CHESTER

Compressed Heat Energy Storage for Energy from Renewable sources

Detailed design of the ORC laboratory prototype

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Glossary, abbreviations and acronyms

ORC	Organic Rankine cycle
TES	Thermal Energy Storage
PTES	Pumped Thermal Energy Storage
CHEST	Compressed Heat Energy Storage
VVT	Variable Valve Timing
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
CHESTER	Compressed Heat Energy Storage For Energy From Renewable Sources
P&ID	Process and Instrumentation Diagram
HFC	Hydro-Fluoro Carbons
HFO	Hydro-Fluoro Olefins
LH	Latent Heat
SH	Sensible Heat
PLC	Programmable Logic Controller

1. Introduction

1.1. Executive Summary

The aim of this document is to provide the reader with the information on the development of an organic Rankine cycle (ORC) test rig to be installed and integrated within a power-to-heat-to-power concept. The ORC system is responsible for the conversion of heat input it receives from the thermal energy storage (TES) system to electrical power. The objective of the study is to provide insight into the design specifics of the ORC prototype in this concept.

Pumped Thermal Energy Storage (PTES), alternatively known as a Carnot battery uses a power-to-heat system to store electrical energy in a thermal energy reservoir (which could be a latent heat storage, sensible heat storage or both), and then use this heat at high temperature to drive a heat engine to produce electricity and optionally, heating at lower temperatures. The Compressed Heat Energy Storage (CHEST) concept utilizes a heat pump combined with a Rankine cycle with latent heat thermal energy storage (TES) for the power-to-heat-to-power operation.

Due to variations in the sink temperature and the strong transient load of the CHEST system, the ORC usually operates at off-design conditions. In the framework of the CHEST system, Ghent University is responsible for the development of a novel off-design expander control for ORC technologies allowing high efficiency at a wide range of operating conditions. To operate efficiently, the expander of the ORC should be able to adapt to these transient loads.

The expander employed in the ORC prototype is a piston expander with variable valve timing (VVT). Varying the timing of the valves creates a varying expansion ratio which can be controlled to match the transient loads received by the CHEST system. R1336mzz(E) is selected as the working fluid. It is a hydro-fluoro-olefin that has very low global warming potential (GWP = 32) and zero ozone depletion potential and operates well within the constraints of the project. Towards this goal, modelling of the ORC system is performed and a test setup is in development to conduct experiments and evaluate the performance of the proposed system.

1.2. Purpose and Scope

The purpose and scope of the study is to develop an ORC system with an expander that can perform well even at off-design conditions. These off-design conditions push the ORC expander towards under- or over-expansion which induce additional losses. To offset this, a controllable expansion ratio over the expander is necessary. This is done by introducing a control algorithm that controls the timing of the exhaust process. For the piston expander considered in the project, this results in a variable valve timing.

Secondly, to achieve an optimal discharge of the TES, the working fluid needs to be specifically tailored to the working conditions of both the low and the high temperature TES. Furthermore there are additional constraints like the environmental impact and the safety.

Finally, the control procedure of the ORC in the CHESTER system needs to be clearly defined for safe and efficient operation.

These results are then used in the development of a laboratory scale ORC prototype of 10 kWe with advanced off-design expander control with high efficiency over a wide range of boundary conditions.

1.3. Methodology

The first step in the design is getting a clear overview of the boundary conditions and possible limitations. Secondly a design model was made to select the optimal working fluid and subsequently this model was used to size the different components.

After initial sizing, the actual components are selected from the manufacturers. At the same time the prototype is developed in CAD. The selection of components, the sizing and the development of the CAD model are an iterative process.

In each of the different steps the input from other partners in the project was crucial to converge to an overall integrated prototype of the CHEST system.

1.4. Structure of the document

The document is divided into 7 sections. Each of these sections elaborate on one particular aspect of the performed work.

Section 1 deals with the overall introduction of the CHEST and ORC systems studied and the purpose and scope of the work undertaken.

Section 2 details about the integration of the ORC system in the CHEST framework. The limitations and the boundary conditions within which the ORC has to operate has been defined. Furthermore, the development of the steady state model of the ORC and the expander is described along with the selection criteria of the working fluid refrigerant.

Section 3 elaborates on the sizing and selection of the components in the ORC prototype. It details the capacity of the heat exchangers, pump and the expander. Information about the safety measures incorporated in the setup are specified and details of the refrigerant lines and electrical cabinet installation has been provided.

Section 4 explains the operational procedure of the ORC system The method adopted for the start-up and shut-down of the ORC system has been elaborated and the strategies to exercise control over the system has been explained.

Section 5 provides with the details of the test rig construction for the ORC system for experimental campaigns. The current test rig on which the preliminary tests have to be performed is presented. The design of the ORC prototype and the details of the equipment and sensors to be installed are provided.

Section 6 concludes the document and provides with an overall summary of the work performed.

Section 7 provides the details of the heat exchanger design as obtained from the heat exchanger manufacturer SWEP.

1.5. Relations with other deliverables

The success of the other work packages and the CHESTER project on the whole heavily depends on the successful operation of the ORC at the desired conditions. Deliverables in work package (WP) 3 and Task 3.3 influences the operation of the individual technologies in WP3 and also for analysis of the CHEST prototype later in WP5. The selection of the working fluid in Task 3.3.1 influences the engine thermodynamic analysis and modelling performed by ECT in Task 3.4.1. The results produced in Task 3.3 will further be used in Task 3.5 for the extension of performance maps from lab scale to real size technologies. WP4 also depends on the results and data obtained from the packages WP2 and WP3 to model the behaviour of all the components of the system as a whole with a high level of detail in order to optimize the CHEST system.

In WP5, Task 5.1 requires preparation of space, and communicate the arrangements for the connection of cooling loops and electricity supply. Arrangements for safe operation procedures and related equipment is also considered along with data acquisition and layout planning. Task 5.2 involves the installation of the ORC in the CHEST system at DLR and its commissioning. The operation and testing campaign of the prototype is taken up in Task 5.3. All these tasks relate to the deliverables D5.1 and D5.2.

2. ORC integration in the CHEST concept

2.1. Working conditions and restrictions

Table 1. Project constraints.

Maximum temperature	133 °C
Minimum temperature	0 °C
Toxicity	A
Flammability	2L or less flammable
Pressure drop expander	Preferably >20 bar
GWP	<150
ODP	0

The working fluids under consideration will need to work under a predefined set of constraints related to the challenges in the CHESTER project. These constraints are detailed in Table 1. A number of potentially promising working fluids were examined for this cause.

2.2. Development of steady state ORC model

The thermodynamic model of the ORC was developed in Python 3.6.0. Working fluid thermodynamic properties were called from the reference fluid property database REFPROP 10. The pressures and inlet temperatures of the hot and cold fluid in the heat exchangers were defined, along with the degree of subcooling and superheating. A non-linear interior point optimization solver called IPOPT is utilized to solve a system of equations with the unknown temperatures and pinch points as variable. With the obtained values, mass flow rates of the working fluid and of the hot and cold fluid are calculated.

An exergy analysis proceeds these calculations

The exergy destruction in the evaporator ($\dot{I}_{lat}$), preheater ($\dot{I}_{sen}$) and the condenser ($\dot{I}_{cond}$) is given as:

$$\dot{I}_{lat} = \left(1 - \frac{T_0}{T_9}\right) \dot{Q}_{lat} - \dot{E}_{in} \tag{1}$$

$$\dot{I}_{sen} = T_0 \left[\dot{m}_{wf}(s_4 - s_3) - m_{h\dot{f},sen}(s_{11} - s_{12})\right] \tag{2}$$

$$\dot{I}_{cond} = T_0 \left[\dot{m}_{wf}(s_7 - s_2) - \dot{m_{cf}}(s_{14} - s_{13})\right] \tag{3}$$

Exergy destroyed in the expander is given as:

$$\dot{I}_{exp} = T_0 \left[\dot{m}_{wf}(s_7 - s_6)\right] \tag{4}$$

Exergy rate transferred to the working fluid is given as:

$$\dot{E}_{in} = T_0 \left[\dot{m}_{hf,sen}(h_{11} - h_{12}) - T_0(s_{11} - s_{12})\right] \tag{5}$$

The total exergy destroyed in the ORC is:

$$\dot{I}_{tot} = \dot{I}_{lat} + \dot{I}_{sen} + \dot{I}_{cond} + \dot{I}_{exp} \tag{6}$$

The cycle efficiency is given as:

$$\eta_{cyc} = \frac{W_{exp} - W_{pump}}{\dot{Q}_{lat} - \dot{Q}_{sen}} \tag{7}$$

The overall exergy efficiency is given as:

$$\eta_{ex} = \frac{W_{exp} - W_{pump}}{\dot{E}_{in}} \tag{8}$$

The exergy destruction ratio is given as:

$$y_{D,component} = \frac{\dot{I}_{component}}{\dot{E}_{in}} \tag{9}$$

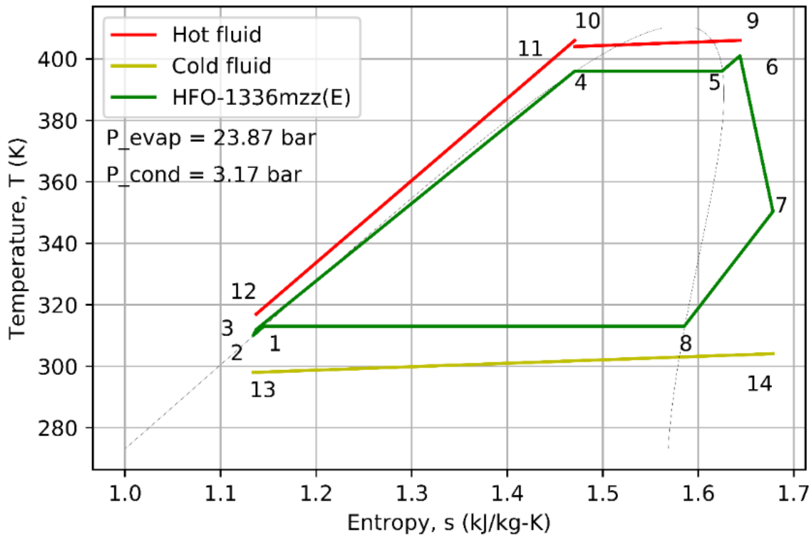

Figure 1. Temperature entropy plot of the simulated ORC system with R1336mzz(E) as working fluid.

where, T_0 is the reference temperature equal to 288 K, $\dot{I}_{component}$ is the exergy destruction of a given component, $\dot{m}_{wf}$ is the mass flow rate of the working fluid $\dot{m}_{hf,lat}$, $\dot{m}_{hf,sen}$ and $\dot{m}_{cf}$ is the mass flow rate of the hot fluid in the latent and sensible heat evaporator and that of the cold fluid respectively, W_{exp} the expander work, W_{pump} the pump work and W_{net} is pump work subtracted from the expander work.

2.3. Refrigerant selection

The selection of the working fluid in Task 3.4.1 is crucial to achieve the maximum engine thermal efficiency and rated power and for determining the design parameters providing the required power of the ORC high-pressure pump (Task 3.3.1) and the storage system (Task 3.2.1). This will be a key point as the cost competitiveness of the CHEST system heavily depends on the roundtrip efficiency that can be achieved.

Some of the potential working fluids for the heat pump and ORC cycle can be hazardous substances. They can, to some extent, be toxic, inflammable or explosive. Most commonly used refrigerants have high GWP (Global Warming Potential) and/or high ODP (Ozone Depletion Potential) and thus have a high environmental impact. Suitable environmentally friendly refrigerants (and cycles) for medium to high temperatures are currently under research and some products are already being commercialized. However, the most analysed conventional working fluid is still R245fa, as such it is also included in this report as a reference working fluid.

Table 2 lists the different working fluids studied for their chemical properties such as toxicity, flammability, GWP and ozone depletion potential (ODP). The data on toxicity, flammability, GWP and ODP are retrieved from the ASHRAE database whenever available. From Table 2, it is seen that many suitable hydrocarbons have a very high GWP and a non-zero ODP and hence these refrigerants cannot be considered for the project. On the other hand, hydro-fluoro-olefins (HFOs) are characterized by very low GWP and zero ODP.

Table 2: Chemical properties of working fluids.

Refrigerant	Toxicity[1]	Flammability[2]	GWP	ODP
R114	A	1	3.9	1
R1234ze(Z)	N.A.	N.A.	<10	0
R124	B	1	610	0.03
R13I1	N.A.	1	1	0
R142b	N.A.	N.A.	2270	0.065
R236ea	B	1	1200	0
R245fa	B	1	1030	0
R40	A	3	13	0.02
R600	A	3	20	0
R600a	B	3	3.3	0
RE 245cb2	N.A.	N.A.	N.A.	N.A.
DR-12	B	1	32	0
DR-14A	B	1	430	0
HFO-1336mzz(Z) (DR-2)	B	1	9.4	0
R1233zd(E)	A	1	1	0
R1224yd	N.A.	1	1	0
R1234yf	A	2L	1	0

Thermodynamic simulations were carried out to rank the performance of the working fluids based on the power output and to determine the pressure drop over the expander as detailed

[1] Toxicity classification:
Class A signifies refrigerants for which toxicity has not been identified at concentrations less than or equal to 400 ppm.
Class B signifies refrigerants for which there is evidence of toxicity at concentrations below 400 ppm.

[2] Flammability classification:
Class 1 indicates refrigerants that do not show flame propagation when tested in air at 21°C and 101 kPa;
Class 2 indicates refrigerants having a lower flammability limit of more than 0.10 kg/m3 at 21°C and 101 kPa and a heat of combustion of less than 19 kJ/kg;
Class 3 indicates refrigerants that are highly flammable as defined by a lower flammability limit of less than or equal to 0.10 kg/m3 at 21°C and 101 kPa or a heat of combustion greater than or equal to 19 kJ/kg.

in Table 3. In the model the pinch point temperature differences within the heat exchangers are taken as 5 K and the degree of superheating and subcooling are 5 K and 3 K respectively. The isentropic efficiency of expander and pump are both 70%. The pressure drop achieved across the evaporator and condenser at temperatures defined by the project goals are calculated. A pressure difference (ΔP) of around 20 bar would be optimal for the reciprocating type piston expander delivered by Viking Heat Engines (VHE) as it reaps higher efficiency at that pressure difference. Power relative to that obtained when using the cycle on the refrigerant R245fa is reported as it is widely used in the literature for study and can be a valuable reference point. Of the HFOs studied, R1234ze(Z) and DR-12 (R1336mzz(E)) are found to be the most promising working fluids that operate within the constraints of the project, while meeting the performance required for the expander and the ORC. While DR-12 is a developmental working fluid produced by Chemours.

Table 3: Simulation results for the working fluids[3].

Refrigerant	ΔP (bar)	Power (Relative to R245fa)
R114	18.77	1.08
R1234ze(Z)	18.79	1.08
R124	31.46	1.51
R13I1	33.1	1.12
R142b	27.15	1.15
R236ea	22.03	1.23
R245fa	18.07	1
R40	41.04	0.91
R600	20.04	1.12
R600a	25.36	1.23
RE 245cb2	20.53	1.32
DR-12	20.95	1.42
DR-14A	**	**
HFO-1336mzz(Z) (DR-2)	10.41	1.08
R1233zd(E)	14.67	0.51
R1224yd	16.24	0.80
R1234yf	**	0.74

[3] ΔP = P(Te) – P(Tc)
Where Te is the evaporator temperature = 120°C
And Tc is the condenser temperature = 20°C
* * Te is above critical temperature
N.A. = ASHRAE data unavailable
P = Power output from the Expander
η = Efficiency of the expander

3. ORC Component sizing and selection

The details of the sizing of each of the components of the ORC system are explained below. These calculations are based on the design values obtained from the steady state model of the ORC. The expander required output is defined by the project´s constraints, and the selected manufacturer was the company Viking Heat Engines. The heat exchangers and the refrigerant pump were selected according to the selection tools from the manufacturers SWEP and Hydra Cell, respectively. The layout of the ORC system can be seen in the process and instrumentation diagram (P&ID) in Figure 2.

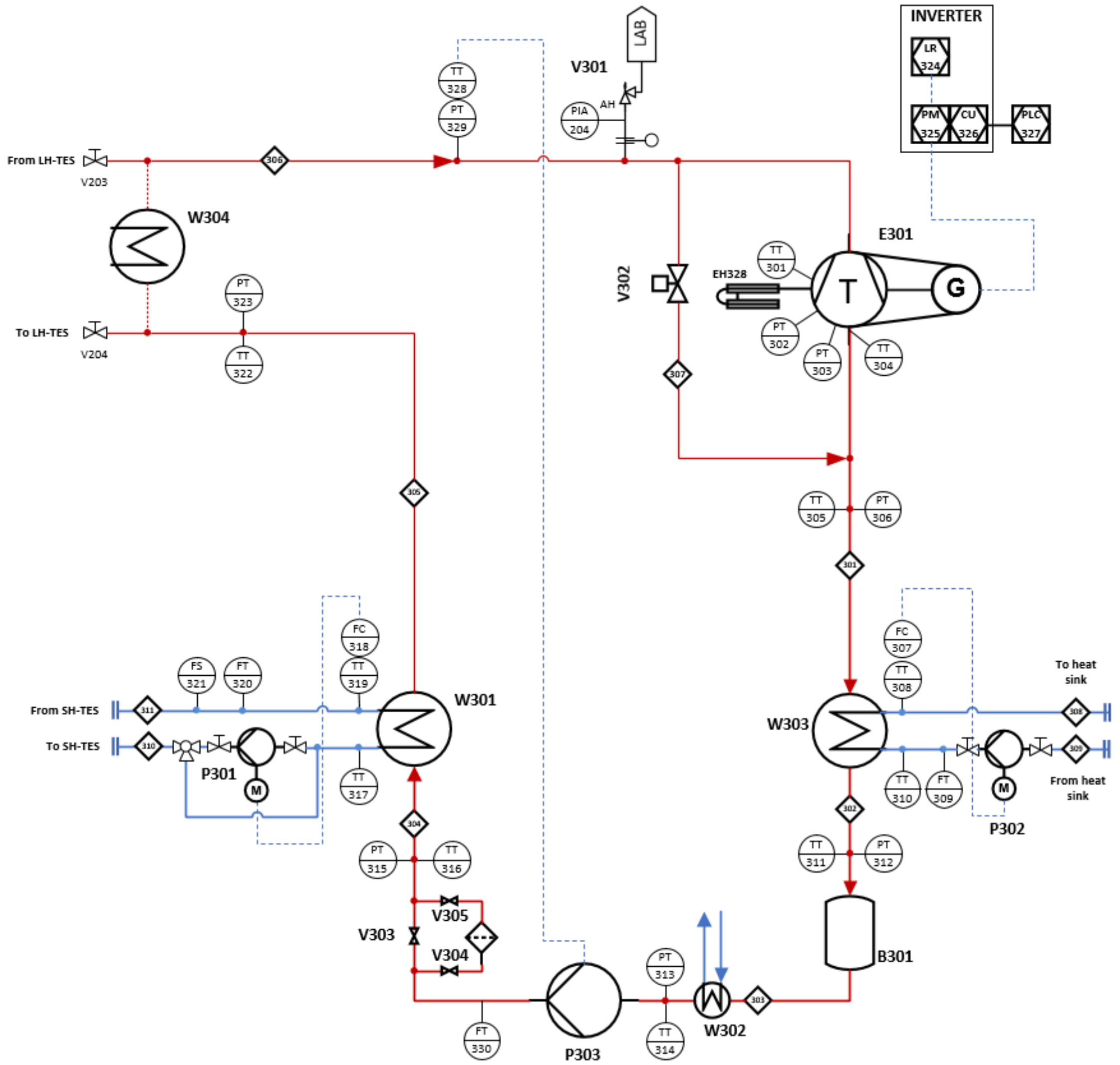

Figure 2. P&ID diagram of the ORC prototype system.

The different components in the P&ID are as seen in Table 4.

Table 4: Component description of the P&ID.

S.No.	Component ID	Description
1.	E301	Piston expander
2.	W301	Preheater
3.	W302	Subcooler
4.	W303	Condenser
5.	W304	Evaporator*
6.	P301	Preheater pump
7.	P302	Condenser pump
8.	P303	Refrigerant pump
9.	B301	Liquid receiver

* The evaporator is only required for tests of the ORC independently from the whole CHEST prototype. Once integrated into the full CHEST system, the evaporation process will take place inside the LH-TES.

3.1. Expander

In the framework of the CHESTER project, the boundary conditions and capacity limits have already been defined. We further look into the capacity of the ORC units produced by these manufacturers and narrow down on the prospective companies that have experience, interest and the required specification needed to meet our project goals. In this regard, an effort is made to list down all the companies that might meet our expectations and the results are tabulated in Table 5.

Table 5: ORC specifications supplied by the manufacturers.

Manufacturer	Temperature Source	Capacity	Reference
ABB	T > 200°C	500 kW	ABB
Viking Heat Engines	80 to 215°C	15.5 kW	
BEP E-Rational	85 to 115°C	500 kWe	E-Rational Low Temp ORC
Calnetix	T > 110°C	125 kW	Access Energy ORC System
Enertime	150 to 200°C	300 to 3000 kWe	Enertime ORC Module
Enogia	80 to 120°C	5 to 100 kWel	Enogia ORC Module
Exergy	80 to 350 °C	100 to 50 MWe	Exergy ORC Technology
Electratherm	77 to 122°C	35 to 110 kWe	Electratherm Power+Generator
Exoes	T < 300°C	4 kW	Exoes EVE
GE Power	T > 155°C	50 - 140 kW	Clean Cycle II R-Series
ORMAT	T > 126°C	45 MW or more	ORMAT ORC
TMEIC	T > 125°C	250 kW	TMEIC ORC
Turboden	-	2 to 10 MWe	Turboden Waste Heat Recovery
gT – Energy Technologies	T > 108°C	170 kWe	gTET ORC generator

The unique selling point for each of these manufacturer is their expander design. The expander used dictates the performance, the temperature levels, the capacity and the part-load operation. Especially for the CHESTER project the part load performance will be of significant importance as the load can vary depending on the conditions of electricity demand (electrical grid). Furthermore the temperature levels at the condenser side are highly variable depending if the rejected heat is used for district heating or just dissipated to the ambience. The few manufacturers (Viking Heat Engines, Exoes) that focus on this topic are working on waste heat recovery for vehicles where the dynamics are large. Upscaling of this technology and further modifying it for the CHESTER project will thus be crucial.

The expander selected for the investigation is a CEE 511 piston expander from Viking Heat Engines which is designed for waste heat recovery applications. The expander is designed to work with all common refrigerants of the 3rd and 4th generation (HFC + HFO). With its integrated synchronous motor it reaches very high efficiency in parallel to the ability to modulate the produced power as required through the variable speed between 500 RPM and 1500 RPM. The rated electrical power output of the expander, as indicated by the manufacturer, is 15.5 kW and can operate at a maximum inlet pressure and temperature of 30 bar and 215 °C.

3.2. Evaporator[4]

According to the preliminary steady state modelling of the ORC system with the selected working fluid, the mass flow rate of the refrigerant was calculated. Depending on the pinch points and boundary conditions the heat input required by the cycle in the evaporator was calculated. For the desired power output of 10 kWe, the working fluid mass flow rate ($\dot{m}_{wf}$) obtained was 0.61 kg/s. For these values, the heat input to the evaporator was 114.5 kW.

For the proposed CHEST system, the heat input in the evaporator is divided into a sensible heat input and a latent heat input. The latent heat input ($\dot{Q}_{lat}$) of 37.29 kW is to be provided by a latent heat thermal energy storage system (LH-TES) designed by DLR, Germany. The sensible heat input ($\dot{Q}_{sen}$) required for the ORC system is estimated to be 77.2 kW, and it will be provided by the sensible heat thermal energy storage system (SH-TES) also designed by DLR, and will consist on a pressurized hot water system. Two SWEP heat exchangers B400TH of the capacity 70 kW and 50 kW (evaporator and preheater) have been acquired for the experiments to obtain a combined capacity of 120 kW. The evaporator will only be used for the tests at UGENT, because the final prototype the evaporation process will take place inside the LH-TES.

- Preheater:
  - Working fluid side pressure drop: 0.585 kPa
  - Heating fluid side pressure drop: 4.93 kPa

- Evaporator:
  - Working fluid side pressure drop: 6.53 kPa
  - Heating fluid side pressure drop: 2.62 kPa

3.3. Condenser

The calculations made for sizing the expander were extended to size the condenser as well. The heat rejected in the condenser ($\dot{Q}_{cond}$) was estimated to be equal to 105 kW. A SWEP heat exchanger B400TH of capacity 120 kW was acquired for the ORC condenser. The heat exchanger design point provided by SWEP was based on calculations that assumed R1336mzz(Z) as working

[4] See Chapter 7 for details.

fluid, instead of R1336mzz(E) which was selected for the project. This is the reason why the heat exchangers acquired are a bit oversized on comparison to the calculated values.

The pressure drop in the condenser as provided by the manufacturer:

- Working fluid side pressure drop: 5.58 kPa
- Cooling fluid side pressure drop: 1.75 kPa

3.4. Refrigerant lines

The pressure drop in the different refrigerant lines in the ORC are theoretically calculated based on the pipe length, density of the working fluid at the given pressure and temperature and the diameter of the pipe. The following equation was used to calculate pressure drop along the length of the pipe:-

$$p_1 + \frac{1}{2}\rho v_1^2 + \rho g z_1 = p_2 + \frac{1}{2}\rho v_2^2 + \rho g z_2 + \sum f\,\rho\frac{Lv^2}{2D} + \sum K_L\frac{1}{2}\rho v^2 \quad (8)$$

Where p is the pressure, ρ is the density, L and D are the length and diameter of the pipe. The design value of the diameter of the pipes is 38.10 mm. The velocity is denoted by v, g is the acceleration due to gravity and z is the height of the pipe from the ground. F is the friction factor calculated from the Moody chart and K_L compensates for all pressure drop at the joints and connections in the pipe. According to the above equation, the estimated pressure drop in the pipes are as shown in Table 6.

Table 6: Pressure drop in the refrigerant lines of the ORC prototype.

S.No.	Line	Δp (Pa)
1.	Pump to Evaporator	6070.67
2.	Evaporator to Expander	62.82
3.	Expander to Condenser	1472.45
4.	Condenser to Liquid Receiver	1494.93
5.	Liquid Receiver to Subcooler	140.32
6.	Subcooler to Pump	16.77

3.5. Subcooler

A subcooling of 5 K will be considered after the condenser. This avoids any cavitation issues that might occur in the pump. According to the calculated mass flow rate of the working fluid within the cycle, the heat rejected in the subcooler is calculated to be 2.09 kW. The mass flow rate of the cooling fluid $\dot{m}_{cf}$) required to achieve this is estimated to be 0.1 kg/s.

A SWEP heat exchanger B28Hx34/1P-SC-S of capacity 7.6 kW with a cooling liquid mass flow rate of 0.2 kg/s will be used as a subcooler.

The pressure drop in the subcooler as provided by the manufacturer:

- Working fluid side pressure drop: 6.21 kPa
- Cooling fluid side pressure drop: 2.91 kPa

3.6. Refrigerant Pump

The refrigerant pump has to operate between the pressure levels of 23.87 bar and 3.17 bar. For this, a high-pressure diaphragm pump from Hydra-Cell is acquired. It can function at a maximum inlet pressure of 17 bar and a maximum discharge pressure of 69 bar at 1450 rpm and can discharge up to 30.6 l/min. An electrical motor of 2.2 kW capacity is used to operate the diaphragm pump. A 3 phase, 5.6A ABB motor drive (ACS355-03E-05A6-4) is used to operate the motor.

3.7. Safety Equipment

Safety equipment employed in the ORC prototype include relief valves, bypass over the expander and PLC controls. A pressure relief valve is placed just before the expander inlet to offset any unusual high pressure increase. A bypass valve is placed across the expander so that the working fluid may flow from the evaporator outlet line to the condenser inlet line. Opening the bypass valve before the expander also aids the start-up of the ORC and helps to make sure that the thermodynamic state of the working fluid prior to the entry in the expander is as desired. Once this is ensured, the bypass can be closed to let the entire mass flow through the expander at the design condition.

The electrical cabinet houses a Siemens programmable logic controller (PLC, Simatic S7) that takes input from the various components controlling the ORC system and continuously monitors their state. This PLC not only controls the flow of information from the input to the output devices, but also acts as a safeguard to ensure the operating limits are respected. It is an effective tool to monitor in real time the inputs and the outputs of the system and provides a safe shutdown of the system.

3.8. Electrical and control cabinet

An electrical cabinet is required to house the modules necessary to operate all the equipment, components, data acquisition and measurement systems. Drives for the motor of the pump and expander are mounted in the cabinet. The ABB drive (ACS355-03E-05A6-4) is used to operate the 2.2 kW motor of the Hydar-Cell G10X pump. Power required to operate the expander is also drawn from the cabinet. The drives, SLM/ALM modules and converters used to operate the expander are as follows:-

Table 7: Details of the drives and modules for the expander.

S.No.	Component	Description
1.	Siemens Sinamics 6SL3000-0BE21-6DA0	Basic Line filter for 16kW SLM/ALM and 20kW BLM
2.	Siemens Sinamics 6SL3120-1TE23-0AC0	Frequency Converter, Single Motor Module 30A
3.	Siemens Sinamics 6SL3130-6TE21-6AA4	Frequency Converter, Smart Line Module 16kW
4.	Siemens Sinamics 6SL3000-0CE21-6AA0	Choke
5.	Siemens Sinamics 6SL3040-1MA01-0AA0	Frequency Converter Equipment, Control Unit CU320-2 PN
6.	Siemens Sinamics 6SL3162-2MB00-0AC0	Connector for Power Cable

Wika pressure sensors, temperature measurements using PT100, flow and torque measurement instruments are connected to National Instruments modules (NI 9208 for pressure and flow and NI 9216 for PT100) that are mounted on the CRIO 9075 chassis on the cabinet. Siemens ALM and SLM units and converters are used to manage the electrical supply to the expander. A Siemens PLC module (Simatic S7) manages the communication of data between all these different control mechanisms to operate the test rig.

4. ORC operation and control

4.1. Control strategy

The ORC operation is based on a constant superheating degree at the inlet of the expander. By monitoring the pressure and temperature of the refrigerant at the inlet of the expander the degree of superheating is calculated. The degree of superheating is kept as low as possible but high enough to ensure safe expander operation away from the two-phase region. In practice this means the superheating is fixed at 5°C. The degree of superheating can be controlled by:

1) Changing the refrigerant mass flow rate
2) Changing the expander speed
3) Changing the heat source temperature or mass flow rate
4) Changing the condenser temperature

The first control action to consider, which has the fastest associated response time (also the mostly used control action), is changing the refrigerant mass flow rate by controlling the refrigerant pump speed. Higher pump speeds result in a lower degree of superheating and vice versa.

Since, the degree of superheating is dependent on pressure (and temperature) it can also be controlled by the expander speed which dictates the pressure level before the expander inlet. By decreasing the expander speed, with constant ORC refrigerant pump speed, the pressure at the inlet will increase but also the mass flow rate of the pump will decrease. With higher inlet pressure the heat input to the ORC will decrease but this will be also partially offset by the reduced pump speed. In practice this means that the level of superheat will decrease.

If adjustments of the heat source characteristics are allowed, increasing the heat source temperature result in a higher degree of superheat.

Changing the condenser conditions is not immediately used as a way to control the superheating as this is highly affected by the heat sink temperature, in our case the outdoor air temperature, but it might serve as a way to control the pressures inside the ORC and hence the degree of superheating.

4.2. Start-up procedure

The start-up procedure of every single ORC needs to be fine-tuned but in general the following actions are taken:

- The heat sink is turned on, meaning cooling water is flowing over the condenser. Also the required subcooler cooling fluid mass flow rate is provided.
- The heat source is connected to the evaporator with the required mass flow rate and the temperature of the heat source is kept stable.
- The bypass over the ORC expander should be open.
- Once the required heat source temperature is obtained, the ORC refrigerant pump is started at a 20-80 % of the nominal speed. The required start up speed depends on the ORC and should be determined experimentally. In some machines this is very low (20%) while in others this can be immediately set to 80 %.

- Simultaneously with starting the ORC refrigerant pump the expander speed is set to ± 25-50 % of the nominal expander speed.
- Once the refrigerant temperature after the evaporator is sufficiently high the bypass is closed.
- Gradually the ORC refrigerant pump speed will be increased to ± 80% while ensuring that at all time the refrigerant exiting the evaporator is in the gas phase. If the start up speed of the ORC refrigerant pump was already 80 % of the nominal speed the next step is executed.
- Simultaneously with increasing the ORC refrigerant pump speed the expander speed will be increased up to 80%. Mainly the pressure before the expander inlet and the degree of superheat determine the ramp up speed.
- Once the pressures and temperatures of the refrigerant are reaching a steady state, the ORC refrigerant pump speed and expander speed can be increased to/or close to the nominal values. The ORC refrigerant pump speed is increased with steps of 10-20 RPM while the expander speed with steps of 50-100 RPM.
- At all-time the heat source temperature is maintained at fixed state.
- Once the ORC is in steady state the heat source characteristics can be changed, as well as cooling loop characteristics.

4.3. Shut-down procedure

The shut-down procedure is easier to execute. Generally, following actions are taken:

- The heat source system should be disconnected while the ORC is still in operation. This will result in decreasing heat source temperatures (heat source mass flow rate is maintained). For the CHEST system, the PCM (LH-TES) cannot be disconnected, but the sensible heat source (SH-TES) can.
- With the decreasing heat source temperature the degree of superheating will decrease and evaporating pressure will decrease. The expander speed is lowered gradually (100 RPM at once) to maintain ORC operation and ensure complete evaporation.
- Simultaneously with decreasing the expander speed also the ORC refrigerant pump speed is lowered gradually (max 50-100 RPM at once).
- Once the pressures inside the ORC decrease drastically or the refrigerant mass flow rate decrease drastically the bypass is opened, the expander speed is set to 0 RPM and the ORC refrigerant pump is stopped.
- Condenser cooling is still provided for 5-15 minutes after ORC shut down to ensure most of the ORC refrigerant is in the liquid phase.

5. Construction of the experimental test-rig

The experimental test rig developed for the integration into the CHEST system is to be shipped to DLR in M24. It was decided that two test rigs were to be developed, one is to be sent to DLR and the other to remain at the home laboratory. Towards this end, a preliminary prototype test rig was developed by modifying the existing ORC test rigs at Ghent University for a preliminary experimental campaign to test the expander performance and gain knowledge on the system control. The preliminary test rig is expected to be in operation beginning of November 2019. At the same time, a new test rig is to be manufactured with all the components that has to be shipped. This prototype will be full tested and characterised before being shipped to DLR, while the preliminary test rig stays at UGent.

5.1. Preliminary prototype test rig

The existing test rig at Ghent University consists of the following components:-

- Preheater
- Evaporator
- Condenser
- Screw Expander
- Vertical Liquid receiver
- Pump
- Subcooler

The setup is installed with a screw expander that facilitates oil flooding as seen in Figure 3. The piping connections are done in a way that extension of the lines are possible and the frame has enough space to mount another expander. The piston expander being considered for the project is seen in Figure 4. The design modifications made to the existing test rig to house the piston expander can be seen in Figure 5.

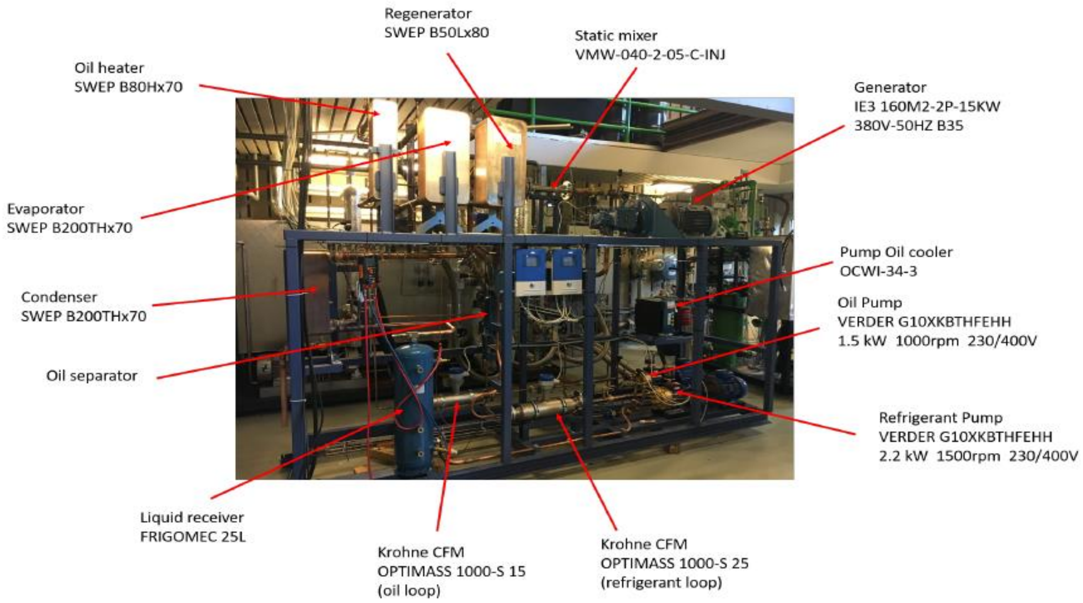

Figure 3. Current test rig at Ghent University.

Figure 4. Piston expander from Viking Heat Engines.

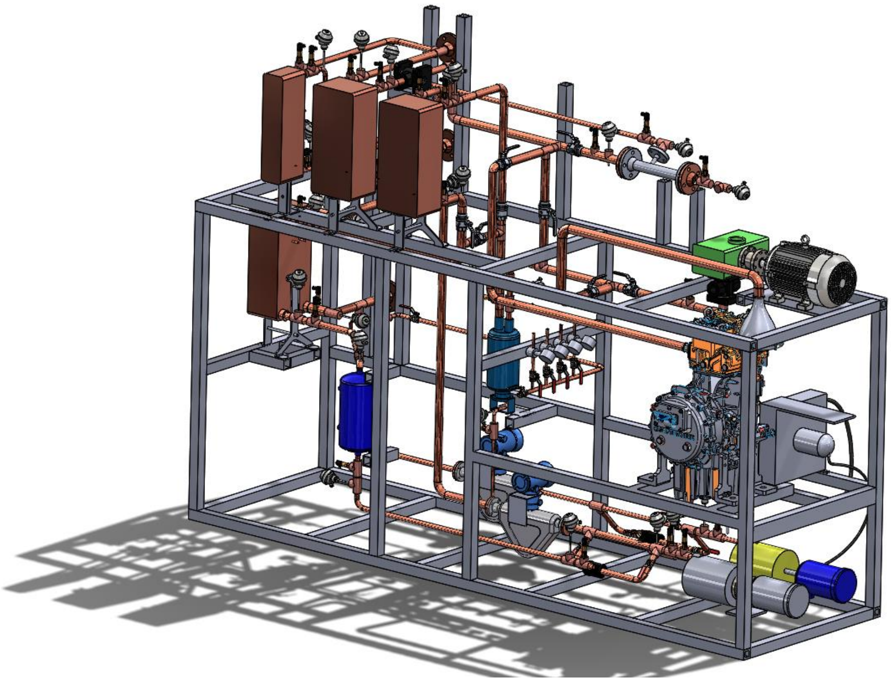

Figure 5. Modified design of the existing test rig to incorporate the new expander.

The current test rig houses Wika pressure sensors and PT100 temperature sensors connected to National Instruments module NI 9216 and 9207 modules for pressure and temperature measurements at various points in the cycle. These modules facilitate the visualization of the data directly into the LabVIEW VI and through the integration of REFPROP module plots a live temperature entropy of the running system.

The newly installed expander provides for the manipulation of the valve timing, creating a varying expansion ratio that can help match the varying load in the CHEST system. The manipulation of the valves of the expander is achieved by a stepper motor attached to the expander assembly. The expander stepper needs a motor controller with clock-direction controlling which is necessary to operate the expander in the different engine loads. A piston cylinder assembly is expected to have vibrations and preliminary tests of the high temperature heat pump compressor at Tecnalia which is supplied by the same manufacturer as the expander at Ghent University was shown to vibrate considerably at low RPMs. To tackle this situation, flexible piping connections are proposed to be used in the new ORC prototype test rig.

5.2. ORC prototype test rig

The design of the new test rig to be built can be seen in Figure 6. The following components will be installed on the prototype test rig:

- Evaporator & Preheater
- Condenser
- Piston Expander
- Horizontal Liquid receiver
- Pump
- Subcooler

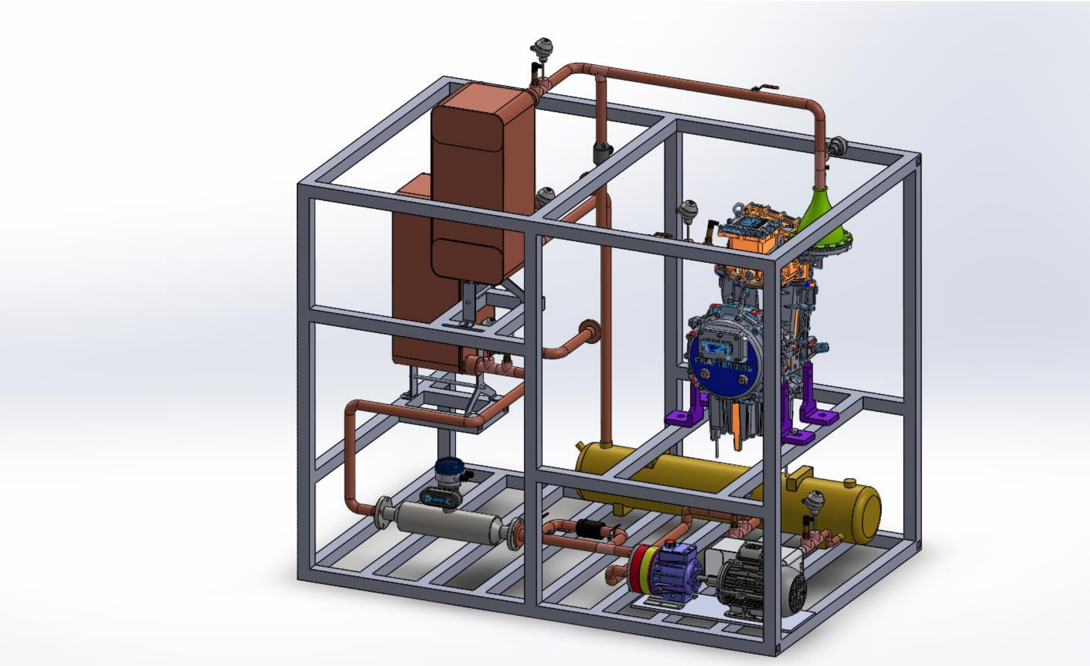

Figure 6. Design of the ORC prototype to be sent to DLR.

The specifications of the acquired components are provided below:

Table 8: Component specifications of the ORC prototype.

S.No.	Component	Company	Description
1.	Evaporator	SWEP	B400THx70/1P. 70.29 kW
2.	Preheater	SWEP	B400THx50/1P, 67.58 kW
3.	Condenser	SWEP	B400THx120/1P, 121.4 kW
4.	Subcooler	SWEP	B28Hx34/1P, 7.6 kW
5.	Expander	Viking Heat Engines	Piston Expander, 15.5 kW @1500RPM
6.	Pump	Hydra-Cell	30.6 l/Min @1450 RPM
7.	Pump Motor	Verder	2.2 kW
8.	Liquid Receiver	ECR-Belgium	40 L

The temperature and pressure sensors will be placed all along the ORC working fluid loop as is seen in the P&ID of the prototype system. A total of 8 points of interest have been identified to take measurements in the system. Wika Pressure sensors of 4 to 20 mA range and PT100 resistance thermometers will be used to make pressure and temperature measurements. These sensors will be connected to National Instruments module NI 9208 and NI 9216 for data acquisition on an NI chassis CRIO 9705.

6. Conclusions

The details of the design of the ORC laboratory prototype has been presented in this document. The purpose and scope of the study was defined as to develop an ORC system with an expander that can perform well even at off-design conditions. To reach this objective, a novel piston expander is employed that provides a control over the exhaust process through a control algorithm. A process methodology was defined based on the constraints of the project and boundary conditions. Initial design and sizing of the system was carried out iteratively and came to a convergence after detailed discussions and input from other project partners.

A steady state model of the ORC system operating within the CHEST concept was developed. A number of refrigerants were examined for the optimal working fluid as the overall efficiency and cost competitiveness of the CHEST system heavily depends on the selected working fluid. Finally, an environment friendly refrigerant with the desirable properties was selected and R1336mzz(E) was adopted. A detailed cycle analysis and exergy analysis of the ORC system was performed and presented.

Working within these constraints, other equipment like heat exchangers and refrigerant pump were sized and ordered. All the necessary instrumentation details and data acquisition systems were acquired. The designs were made with safety procedures in mind and necessary precautions were taken.

7. Annex

The details of the heat exchanger design as provided by the manufacturer SWEP is shown below:

Evaporator: SWEP B400THx70 / 1P			
		Side 1	**Side 2**
Flow direction		Counterflow	
Capacity	kW	70.29	
Mass flow	kg/s|m³/h	0.61	1,450
Total heat exchanging surface	m²	10.4	
Heat Flux	kW/m²	6.49	
Pressure drop (Total)	kPa	0.585	4.93
Number of plates		70	
Heat Transfer coefficient	W/m².°C	2990	

Preheater: SWEP B400THx50 / 1P			
		Side 1	**Side 2**
Flow direction		Counterflow	
Capacity	kW	67.58	
Mass flow	kg/s|m³/h	0.61	1,450
Total heat exchanging surface	m²	14.8	
Heat Flux	kW / m²	4.76	
Pressure drop (Total)	kPa	6.53	2.62
Number of plates		50	
Heat Transfer coefficient	W / m² °C	746	961

Condenser: SWEP B400THx120 / 1P			
		Side 1	**Side 2**
Flow direction		Counterflow	
Capacity	kW	121.4	
Mass flow	kg/s|m³/h	0.61	7,326
Total heat exchanging surface	m²	25.6	
Heat Flux	kW / m²	4.74	
Pressure drop (Total)	kPa	5.58	1.75
Number of plates		120	
Heat Transfer coefficient	W / m² °C	2200	2560