits. In this case the hot source temperature is lower; only 300°C and the degree of superheating is zero. This resulted saturated vapour being directly expanded. The expansion of retrograde fluids at saturation results superheated vapours at lower pressure.
Figure 3.21 T-s diagram of a regenerative ORC with retrograde working fluid: Toluene.

Figure 3.22 ORC cycle with expansion of saturated vapour of retrograde fluid: Toluene.
3.4.3 Supercritical ORC

Another ORC configuration is the supercritical ORC, which operates above the critical point of the fluid. The diagram for this configuration is similar to that one depicted in Figure 3.23. During the cycle, the working fluid reaches sub-critical points, as it is expanded, cooled and condensed before pressurizing it again. The passage of the working fluid to sub-critical points and then to supercritical points justifies the appellative of “trans-critical”.

An example of supercritical ORC is illustrated in Figure 3.23 and plotted with the EES software. This had shown that in a supercritical ORC, there is no constant temperature profile during heating of the working fluid due to the non-existence of boiling at supercritical pressures. In the example, R404A had been chosen as working fluid. The high pressure in this cycle is 38 bar while the low pressure is 13 bar, making a pressure ratio of 2.9 which is applicable to positive displacement expanders. The temperature profiles match very well proving that there is reduced exergy destruction in a supercritical ORC. In the regenerative heat exchanger a pinch point appears due to the non-linear temperature profile of the supercritical fluid.

![Figure 3.23 Example of a supercritical ORC.](image-url)
Coupling the ORC to the heat source is one of the main issues regarding a proper design. Depending on the nature of the heat sources, an appropriate heat exchanger must be selected or designed to minimize temperature differences and further minimize the pinch point problem if it occurs. Several cases may be contemplated as follows:

- If the heat source is a hot liquid (geothermal brine, waste heat stream, etc), then a plate-type heat exchanger could be installed between the hot fluid and the working fluid. Another option could be shell and plate or shell and tube heat exchangers.
- If the heat source is a hot gas (flue gas from cement or metal foundry), then an extended surface heat exchanger could be used. Finned surfaces could be employed in this case.
- If the heat source is thermal radiation (concentrated solar radiation), then either the heat exchanger is constructed in the form of a solar receiver. Another option could be that a heat transfer fluid may be used to “carry” the heat from the solar “field” to the ORC engine.

Coupling ORC with solar collectors has previously been described by Delgado-Torres and Garcia-Rodriguez (2010). The system was equipped with an external heat transfer loop as shown in Figure 3.24. In this configuration, the fluid was superheated to some degree and after expansion the fluid remained in a vapour state. A regenerator is used to preheat the liquid stream thus extracting much heat to increase the efficiency of the cycle.

Several kinds of heat transfer fluids are presently available for use in an ORC. For the low temperature range various heat transfer oils are available. At temperatures up to 400°C one can use siloxanes as heat transfer fluids. For higher temperatures, various kinds of molten salts are recommended.
3.4.4 Trilateral-Flash ORC

The concept of trilateral flash Rankine cycle introduced above in Section 3.3.3 with ammonia-water is discussed here again as operating with any organic fluid. Based on the study by Zamfirescu and Dincer (2008), the energy and exergy efficiency of this cycle operating with a hot brine of 150°C is indicated in Table 3.3 for 4 refrigerant working fluids.

A trilateral-flash ORC may be a good alternative to a supercritical ORC for some specific conditions. For example, from a technical and economical point of view, a trilateral flash ORC does not require supercritical pressures to operate. In addition to this, its configuration is of
“basic” kind which is simpler and more cost effective as it would not require a regenerative heat exchanger.

Table 3.3 Trilateral-flash ORC performance with various working fluids.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>R141b</th>
<th>R123</th>
<th>R245ca</th>
<th>R21</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta$, %</td>
<td>10</td>
<td>9</td>
<td>9</td>
<td>9</td>
</tr>
<tr>
<td>$\psi$, %</td>
<td>13</td>
<td>16</td>
<td>16</td>
<td>13</td>
</tr>
</tbody>
</table>

Source: Zamfirescu and Dincer (2008)

### 3.5 Working Fluids for ORC

In the selection of organic fluid for use with low temperature heat sources in ORC heat engines, attention should be given to obtain higher cycle efficiency. Simultaneously safety criteria, environmental impact, cost and availability should also be considered. The important parameters are as follows:

- If an ORC is required to operate with a low temperature heat source, working fluid with low boiling point is preferred. However, a very low boiling point at atmospheric pressure may require a low condensing temperature;
- A lower freezing point below the heat sink temperature is desired to prevent freezing of the working fluid;
- A fluid with relatively low specific heat capacity should be used to offset the possibility of condenser overloading;
- At high pressure and high temperature organic working fluids usually suffer from chemical deteriorations and decompositions. This factor should be considered during working fluid selection;
- A working fluid with a high latent heat of vaporization can absorb more heat during evaporation. Therefore a fluid with high latent heat of vaporization is preferred to enhance heat recovery and by extension further increase the efficiency of the system;
- In selection of a working fluid, Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) rating of the working fluids should be considered;
- A fluid with low toxicity is required.
Table 3.4 Physical, safety and environmental data of working fluids.

<table>
<thead>
<tr>
<th>Substance</th>
<th>Mol. mass (kg/Kmol)</th>
<th>(T_{bp}) (°C)</th>
<th>(T_{crt}) (°C)</th>
<th>(P_{crt}) (MPa)</th>
<th>Safety data ASHRAE</th>
<th>Atm. life time, (yr)</th>
<th>ODP</th>
<th>GWP (100 yr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>RC318</td>
<td>200.03</td>
<td>-6.0</td>
<td>115.2</td>
<td>2.778</td>
<td>A1</td>
<td>3200</td>
<td>0</td>
<td>10,250</td>
</tr>
<tr>
<td>R600a</td>
<td>58.12</td>
<td>-11.7</td>
<td>135</td>
<td>3.647</td>
<td>A3</td>
<td>0.019</td>
<td>0</td>
<td>~20</td>
</tr>
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<td>145.7</td>
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<td>1.000</td>
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<td>R600</td>
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<td>3.796</td>
<td>A3</td>
<td>0.018</td>
<td>0</td>
<td>~20</td>
</tr>
<tr>
<td>R601</td>
<td>72.15</td>
<td>36.1</td>
<td>196.5</td>
<td>3.364</td>
<td>-</td>
<td>0.01</td>
<td>0</td>
<td>~20</td>
</tr>
<tr>
<td>R113</td>
<td>187.38</td>
<td>47.6</td>
<td>214.1</td>
<td>3.439</td>
<td>A1</td>
<td>85</td>
<td>1.000</td>
<td>10,130</td>
</tr>
<tr>
<td>Cyclohexane</td>
<td>84.16</td>
<td>80.7</td>
<td>280.5</td>
<td>4.075</td>
<td>A3</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>R290</td>
<td>44.10</td>
<td>-42.1</td>
<td>96.68</td>
<td>4.247</td>
<td>A3</td>
<td>0.041</td>
<td>0</td>
<td>~20</td>
</tr>
<tr>
<td>R407c</td>
<td>86.20</td>
<td>-43.6</td>
<td>86.79</td>
<td>4.597</td>
<td>A1</td>
<td>N/A</td>
<td>0</td>
<td>1,800</td>
</tr>
<tr>
<td>R32</td>
<td>52.02</td>
<td>-51.7</td>
<td>78.11</td>
<td>5.784</td>
<td>A2</td>
<td>4.9</td>
<td>0</td>
<td>675</td>
</tr>
<tr>
<td>R500</td>
<td>99.30</td>
<td>-33.6</td>
<td>105.5</td>
<td>4.455</td>
<td>A1</td>
<td>N/A</td>
<td>0.738</td>
<td>8,100</td>
</tr>
<tr>
<td>R152a</td>
<td>66.05</td>
<td>-24.0</td>
<td>113.3</td>
<td>4.520</td>
<td>A2</td>
<td>1.4</td>
<td>0</td>
<td>124</td>
</tr>
<tr>
<td>R717</td>
<td>17.03</td>
<td>-33.3</td>
<td>132.3</td>
<td>11.333</td>
<td>B2</td>
<td>0.01</td>
<td>0</td>
<td>&lt;1</td>
</tr>
<tr>
<td>Ethanol</td>
<td>46.07</td>
<td>78.4</td>
<td>240.8</td>
<td>6.148</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Methanol</td>
<td>32.04</td>
<td>64.4</td>
<td>240.2</td>
<td>8.104</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>R718</td>
<td>10.2</td>
<td>100</td>
<td>374</td>
<td>22.064</td>
<td>A1</td>
<td>N/A</td>
<td>0</td>
<td>&lt;1</td>
</tr>
<tr>
<td>R134a</td>
<td>102.03</td>
<td>-26.1</td>
<td>101</td>
<td>4.059</td>
<td>A1</td>
<td>14.0</td>
<td>0</td>
<td>1,430</td>
</tr>
<tr>
<td>R12</td>
<td>120.91</td>
<td>-29.8</td>
<td>112</td>
<td>4.114</td>
<td>A1</td>
<td>100</td>
<td>1.000</td>
<td>10,890</td>
</tr>
<tr>
<td>R123</td>
<td>152.93</td>
<td>27.8</td>
<td>183.7</td>
<td>3.668</td>
<td>B1</td>
<td>1.3</td>
<td>0.020</td>
<td>77</td>
</tr>
<tr>
<td>R141b</td>
<td>116.95</td>
<td>32.0</td>
<td>204.2</td>
<td>4.249</td>
<td>N/A</td>
<td>9.3</td>
<td>0.120</td>
<td>725</td>
</tr>
</tbody>
</table>

Source: Tchanche et al. (2008); N/A: non-available; \(T_{bp}\): normal boiling point; \(T_{crt}\): critical temperature; \(P_{crt}\): critical pressure; ODP: ozone depletion potential, relative to R11; GWP: global warming potential.

Tchanche et al. (2008) performed research in the area of working fluid selection for low temperature ORCs and listed 20 suitable fluids; these are shown Table 3.3. The selection and/or sizing of the expander is very much related to the properties of the working fluid as detailed by Fraas (1982). Mago et al. (2007, 2008) had concluded that the increased degree of superheating of retrograde working fluids leads to a decrease in cycle efficiency.

Attention must be paid during the selection of the equation state of the working fluid for design, optimization and modeling purposes of the ORC. This is due to the accuracy of its predictions being very important during operation in the two-phase region and close to the two-
phase region. Expansion of complex molecular fluid in the vicinity of the vapour saturation curve can lead to the occurrence of non-linear gas-dynamic phenomena. These include shock formation and expansion waves. Zamfirescu et al. (2008) demonstrated the existence of a special gas-dynamic region of dense gases of some organic working fluids where weak shocks may occur in expanding flows under some conditions. These processes may affect the design of the expander and the rational selection of the operating condition for the expansion process as commented in Colonna et al. (2008).

3.6 Expanders
In an ORC heat engine, the expansion device is the most important component of the cycle. The performance and efficiency of the cycle strongly depend on the expander. Consequently, this correlates with working fluid, operating conditions, and heat source characteristics. Expanders are broadly divided into two categories: turbo-machines and positive displacement machines. Quoilin (2007) and Quoilin et al. (2010) had demonstrated that the use of positive displacement expanders were advantageous compared to turbomachines for use in low temperature applications. He had illustrated that due to the performance criteria, turbomachines should have high tip speeds compared to smaller positive displacement machines like scroll machines.

In order to have comparative performance, a smaller turbomachine needs to rotate at much higher speeds. This is likely to generate high mechanical stress, higher bearing friction and require additional reduction gearing. Hung (1997) described that turbomachines have relatively low pressure ratios per expansion stage whereas positive displacement machines could be built with much higher pressure ratios per stage. Also, positive displacement machines were found to be much more resistant to operate in the two-phase region. This would allow the expansion into the two-phase region. Positive displacement machines are also available in a wide variety of types for use as a compressor. Among them are: reciprocating, rolling piston and rotating vane. Additionally, a positive displacement compressor operating in reverse can function as an expander.
3.7 Scroll Expander

The scroll compressor was first developed in 1905. It consists of two identical intermeshing spiral elements placed in a drum-like, air-tight chamber with a phase difference of 180°. One of the scrolls is rigidly attached to the drum, while the other one orbits within it during operation. In a compression mode, this orbiting motion performs the function of trapping, compressing and discharging of the fluid between the scrolls. A rotor shaft is attached to the moving scroll to couple the machine either to a motor or to a generator. The set orbiting position of the moving scroll is maintained by a special device known as “Oldham coupling”. With the movement of the inner scroll, the outer periphery forms a line of contact with the fixed scroll forming a crescent shaped pocket. This pocket acts as the cavity of a positive displacement compressor. The size of the cavity would depend purely on the geometry of the scroll wraps.

During operation as a compressor, low pressure gas enters and fills the pockets to start the first orbit. With the rotation of the drive shaft the orbiting scroll closes the pair of pockets at the completion of the first orbit. As the first orbit ends, the first pair of pockets moves inward and the ends of the scroll start opening again to allow a fresh intake of gas. The second orbit moves the gas pockets inwards, decreasing its volume and consequently increasing its pressure. The third orbit further reduces the volume of the gas and finally breaks the inner tip contacts and discharges the compressed vapour through the centre discharge port. This completes one cycle of operation.

The operation of the scroll expander is described with the help of Figure 3.23. The high pressure gas enters through a port in the scroll centre. This high pressure gas generates forces on the scroll vanes which translates into torque and produces an orbiting movement. The orbiting scroll with the movement transfers the gas to two adjacent vanes and forms two symmetrical pockets. The trapped gas further expands, forcing the orbiting scroll to move around the centre of the fixed scroll thus transmitting the rotating motion to an eccentric shaft. The pockets finally break up at the periphery of the scroll and discharge through the exhaust port. Several pairs of symmetrical pockets of increasing volumes co-exist at any time during the expander operation. The number of pockets pair is dictated by the scroll rolling angle. In the example from Figure 3.25, the rolling angle is taken as $8\pi$, thus 4 symmetrical pockets can coexist at the same time.
Scroll geometry is one of the key components affecting the efficiency of scroll machines. The built-in volume ratio which is an important parameter in scroll machine operation depends on the scroll geometry. Scroll geometry is defined with the following parameters:

- $r_o$, orbiting angle.
- $r_b$, radius of the basic circle of the scroll.
- $h$, height of scroll vanes.
- $\phi_{0,0}$, initial angle of the outer involute.
- $\phi_{i,0}$, initial angle of the inner involute.
- $\phi_{o,s}$ starting angle of the outer involute.
- $\phi_{i,s}$ starting angle of the inner involute.

The orbiting angle describes the number of turns that the orbiting scroll must do for a complete cycle of compression or expansion. The built-in volume ratio is defined as the ratio of the volume of the expansion chamber at the end of the expansion process to the volume of the intake chamber at the beginning of the process. The built-in volume ratio is the most important parameter of scroll machine because it directly influences the capacity and operation of a scroll machine.

Ideally, the two halves of a scroll compressor remain perfectly in contact as they rotate. In reality, it is not practical to machine them accurately enough for this to be the case. Due to required tolerance, there remains a narrow gap. Typically the gap is approximately one micron across. That may be increased by wear over time and poor machining. It is known that if it reaches around eight microns, the compressor becomes useless. There are two types of by-pass flow (in the form of leakage) that occur in a scroll expander: radial and axial. Radial leakage manifests itself between adjoining flanks of the vanes whereas axial leakage occurs between the vane tip and the base plate of the opposite scroll. This is illustrated in Figure 3.26. The gas leakage occurs from the higher pressure side to the lower pressure side directing the flow toward the discharge side. The existence of by-pass flow decreases the net generated power by the expander.

![Figure 3.26 By-pass (leakage) flows in scroll expander.](image-url)
Scroll expanders have numerous advantages, namely:

- More efficient over their entire operating range
- Operate at lower sound and vibration levels than traditional compressors
- Fewer moving parts
- Ability to start under any system load, without assistance during starting
- Easy to service and maintain due to their compact size, light weight, and simple design
- No complex internal suction and discharge valves
- Quieter operation and higher reliability due to fewer moving parts
- Since high pressure gas exerts pressure in all directions (tangentially, radial and axially) the requirement for axial bearing is omitted.
CHAPTER 4: EXPERIMENTAL SYSTEMS

4.1 Introduction
Refrigeration scroll compressors operating, geometrical and performance parameters are specified in different manufacturer’s catalogue. Two units are procured and modified to act as expanders. For each expander a specific test bench was designed and built to determine their performance through measurements. Eventually, one of the expanders was to be incorporated into a specially designed ORC test bench. We aim to determine expander performance and ORC system performance under various operating conditions. In this chapter, illustrations of the experimental systems, the components, the measuring equipment and the experimental procedures are presented.

4.2 Selection of the Scroll Machine
An extensive search through catalogues for refrigeration scroll compressors from various manufacturers was performed. The purpose of this was to identify appropriate units that could operate in reverse as expander for use in an ORC. The power range for the scope of the present work is set to 1-5 kW electric. The range of manufacturers was narrowed down to 5 choices: Copeland, Bristol, Hitachi, Sanyo, Bitzer. Several criteria were identified and applied for the selection of the compressor type. Some of which were:

- **Source temperature range**
  The experimental investigation is to be restricted to lower than 200°C temperature range for two reasons. Firstly, the majority of renewable sources are below this temperature. Secondly, at lower temperatures, there are fewer mechanical problems concerned with thermal expansion. Finally, one must safeguard against the possibility of oil and refrigerant thermal decomposition.

- **Appropriate pressure ratio**
  In expander operation, the pressure ratio must be lower than in compressor operation. Precaution must be taken to ensure that the work output from the expander is reasonably sufficient.
• **Maximum pressure into the system**
  Maximum pressure must be lower than to be lower than 4000 kPa for tube safety and better structural integrity.

• **Built-in volume ratio**
  Since the built-in volume ratio affects the expansion process, it should be chosen such that the ORC condensation temperature is higher than the sink temperature to ensure heat transfer.

• **Ability to modify**
  It should be easily modifiable to an expander.

• **Motor type**
  A motor capable of working as a generator in reverse operation is preferred.

• **Oil circuit**
  Lubrication system working in reverse operation is required.

• **Refrigerant type**
  The vapour pressure at standard temperature, the normal boiling point, the critical pressure and temperature, the Ozone Depleting Potential (ODP), the Global Warming Potential (GWP), the flammability and toxicity of the working fluid should all be considered for each potential working fluid.

• **Cost criterion**
  It should be effective and easily available.

The main characteristics of the selected scroll compressors are indicated on Table 4.1. Based on performance characteristics of the compressors under “nominal” conditions, the scroll compressors have been analyzed in light of selection. The following parameters are the most important for determining the characteristics of the scroll unit. These are the temperatures of evaporation, condensation, vapour superheating and liquid sub-cooling, mass flow rate, electric power consumption, angular velocity and the displaced volume (given in cm³ of gas pocket at suction per shaft revolution). The displaced volume has been determined from the displacement (i.e., volumetric flow rate at suction in nominal conditions, see in Table 4.1), the nominal turning speed (NTS) and according to:
\[ V_d = 60 \times \frac{\dot{V}_d}{NTS} \]  

(4.1)

where \( \dot{V}_d \) is the volumetric flow rate (m\(^3\)/s).

Table 4.1 Selected scroll compressor units and their main characteristics.

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Model</th>
<th>Type</th>
<th>Motor Power, (W)</th>
<th>Motor Type</th>
<th>Refrigerant</th>
<th>Displacement, (m(^3)/h)</th>
<th>Pressure, (kPa)</th>
<th>Pressure Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bristol</td>
<td>H20R483DBE</td>
<td>H</td>
<td>4150</td>
<td>TFI</td>
<td>R22</td>
<td>13.60</td>
<td>2149</td>
<td>626</td>
</tr>
<tr>
<td>Copeland</td>
<td>ZF06K4E-PFV</td>
<td>H</td>
<td>1730</td>
<td>SFI</td>
<td>R404A</td>
<td>7.14</td>
<td>2540</td>
<td>267</td>
</tr>
<tr>
<td>Hitachi</td>
<td>G300DL</td>
<td>H</td>
<td>3750</td>
<td>TFI</td>
<td>R407C</td>
<td>9.87</td>
<td>2461</td>
<td>710</td>
</tr>
<tr>
<td>Sanyo</td>
<td>C-SBN303LBA</td>
<td>H</td>
<td>4450</td>
<td>TFI</td>
<td>R404A</td>
<td>14.02</td>
<td>2313</td>
<td>443</td>
</tr>
<tr>
<td>Bitzer</td>
<td>ECH209Y-02G</td>
<td>SH</td>
<td>1500</td>
<td>TFI</td>
<td>R134A</td>
<td>6.21</td>
<td>1471</td>
<td>377</td>
</tr>
</tbody>
</table>

SFI=single-phase induction motor; TFI=three-phase induction motor; H=hermetic; SH=semi-hermetic.

The nominal thermodynamic cycle in T-s diagram for each unit is generated with EES software (Klein, 2010). The higher and lower pressures in the system were calculated based on the condensation and evaporation temperatures, respectively. The flow enthalpy at the compressor discharge is approximated based on energy balance equation on “electrical side” and “fluid side” written as follows:

\[ \dot{W}_{el} \cong \dot{m} \times (h_2 - h_1) \]  

(4.2)

where \( \dot{m} \) is the mass flow rate, \( h \) is the enthalpies at state 1 (suction) and state 2 (discharge). Based on enthalpy and pressure at discharge, the temperature and the specific entropy are determined with EES by solving the equations of state for each fluid. The volume ratio has been determined for each case as the ratio of specific volumes at suction vs. discharge according to:

\[ \text{VR} = \frac{v_2}{v_1} \]  

(4.3)

The volume ratio in nominal conditions is an approximation of the built-in volume ratio which is the geometrical characteristic of positive displacement machines that defines their operation as both a compressor and an expander.
Figure 4.1 Performance parameters and T-s diagram of Bristol H20R483DBE scroll compressor in nominal operation conditions.

Manufacturer catalogue data for nominal operation

\( T_c = 54.4^\circ C \)
\( T_e = 7.2^\circ C \)
\( T_{sc} = 46.1^\circ C \)
\( T_{sh} = 18.3^\circ C \)
\( m = 87.9 \text{ g/s} \)
\( W_e = 4.153 \text{ kW} \)
Speed: 3500 RPM
\( V_D = 64.9 \text{ cm}^3/\text{rev} \)

Figure 4.2 Performance parameters and T-s diagram of Copeland ZF06K4E-PFV scroll compressor in nominal operation conditions.

Manufacturer catalogue data for nominal operation

\( T_c = 43.3^\circ C \)
\( T_e = -23.3^\circ C \)
\( T_{sc} = 43.3^\circ C \)
\( T_{sh} = 18.3^\circ C \)
\( m = 20.96 \text{ g/s} \)
\( W_e = 1946 \text{ W} \)
Speed: 3500 RPM
\( V_D = 34 \text{ cm}^3/\text{rev} \)
Figure 4.3 Performance parameters and T-s diagram of Hitachi G300DL scroll compressor in nominal operation conditions.

Manufacturer catalogue data for nominal operation

- \( T_c = 54.4°C \)
- \( T_e = 7.2°C \)
- \( T_{sc} = 54.4°C \)
- \( T_{sh} = 21.7°C \)
- \( m = 73.51 \text{ g/s} \)
- \( W_e = 3750 \text{ W} \)
- Speed: 3500 RPM
- \( V_D = 47 \text{ cm}^3/\text{rev} \)

Hitachi G300DL
Refrigerant: R407C
VR=3.31

Figure 4.4 Performance parameters and T-s diagram of Sanyo C-SBN303LBA scroll compressor in nominal operation conditions.

Manufacturer catalogue data for nominal operation

- \( T_c = 50.0°C \)
- \( T_e = -15.0°C \)
- \( T_{sc} = 50.0°C \)
- \( T_{sh} = 18.3°C \)
- \( m = 79.21 \text{ g/s} \)
- \( W_e = 4450 \text{ W} \)
- Speed: 3500 RPM
- \( V_D = 66.8 \text{ cm}^3/\text{rev} \)

Sanyo C-SBN303LBA
Refrigerant: R404A
VR=5.77
Figures 4.1 to 4.5 present the calculated thermodynamic cycles of 5 selected scroll units in \( T-s \) coordinates. The determined nominal volume ratio is indicated on each diagram. As seen in these figures, the temperature at the compressor discharge varies from 90°C -120°C. This temperature imposes the maximum temperature in the ORC with no significant modifications of the unit’s design. Since scroll units in the air conditioning industry can handle higher temperatures, the test units should be chosen from there. Another observation is the range of volume ratios, which varied from 2.8 to 5.8. This range is reasonable for the chosen machine to operate in reverse as a single stage expander for an ORC. This facilitates the design of a compact organic Rankine power generation system.

The selected units are from Bristol and Bitzer. These units have approximately the same range of pressure and volume ratio. The Bristol one has a power rating 300% greater than the Bitzer unit. This selection allows for testing at lower and higher power ranges with units having similar characteristics. Recall that the power range for the present investigation is set to 1.0-5.0 kW. Table 4.1 depicts that the Bristol unit is hermetic, while the Bitzer one is semi-hermetic.
Moreover, the Bitzer unit incorporates a permanent magnet motor which can be reversed to work as generator. These facts are an excellent advantage with regards to an experimental design.

4.3 Modifications of the Selected Scroll Units for Operation as an Expander

The hermetic Bristol H20R483DBE is carefully cut to remove the check valve and to gain access to the output shaft. The stator of the electric drive is removed because it is not required to operate the unit in expander mode. The expander in this modified version later incorporated on a dynamometer bench.

A cut-away view of Bristol scroll compressor with major components labelled is shown in Figure 4.6. The scroll unit is located in the top portion of the shell. The fixed scroll is rigidly attached to the shell while the meshing orbiting scroll is fitted to an eccentric shaft. The motor is bottom mounted and the rotor of the motor is shrink-fitted to the shaft. On the other hand the stator is attached to the shell wall by compression fitting. The shaft is supported by two bearings, one is in the crank case and the other is below the motor. The crankshaft transmits a torque to the orbiting scroll from the electrical motor.

Figure 4.6 Cut-away view of the hermetic Bristol H20R483DBE compressor.
Figure 4.7 End cap of the Bristol unit showing the displacement of the check valve.

Figure 4.8 illustrates top and bottom view of the scroll unit. The scroll wrap and the intake port are visible in the top view. In the bottom view, the journal bearing is visible where the motor shaft is inserted in the bushing of the orbiting scroll in an eccentric position. The eccentric shaft which drives the orbiting scroll is shown in Figure 4.9.

Figure 4.8 Top and bottom views of Bristol scroll unit.
The geometrical dimensions of the scroll wrap (thickness, height, pitch and rolling angle) are measured. Based on these parameters all the geometrical characteristics of the scroll unit can be calculated. The geometrical parameters are further used for calculation of the built-in volume ratio and the leakage paths. These are both important quantities for the thermodynamic modeling of the expander.

Table 4.2 Measured geometric dimensions of the Bristol scroll wrap.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Measured value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height of the scroll wrap</td>
<td>$h$</td>
<td>29 mm</td>
</tr>
<tr>
<td>Thickness of the scroll wrap</td>
<td>$\delta$</td>
<td>3 mm</td>
</tr>
<tr>
<td>Scroll pitch</td>
<td>$p$</td>
<td>16 mm</td>
</tr>
<tr>
<td>Rolling angle</td>
<td>$\varphi_e$</td>
<td>$8\pi$</td>
</tr>
</tbody>
</table>

The cut-away view of the Bitzer scroll unit is shown in Figure 4.10. This unit is designed for air conditioning applications in vehicles. It also includes a low voltage motor with 26 V DC power supply. A permanent magnet motor is incorporated in the same housing as the scroll unit. There is also an electronic block which has an inverter that converts the DC current into three-phase AC current. This is required to drive the unit when it is in compressor mode. In expander-
mode operation, a three-phase AC current is generated as the shaft rotates and the inverter plays the role of a rectifier to transform the AC current to DC current.

Figure 4.10 Cut-out view Bitzer ECH209Y-02G of the semi-hermetic compressor.

Figure 4.11 Side view of Bitzer ECH209Y-02G scroll unit.
The Bitzer scroll unit is shown in Figure 4.11. The check valve of the Bitzer unit is turned aside as indicated in Figure 4.12 in order to let the unit to operate as an expander.

![Diagram of a Bitzer unit with labeled parts: Check valve displaced, Intake port, Gasket channel.]

Figure 4.12 End view of the Bitzer unit showing the displacement of check valve.

**4.4 Expander-Dynamometer Experimental System**

The first experimental system used within this work is the expander-dynamometer test bench depicted in Figure 4.13. The objective of this system is to test expander operation with compressed air. The reason of using compressor air in expander testing is that running the tests in open loop system, it’s easy to adjust pressure ratio with the help of a forward pressure regulator. Moreover, air is a substance with well known thermodynamic properties; which for the range of experimental conditions can be assumed to behave close to ideal gas. This increases both simplicity and accuracy in experimental data processing. Testing low power positive displacement machines with compressed air in open loops is a common practice (see Prins and Zaytsev, 2001). One practical problem with open loop testing system has been identified as that of lubrication system. The normal lubrication system has been ceased because of modification. In the present experimental system, the lubrication is performed by splashing oil generously to the moving components before each run.
The conceived test bench is simple. It comprises a commercially available dry air cylinder compressed to 190 atm. and contains approximately 80 standard cubic meters of air. This quantity of compressed air is sufficient for running lab-scale tests with small expander units. The pressure regulator allows for the setting of the operating pressure at which the test is performed. Since the fact that the expander discharges to the open atmosphere, one can obtain the desire pressure ratio for testing by adjusting the upper pressure. The flow rate of air is measured by a Rotameter (which typically gives the readings in liter per minute). In order to accurately determine the mass flow rate of air, pressure measurements are performed before and after the flow meter. Finally, the temperature of air is measured at the discharge with a thermocouple. Based on the average pressure over the flowmeter and the air temperature, air density is determined. Mass flow rate is calculated by multiplying density and volumetric flow rate. The expander shaft is coupled to a simple dynamometer to determine the torque generated. The rotational speed of the shaft is measured in RPM with a tachometer. Based on the rotational speed and the torque, the shaft power is determined. The mass flow rate, upper and lower pressure, and the gas temperature measurements allow for the computation of the power delivered by the gas. Through comparison of the power generated at the shaft and the power produced by the gas, the expander efficiency can be determined. The experimental system is
versatile and allow for testing over a good range of conditions. A picture of this experimental system is shown in Figure 4.14.

![Figure 4.14 Expander-dynamometer test bench: front and side views.](image)

The dynamometer is constructed with a set of linkages and a brake pad to apply controllable brake load on the shaft of the expander unit. The brake pad is made of rubber and affixed on a pivoted lever. Attached to the lever a weighing scale is positioned in such a way that its spring can apply tension load to the lever.

This expander-dynamometer system is set up to determine the torque with the applied load on the expander shaft as shown in Figure 4.15. The reaction force $R_x$ can be calculated by taking the moment as

$$R_x \times (6\text{cm}) = F_2 \times (11\text{cm}) \rightarrow R_x = F_2 \times \frac{11}{6}$$

(4.4)

where $F_2 = m \times g$, therefore,

$$R_x = mg \times \frac{11}{6}$$

(4.5)
The coefficient of friction of the rubber pad in an aluminum surface was measured with an inclined plane. This inclined plane is presented in Figure 4.16.

A piece of rubber from the same material as used in the brake shoe is attached to a piece of iron bar of 0.25 kg and placed in an aluminum channel with a sliding surface of one meter in length. The inclination of the channel is then slowly increased from the horizontal position till the piece of rubber started to slide. At this position the gravitational component is just equal to the resistive force of friction and can be written as \( Fr = \mu N \), where \( N \) is the normal force on the inclined plane. The force balance is written as

\[
W \sin \theta = \mu \times W \cos \theta
\]  

(4.6)

where \( \mu = \tan \theta \), the coefficient of friction. The determined value of \( \mu \) is found to be approximately 1.04. A detailed view of the dynamometer setting is shown in Figure 4.17.
The main steps followed to conduct an experimental run are:

- Fully open the flow meter control valve.
- While the line flow control valve is fully closed, opened the cylinder supply valve slowly to achieve a desired regulated pressure.
- The inline flow control valve is then opened slowly to start the scroll expander and placed at full open position.
- With expander running the rotational speed in RPM is measured first with no dynamometer load and then with application of braking loads.
- Braking load is applied with varying magnitude and in each case after stabilization of shaft rotational speed data recorded.

### 4.5 Expander-Generator Experimental System

The second experimental system developed in this work is an expander generator test bench, where a scroll expander turns an electric generator that is connected to an adjustable electrical load. As shown in Figure 4.18 the system is similar to the previous expander-dynamometer setup. The difference is that, instead of measuring expander output by a dynamometer, this system facilitates the measurement of output power by the built-in generator output driven by the expander.
A 3-phase generator was chosen because it is more versatile with respect to RPM-torque characteristics. However, the output of a three-phase generator is not easy to measure with regular electric instruments (Multimeter). The reason of this is due to the fact that the frequency of the generated AC current is proportional to the RPM. Since ammeters are calibrated to 60 Hz, their measurement capability with variable frequencies is not accurate. It was decided to rectify the AC current and take the electrical measurements in DC. As indicated in Figure 4.18 a rectifier is included in the experimental system. The electrical load is of resistive kind and adjustable in steps with switches. A simple DC ammeter and voltmeter are connected in the loop to determine the electric power. Moreover, the measured voltage is proportional to the RPM. Knowing the manufacturer characteristics of the generator – especially its nominal voltage and RPM – one can approximate the actual RPM through voltage measurements. From these data, the torque generated can be determined. Regarding the measurements on the “Gas side”, procedure is same as that previously described. Based on the power determination on the “gas side” and “electric side”, the efficiency of the expander is calculated. Since the test bench allows for conducting experiments for a range of operating conditions, a comprehensive expander investigation can be completed.

In this experimental set-up, Bitzer ECH209Y-02G expander-generator unit has been incorporated to the system. Dry compressed air is supplied through the hose and via the flow meter to the expander. The procedures followed are:
• The in-line valve was opened at the desired pressure. The effect of this was that the expander started to turn immediately.
• When the pressure and flow rate appeared to be steady, the electric lamp switch was turned on to apply a load to the system.
• With this configuration voltage, current, fluid flow, pressure and temperature were recorded.

4.6 ORC Bench Experimental System
The third experimental system is the ORC test bench depicted in Figure 4.19. The test bench is a closed loop configuration comprising an expander, an air cooled condenser, a compressor, a boiler and other auxiliary components. The compressor is a reciprocating type for refrigeration application capable to operate with high pressure ratio and high discharge pressure. Such previsions are taken to have wide flexibility in adjusting the operating parameters during experiments.

The heater is a radiant electric heater and composed of six heating elements connected in parallel to operate individually through switches. The heater is designed to operate manually and to adjust the fluid temperature up to 200 °C. The reason behind this is to simulate low temperature heat sources that can be obtained from renewable or waste heat sources.

There are two bypass lines; one is in the liquid side and the other is in the vapour side. The liquid side throttle valve is installed to manipulate flow rate through heater and expander. The vapour by-pass line has the ability to completely isolate the expander and the installed throttle valve can act as an expansion valve in the cycle.

A number of thermocouple probes, four pressure gauges and a liquid flow meter are installed within the system as shown in the process and instrumentation diagram (Figure 4.19). A fluid filter dryer is also placed within the system to prevent any solid material incidentally present during test bench construction. A data acquisition system is used to record the temperature and flow meter reading during operation.
The expander of the ORC system is coupled to an electric generator. The measurement of the electric generator outputs is extremely important for the experimental investigation. This is because the electric power is the desired output of the heat engine. The electrical diagram and electrical measurements are indicated in Figure 4.20. The three-phase current is rectified and a resistive load is connected to the DC circuit. The voltage potential and current are measured to determine the power. Optionally, the current can be measured for each phase in AC. circuit ($I_1$, $I_2$, $I_3$ in Figure 4.20) and the voltage can be measured between every two phases ($V_{12}$, $V_{23}$, $V_{31}$). Since the focus of the experiments with the ORC test bench is on the expander performance,
valuable information can be drawn with regards to the overall ORC operation. In this configuration, the Bitzer ECH209Y-02G expander-generator is found as a suitable candidate for the test bench.

Figure 4.21 Photograph of the ORC experimental system (front view).

Figure 4.22 Photograph of the ORC experimental system (side view).
The overall view of the ORC test bench is presented in Figures 4.21 and 4.22. It incorporates the Bitzer ECH209Y-02G expander-generator unit. Other components are the boiler and the condensing unit (Tecumseh AWA2460ZXD). These are discussed in the subsequent paragraphs.

The boiler design is indicated in Figure 4.23 and consists of a copper coil through which the working fluid flows. Twelve 300 W electrical resistant heaters placed inside of the coil in a circumference without touching the coil. The whole assembly is placed in a cylindrical shell of which the outer side is insulated with glass wool. The copper coil radius is 150 mm while its tube outer diameter is 3/8 inch. The heating elements are supplied with 110V AC electrical power.

![Figure 4.23 Sketch of the cut away view of the boiler.](image-url)
The inside view of the boiler is shown in Figure 4.24 and a picture of the boiler is presented in Figure 4.25.

Figure 4.24 Inside view of the boiler shell.

Figure 4.25 Front view of the boiler.
The electric circuit for the heater elements is indicated in Figure 4.26. As shown in the circuit diagram the pair of heaters is connected to live line via a manual switch. Additionally an auto transformer is connected to the sixth pair to control part of the load.

![Electrical circuit of the boiler heater](image)

Figure 4.26 Electrical circuit of the boiler heater.

The boiler is designed based on heat transfer calculations that account for combined heat transfer by convection and radiation between the electric heaters and the coil tube assumed at constant (boiling) temperature. A separate heat transfer calculation is made on the working fluid side under the assumption of constant heat flux at the wall and forced convection boiling in tube. The lateral pipe of the tube coil has been determined as \( A = 0.296 \text{ m}^2 \), and the heat transfer coefficient at natural convection inside the boiler (between the heaters and the coil) is estimated to be approximately \( U = 10 \text{ W/m}^2\text{K} \). The heat transfer by convection is calculated with

\[
Q_c = U \times A \times (T_r - T_w)
\]  

(4.7)

where \( T_r \) represents the temperature of the heating element at its surface and \( T_w \) is the temperature of the outer side of the coil tube. The radiation component of heat transfer is calculated with

\[
Q_r = \sigma \times A \times (T_r^4 - T_w^4)
\]  

(4.8)

where \( \sigma \) is the Stefan-Boltzmann constant and \( \sigma = 5.6703 \times 10^{-8} \text{ (W/m}^2\text{K}^4) \).
For the considered design conditions, the average temperature of the tube has been determined based on the enthalpy of the working fluid. Namely, the working fluid temperature increases from $T_{sc}$ that corresponds to the subcooled state ($h_{sc}$) to saturation temperature $T_b$ at which the enthalpy of the saturated liquid is $h_{liq}$ and that of saturated vapour is $h_{vap}$, up to the temperature of superheated vapour $T_{sh}$ for which the enthalpy is $h_{sh}$. Thus the average temperature of the working fluid in the boiler is taken as

$$\bar{T} = \frac{0.5(h_{liq}-h_{sc}) \times (T_{sc}+T_b) + (h_{vap}-h_{liq}) \times T_b + (h_{sh}-h_{b}) \times (T_{sh}+T_b)}{h_{sh}-h_{sc}}$$ (4.9)

Assuming the extreme heating of the working fluid to be 200°C, the average temperature of the working fluid (R134a) is calculated at 105°C. Due to this, the total heating power of the boiler heating system is calculated to be 3.6 kW. Assuming that all this power is delivered to the working fluid and the temperature of the wall can be approximated to the average temperature of the working fluid, then the radiation heat transfer is determined as 2.775 kW. The convective heat transfer component is only 0.825 kW. The temperature at the surface of the heater elements is estimated at 380°C. As stated above, the heat transfer analysis has also been conducted for convective boiling. This has been done both for experimental system design and data processing.

Figure 4.27 illustrates the condensing unit (Tecumseh, AWA2460ZXD). This unit is installed in the ORC experimental system. The compressor of this unit was found to have
complimented its use together with the Bitzer expander-generator within the same loop. The Tecumseh unit can satisfy the required circulated volumetric flow rate and pressure ratio of the Bitzer expander. The volumetric flow rate of the Tecumseh unit is 12.3 m$^3$/h, at a pressure ratio of 8, while the Bitzer unit necessitates 6.21 m$^3$/h at a pressure ratio of 3.7. Due to this, the Tecumseh unit can sustain the operation of the ORC expander with higher flexibility. The condensing unit comprises several elements. They are compressor, air cooled condenser, liquid receiver and accumulator. Despite the fact that the condensing unit is originally designed for R404A, it is compatible with R134a.

The ORC plant is constructed with copper piping and insulated thoroughly. It is leak-tested with pressurized nitrogen and then vacuumed with a double stage vacuum pump. The quantity of required working fluid is calculated based on the volume of the ORC plant. Only the pipe segments and equipments filled with liquid are considered for filling. R134a is purchased and scaled to meter the quantity of the charged fluid. The charged fluid mass was determined to be 3.5 kg.

The ORC system running and experimentation procedures are given as follows:

- Initially the two expander isolation valves are fully closed.
- The by-pass valve TV$_2$ is opened.
- The compressor is started.
- Three of six heaters are turned on.
- When the temperature of the heater reaches approximately 120$^\circ$C the expander is started by simultaneously opening the isolation valves and closing the by-pass valve.
- When the temperatures and pressures have stabilized, the generator power is connected to the load by switching on the electric lamps.
- During the experiments, liquid fluid flow is adjusted by the by-pass valve TV$_1$ to manipulate the operating parameters.
- The boiler outlet temperature, expander inlet temperature is adjusted by changing the liquid flow and manually turning on/off heating element.
4.7 Measuring Instruments

The experimental activities in this project are conducted in sequence, starting with the expander-dynamometer system, then the expander-generator system and finally the ORC system. This strategy permitted the use of the same set of measuring instruments for all experiments. In this section all the measuring instruments used are described.

The flowmeter is known as “rotameter”. It is used to measure the flow rate of the compressed gas. In the rotameter, the fluid flow raises afloat in a tapered tube. The float moves up or down in proportion to the fluid flow rate. The inlet of the flow meter is near the bottom of the tube and the outlet is near the upper end. When the lifting force of the gas equals the gravitational force exerted by the weight of the float, it reaches a stable position to give a constant flow rate. A linear scale reading of the float is converted to flow rate unit from a chart supplied by the manufacturer. The selected flowmeter is Omega FLR – 5571G with maximum airflow (end of scale) of 9180 cm³/min, (see Figure 4.28). The scale length is 150 mm with magnifying glass and the accuracy is ±3% at full scale (±275 cm³/min). The maximum operating pressure and temperature is 13 bar and 93°C, respectively.

![Figure 4.28 Gas flowmeter (Omega FLR – 5571G).](image)

The selected flowmeter (Figure 4.29) for measurement of the working fluid flow rate (R134a) is Omega FLR1010ST having ±1% accuracy from full scale and up to ±0.2%
repeatability. The measurement range is 0.1-1 (l/min) and the reading accuracy is ±10 ml/min. The flow meter signal in the form of voltage within the range 0-5 V is sent to data acquisition. A conversion chart is used to determine the flow in l/min. The pressure drop across the flowmeter is 0.41 bar at maximum flow rate. The flowmeter is supplied by an external source of 12.5 VDC.

![Liquid flowmeter (Omega FLR101ST)](image)

Figure 4.29 Liquid flowmeter (Omega FLR101ST).

The digital tachometer used in this experiment is a hand held (Figure 4.30). A fluorescent light points on the rotating surface to measure the rotational speed in RPM. A shiny tape is affixed on the shaft to track the rotation by the tachometer.

![Hand held digital tachometer](image)

Figure 4.30 Hand held digital tachometer.
For electrical measurements, a regular clamp-on ammeter is used which measures the DC or AC electric current. Note that this instrument, when used for AC current, is calibrated for 60 Hz; thus it must be used with care when measuring the generator as if the generator does not turn at nominal speed, the current frequency differs from 60 Hz. The electrical voltage has been measured with a regular multimeter. The ammeter and multimeter used are shown in Figure 4.31.

For temperature measurement, K-type thermocouple wires are used. The thermocouple wire is of 0.1 mm diameter insulated with silicon-based coating. In total 8 thermocouples are used. These are shown in Figure 4.32. The thermocouples are connected to the data acquisition system to record the temperature data. Omega DAQ PRO 5300 data acquisition system is used (Figure 4.32). The data acquisition system has 8 programmable input channels. The thermocouple reading is done with ±0.5% accuracy. The data acquisition has been also used to measure the flowmeter signal in the input range of 0-10 V with an accuracy of ±0.5%.
The pressures are measured with bourdon tube manometers. These are typically used for refrigeration applications. Two manifolds of this kind have been purchased, each of them comprising two manometers with different scale. The blue dial is the “low scale” and the red dial is the “high scale” (Figure 4.33). The blue dial has two scales, one measures vacuum gauge pressure from 0 to 30 mm Hg, while the other measures atmospheric gauge pressure from 0 to 500 psi. The red dial measures in the range of 0 to 800 psi gauge pressure. The manifolds are used also for various procedures like charging the refrigerant, drawing vacuum and pressurisation with nitrogen for leak test.

Figure 4.33 Manifold gauges for pressure measurement (Yellow Jacket).

Throughout the experimental procedures extra precautions are taken not to damage or change the calibration of the equipment.
CHAPTER 5: THERMODYNAMIC ANALYSIS

Thermodynamic analysis of an ORC and its components requires special consideration of multiple aspects, such as fluid dynamics, gas-dynamics and thermodynamics of expanding flows with non-linear behaviour, heat transfer, geometrical modeling of prime movers, and dynamics of rotating systems. In this chapter the thermodynamic analysis of a small scale ORC and its components are discussed. Emphasis is given to the modeling of the positive displacement machines, especially the scroll expander.

5.1 Thermodynamic Analysis of the ORC
The thermodynamic analysis of an ORC is performed for the purpose of design and optimizing the design during steady state operation. The cycle is analyzed in the T-s diagram for a convenient design point. Based on the cycle analysis, the required performance parameters of all cycle components are determined.

5.1.1 Cycle Modeling
Mass, energy, entropy and exergy balance equations must be written for all cycle components and for the overall cycle under the assumption of steady state operation. This is done in regard to a selected working fluid in a given cycle configuration and imposed operating conditions of sink and source. The general energy balance equations for thermodynamic analysis on control volumes (viz. cycle components) are given next:

- Mass balance equation:
  \[ \sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \] (5.1)
- Energy balance equation:
  \[ \sum \dot{m}_{\text{in}} h_{\text{in}} + \sum \dot{Q}_{\text{in}} + \sum \dot{W}_{\text{in}} = \sum \dot{m}_{\text{out}} h_{\text{out}} + \sum \dot{Q}_{\text{out}} + \sum \dot{W}_{\text{out}} \] (5.2)
- Entropy balance equation:
  \[ \sum \dot{m}_{\text{in}} s_{\text{in}} + \sum \left( \frac{\dot{Q}}{T} \right)_{\text{in}} + \dot{S}_{\text{gen}} = \sum \dot{m}_{\text{out}} s_{\text{out}} + \sum \left( \frac{\dot{Q}}{T} \right)_{\text{out}} \] (5.3)

where \( T \) is the temperature at which heat fluxes cross the system boundary.
- Exergy balance equation:
\[ \sum \dot{m}_{\text{in}} e_{\text{in}} + \sum \dot{W}_{\text{in}} = \sum \dot{m}_{\text{out}} e_{\text{out}} + \sum \dot{W}_{\text{out}} + \dot{E} \dot{x}_d \]  
(5.4)

where \( \dot{E} \dot{x}_d \) is the exergy destroyed.

### 5.1.2 Cycle Efficiency Definitions

The cycle efficiency can be defined in various ways, depending on what is important for the actual design. The most common approach is to calculate the ratio of net work output and the heat input. Assuming that the net work output is the work generated by the expander less the amount of work required to power the pump, the energy efficiency of the ORC is:

\[ \eta = \frac{\dot{W}_{\text{net}}}{\dot{q}_{\text{in}}} = \frac{\dot{W}_{\exp} - \dot{W}_{\text{pump}}}{\dot{q}_{\text{in}}} \]  
(5.5)

The exergy efficiency of the cycle can be calculated as the net work output divided by the thermal exergy input as follows:

\[ \psi = \frac{\dot{W}_{\text{net}}}{\dot{q}_{\text{in}} \times (1 - T_0/T_{\text{eq}})} \]  
(5.6)

In Eq. (5.6), the equivalent temperature \( T_{\text{eq}} \) can be determined as described below by Eq. (5.9), if we denote it with subscript “in”, which is representative of source temperature. Also the subscript “out” is representative of the sink temperature. With these definitions, the thermal exergy is defined as

\[ \dot{E} x_{\text{in}} = \dot{m} \times [h_{\text{in}} - h_{\text{out}} - T_0 \times (s_{\text{in}} - s_{\text{out}})] \]  
(5.7)

Denoting \( \dot{Q}_{\text{in}} = \dot{m} \times (h_{\text{in}} - h_{\text{out}}) \), Eq. (5.7) becomes

\[ \dot{E} x_{\text{in}} = \dot{Q} \times \left(1 - \frac{T_0}{h_{\text{in}} - h_{\text{out}}} \right) \]  
(5.8)

The expression (5.8) identifies the equivalent temperature of the thermal energy source as

\[ T_{\text{eq}} = \frac{h_{\text{in}} - h_{\text{out}}}{s_{\text{in}} - s_{\text{out}}} \]  
(5.9)

In the case that the heat engine generates power and heat, there becomes two useful outputs. These are not of the same nature, so we have the inability to add them. However, a “fuel” or thermal energy source utilisation efficiency can be defined as

\[ \eta_{\text{cog}} = \frac{\dot{W}_{\text{net}} + \dot{q}_{\text{net}}}{\dot{q}_{\text{in}}} \]  
(5.10)
where with $\dot{Q}_{\text{net}}$ one denotes the useful heat co-generated. Note that due to the arguments presented above, it is not appropriate that the $\eta_{\text{cog}}$ bear the name of energy efficiency. A more accurate calculation of the co-generation efficiency associated with an ORC is the ratio of output to input exergy. If we define $T_{\text{eq,in}}$ as the equivalent temperature of the heat source and $T_{\text{eq,cog}}$ the equivalent temperature of the co-generated heat, then the exergy efficiency of the ORC system can be written as

$$\psi_{\text{cog}} = \frac{w_{\text{net}}+\dot{Q}_{\text{net}}\times(1-\frac{T_0}{T_{\text{eq,cog}}})}{\dot{Q}_{\text{in}}\times(1-\frac{T_0}{T_{\text{eq,in}}})} \quad (5.11)$$

The total exergy destruction of the ORC can be calculated based on the exergy efficiency (either with or without cogeneration) as follows:

$$\dot{E}_{x,d,\text{ORC}} = \dot{E}_{x,\text{in}}(1-\psi) \quad (5.12)$$

Also, the specific equations (not balance equations) for the basic components of the ORC are summarized in Table 5.1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Specific equation</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump</td>
<td>Isentropic efficiency: $\eta_P = \frac{h_{\text{outs}}-h_{\text{in}}}{h_{\text{out,a}}-h_{\text{in}}}$</td>
<td>$h_{\text{in}}$ – for pump input $h_{\text{outs}}$ – for isentropic discharge $h_{\text{out,a}}$ – for actual discharge</td>
</tr>
<tr>
<td>Heat exchanger</td>
<td>Exergy efficiency: $\psi_{\text{HE}} = \frac{e_{x,\text{out,cold}}-e_{x,\text{in,cold}}}{e_{x,\text{in,hot}}-e_{x,\text{out,hot}}}$</td>
<td>cold – cold stream hot – hot stream</td>
</tr>
<tr>
<td>Expander</td>
<td>Isentropic efficiency: $\eta_E = \frac{h_{\text{in}}-h_{\text{out,a}}}{h_{\text{in}}-h_{\text{outs}}}$</td>
<td>$h_{\text{in}}$ – for expander input $h_{\text{outs}}$ – for isentropic discharge $h_{\text{out,a}}$ – for actual discharge</td>
</tr>
<tr>
<td>Condenser</td>
<td>Exergy efficiency: $\psi_C = \frac{e_{x,\text{out,coolant}}-e_{x,\text{in,coolant}}}{\dot{Q}_{C}\times(1-\frac{T_0}{T_C})}$</td>
<td>$\dot{Q}<em>{C}$ – condenser capacity $T</em>{C}$ – condensation temperature</td>
</tr>
</tbody>
</table>

5.2 Scroll Expander Analysis and Modeling

The expander is the component that requires the most attention during ORC cycle design and optimization. Several aspects such as geometrical modeling, governing equations, leakage flow
prediction, heat dissipation, and thermodynamic modeling must be considered. Since the scroll expander is the focus of this present thesis work, the discussion in this section is restricted to it. Thorough geometrical analysis is also discussed in this section.

### 5.2.1 Geometric Modeling

Geometric modeling is the core to any positive displacement expander analysis. In positive displacement machines the gas is enclosed in a cavity with variable volume. As opposed to piston-cylinder expanders it has crescent shaped working volume. It is important to know the volume of the cavity as the expander turns. The scroll profile is an involute spiral as indicated in Figure 5.1. The rolling angle \( \phi_e \) of the scroll represents the number of turns around its centre. Basically at any given rotation there are three types of symmetrical gas pockets: intake (at high pressure in the scroll center), expanded-crescent (at intermediate pressure), and discharge (at low-pressure).

![Figure 5.1 Geometrical descriptions of the scroll involutes.](image-url)
The equation describing the involute in polar coordinates is given as (Wang, 2005):

\[ r = r_b \times \sqrt{1 + (\varphi - \varphi_0)^2} \]  

(5.13)

where \( \varphi_0 \) is the initial angle of the involute, \( \varphi \) is the polar angle, and \( r \) is the polar radius; the \( \varphi_0 \) can refer to the inner (\( \varphi_0 = \varphi_{i0} \)) or outer (\( \varphi_0 = \varphi_{o0} \)) involutes, depending on the case.

The rolling angle is the ending angle of the involute and it influences the built-in volume ratio of the expander. During the intake process, the gas enters the space between the two mated scrolls at the center. The pocket size increases as the scroll orbits. After a certain angle when the intake process ends, the initial pocket is divided into two small crescents locked between scrolls. During this process, the intake cavity volume becomes (Wang, 2005):

\[ V_{ei}(\theta) = h_r r_0 \theta \times (\theta - \varphi_{i0} - \varphi_{o0} + 3\pi) \]  

(5.14)

where \( r_0 \) is the orbiting radius, and \( \theta \) is the orbiting angle.

The intake process takes one full turn, thus the Eq. (5.14) is valid for \( 0 \leq \theta < 2\pi \). After being locked between the scrolls the gas starts to expand as its volume increases according to:

\[ V_{ee}(\theta) = 2\pi h_r r_0 (2\theta - (\varphi_{i0} - \varphi_{o0} - \pi)) \]  

(5.15)

The number of turns for the expansion process to complete depends on the length of the spiral, which is given by the rolling angle; the larger the rolling angle, the longer is the expansion time. Thus the expansion process occurs for \( 2\pi \leq \theta < \varphi_e - 2\pi \). The discharge process corresponds to the last turn, \( \varphi_e - 2\pi \leq \theta < \varphi_e \) and is given by:

\[ V_{ed}(\theta, \varphi_e) = h_r r_0 \times (2\varphi_e - 2\theta) \varphi_e - (\varphi_e - \theta)^2 - (\varphi_e - \theta) \times (\varphi_{i0} + \varphi_{o0} + \pi) + 2 \times (1 - \cos(\varphi_e - \theta)) - 2(\varphi_e - \pi) \sin(\varphi_e - \theta) - \frac{\pi}{4} \sin(2(\varphi_e - \theta)) \]  

(5.16)

The volume of the pocket at the beginning of the discharge process (\( \theta = \varphi_e - 2\pi \)) divided by the volume of the pocket at the end of the intake process (\( \theta = 2\pi \)) represents the built-in volume ratio (BVR) of the expander. Based on Eqs. (5.14, 5.15), one obtains the expression for the BVR:

\[ \text{BVR} = \frac{V_{ee}(\varphi_e - 2\pi)}{V_{ei}(2\pi)} = \frac{2\varphi_e - \varphi_{i0} + \varphi_{o0} - 3\pi}{5\pi - \varphi_{i0} + \varphi_{o0}} \]  

(5.17)
5.2.2 Governing Equations

A positive displacement expander can be represented for thermodynamic modeling purposes as a reciprocating piston-cylinder. The gas pocket entrapped between the expander scroll can be represented as a control volume, as indicated in Figure 5.2. From the higher pressure side there are flow leakages into the control volume, while other flow leaks-out toward the lower pressure regions.

The mass balance equation in differential form reads: $dm = (dm)_{in} - (dm)_{out}$. If one replaces the control volume mass with $m = V/v$, where $V$ is the volume of the control volume and $v$ is the specific volume of the fluid entrapped inside, then the mass balance equation can be written in function of the scroll orbiting angle as follows:

$$\frac{dv}{d\theta} = -\frac{v}{v} \left( v \times \left[ \frac{(dm)_{in}}{d\theta} - \frac{(dm)_{out}}{d\theta} \right] - \frac{dv}{d\theta} \right)$$

(5.18)

Figure 5.2 Representation of the control volume “expansion cavity”.

The energy conservation equation for control volume is written as:

$$\delta Q + (h \times dm)_{in} - h \times (dm)_{out} = d(m \times u) + \delta W$$

(5.19)

where $\delta Q$ is the heat transferred across the control volume boundary with the exterior during an infinitesimal scroll orbiting $d\theta$ and is assumed positive as heat is added to the control volume. The second term: $(h \times dm)_{in}$ is the sum of all leakage flows, $h$ in the third term of the equation is the actual fluid enthalpy, $(dm)_{out}$ is the sum of all leakages out of the system, $u$ is the internal energy of the control volume and $\delta W$ is the work generated by the expansion process which can be expressed as $\delta W = P \times dV$; thus Eq. (5.19) can be transformed into:

$$\frac{du}{d\theta} = -\frac{v}{v} \times \left[ P \times \frac{dv}{d\theta} + (u + Pv) \times \frac{(dm)_{out}}{d\theta} - \left( h \times \frac{dm}{d\theta} \right)_{in} - \frac{\delta Q}{d\theta} \right]$$

(5.20)
Eqs. (5.18, 5.20) must be accompanied by an equation of state specific to the working fluid, in order to be solved. The equation of state relate the specific volume and internal energy to enthalpy, entropy, pressure, temperature which are quantities needed to estimate various terms in Eq. (5.18) and Eq. (5.20). For the scroll expander case, the derivative $dV/d\theta$ that appears in Eq. (5.20) can be extracted from Eq. (5.15) and becomes:

$$\frac{dV}{d\theta} = \frac{d}{d\theta} [V_{ee}(\theta)] = 4\pi hr_tr_o$$  \hspace{1cm} (5.21)

The terms $\left(\frac{d\dot{m}_{out}}{d\theta}\right)$, $\left(h \times \frac{d\dot{m}}{d\theta}\right)_in$ and $\frac{\delta Q}{d\theta}$ in Eq. (5.20) represent the major irreversibilities in positive displacement expanders that are due to leakage flows and heat dissipation respectively. These terms are treated in the next section.

5.2.3 Irreversibilities

The leakage (by-pass) flow in the scroll expander is similar to orifice flow. In orifice, gas is forced to pass from high pressure to low pressure through a small passage area. The leakage flow in scroll machines sharply changes its direction. The physical mechanism flow acceleration causes pressure reduction. Additional pressure drop occurs due to the directional change of the fluid stream. In principle, the leakage flow regime is between a Fanno flow and isentropic flow.

The schematic from Figure 5.3 illustrates the leakage flow pattern between the fixed scroll and the orbiting scroll. If the area of the “vena contracta” is smaller than the area of the leakage path, then the flow velocity at the gap entrance can be neglected with respect to the velocity of the flow at exit.

Figure 5.3 Leakage flow path and the “vena contracta” [adapted from Zamfirescu et al. (2004b)].
Trutnoski and Komotori (1981) had shown that the leakage mass flow rate can be calculated using the area of the vena contracta as follows:

\[
\dot{m}_{\text{Leak}} = A_{\text{vc}} \sqrt{\frac{2\Delta P}{\bar{\nu}}}
\]  

(5.22)

where \( \bar{\nu} \) is the average specific volume of the flow, \( A_{\text{vc}} \) is the area of the vena contracta, and \( \Delta P \) is the pressure drop across the gap. The area of the vena contracta for non-choked flow is approximately 70-80% of the area of the gap. For modeling the leakages in the scroll machine, Oralli et al. (2010a, b) used a leakage coefficient defined based on Eq. (5.23) according to:

\[
\zeta = \dot{m}_{\text{Leak}} \times \left( \frac{p_2}{v_2} - \frac{p_3}{v_4} \right)^{-0.5}
\]  

(5.23)

Zamfirescu et al. (2004a) established a calculation method of leakages within positive displacement machines operating with multiple working cavities and leakages paths. A flow network problem must be solved to determine all leakages simultaneously. The sensitivity of the pressure drop in this system is high.

During the expansion process in an ORC positive displacement expander, the gas goes from a state of high temperature to low temperature with a pressure drop. The location of the highest temperature is at the scroll center which is inlet of the expander. Heat transfer due to gas and rotating metal surface thermal interactions can be estimated with \( Nu = 0.023Re^{0.8} \) as indicated by Stosic et al. (1992). This kind of correlation was used for screw kind machines and reciprocating machines and internal combustion engines (Fagotti et al., 1994).

### 5.2.4 Torque-Angular Velocity Relationship

The torque-RPM relationship is very important for expander modeling because it relates the pressure forces to the shaft torque, and the angular velocity and the fluid acceleration. In a scroll expander, the gas pocket at the intake has zero radial velocity. As the scroll orbits, the fluid is accelerated toward the scroll periphery. For the scroll compressor, the flow acceleration is in the reverse direction as of the expander.

The force balance on the pocket of gas trapped in the case of scroll operation as compressor, which is accelerated from zero velocity (exterior of the scroll) to a maximum velocity (scroll core) under the influence of motor force \( F_{\text{mot}} \), adverse pressure gradient force and friction force \( F_{\text{fr}} \) can be represented as:
\[ F_{\text{mot}} = F_{\text{pf}} + ma \] (5.24)

where \( m \) is the mass of the gas pocket, and \( a \) is the acceleration.

The duration of the compression cycle can be approximated by assuming uniform acceleration of the pocket for a distance \( R \) equal with the scroll radius, and is given by:

\[ t_c = \sqrt{\frac{2R}{a}} \] (5.25)

As the scroll orbits, the motor force, pressure and frictional forces generate torques with the arm equal to the orbiting radius \( r_o \) and can be represented by: \( T = r_o \times F \). By this definition, for one cycle the time period,

\[ t_c = \frac{2Rr_o m}{J_{\text{mot}} - J_{\text{pf}}} \] (5.26)

The number of revolutions per second is the inverse of cycle period defined in Eq.(5.26)

\[ RPS = \frac{1}{t_c} \] (5.27)

Thus the angular velocity is defined as:

\[ \omega = 2\pi RPS = 2\pi \frac{J_{\text{mot}} - J_{\text{pf}}}{2Rr_o m} \] (5.28)

From Eq. (5.24) one can derive here torque component to overcome pressure forces \( T_{\text{pf}} \) as

\[ T_{\text{pf}} = T_{\text{mot}} - T_{\text{acc}} = T_{\text{mot}} - 2Rr_o RPS \] (5.29)

where \( T_{\text{acc}} = 2Rr_o RPS \), is the torque applied to fluid pocket to accelerate it. It is reasonable to assume there is a relationship between torque \( T_{\text{pf}} \) and pressure forces. One can define the torque-pressure drop coefficient as

\[ K_p = \frac{T_{\text{pf}}}{\Delta P} \] (5.30)

In a similar manner one defines the angular velocity coefficient for fluid pocket acceleration as

\[ K_\omega = \frac{T_{\text{acc}}}{\omega^2} \] (5.31)

The power transmitted to the fluid to raise the pressure under a constant torque motor driving becomes

\[ \dot{W}_{\text{pf}} = \omega \times T_{\text{pf}} = 2\pi T_{\text{pf}} \sqrt{\frac{J_{\text{mot}} - J_{\text{pf}}}{2Rr_o m}} \] (5.32)
Note from Eq. (5.32) that the useful power component transmitted to the fluid to overcome the pressure forces has a maximum value at a specific torque $T_{pf}$ or pressure difference. Assuming compressor operation under the same suction conditions, the optimum $T_{pf}$ can be found by differentiating Eq. (5.32) and equating to zero. The result was found to be

$$T_{pf,\text{opt}} = \frac{2}{3} T_{\text{mot}}$$  \hspace{1cm} (5.33)

This expression which gives the optimum acceleration torque as: $T_{\text{acc, opt}} = \frac{1}{3} T_{\text{mot}}$.

It is reasonable to assume that the compressor is designed to produce maximum useful effect at rated conditions. This means that at rated conditions, the work transmitted to the fluid to raise the gas pocket pressure must have a maximum value. Denoting the rated torque and angular velocity with $T_r$ and $\omega_r$, respectively, one obtains expressions for the torque-pressure and the angular velocity coefficients as follows:

$$K_p = \frac{2}{3} \frac{T_r}{(\Delta P)_r}$$
$$K_\omega = \frac{1}{3} \frac{T_r}{\omega_r^2}$$  \hspace{1cm} (5.34)

Thus the maximum transmitted power to the fluid to overcome pressure forces is

$$W_{pf,\text{max}} = \frac{2}{3} T_r \times \omega_r$$  \hspace{1cm} (5.35)

The angular velocity becomes maximum when the gas pocket acceleration is maximum. This occurs when all motor power is used for acceleration and none for pressurization. In this case, the maximum angular velocity becomes

$$\omega_{\text{max}} = \omega_r \times \sqrt{3 \frac{T_{\text{mot}}}{T_r}}$$  \hspace{1cm} (5.36)

As an expander, the energy balance is written differently as

$$T_{pf} = T_{\text{acc}} + T_{\text{out}}$$  \hspace{1cm} (5.37)

This shows that the balance reads as follows: the energy developed by the working fluid is used partly to accelerate the working fluid and is delivered partly as useful energy in the form of shaft work. All torque-angular velocity equations for expander operation are similar to those for a compressor. The only difference being that the work developed by the pressure forces is the “motor” work, while the generated shaft work is the “useful” work.
5.3 Thermodynamic Analysis of Compressor Operation

In this section, the modeling procedure and results for a number of scroll compressors is presented. The compression process in any kind of positive displacement compressor implies closing the volume of gas in a compression chamber which reduces its volume according to the built-in volume ratio. In an ideal case there would be no leakage occurring and all the flow experiences an isentropic compression process from state 1 to state 2s as shown in Figure 5.4. Figure 6.2 illustrates the process on a $T$-$s$ diagram (presented in the results and discussion section). The amount of work required for this part of compression process is denoted $W_s$. In real compression process the gas does not compress isentropically. Friction and other irreversible processes require additional work input. This additional work which is substituted from the work input is transformed into heat that increases the entropy and enthalpy of the gas. Therefore, after the isentropic process 1-2s it follows an isochoric pressure rising process driven by heat addition. This added heat dissipates into the main flow thus increasing the pressure to reach state ‘2a’. But in a compression process the successive pressure being higher and due to the built-in gap between the moving and fixed parts, a portion of the fluid stream flows back to the suction side as a leakage flow. This leakage flow is a source of loss in the compressor performance and it varies with the variation of pressure ratio. The leakage flow rate can be estimated from energy balance applied to a simplified model. The simplified leakage flow model is shown along state 2a-3-4.

![Figure 5.4 Model of fluid flow through a positive displacement compressor.](image-url)
An isothermal expansion process is considered to merge this stream to a lower pressure stream of fluid. A mass balance and energy balance equations can be derived considering the above phenomena:

\[ \dot{m}_1 + \dot{m}_4 = \dot{m}_{1a} \]  

(5.38)

The enthalpies at states 2a, 2, 3 and 4 are equal as no process of heat addition or rejection is occurring along the states. Therefore:

\[ h_{2a} = h_2 = h_3 = h_4 \]

Also the specific volumes at 2a, 2, and 3 are equal as no thermodynamic changes are involved here:

\[ v_{2a} = v_2 = v_3 \]

An energy balance equation can be written as

\[ \dot{m}_1 h_1 + \dot{m}_4 h_4 = \dot{m}_{1a} h_{1a} \]  

(5.39)

The motor shaft work rate and the work rate for overcoming the friction can be expressed as

\[ W_s = \dot{m}_{1a} \times (h_{2sv} - h_{1a}) \]  

(5.40)

\[ W_{\text{diss}} = \dot{m}_{1a} \times (h_{2a} - h_{2sv}) \]  

(5.41)

where

\[ s_{2sv} = s_{1a} \]

\[ v_{2sv} = v_{2a} \]

\[ h_{2sv} = h(s_{1a}, v_{2a}) \]

\[ P_{2sv} = P(s_{1a}, v_{2a}) \]

\[ W_{\text{inp}} = W_s + W_{\text{diss}} \]  

(5.42)

The work required for isentropic compression and the work required to overcome friction can be expressed by a dimensionless parameter introduced here and denoted as isochoric pressure building coefficient:

\[ \Pi = \frac{W_{\text{diss}}}{W_s} = \frac{h_{2a} - h_{2sv}}{h_{2sv} - h_{1a}} \]  

(5.43)

The isentropic efficiency of the compressor is defined as

\[ \eta_s = \frac{h_{2a} - h_1}{h_2 - h_1} \]  

(5.44)

The built-in volume ratio (BVR) can be approximated by certain thermodynamic parameters, namely the specific volumes of the gas before and after the compression, as
BVR = \frac{v_{1a}}{v_{2sv}} \tag{5.45}

The volumetric efficiency is defined by

\eta_V = \frac{\dot{m}v_1}{V_D} \tag{5.46}

where \( V_D \) is the volume displacement rate given by

\dot{V}_D = \frac{V_D \times RPM}{60} \tag{5.47}

Here, \( V_D \) is defined as the volume of the cavity at the beginning of the compression process. This is the volume that will be displaced at high pressure if no back-flow leakages occur.

### 5.4 Thermodynamic Analysis of Expander Operation

The expander converted from a positive displacement compressor with the same geometrical features; in this case the scroll compressor works in a similar manner but in reverse. The fluid flows through the discharge port of the compressor, is expanded and exits from the suction port. When the flow is reversed to facilitate the expander operation, the direction of the orbital scroll movement also reverses itself. As the operating and functional parameters are not specified for expander, model derivation is not straight forward in this expansion process. A number of possibilities of operational characteristics must be considered. Three specific conditions are considered with reasonable assumptions.

- Case 1: Under expansion.

![Figure 5.5 Representation of under expansion process.](image_url)
Figure 5.5 shows the expander with a control volume boundary. In this expansion process high pressure fluid enters at state 1 through the expander inlet, which is the entry of the expander that represents the inlet port. If there are no frictional or leakage loss, fluid should exit state 2 at the condenser pressure after isentropic expansion. But in a “real” positive displacement expander, there exist both frictional and leakage losses. A flow stream is shown by passing the main flow as leakage and merges the low pressure discharge stream which is represented by path 3-4. This high pressure leakage flow has to reduce its pressure to condenser pressure which can be modelled as an isenthalpic expansion within this channel. To categorize under expansion, pressure ratio is adjusted by trial and error to reach a point ‘2sv’ in a process of isentropic expansion, higher than condenser pressure as demonstrated under expansion. The frictional loss which is compensated by a portion of the expander work ultimately generates heat and dissipates within the control volume. This heat adds to the fluid and increases the pressure of the main stream to state ‘2v’. Finally, the pressure has to reduce to the condenser pressure which can be modelled by another isenthalpic process shown by an expansion valve to reach the state ‘2a’. In equation form the work distribution for this model can be written as:

\[ W_s = W_{diss} + W_{out} \]  

(5.48)

- Case 2: Over expansion.

![Diagram of over expansion process](image-url)
Figure 5.6 is a schematic of the over expansion case. Over expansion can be thought as the expander outlet pressure dropping below the isentropic expansion pressure ‘2s’. In a closed loop system this state cannot exist. The final state of the exhaust in the expander should be equal to that of the condenser pressure. An internal pressure rise is required to elevate the pressure up to condenser pressure. The heat dissipation can fully or partially compensate to reach the condenser pressure at ‘2a’. In this case one may assume that after isochoric pressurization occurs due to the heat addition it reaches state ‘2v’. To further increase the pressure to the condenser pressure, additional work is required. This is taken from the isentropic work ‘W_s’. The net work output occurs in path 1-2a. However, the flow merges with leakage flow to reach the state ‘2’ which is the same as ‘2a’. The work generated after isentropic expansion can be written as:

\[ W_s = W_{diss} + W_{dp} + W_{out} \]  \hspace{1cm} (5.49)

- Case 3: Optimal expansion.

Figure 5.7 represents the optimal expansion process. As mentioned in the previous case about expansion, if the heat dissipation is enough to build up a pressure ‘2v’ to ‘2a’ the process would depict an optimal expansion process. In this case a portion of the work is compensated for only frictional resistances. Maximum work output is generated in this case. This process is defined as the ideal expansion process.
CHAPTER 6: RESULTS AND DISCUSSION

The research methodology followed in this work has been of two kinds; thermodynamic modeling in steady state operation and experimental investigation. The purpose of the experiments is validation of the theoretical models. This chapter reports and discusses the experimental results. In the first section the numerical results of thermodynamic modeling of the scroll machine and the ORC operation are presented. Thereafter, the experimental activity and results are comprehensively explained and reported. The last section refers to the validation of the models against the experimental results and evidence.

6.1 Thermodynamic Modeling and Parametric Studies

6.1.1 Modeling of Scroll Machine Operation as Compressor

In this section, the scroll machine is modelled as compressor and numerical results regarding its operation are obtained. Prediction of scroll machine operation as compressor has general importance for analysis and design of systems and equipment. In particular, when a scroll compressor is modified to be used as an expander, the simulation of the compressor operation serves as a means to identify and determine the required parameters for predicting the reverse operation. Two compressor models were introduced in the analysis section, one based on governing equation and the other based on thermodynamic modeling with equivalent diagram. The first approach is dismissed for this project as it is considered too expensive with regards to the outcomes. Also, that approach requires thorough geometrical modeling and calculation of leakage paths at any angle. Furthermore, it requires fluid dynamic, thermodynamic and heat transfer modeling. A thermodynamic modeling of compressor is found to be appropriate for this work.

Thermodynamic modeling requires writing the mass and energy balance equations for the compressors and their equivalent diagrams. The complete set of equations is already mentioned in the analysis section. A computer program has been implemented to solve it starting from the following set of parameters with known (imposed) values:

- Temperatures: evaporation, superheating, condensation, subcooling \((T_{ev}, T_{sh}, T_c, T_{sc})\).
- Evaporation heat rate \((\dot{Q}_e)\).
- Compressor motor work rate \( (\dot{W}_{\text{mot}}) \).
- Displaced volume \( (V_d) \).
- Shaft rotational speed (SRS).
- Working fluid type.

A program was written in EES to solve the set of equations for numerical simulation of the compressor operation based on equivalent diagrams. This program is included in the Appendix. A first example of simulated results with this computer code is shown in Figure 6.1. Based on the imposed evaporation and condensation temperatures, the solution algorithm determines the corresponding pressures. These are denoted as \( P_1 \) and \( P_2 \) respectively. State 1 represents compressor suction and state 2 the discharge (see Figure 6.1).

The imposed thermodynamic conditions at the compressor suction are the pressure and the temperature (given as \( T_{\text{sh}} \)). Based on the equivalent diagram approach, the compression process occurs isentropically from state 1-2sv and the isochorically from state 2sv-2a. The mass balance equation is written at suction and discharge as follows:

\[
\begin{align*}
\dot{m}_1 + \dot{m}_{1b} &= \dot{m}_{1a} \\
\dot{m}_{2a} &= \dot{m}_2 + \dot{m}_{2b}
\end{align*}
\]  
(6.1)
The energy balance equation is written at the suction side as
\[ \dot{m}_1 h_1 + \dot{m}_{1b} h_{1b} = \dot{m}_{1a} h_{1a} \] (6.2)

Figure 6.2 Compression process representation for Bristoll H20R483DBE on T-s diagram.

Recall that each process in the equivalent diagram is discussed in the analysis section. The calculated results include the unknown state parameters in each point of the equivalent diagram (see Figure 6.1) and additional important quantities, as introduced in the analysis chapter, namely:

- Built-in volume ratio (BVR).
- Isochoric pressure building work ratio (Π).
- Leakage coefficient (ζ).
- Angular velocity coefficient (Kω).
- Torque pressure coefficient (Kp).
- Isentropic efficiency (ηs).
- Volumetric efficiency (ηv).
- Torque to accelerate the fluid pocket (Tace).
- Torque to build pressure ($T_{pf}$).
- Motor torque ($T_{mot}$).

**Scroll compressor model: Copeland ZF06k4E-PVF**

**Manufacturer data**
- Working fluid: R404A
- $T_c = 23.3^\circ C$
- $T_1 = 45.3^\circ C$
- $T_1 = 12.2^\circ C$
- $Q = 2642$ W
- $P_{mot} = 1946$ W
- $V_p = 34$ cm³ rev
- SRS = 3500 RPM

**Determined parameters**
- $BVR = 6.1$
- $\zeta = 3.979E-07$
- $K_m = 2.994E-02$
- $K_s = 2.018E-06$
- $\eta = 81.67%$
- $\mu = 1.93$ Nm
- $\tau = 3.5$ Nm
- $T_{max} = 5.3$ Nm

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Figure 6.3 Modeling of Copeland ZF06k4E-PVF compressor in nominal operating conditions.

The process illustrated with numerical values on the equivalent compressor diagram in Figure 6.1 and also represented on a $T-s$ diagram as shown in Figure 6.2. The process 1a-2sv is the isentropic compression of the gas pocket locked between the compressor scrolls. The process 2sv-2a is the isochoric pressure build-up due to irreversibilities. The process 2b-1b is the representation of the isenthalpic internal leakage flow. At the compressor suction side it takes place an internal flow mixing process between the flow at the intake (state 1) and the leakage back-flow (state 1b). This summation forms the actual intake at state 1a.

Numerical calculations have been done in a similar manner with other 4 scroll machines selected above (Copeland, Hitachi, Sanyo, Bitzer). They all showed similar results. These results are presented in Figures 6.3, 6.5, 6.7 and 6.9 with pressures, temperatures, specific enthalpies, specific entropies and other relevant parameters indicated on the compressor’s equivalent diagram. The corresponding thermodynamic processes for nominal operation of the mentioned scroll units are represented on the $T-s$ diagrams in Figures 6.4, 6.6, 6.8 and 6.10.
The processes are similar for all considered cases that cover a range of temperatures from 20°C to 120°C, and pressures from 200 kPa to 2300 kPa. The calculated isentropic efficiencies of the studied scroll compressors are in the range of 46% to 69%; the smallest being that of Copeland unit and the highest being that of the Bristol unit. The Bristol unit has the lowest built-in volume ratio while the Copeland one has the highest. This seems to imply a relationship between BVR and $\eta_s$.

![Figure 6.4 Compression process representation for ZF06k4E-PVF on T-s diagram.](image-url)
**Scroll compressor model: Hitachi G300DL**

**Manufacturer data**
- Working fluid: R407C
- \( T_r = 72 \) °C
- \( T_i = 54 \) °C
- \( T_{in} = 21.7 \) °C
- \( Q = 9600 \) W
- \( \eta_m = 3750 \) W
- \( \eta_r = 47 \) cm rev
- \( SRS = 4072 \) RPM

**Detected parameters**
- \( BVR = 3.3 \)
- \( \eta_{V} = 0.49 \)
- \( \zeta = 7.603E-07 \)
- \( k_o = 1.529E-02 \)
- \( k_i = 3.625E-06 \)
- \( \eta = 59.3 \% \)
- \( \eta_{r} = 81.14 \% \)
- \( m_o = 2.9 \) Nm
- \( m_{r} = 5.9 \) Nm
- \( T_{ref} = 8.5 \) Nm

(\(^{(*)}\)) motor driving at 70 Hz, 3% slip

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**Figure 6.5** Simulation of Hitachi GL300DL compressor in nominal operating conditions.

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**Figure 6.6** Compression process representation for Hitachi GL300DL on T-s diagram.

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Working fluid: R407C

1-2: Actual process
1a-2sv: \( s = c_t \) process
2sv-2a: \( v = c_t \) process
2b-1b: \( h = c_t \) process

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Figure 6.7 Simulation of Sanyo C-SBN303LBA compressor in nominal operating conditions.

Figure 6.8 Compression process representation for Sanyo C-SBN303LBA on T-s diagram.
Figure 6.9 Modeling of Bitzer ECH209Y-02G compressor in nominal operating conditions.

Figure 6.10 Compression process representation for Bitzer ECH209Y-02G on T-s diagram.
Table 6.1 indicates the obtained values for the following parameters for each of the compressors: built-in volume ratio, isochoric pressure building work ratio, leakage coefficient, angular velocity coefficient, and torque-pressure coefficient. These parameters are relevant for scroll machine modeling as either a compressor or expander. One can observe that the built-in volume ratio varies from 2.9 to 6.1, the isochoric pressure building coefficient varies from 0.38 to 0.72, the leakage coefficient is approximately $5 \times 10^{-7}$, the angular velocity coefficient is around 0.02, while the pressure-torque coefficient varies around $4 \times 10^{-6}$. It is also observed the correlation between BVR and $\Pi$, which is illustrated graphically in Figure 6.11.

Table 6.1 Selected scroll compressor units and their main characteristics.

<table>
<thead>
<tr>
<th>Model</th>
<th>Parameter</th>
<th>BVR</th>
<th>$\Pi$</th>
<th>$\zeta$</th>
<th>$K_\omega$</th>
<th>$K_p$</th>
</tr>
</thead>
<tbody>
<tr>
<td>H20R483DBE</td>
<td></td>
<td>2.9</td>
<td>0.38</td>
<td>$5.67 \times 10^{-7}$</td>
<td>0.0173</td>
<td>$4.96 \times 10^{-6}$</td>
</tr>
<tr>
<td>ZF06K4E-PFV</td>
<td></td>
<td>6.1</td>
<td>0.71</td>
<td>$3.98 \times 10^{-7}$</td>
<td>0.0299</td>
<td>$2.08 \times 10^{-6}$</td>
</tr>
<tr>
<td>G300DL</td>
<td></td>
<td>3.3</td>
<td>0.49</td>
<td>$7.60 \times 10^{-7}$</td>
<td>0.0153</td>
<td>$3.63 \times 10^{-6}$</td>
</tr>
<tr>
<td>C-SBN303LBA</td>
<td></td>
<td>5.4</td>
<td>0.72</td>
<td>$2.63 \times 10^{-7}$</td>
<td>0.0287</td>
<td>$4.19 \times 10^{-6}$</td>
</tr>
<tr>
<td>ECH209Y-02G</td>
<td></td>
<td>3.5</td>
<td>0.61</td>
<td>$7.23 \times 10^{-7}$</td>
<td>0.0593</td>
<td>$4.30 \times 10^{-6}$</td>
</tr>
</tbody>
</table>

Figure 6.11 Correlation between the built in volume ratio (BVR) and the isochoric pressure building work ratio ($\Pi$).
Angular velocity and pressure difference across the compressor and the torque are all related to each other. Figure 6.12 indicated their relationship as calculated for the Bristol compressor. The above determined angular velocity coefficient ($K_{\omega}$) and pressure-torque coefficient ($K_p$) are used to draw the diagrams (Fig. 6.12 to 6.16) according to the theory developed in Chapter 5. It is assumed compressor driving with a constant torque motor.

Figure 6.12 Pressure difference vs. angular velocity and power for Bristol compressor.

Figure 6.13 Pressure difference vs. angular velocity and power for Copeland compressor.
Figure 6.14 Pressure difference vs. angular velocity and power for Hitachi compressor.

The compressor operation is influenced by the external (boundary conditions) conditions which are the suction pressure, discharge pressure and the suction temperature. These parameters must be specified.

Figure 6.15 Pressure difference vs. angular velocity and power for Sanyo compressor.
When the pressure difference across the compressor is small, a large proportion of the shaft power is used to accelerate the fluid. By consequence, the angular velocity and mass flow rate are higher as well. Under this condition, the power transmitted to the working fluid to increase its pressure which is the useful power of the compressor is negligible.

As the pressure difference increases, more power is consumed from the motor shaft to increase the pressure and less power is consumed to accelerate the flow. Consequently, the angular velocity decreases. Eventually the angular velocity will approach its minimum value, while the pressure difference approaches its maximum value. Under this condition, there is no flow, but maximum torque is developed to withstand the pressure forces. Since there is no flow, the power transmitted to the working fluid to increase its pressure again approaches to a minimum value.

It can be observed that in between the two extreme situations explained above, the compressor performance reaches some optimum value. At this point, the power transmitted to the working fluid to increase its pressure becomes maximum. The compressor must be sized around this optimum point. Figure 6.12 to 6.15 present the “simulated” angular velocity vs. pressure difference diagram for the selected compressors. Recall: these diagrams were made
based on the compressors parameters listed in Table 6.1. The same parameters are used in the next section to model and predict the operation of the scroll machine as an expander.

6.1.2 Modeling the Scroll Machine Operation as Expander

In expander operation, the power that turns the shaft is generated by fluid expansion across the scroll. Therefore, the energy balance can be stated with words as follows: the power generated by the expanding fluid is used partially to accelerate the fluid pocket and partially to generate useful shaft power. The acceleration of fluid pocket is necessary because the fluid enters the expander axially at the intake port placed in the center of the scroll. Thus, the fluid pocket – entrapped within the scroll vanes, has initially no radial and tangential velocity. It must be then accelerated from zero velocity and moved toward the scroll periphery. This action requires energy. Obviously, the energy to displace the pocket is taken from the work rate developed by the expanding fluid; this diminishes the net work output at expander shaft, but it is an unavoidable power loss for sustaining the scroll expander operation. The calculations made for Bristol scroll unit assuming 30°C vapour superheating at expander inlet are presented in Figure 6.16.

For the simulations presented in Figure 6.16 the pressure ratio is 4.0. Under this condition the expander operation has been found in the “under-expansion” regime. The process is demonstrated in the figure in a detailed manner. As discussed in the “Thermodynamic analysis” chapter, the overall expansion process is decomposed into a number of thermodynamic processes. The working fluid trapped inside the expansion cavity formed between mated scrolls and it increases its volume in according to the built-in volume ratio of the expander. This process is 1a to 2sv as indicated in the Figure 6.17. As mentioned before, the flows at the suction state divide into the main flow and the bypass flow. The main flow passes through the expander where it produces work, and by consequence generates entropy. The process is modeled as a succession of two “ideal” processes, the isentropic expansion (1a-2sv) followed by the isochoric pressure building (2sv-2v). At state 2v the flow pressure is still higher than pressure at the expander boundary; therefore the flow continues to expand by an isenthalpic process which may be considered a throttling process. The by-pass leakage flow is from state 3 to state 4 and it combines with main flow at state 2a through an isobaric mixing process that produces state 2.
Figure 6.17 Modeling of under-expansion for Bristol H20R483DBE scroll unit.
The results of the exergy destruction analysis for the under-expansion process discussed above are presented in Figure 6.18. The exergy destruction has been calculated for each subprocess indicated in the equivalent diagram in Figure 6.16. Note that the sub-process 1a-2sv doesn’t destroy exergy because it is isentropic. All other processes destroy exergy as they generate entropy as follows:

- Isochoric pressure building: \( \dot{m}_{2sv} x_{2sv} + W_{\text{diss}} = \dot{m}_{2v} \times e_{x_{2v}} + \dot{E}x_{d,\text{v} = ct} \).
- Flow throttling at discharge: \( \dot{m}_{2v} \times e_{x_{2v}} = \dot{m}_{2a} \times e_{x_{2a}} + \dot{E}x_{d,\text{throttling}} \).
- Internal leakage: \( \dot{m}_{3} \times e_{x_{3}} = \dot{m}_{4} \times e_{x_{4}} + \dot{E}x_{d,\text{leakage}} \).
- Discharge flow mixing: \( \dot{m}_{2a} \times e_{x_{2a}} + \dot{m}_{4} \times e_{x_{4}} = \dot{m}_{2} \times e_{x_{2}} + \dot{E}x_{d,\text{mixing}} \).

From the exergy analysis, it shows that the most of irreversibilities come from the improper adaptation of the expander operation to the external condition. The expander operates in under-expansion regime, which means that it expands less than required. Another important irreversibility is due to internal frictions which made expansion process non-isentropic.

The exergy efficiency of the expander is calculated with

\[
\psi = 1 - \frac{\dot{E}x_{d}}{m_{1} \times (e_{x_{1}} - e_{x_{2}})}
\] (6.3)
where $\dot{E}_{x_d}$ is the sum of exergy destructions for all processes. For the analyzed case the exergy destruction becomes:

$$\dot{E}_{x_d} = \dot{E}_{x_d,v=ct} + \dot{E}_{x_d,throttling} + \dot{E}_{x_d,leakage} + \dot{E}_{x_d,mixing}.$$
Figure 6.20 Exergy destructions during the over-expansion process as modeled above.

Figure 6.21 Modeling of optimal expansion for Bristol H20R483DBE scroll unit.
The other extreme case of expander operation is the “over-expansion”. Figure 6.19 presents the over-expansion process that has been modeled starting from the same intake conditions as those for the under-expansion case previously presented. The only difference is that of a lower pressure ratio. In this case the pressure ratio is 2.70. Due to over-expansion, the pressure after the expansion process is lower than the external pressure. Thus, in order to discharge the flow, the expander must consume additional work to pressurize the displaced working fluid pocket during discharge. This additional work is indicated in Figure 6.18 with $W_{dp}$ and is assumed to occur during a constant isochorically.

![Diagram of over-expansion process](image)

**Figure 6.22 Exergy destructions during the optimal-expansion process as modeled above.**

The exergy destructions for the over-expansion case are different than for the case of under-expansion; they are as follows:

- Isochoric pressure building: $\dot{m}_{2v}e_{2v} + W_{diss} = \dot{m}_{2v} \times e_{2v} + \dot{E}_{x_d,v=ct}$.
- Flow pressurizing at discharge: $\dot{m}_{2v} \times e_{2v} = \dot{m}_{2a} \times e_{2a} + \dot{E}_{x_d,\text{pressurizing}}$.
- Internal leakage: $\dot{m}_3 \times e_3 = \dot{m}_4 \times e_4 + \dot{E}_{x_d,\text{leakage}}$.
- Discharge flow mixing: $\dot{m}_{2a} \times e_{2a} + \dot{m}_4 \times e_4 = \dot{m}_2 \times e_2 + \dot{E}_{x_d,\text{mixing}}$.

Figure 6.21 illustrates the third case: “optimal expansion”. The same intake conditions are assumed as for the previous two cases. The pressure ratio in this case is 2.825.
ratio falls very close to the built-in volume ratio that has been predetermined as 2.9. In this case, the pressure inside the cavity at the moment of its opening is the same as the external pressure. Due to this, pressure rebuilds or throttling to adjust the expander operation to the imposed external conditions do not require. The exergy destructions for this case are indicated in Figure 6.22. The isentropic efficiency is at a maximum value of 66%. This is higher when compared to the 50% associated with under-expansion and the 37% associated with over-expansion.

![Figure 6.23 Isentropic ($\eta_s$) and exergetic ($\psi$) efficiency of Bristol expander as a function of pressure ratio.](image)

The maximum exergy efficiency of the expander is 64.4% under optimal-expansion. This is higher than the 48.4% associated with under-expansion and the 37.9% associated with over-expansion. All three above cases are summarized in Figure 6.23. Notice that the optimum operation point of the expander is sharp and that the vicinity around the optimum the isentropic efficiency experiences fast degradation.

It is interesting to remark that there is a sharp optimum operation point with respect to the degree of superheating. Figure 6.24 illustrates this with $20^\circ$C of superheating. All three curves are drawn assuming the same intake pressure. A conclusion of this study is that there is no notable benefit associated with the increased degree of superheating over certain value. Beyond
this critical value isentropic efficiency decreases rapidly. This degradation can be explained via thermodynamic reasons. If one fixes the intake and discharge pressures for an expansion process, the enthalpy difference between the high pressure and low pressure states becomes lower as the degree of superheating increases. The result is a decrease in specific work.

Another aspect studied with regards to expander operation refers to the intrinsic relationship between angular velocity, torque and pressure ratio. Starting with the same intake conditions as above, the pressure ratio has been varied from 2.6 to 8.0 and the results found are plotted in Figure 6.25. The torques plotted against pressure ratio are:

- The torque developed by the pressure forces.
- The torque consumed to accelerate the working fluid.
- The torque available at the expander shaft.

The optimum expansion condition is indicated in Figure 6.25. It is interesting to remark that the output torque (useful torque) remains constant throughout the under-expansion regime. Due to this, there is no reason to increase the pressure ratio beyond certain value which is regulated by optimal expansion process. Also, in the ‘under expansion’ regime, the flow rate

Figure 6.24 Influence of vapour superheating degree and pressure ratio on the expander’s isentropic efficiency (reversed Bristol H20R483DBE unit case).
increases with pressure ratio. This is due to higher angular velocities. Finally, in the “under expansion“ regime, the shaft power increases due to increased mass flow rate. The best operating conditions of the expander can be found at the transition between the over and under expansion regimes. In this part of the diagram, the shaft torque and the angular speed have opposite variation trends with respect to pressure ratio (see Figure 6.26).

![Diagram showing Expander torques and shaft rotational speed vs. pressure ratio](image)

Figure 6.25 Expander torques and shaft rotational speed vs. pressure ratio for the Bristol unit.

Figure 6.26 further clarifies expander operation. For shaft torque between 2.0-11.0 N.m, the expander rotational speed decreases from about 3600 RPM to 2000 RPM. The generated power is at a maximum of 2500 W around 2400 RPM and 10.4 N.m. The point for maximum power generation is not coincidental with the point of maximum isentropic efficiency. The conclusion is that for generating maximum power, the expander must operate slightly in the over-expansion regime.

The previous 10 diagrams drawn for the Bristol expander can be repeated for the other expanders and one can find similar results. These kinds of results are relevant with respect to the ORC cycle design. The main conclusion is that superheating should not be very high (20-80°C is sufficient). Also, the expander must operate close to the optimum expansion.
6.1.3 Parametric Studies of ORC with Scroll Expander

In this section, the ORC cycle with scroll expander operating in optimal conditions is studied. A computer program has been written to calculate all thermodynamic properties of an ORC from an initial condensation temperature. The computer program solves for the optimum intake conditions for the expander by determining the optimum pressure ratio and degree of superheating associated with maximum cycle efficiency.

The results for Bristol expander are shown in Figure 6.27. The optimum temperature at the expander inlet has been found to be 120°C. Therefore, source temperature must be around 130°C. The cycle energy and exergy efficiencies were found to be 6.4% and 58.3%, respectively. The expander isentropic efficiency is 66% while the generated power was found to be 4.2 kW. Note also that the assumed ORC configuration was found to be the regenerative one. The study from Figure 6.28 shows the influence of the degree of superheating on the expander and the cycle efficiencies. The deviation from the optimum degree of superheating makes the efficiencies decrease rapidly.
Figure 6.27 ORC with the Bristol expander.

Figure 6.28 Influence of the degree of superheating on the expander isentropic and cycle efficiency for the Bristol-based ORC.
Figure 6.29 Influence of the degree of superheating on the expander isentropic and cycle efficiency for the Copeland-based ORC.

Figure 6.30 Influence of the degree of superheating on the expander isentropic and cycle efficiency for the Hitachi-based ORC.
Figure 6.31 Influence of the degree of superheating on the expander isentropic and cycle efficiency for the Sanyo-based ORC.

Figure 6.32 Influence of the degree of superheating on the expander isentropic and cycle efficiency for the Bitzer-based ORC.
Other cycle modeling results are presented in Figures 6.2 through 6.3, inclusive. These results refer to ORC cycles constructed with Copeland, Hitachi, Sanyo and Bitzer expanders, respectively. They show the cycle’s energy and exergy efficiency and the isentropic efficiency of the expander. For all cases studied, degrees of superheat have a positive effect on energy efficiency, exergy efficiency and isentropic efficiency of the expander. However, in all cases excessive superheating lowers all efficiencies. For the case of Copeland and Sanyo expanders the initial working fluid as per manufacturer specifications is R404A. This refrigerant is suitable for low temperature application but it is not found suitable for ORC. Thus, the working fluid has been replaced in those cases with Siloxane-1. This is suitable for ORC applications because it has a critical temperature of 245°C.

6.1.4 Modeling of the Experimental Systems
In this section the experimental systems are modelled based on thermodynamic analysis of expansion process of the expander.

6.1.4.1 Expander-Dynamometer and Expander-Generator Systems
The expander-dynamometer system has been modeled thermodynamically based on energy balance equations. Basically the expander is represented in the form of a “black box” according to the sketch as shown in Figure 6.33. The inlet is the high pressure air flow at a set temperature, pressure and mass flow rate. The outputs are three in number: the expanded flow (carrying stream enthalpy), the generated shaft work (denoted with $W_{mechanic}$) and the work losses (denoted with $W_{losses}$).

![Figure 6.33 Thermodynamic model for the expander-dynamometer test bench.](image)
The energy balance can be written as
\[ \dot{m} \times h_1 = \dot{W}_{\text{mechanic}} + \dot{W}_{\text{losses}} + \dot{m} \times h_2 \]  
where the mass flow rate is determined through:
\[ \dot{m} = \rho(T_1, P_1) \times \dot{V} \]  
Here, \( \dot{V} \) is the volumetric flow rate in state 1.

The ideal work output which assumes isentropic expansion can be calculated by the following equation:
\[ \dot{W}_s = \dot{m} \times [h_1 - h_2(P = P_0, s = s_1)] \]  
The temperature at the expander discharge is bounded by two physical limits; the lowest which corresponds to the isentropic expansion process:
\[ T_{2s} = T(P = P_0, s = s_1) \]  
The highest that corresponds to isenthalpic process (no work generation in this case) is
\[ T_{2h} = T(P = P_0, h = h_1) \]  
The expander-dynamometer and expander-generator systems are modelled in the same way as the experimental arrangement. The only difference being that in one case the expander is coupled to a dynamometer and in the other the expander is coupled to a generator. In the first case, the generated power is calculated with the help of shaft torque and rotational speed as
\[ \dot{W}_{\text{shaft}} = \omega \times T \]  
while in the second case shaft power can be expressed as
\[ \dot{W}_{\text{shaft}} = I \times V \]

Figure 6.34 shows the predicted torque vs. angular velocity diagram with expander-dynamometer system operating with Bristol expander with air. Also, there are shaft power profiles for all cases. The inlet temperature was assumed 25°C for all cases. In the diagram, the shaft power profiles which present maxima at specific torques are also indicated. The output torque is imposed with the application of dynamometer load. Once the torque is produced the angular velocity is determined based on torque balance. The torque generated by the expander gas force must be balanced by the torque needed to accelerate air and the torque delivered “outside”. Due to the fact that fluid acceleration is required by expander operation the useful energy delivered at the shaft is always smaller than that generated by working fluid expansion. This introduces the concept of mechanical efficiency. One must consider that the output power is
smaller than the thermodynamic output which is: \( \dot{W}_{\text{out}}^{\text{th}} = \dot{m}_1 \times (h_1 - h_2) \). The actual work output is \( \dot{W}_{\text{out}}^{\text{shaft}} = \omega \times T_{\text{out}} \), while the work generated by the fluid during the isentropic expansion process is given by \( W_s = \dot{m}_1 \times (h_1 - h_{2s}) \).

Figure 6.34 Predicted torque-angular velocity diagram for the expander-dynamometer system.

Figure 6.35 Predicted isentropic and mechanical efficiencies in function of pressure ratio for the expander-dynamometer experiment.
The isentropic efficiency is independent of angular velocity. It is solely a function of thermodynamic properties. It can be written as:

\[
\eta_s = \frac{\dot{W}_{\text{th}}}{\dot{W}_s}
\]  

(6.9)

Since the actual work output is generated at the expander shaft, the mechanical efficiency of the expander is defined as:

\[
\eta_m = \frac{\dot{W}_{\text{shaft}}}{\dot{W}_{\text{th}}}
\]  

(6.10)

The predicted efficiencies for the expander-dynamometer experiment are indicated in Figure 6.35 for three rotational speeds. It can be observed that the increase in rotational speed leads to the reduction of the mechanical efficiency. Basically, the energy needed to accelerate the fluid increases with the third power of the rotation speed (or angular velocity). For example, if the rotational speed increases three times, then it would require 27 times more energy to accelerate the fluid pocket. This energy is extracted from the same flow. Figure 6.35 also shows that after a certain threshold value of pressure ratio, the mechanical efficiency remains constant.
Regarding the expander generator, the electric generator actually used in the experiment has a nominal rotational speed of 2000 RPM for which it generates 26 V. It is possible to convert in this case the torque vs. angular velocity diagram to a current vs. voltage diagram for the expander generator system. Figure 6.36 shows the predicted angular velocity vs. torque diagram for three values of pressure ratio. The developed shaft power is also indicated there. This diagram is converted to the current-voltage diagram based on nominal output information for the generator. Basically, the generated voltage is proportional with the rotation speed. One can thus determine the voltage:

\[ V = V_n \times \frac{SRS}{SRS_n} \]  

(6.11)

where \( V_n \) is the nominal voltage, \( SRS \) is the actual shaft rotational speed and \( SRS_n \) is the nominal shaft rotational speed. The results of this conversion are presented in Figure 6.37. Also in Figure 6.38 the predicted efficiency is indicated.
6.1.4.2 ORC Test Bench System

In this section, the thermodynamic modeling of the ORC test bench system is discussed. This modeling is required to facilitate thorough understanding of the processes and to help the processing of the experimental data for comparison purposes. The thermodynamic modeling involves writing the mass and energy balance equations for each component of the test bench system. The modeling is done in EES for two cases:

Case I: Expander operation in superheated region (shown in Figure 6.39).
Case II: Expander operation in the two-phase region (shown in Figure 6.40).

As shown in Figure 6.39, the fluid is superheated before entering the expander. Multiple degrees of superheating were conducted in experiments to show the effect of superheating. The expander operation in two-phase region is shown in Figure 6.40 where the fluid undergoes expansion process at or within saturated vapour curve. Models are developed to investigate the expander performance with dry and wet fluid expansion process.
Figure 6.39 Modeling results of ORC test bench operation in superheated region.

Figure 6.40 Modeling results of ORC test bench operation in two-phase region.
6.2 Experimental Systems

As mentioned earlier a series of experiments were carried out with three set-ups. The first experiment with expander-dynamometer test bench was performed to achieve a preliminary investigation of the scroll machine operation as an expander. A visual observation of the inner component movement also helped to identify the frictional surfaces, the actual mechanism of operation and any possible system modifications. The experimental knowledge on this open-loop system had allowed for further design and planning of the subsequent experiments. The expander-generator and ORC test bench experiments were performed eventually in a comprehensive, planned manner. The experimental data acquired within this work is reported in tabular form, denoting each experiment in rational order with “Experiment $i$”, were $i$ is the number of experimental session; the total number of experimental sessions is 14. The experiments are numbered and summarized in Table 6.2 as indicated below.

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<th>Experiment #</th>
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<th>Remarks</th>
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<td>ORC experiment</td>
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6.2.1 Expander-Dynamometer System

There are two sets of experiments performed with this setup. The first experiment has been conducted at lower pressure ratios. However, the outlet pressure of the expander could not be measured as the system was operated in an open loop and the discharge flow merges to the atmosphere. Also the temperature of the expanded flow was not recorded accurately in this session because of difficulties placing the thermocouple properly. However, an infra-red thermometer was used to get an approximate value of the temperature which was noted 7°C- 8°C. The flow rate of the fluid was not carefully monitored during each set of runs at constant pressure.

There are two sets of experiments performed with this setup. The first experiment has been conducted at lower pressure ratios. However, the outlet pressure of the expander could not be measured as the system was operated in an open loop and the discharge flow merges to the atmosphere. Also the temperature of the expanded flow was not recorded accurately in this session because of difficulties placing the thermocouple properly. However, an infra-red thermometer was used to get an approximate value of the temperature which was noted 7°C- 8°C. The flow rate of the fluid was not carefully monitored during each set of runs at constant pressure.

The second experiment has been conducted at higher pressure ratios. An additional run has been made to measure the outlet temperature by placing a thermocouple at the discharge port of the expander.

The raw data of both the experiments are presented without any unit conversion in Table 6.3 and Table 6.4. It can be seen that in Run 11 of the second experiment, the shaft RPM was zero because the unit stalled. The raw data was then processed to determine the angular velocity,

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Notes: Ambient temperatures18°C, (*) Flowmeter conversion chart to be used to determine flow rate.

The raw data of both the experiments are presented without any unit conversion in Table 6.3 and Table 6.4. It can be seen that in Run 11 of the second experiment, the shaft RPM was zero because the unit stalled. The raw data was then processed to determine the angular velocity,
torque, flow rate and mechanical work. Also, from these parameters further calculations were made to determine the work distribution, isentropic efficiency and leakage factor. It is seen that a major percentage of work converted goes to the work loss factor. This is justified due to the fact that the expander of this setup was made from a hermetically sealed refrigeration compressor with proper lubrication system. But during conversion, the normal lubrication system was damaged and an alternate means was adopted for lubrication. Oil was manually splashed on the bearing before each run. This was determined to be inadequate as centrifugal forces drive away the oil from bearing surfaces.

Table 6.4 Experiment-2: Expander-Dynamometer system.

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Notes: Ambient temperature 19°C; (*) Flowmeter conversion chart to be used to determine flow rate.

The processed parameters of expander-dynamometer experiments are shown in Figure 6.41 to Figure 6.43. In Figure 6.41, the shaft rotational speed and torque relationship are presented. At zero torque, the shaft rotational speed is highest and the expansion work is used for overcoming the frictional forces. With the application of dynamometer load, the torque increases and the shaft work is used for production of useful work and balancing the frictional forces. With the increase of torque, the shaft rotational speed decreases at a higher rate. This ultimately stalls the expander just above maximum torque generation. This phenomenon was observed during experiment-2 at run 11.
Figure 6.41 Torque vs. Shaft rotational speed diagram of the Expander-Dynamometer operation.

Figure 6.42 Angular velocity vs. Power diagram of the Expander-Dynamometer operation.
Table 6.5 Processed data of the expander-dynamometer experiments.

<table>
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<th>Experiment #</th>
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<th>( R_x ) [N]</th>
<th>( \omega ) [rad/s]</th>
<th>Flow rate, ( \dot{m} ) [ml/s]</th>
<th>Torque, ( T ) [Nm×10^{-3}]</th>
<th>( T_{in} ) [°C]</th>
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Note: experiment 2, run 11 – shaft stalled – not included

Continued
Table 6.5 Processed data of the expander-dynamometer experiments (continued).

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Figure 6.43 Inlet pressure vs. flow rate diagram of the expander-dynamometer system.
Figure 6.44 Inlet pressure vs. work loss diagram of the expander-dynamometer system.

Figure 6.45 Influence of inlet pressure on leakage factor and isentropic efficiency of the expander-dynamometer system.

In Figure 6.42, one can note that with the increase of angular velocity shaft power increases up to a certain value, and after that it decreases. Power generated by the expansion
process is converted to useful work and a portion is used to accelerate the fluid as mentioned in the analysis chapter. After a certain angular velocity, more energy is used to accelerate the fluid in comparison to producing useful work. This phenomenon is explained in section 6.2.4.1. Also it validates the torque output trend of Bristol expander as shown in Figure 6.25. In Figure 6.42 loss of useful work due to this process is shown graphically. Figure 6.43 illustrates the influence of inlet pressure over flow rate of the expander. The flow rate increases with the increase of inlet pressure, this can best be defined as leakage flow rate increases at a higher rate with the increase of inlet pressure. This can also be noted in Figure 6.45.

6.2.2 Expander-Generator System
In the Expander-generator experiments, the power generated by the built-in generator is measured in the form of current and voltage. The input gas pressure varied in each run and the corresponding output parameters are recorded. Table 6.6 through Table 6.9 represent the raw data whereas the processed data are presented in Table 6.10. Graphical representations of the processed data are shown in Figure 6.46 through Figure 6.48. Figure 6.46 illustrates the influence of inlet pressure over power which shows that power increases with the increase of inlet pressure. Initially the slope of the curves is shallow and gradually it becomes steeper. This is due to the fact that total power generated by the expander is used to overcome the frictional resistance and the rest converted to useful work. Frictional resistance being constant, at higher total power the major portion goes for useful work. Figure 6.47 depicts the same concept of optimal expansion as discussed earlier. Similar trend as that of expander-dynamometer experiment for flow rate variation with inlet pressure is observed in Figure 6.48.
Table 6.6 Experiment-3: Expander-generator system.

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Notes: Ambient temperature 20°C; two resistive loads connected in series.
* (*) Flowmeter conversion chart to be used to determine the flow rate.

Table 6.7 Experiment-4: Expander-generator system.

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Notes: Ambient temperature 23°C; two resistive loads connected in parallel.
* (*) Flowmeter conversion chart to be used to determine the flow rate
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Notes: Ambient temperature: 20°C; Open circuit generator (without load). (*) Temperature reading inconsistent.

Table 6.9 Experiment-6: Expander-generator system.

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Notes: Ambient temperature: 20°C; Expander discharge throttled; (*) Temperature reading inconsistent.
Table 6.10 Processed data of the expander-generator experiments.

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Table 6.10 Processed data of the expander-generator experiments (continued).

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Figure 6.46 Pressure vs. power diagram of the expander-generator system.

Figure 6.47 Pressure ratio vs. isentropic efficiency diagram of the expander-generator system.
6.2.3 ORC Test Bench System

The ORC experiments were performed in a versatile way to obtain as much data as possible for better and accurate evaluation of the system. Initially, air was used as the working fluid to perform a preliminary test of the experimental set-up. Eventually, experiments were performed with R134a at different operating conditions. Twelve sets of experiments were conducted with ORC test bench. Graphical representation of pressure ratio-power and pressure difference-power of experiment-6 are made in Figure 6.49 and Figure 6.50 respectively. Figure 6.49 and 6.50 clearly defines the optimal expansion point as represented graphically in Figure 6.23, the result of the model analysis of Bristol compressor. Pressure difference vs. power graph of Figure 6.51 also signifies the optimal expansion phenomenon of the expander.
Figure 6.49 Pressure ratio vs. power diagram of ORC system.

Figure 6.50 Influence of pressure ratio on ORC isentropic efficiency.
Figure 6.51 Pressure difference vs. power diagram of ORC system (experiment-6).

Table 6.11 Experiment-7: ORC system with air.

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Notes: Ambient temperature 20°C; Heat added to the system; Compressor discharge at 100°C.

The experimental data have been processed as indicated in Table 6.12. Runs 2 and 4 were ignored as they are repeated runs. The temperature for isentropic discharge is denoted in the table with $T_{2s}$. The isentropic efficiency has been determined to around 45% while the mechanical efficiency was 27-32%. For higher efficiency it is important to operate as close as possible to the nominal shaft rotation speed (SRS). The next experiment is presented in Table 6.13. In this case, the volume flow rate has been measured as well. The experiment-9 is similar to experiment-8 with a difference that the air has been heated up to 62°C.
Table 6.12 Processed data for experiment-7.

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<th>$W_s$ [W]</th>
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Table 6.13 Experiment-8: ORC system with air and without heat addition.

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Note: $P_{ch}$ is the discharge pressure of the compressor.

Experiment-9 operates with high over-expansion as the pressure ratio has been below 3. During this experiment a “calibration” of the pressure-drop along the working fluid circuit has been done. The calculated pressure drop is illustrated on the diagram in Figure 6.51.

![Diagram](image)

Figure 6.52 Pressure drop measurements during calibration tests.
Table 6.14 Experiment-9: ORC system with air and heat addition.

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The next two experiments, 10 and 11 were done with the purpose to calibrate the throttling valves TV1 and TV2. These experiments are explained schematically with the help of the diagrams in Figures 6.52 and 6.53 and the data is presented in Tables 6.15 and 6.16.

Table 6.15 Experiment-10: ORC system pressure drop calibration using TV1.

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<th>$P_{l}$, [kPa]</th>
<th>$P_{\text{mix}}$, [kPa]</th>
<th>$T_{\text{mix}}$, [°C]</th>
<th>TV1 Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>694</td>
<td>673</td>
<td>142</td>
<td>142</td>
<td>-24.6</td>
<td>11’O clock</td>
</tr>
<tr>
<td>2</td>
<td>694</td>
<td>652</td>
<td>135</td>
<td>136</td>
<td>-26.4</td>
<td>9’O clock</td>
</tr>
<tr>
<td>3</td>
<td>673</td>
<td>583</td>
<td>0</td>
<td>0</td>
<td>-23.5</td>
<td>6’O clock</td>
</tr>
<tr>
<td>4</td>
<td>673</td>
<td>521</td>
<td>0</td>
<td>0</td>
<td>-22.1</td>
<td>3’O clock</td>
</tr>
<tr>
<td>5</td>
<td>673</td>
<td>501</td>
<td>0</td>
<td>0</td>
<td>-21.5</td>
<td>12’O clock</td>
</tr>
<tr>
<td>6</td>
<td>673</td>
<td>480</td>
<td>108</td>
<td>108</td>
<td>-21</td>
<td>Fully open</td>
</tr>
</tbody>
</table>

Figure 6.53 Diagram explaining the experiment-10 (Table 6.15).
Table 6.16 Experiment-11: ORC system pressure drop calibration using TV₂

<table>
<thead>
<tr>
<th>Run</th>
<th>$P_c$ [kPa]</th>
<th>$P_h$ [kPa]</th>
<th>$P_l$ [kPa]</th>
<th>$P_s$ [kPa]</th>
<th>Flow reading</th>
<th>$T_{bi}$ [°C]</th>
<th>$T_{bo}$ [°C]</th>
<th>$T_{mix}$ [°C]</th>
<th>Valve O'Clock</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>659</td>
<td>660</td>
<td>68</td>
<td>68</td>
<td>0.3</td>
<td>21.5</td>
<td>21.6</td>
<td>-22.5</td>
<td>11</td>
</tr>
<tr>
<td>2</td>
<td>639</td>
<td>625</td>
<td>68</td>
<td>68</td>
<td>0.28</td>
<td>22.9</td>
<td>20.4</td>
<td>-28.4</td>
<td>10</td>
</tr>
<tr>
<td>3</td>
<td>701</td>
<td>618</td>
<td>0</td>
<td>0</td>
<td>0.4</td>
<td>22.8</td>
<td>22.2</td>
<td>-31.1</td>
<td>9</td>
</tr>
<tr>
<td>4</td>
<td>667</td>
<td>549</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
<td>18.9</td>
<td>18.6</td>
<td>-28.0</td>
<td>6</td>
</tr>
<tr>
<td>5</td>
<td>667</td>
<td>412</td>
<td>0</td>
<td>0</td>
<td>2.0</td>
<td>10.8</td>
<td>9.5</td>
<td>-25.9</td>
<td>3</td>
</tr>
<tr>
<td>6</td>
<td>667</td>
<td>356</td>
<td>0</td>
<td>0</td>
<td>2.4</td>
<td>8.0</td>
<td>6.2</td>
<td>-24.8</td>
<td>12</td>
</tr>
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<td>7</td>
<td>680</td>
<td>315</td>
<td>3</td>
<td>3</td>
<td>2.6</td>
<td>4.5</td>
<td>2.5</td>
<td>-24.5</td>
<td>Full</td>
</tr>
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Figure 6.54 Diagram explaining the experiment-11 (Table 6.16).

Table 6.17 Experiment-12: ORC system with R134a.

<table>
<thead>
<tr>
<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{eh}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [°C]</th>
<th>$T_{bi}$ [°C]</th>
<th>$T_{bo}$ [°C]</th>
<th>$T_{em}$ [°C]</th>
<th>$T_{eo}$ [°C]</th>
<th>$T_{mix}$ [°C]</th>
<th>Voltage [V]</th>
<th>Current [A]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>701</td>
<td>129</td>
<td>274</td>
<td>150</td>
<td>2.578</td>
<td>122</td>
<td>-0.4</td>
<td>34.7</td>
<td>3.7</td>
<td>1.6</td>
<td>-17</td>
<td>4</td>
<td>2.5</td>
</tr>
<tr>
<td>2</td>
<td>701</td>
<td>122</td>
<td>274</td>
<td>150</td>
<td>2.205</td>
<td>100</td>
<td>-0.3</td>
<td>28.6</td>
<td>6.5</td>
<td>6</td>
<td>-16.5</td>
<td>4.5</td>
<td>3</td>
</tr>
<tr>
<td>3</td>
<td>721</td>
<td>115</td>
<td>281</td>
<td>150</td>
<td>2.411</td>
<td>126</td>
<td>0.3</td>
<td>47.7</td>
<td>8.6</td>
<td>7.2</td>
<td>-16.5</td>
<td>3.6</td>
<td>3.9</td>
</tr>
</tbody>
</table>
### Table 6.18 Experiment-13: ORC system with R134a.

<table>
<thead>
<tr>
<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{eh}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [$^\circ$C]</th>
<th>$T_{hi}$ [$^\circ$C]</th>
<th>$T_{bo}$ [$^\circ$C]</th>
<th>$T_{em}$ [$^\circ$C]</th>
<th>$T_{eo}$ [$^\circ$C]</th>
<th>$T_{mix}$ [$^\circ$C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>766.3</td>
<td>149.6</td>
<td>356.4</td>
<td>163.4</td>
<td>1.2</td>
<td>160</td>
<td>6.7</td>
<td>105</td>
<td>46.2</td>
<td>36.4</td>
<td>-10</td>
</tr>
<tr>
<td>2</td>
<td>766.3</td>
<td>149.6</td>
<td>342.7</td>
<td>170.5</td>
<td>1.5</td>
<td>220</td>
<td>6.4</td>
<td>109</td>
<td>47</td>
<td>38</td>
<td>-11</td>
</tr>
<tr>
<td>3</td>
<td>766.3</td>
<td>149.6</td>
<td>342.7</td>
<td>184</td>
<td>1.4</td>
<td>140</td>
<td>3.7</td>
<td>88.1</td>
<td>58.7</td>
<td>47.4</td>
<td>-9</td>
</tr>
<tr>
<td>4</td>
<td>766.3</td>
<td>149.6</td>
<td>342.7</td>
<td>204.8</td>
<td>1.5</td>
<td>104</td>
<td>0.4</td>
<td>38.3</td>
<td>60.4</td>
<td>52.1</td>
<td>-9</td>
</tr>
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(continued)

<table>
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<tr>
<th>Run</th>
<th>$V_{dc}$ [V]</th>
<th>$I_{dc}$ [A]</th>
<th>$P$ [W]</th>
<th>$V_{12}$ [V]</th>
<th>$V_{23}$ [V]</th>
<th>$V_{31}$ [V]</th>
<th>$I_{12}$ [A]</th>
<th>$I_{23}$ [A]</th>
<th>$I_{31}$ [A]</th>
<th>$I_{mix}$ [A]</th>
<th>Lamp/HTR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5.9</td>
<td>6.7</td>
<td>26.1</td>
<td>4.1</td>
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<td>4.1</td>
<td>5.5</td>
<td>5.5</td>
<td>5.3</td>
<td>4/All</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>7.1</td>
<td>3.7</td>
<td>21</td>
<td>5.1</td>
<td>5.1</td>
<td>5.1</td>
<td>3</td>
<td>3</td>
<td>3.9</td>
<td>2/All</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>8.4</td>
<td>2.1</td>
<td>17.6</td>
<td>7.2</td>
<td>7.2</td>
<td>7.2</td>
<td>1.8</td>
<td>1.8</td>
<td>1.7</td>
<td>1/Nil</td>
<td></td>
</tr>
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<td>4</td>
<td>9.2</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
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</table>

### Table 6.19 Experiment-14: ORC system with R134a.

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<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{eh}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [$^\circ$C]</th>
<th>$T_{hi}$ [$^\circ$C]</th>
<th>$T_{bo}$ [$^\circ$C]</th>
<th>$T_{em}$ [$^\circ$C]</th>
<th>$T_{eo}$ [$^\circ$C]</th>
<th>$T_{mix}$ [$^\circ$C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>742.6</td>
<td>149.6</td>
<td>294</td>
<td>163</td>
<td>1.5</td>
<td>155</td>
<td>2.5</td>
<td>37.2</td>
<td>18</td>
<td>21.8</td>
<td>-11</td>
</tr>
<tr>
<td>2</td>
<td>715.0</td>
<td>149.6</td>
<td>363</td>
<td>163</td>
<td>1.5</td>
<td>212</td>
<td>4.2</td>
<td>73.7</td>
<td>20.1</td>
<td>21.2</td>
<td>-12</td>
</tr>
<tr>
<td>3</td>
<td>715.0</td>
<td>149.6</td>
<td>371</td>
<td>163</td>
<td>1.2</td>
<td>194</td>
<td>8</td>
<td>119</td>
<td>32</td>
<td>25.8</td>
<td>-12</td>
</tr>
<tr>
<td>4</td>
<td>756.4</td>
<td>149.6</td>
<td>371</td>
<td>163</td>
<td>1.1</td>
<td>181</td>
<td>7.7</td>
<td>100.7</td>
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<td>31.8</td>
<td>-12</td>
</tr>
</tbody>
</table>

(continued)

<table>
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<tr>
<th>Run</th>
<th>$V_{dc}$ [V]</th>
<th>$I_{dc}$ [A]</th>
<th>$P$ [W]</th>
<th>$V_{12}$ [V]</th>
<th>$V_{23}$ [V]</th>
<th>$V_{31}$ [V]</th>
<th>$I_{12}$ [A]</th>
<th>$I_{23}$ [A]</th>
<th>$I_{31}$ [A]</th>
<th>Lamp/HTR</th>
</tr>
</thead>
<tbody>
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<td>1.9</td>
<td>11.2</td>
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<td>5.2</td>
<td>5.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>1/3</td>
</tr>
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<td>4.3</td>
<td>3.4</td>
<td>14.6</td>
<td>4.2</td>
<td>4.3</td>
<td>4.3</td>
<td>0.15</td>
<td>0.15</td>
<td>0.15</td>
<td>2/All</td>
</tr>
<tr>
<td>3</td>
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<td>8.8</td>
<td>17.4</td>
<td>3.2</td>
<td>3.2</td>
<td>3.2</td>
<td>0.4</td>
<td>0.4</td>
<td>0.4</td>
<td>5/4</td>
</tr>
<tr>
<td>4</td>
<td>2.3</td>
<td>8.2</td>
<td>18.8</td>
<td>2.8</td>
<td>2.8</td>
<td>2.8</td>
<td>6.7</td>
<td>6.7</td>
<td>6.7</td>
<td>4/4</td>
</tr>
</tbody>
</table>
Table 6.20 Experiment-15: ORC system with variable fan speed with R134a at higher pressure ratio.

<table>
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<tr>
<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{eh}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [°C]</th>
<th>$T_{hi}$ [°C]</th>
<th>$T_{bo}$ [°C]</th>
<th>$T_{em}$ [°C]</th>
<th>$T_{eo}$ [°C]</th>
<th>$T_{mix}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1031</td>
<td>0</td>
<td>133</td>
<td>0</td>
<td>0.2</td>
<td>114</td>
<td>38</td>
<td>38</td>
<td>14.3</td>
<td>8.9</td>
<td>-25</td>
</tr>
<tr>
<td>2</td>
<td>1134</td>
<td>117</td>
<td>535</td>
<td>212</td>
<td>0.4</td>
<td>139</td>
<td>23</td>
<td>19/5</td>
<td>3.1</td>
<td>1.4</td>
<td>-9.3</td>
</tr>
<tr>
<td>3</td>
<td>1720</td>
<td>2/3</td>
<td>1/6</td>
<td>134</td>
<td>1.02</td>
<td>191</td>
<td>35.6</td>
<td>31.8</td>
<td>13.6</td>
<td>12.6</td>
<td>0.1</td>
</tr>
<tr>
<td>4</td>
<td>1996</td>
<td>2/3</td>
<td>1/6</td>
<td>535</td>
<td>1.3</td>
<td>149</td>
<td>34.9</td>
<td>30.9</td>
<td>10.2</td>
<td>10.1</td>
<td>-0.2</td>
</tr>
</tbody>
</table>

(continued)

<table>
<thead>
<tr>
<th>Run</th>
<th>By-pass</th>
<th>$I_{dc}$ [A]</th>
<th>$V_{dc}$ [V]</th>
<th>$P$ [W]</th>
<th>Condenser Fan Dimmer Position</th>
<th>Lamp</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>9 O Clock</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>Full</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>Closed</td>
<td>8.1</td>
<td>2.3</td>
<td>18.6</td>
<td>Full</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>Closed</td>
<td>10.5</td>
<td>3.8</td>
<td>39.5</td>
<td>12 O Clock</td>
<td>4</td>
</tr>
<tr>
<td>4</td>
<td>Closed</td>
<td>12.6</td>
<td>5.5</td>
<td>66.8</td>
<td>3 O Clock</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 6.21 Experiment-16: ORC system with R134a.

<table>
<thead>
<tr>
<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [°C]</th>
<th>$T_{hi}$ [°C]</th>
<th>$T_{bo}$ [°C]</th>
<th>$T_{em}$ [°C]</th>
<th>$P_{mix}$ [°C]</th>
<th>$I_{dc}$ [A]</th>
<th>$V_{dc}$ [V]</th>
<th>Lamp/HTR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2168</td>
<td>790</td>
<td>395</td>
<td>2.4</td>
<td>140</td>
<td>41.5</td>
<td>39.3</td>
<td>18.4</td>
<td>17.7</td>
<td>7</td>
<td>15</td>
<td>4/5.5</td>
</tr>
<tr>
<td>2</td>
<td>2168</td>
<td>859</td>
<td>395</td>
<td>N/A</td>
<td>155</td>
<td>41.6</td>
<td>37.7</td>
<td>18.5</td>
<td>18.5</td>
<td>8</td>
<td>14.3</td>
<td>5.8</td>
</tr>
<tr>
<td>3</td>
<td>2168</td>
<td>790</td>
<td>411</td>
<td>5.2</td>
<td>138</td>
<td>42.8</td>
<td>37.7</td>
<td>21</td>
<td>18</td>
<td>10.3</td>
<td>10</td>
<td>17</td>
</tr>
</tbody>
</table>

Note: In run two by-pass valve TV1 positioned to 12 O clock.

Table 6.22 Experiment-17 ORC system with variable fan speed (R134a).

<table>
<thead>
<tr>
<th>Run</th>
<th>$P_{ch}$ [kPa]</th>
<th>$P_{cl}$ [kPa]</th>
<th>$P_{eh}$ [kPa]</th>
<th>$P_{el}$ [kPa]</th>
<th>Flow meter [V]</th>
<th>$T_h$ [°C]</th>
<th>$T_{hi}$ [°C]</th>
<th>$T_{bo}$ [°C]</th>
<th>$T_{em}$ [°C]</th>
<th>$T_{eo}$ [°C]</th>
<th>$T_{mix}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1031</td>
<td>0</td>
<td>133</td>
<td>0</td>
<td>0.2</td>
<td>114</td>
<td>38</td>
<td>38</td>
<td>14.3</td>
<td>8.9</td>
<td>-25</td>
</tr>
<tr>
<td>2</td>
<td>1134</td>
<td>117</td>
<td>535</td>
<td>212</td>
<td>0.4</td>
<td>139</td>
<td>23</td>
<td>19/5</td>
<td>3.1</td>
<td>1.4</td>
<td>-9.3</td>
</tr>
<tr>
<td>3</td>
<td>1720</td>
<td>2/3</td>
<td>1/6</td>
<td>134</td>
<td>1.02</td>
<td>191</td>
<td>35.6</td>
<td>31.8</td>
<td>13.6</td>
<td>12.6</td>
<td>0.1</td>
</tr>
<tr>
<td>4</td>
<td>1996</td>
<td>2/3</td>
<td>1/6</td>
<td>535</td>
<td>1.3</td>
<td>149</td>
<td>34.9</td>
<td>30.9</td>
<td>10.2</td>
<td>10.1</td>
<td>-0.2</td>
</tr>
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</table>

(Continued)
Table 6.22 (continued)

<table>
<thead>
<tr>
<th>Run</th>
<th>Voltage [V]</th>
<th>Current [A]</th>
<th>Condenser Fan Dimmer Position</th>
<th>Lamp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intl.</td>
<td>Full</td>
<td>Full</td>
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</tr>
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<td>1</td>
<td>2.3</td>
<td>8.1</td>
<td>Full</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>3.8</td>
<td>10.5</td>
<td>12/0 clock</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>5.3</td>
<td>12.6</td>
<td>3/0 clock</td>
<td>4</td>
</tr>
</tbody>
</table>

Figure 6.54 represents the influence of degrees of superheating over isentropic efficiency. It is clear that superheating has positive impact on expander isentropic efficiency up to a certain value. After that, the isentropic efficiency decreases sharply. This phenomenon is discussed in section and graphically represented in Figure 6.32 with the result of the model analysis of Bitzer expander in ORC. The experimental result thus validates the concept that after a certain level of superheating it involves wastage of thermal energy.

In Figures 6.55 and 6.56, the experimental results of lower pressure ratio and higher pressure ratio are graphically represented. The isentropic efficiency of the expander varied from 40% to about 70%. The ORC cycle efficiency while operating in a regenerative configuration has an isentropic efficiency of approximately 5% while the exergy efficiency is approximately 30%.

Figure 6.55 Degree of superheating vs. isentropic efficiency diagram.
Figure 6.56 Expansion process at lower pressure ratio.

Figure 6.57 Expansion process at higher pressure ratio.
6.3 Uncertainty Analysis

In every experiment there exists uncertainty due to measurement and data processing error. An uncertainty analysis is required to determine the uncertainty of the measured parameters and the percentage of relative uncertainty of the parameters derived from measured data. In this experimental process temperature, pressure, voltage, current and flow rate are the measured parameters. The accuracy of the measured parameters is determined based on the catalogue data of the measurement equipment. The measurement accuracy data is further used in uncertainty analysis. The value of the measured parameters and the estimated value of the uncertainties are as follows:

\[ I_{\text{max}} = 10 \, \text{A}, \quad V_{\text{max}} = 15 \, \text{V}, \quad \varepsilon_{I} = 0.05 \, \text{A}, \quad \varepsilon_{V} = 0.1 \, \text{V}, \quad \varepsilon_{\rho} = 1 \, \text{kg/m}^3, \quad \varepsilon_{\nu} = 18 \, \text{ml/min}, \quad \varepsilon_{T} = 1 \, \text{K} \]
\[ \varepsilon_{p} = 10 \, \text{kPa}, \quad C_{p} = 1.5 \, \text{kJ/kgK}, \quad \frac{\partial h}{\partial P} = 0.01 \, \text{kJ/kg kPa}, \quad T = 150 \, ^\circ\text{C}, \quad P = P_{\text{sat}}(50 \, ^\circ\text{C}) \]

Table 6.23 Results of the uncertainty analysis.

<table>
<thead>
<tr>
<th>Quality</th>
<th>Uncertainty calculation</th>
<th>Relative uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric power</td>
<td>[ \dot{W} = V \times I ] [ d\dot{W} = I , dV + V , dI ] [ \varepsilon_{W} = I_{\text{max}} \times \varepsilon_{V} + V_{\text{max}} \times \varepsilon_{I} ]</td>
<td>[ \frac{\varepsilon_{W}}{W} = 1.2 % ]</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>[ \dot{m} = \dot{V} \times \rho ] [ \rho = \rho_{\text{liq}}(T_{\text{sat}}) ] [ d\dot{m} = \dot{V} , d\rho + \rho , d\dot{V} ] [ \dot{V} = 100 + 180 , \text{V , [ml/min]} ] [ \varepsilon_{\dot{m}} = \dot{V} \varepsilon_{\rho} + \rho_{\text{max}} \times \varepsilon_{\dot{V}} ] [ \varepsilon_{\dot{V}} = 180 \varepsilon_{\nu} ]</td>
<td>[ \frac{\varepsilon_{\dot{m}}}{\dot{m}} = 4.1 % ]</td>
</tr>
<tr>
<td>Specific enthalpy of working fluid</td>
<td>[ h = h(T, \ P) ] [ dh = C_{p} , dT + \left( \frac{\partial h}{\partial P} \right)<em>{T} \times dP ] [ \varepsilon</em>{h} = C_{p} , \varepsilon_{T} + \left( \frac{\partial h}{\partial P} \right)<em>{T,\text{max}} \times \varepsilon</em>{p} ]</td>
<td>[ \frac{\varepsilon_{h}}{h} = 2 % ]</td>
</tr>
</tbody>
</table>
In Table 6.23 the result of the uncertainty analysis for the most important quantities determined within (electric power, mass flow rate and specific enthalpy of the working fluid) this work are shown. The highest relative uncertainty is at the determination of the mass flow rate with around 4%, while the lowest for electric power. All uncertainties are within 95% of confidence level.
CHAPTER 7: CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

7.1 Conclusions

The experimental investigation of the scroll expanders has been performed in three test bench set-ups that facilitated the acquirement of a set of well-defined experimental data for performance analysis. Starting with a preliminary experiment in an expander-dynamometer test bench the thesis work was concluded by a detailed experimental process in a modified Organic Rankine cycle test bench. A total number of seventeen experiments were conducted, eleven were with air and the rest were with R134a. The expander-dynamometer experiments represented validated the modelled Torque-RPM relationship.

The organic Rankine cycle test bench was designed in a unique configuration that helped to imply a wide range of operating parameters to evaluate the performance of the scroll expander more precisely. The parameters that influence the expander operation were inlet temperature, inlet pressure, mass flow rate, degree of superheat, and electrical load to the generator. The experimental result shows a maximum isentropic efficiency of 66% and an overall energy efficiency of maximum 5% at 120°C source temperature, while the exergy efficiency is about 30%. Initially, difficulties arose in controlling the boiler temperature with the on-off heater switches, however after attaining experience subsequent experiments were much smoother. Another problem encountered in maintaining the high vapour side pressure upstream of the expander because of pressure loses in the systems. This problem was solved by installing a dimmer in the fan circuit of the condenser to maintain a fan speed of lower side.

Despite few difficulties, the overall performance of the expander in the Rankine cycle was in good agreement with the modelled and the predicted value.
7.2 Recommendations for Future Work

A few recommendations are brought to the attention of future researchers for further analysis:

- Modification of the scroll expander to drive an electric generator outside the scroll unit. This feature would facilitate measuring the RPM accurately and replacing the electric generator of varying capacity for diverse experimental investigation.
- Improvement of the experimental set up by replacing the electric boiler with a fin-tube hot air heat exchanger.
- Addition of auto temperature control and working fluid auto flow control could increase the performance of the experimental set up.
- Installation of electric power analyzer to determine the instantaneous load characteristics could precisely identify the influence of operating parameter changes in a high level of accuracy.
References


Quoilin S. 2007. Experimental study and modeling of a low temperature Rankine cycle for small scale cogeneration. BSc Thesis, Liege University, Belgium.


Appendix: EES Code for Expander Modeling

//Bristol Coeff --> Functions
Function dH(i)
dH=lookup('C',i,'Q')
end
Function P(i)
P=lookup('C',i,'W')
end
Function A(i)
A=lookup('C',i,'I')
end
Function M(i)
M=lookup('C',i,'M')
end
heat=dH(1)+dH(2)*S+dH(3)*D+dH(4)*S^2+dH(5)*S*D+dH(6)*D^2+dH(7)*S^3+dH(8)*D*S^2+dH(9)*S*D^2+dH(10)*D^3
Q=0.2931*heat"W"
current=A(1)+A(2)*S+A(3)*D+A(4)*S^2+A(5)*S*A(6)*D^2+A(7)*S^3+A(8)*D*S^2+A(9)*S*D^2+A(10)*D^3
power=P(1)+P(2)*S+P(3)*D+P(4)*S^2+P(5)*S*D+P(6)*D^2+P(7)*S^3+P(8)*D*S^2+P(9)*S*D^2+P(10)*D^3
mFlow=M(1)+M(2)*S+M(3)*D+M(4)*S^2+M(5)*S*D+M(6)*D^2+M(7)*S^3+M(8)*D*S^2+M(9)*S*D^2+M(10)*D^3
m/0.000125997881=mFlow "kg/s"
COP=Q/power

//Scroll compressor Bristol --> Input data
R$='R22'
S=45
D=130
R=S+20 "return gas"
L=D-15
Te=(S-32)*5/9
Tc=(D-32)*5/9
Tsh=(R-32)*5/9
Tsc=(L-32)*5/9

//State1
T[1]=Tsh
s[1]=Entropy(R$,T=T[1],P=P[1])
h[1]=Enthalpy(R$,T=T[1],P=P[1])
x[1]=2
v[1]=Volume(R$,T=T[1],P=P[1])

//State2
w=h[2]-h[1]
power=w*m*1000
h2s=Enthalpy(R$,P=P[2],x=s[1])
h2s-h[1]=etaS*(h[2]-h[1])
//etaS=0.7
s[2]=Entropy(R$,P=P[2],h=h[2])

//State3
T[3]=Tc
P[3]=Pressure(R$,T=T[3],x=x[3])
h[3]=Enthalpy(R$,T=T[3],x=x[3])
s[3]=Entropy(R$,T=T[3],x=x[3])
x[3]=1

//State4
h[4]=Enthalpy(R$,T=T[4],x=x[4])
s[4]=Entropy(R$,T=T[3],x=x[4])
x[4]=0

//State5
T[5]=Tsc
h[5]=Enthalpy(R$,T=T[5],P=P[5])
s[5]=Entropy(R$,T=T[5],P=P[5])

//State6
T[6]=Te
s[6]=Entropy(R$,T=T[6],h=h[6])

//State7
T[7]=Te
P[7]=Pressure(R$,T=T[7],x=x[7])
h[7]=Enthalpy(R$,T=T[7],x=x[7])
x[7]=1
s[7]=Entropy(R$,T=T[7],x=x[7])

//State8
P[8]=P[1]
s[8]=s[1]

//Plot the cycle
duplicate i=1,7
ss[i]=s[i]
TT[i]=T[i]
end
ss[8]=s[1]

//Compressor parameters
//1. Pressure ratio
pr=P[2]/P[1]

//2. Volume ratio
VR=v[1]/v[2]

//3. Volumetric efficiency
Displacement=13.627/3600 "m3/s"
Vd=Displacement/RPS*1000000
etaV=m*v[1]/Displacement
LeakRate=(1-
etaV)*Displacement*3600"m3/s"

//4. Isentropic work
Wss=mCmp*(h2sv-h1a)
Wsv=Wss*1000

//5. Dissipated work
Wlosss=(m+mLeak)*(h2a-h2sv)
Wdiss=Wlosss*1000

//6. Compressor driving work
Win=Wsv+Wdiss

//7. Isochoric pressure building work ratio
PIE=Wlosss/Wss

//8. Built-in volume ratio
BVR=v1a/v2sv

mLeak=etaV*Displacement*3600"m3/s"
eta_SV=(enthalpy(R$,s=s[1],v=v[2]-h[1])/h[2]-h[1])

//10. rated shaft torque
RPM=3500
RPS=RPM/60
omega=2*Pi#*RPS
power=omega*Trq

//11. Torque for fluid acceleration
TrqAcc=Trq/3

//12. Torque for pressure build --> useful work
TrqP=2/3*Trq

//12. Angular velocity coefficient
Kw=TrqAcc/mRated/omega^2

//13. Torque-pressure coefficient
Kp=TrqP/dP

//14. Rated displaced mass
mRated=mCmp/RPS

//Model of the compressor
h1=h[1]

h2a=h[2]

//State 1a
mCmp=m+mLeak
mCmpDispl=mCmp*1000
m*h1+mLeak*h2a=mCmp*h1a
mDispl=1000*m
mLdispl=1000*mLeak
P1=P[1]
P1a=P1
s1a=entropy(R22,P=P1,h=h1a)
v1a=volume(R22,P=P1,h=h1a)
T1a=temperature(R22,P=P1,h=h1a)

//State 1b
h1b=h2a
P1b=P1
T1b=temperature(R22,P=P1b,h=h1b)
s1b=entropy(R22,P=P1b,h=h1b)

//State 2sv
h2sv=enthalpy(R22,v=v2sv,s=s1a)
P2sv=pressure(R22,v=v2sv,s=s1a)
s2sv=s1a
T2sv=Temperature(R22,v=v2sv,s=s1a)
v2sv=v2a
v2a=v[2]

//State 2a
h1b=h2a
P2a=P[2]
T2a=T[2]
s2a=s[2]

//Display eta
etaSperc=100*etaS
etaVperc=100*etaV

//Torque-RPM diagram
Trq=kw*mRated*oMax^2
o=300
Tacc=Kw*mRated*o^2
Tpf=Trq-Tacc
Tpf=Kp*dPo*10^5
Wpf=o*Tpf
Wacc=o*Tacc
Wmot=o*Trq
//Draw compressor operation in T-s
//line 2b--->1b
sCmp[1]=s2b
Tcmp[1]=T2b
sCmp[2]=s1b
Tcmp[2]=T1b
//low pressure line
dTlow=(T1b+20-Tcmp[3])/20
duplicate i=4,24
Tcmp[i]=Tcmp[3]+(i-3)*dTlow
sCmp[i]=entropy(R22,T=Tcmp[i],P=P[7])
end
//high pressure line
sCmp[25]=s[3]
dThigh=(T[2]+20-T[3])/20
duplicate i=26,41
Tcmp[i]=Tcmp[25]+(i-25)*dThigh
sCmp[i]=entropy(R22,T=Tcmp[i],P=P[2])
end
//line 1a-2sv-2a
sCmp[42]=s1a
sCmp[43]=s2sv
sCmp[44]=s[2]
Tcmp[42]=T1a
Tcmp[43]=T2sv