# THERMODYNAMIC SIMULATION AND EXPERIMENTAL VALDIATION OF A CASCADED TWO-STAGE ORGANIC RANKINE CYCLE

Frithjof H. Dubberke*, Klaus-Peter Priebe*, Jadran Vrabec*, Maximilian Roedder‡ and Matthias Neef‡

*University of Paderborn,
Warburger Str. 100, 33098 Paderborn, Germany
E-mail: f.dubberke@thet.upb.de
Web page: http://thet.uni-paderborn.de

‡University of Applied Sciences Düsseldorf,
Josef-Gockeln-Str. 9, 40474 Düsseldorf, Germany
Web page: http:// http://www.fh-duesseldorf.de

## ABSTRACT

Employing a zeotropic mixture as a working fluid in organic Rankine cycles (ORC) allows for an exergetically favorable heat transfer to the evaporator due to its temperature glide. However, during heat discharge from the condenser, the temperature glide could become a disadvantage. Therefore, a cascaded combination of a two-staged ORC, where the high temperature (HT) cycle is operated with a zeotropic mixture and the low temperature (LT) cycle is operated with a pure fluid in supercritical mode, facilitates both favorable heat uptake from sensible heat sources as well as heat discharge to the environment [6].

As a test rig for according two-stage cycle innovations, an electrically heated cascaded organic Rankine cycle (CORC) was designed and commissioned at the University of Paderborn. To achieve a high efficiency in each cycle, the design strongly depends on the temperature level of the heat source. The integration of four electrical heating rods as a primary heat source into the HT cycle – each with 50 kW and one of them adjustable – the design enables for the specification of different temperature levels. The LT cycle is supplied with the discharged thermal energy of the HT cycle.

After successful commissioning of this two-stage CORC, experimental results are used to evaluate cycle and component performance to study the intended design. For this purpose, a detailed cycle simulation is performed with *EBSILON®Professional* [13], which can be fed with the operating parameters. The aim is to complement the flexible test rig with a suitable thermodynamic model that allows for the study of cycle variations, such as working fluid changes, hardware design improvements, etc.

First results on modeling and experimental validation are presented for a combination of two pure fluids that exemplify the heat integration between the HT and LT cycles. With a validated simulation tool based on energy and mass balances as well as suitable equations of state, the optimization of individual components of the CORC test rig, such as heat exchangers, pumps, condensers, turbines, as well as working fluids can be carried out rapidly and at low cost. The long-term goal of the present project is to put a two-stage CORC system into practical application.

## 1. INTRODUCTION

Many types of organic Rankine cycles (ORC) aiming at the recovery of exergy from waste heat have been described in the literature. Different cycle layouts and numerous components were studied for a wide variety of working fluids to obtain a better power cycle efficiency for the given heat source characteristics. Depending on the size and the economic situation of an individual project, cycle construction can be more sophisticated and complex to reach a higher power efficiency. The objective of the present experimental setup was to design and construct an ORC test rig that can help to identify combinations of working fluids and operating conditions that exploit the maximum of the incoming exergy from a waste heat source and thus achieve the maximum achievable utilization efficiency.

The cascaded ORC process (CORC) couples two organic Rankine cycles in a subsequent manner (cf. Fig. 1). It is intended to reach with this setup a higher exergetic performance than with conventional ORC by reducing dissipation losses [3].

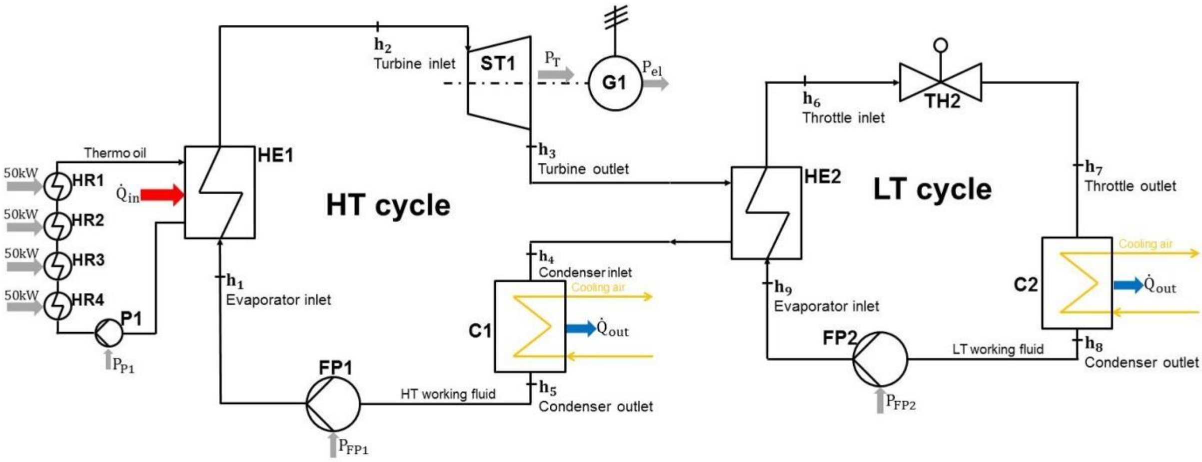


**Figure 1:** The two-stage ORC couples two organic Rankine cycles as a cascade (CORC).

To maximize the performance of the CORC process, zeotropic mixtures appear to be a reasonable choice as working fluids due to potentially low exergetic losses in the evaporator [9]. Having an almost constant temperature gradient $\Delta T$ as a driving force for the heat transfer from waste heat source to working fluid, because of the temperature glide of mixtures during evaporation, may significantly minimize exergetic losses there. Moreover, mixtures, depending on their components and composition, allow that their temperature glide is adjusted to the heat source characteristics. Although the heat sink for condensation is not isothermal [1], its difference in temperature is usually much smaller than the temperature glide of the zeotropic mixture. Therefore, the temperature glide during condensation towards a heat sink could be exergetically unfavorable and would neutralize a large part of the advantage during evaporation. Therefore, the second, i.e. low temperature (LT), stage of the present CORC utilizes the residual exergy from the condenser of the first, i.e. high temperature (HT), cycle. Transferring the process-flow sheet in Fig. 1 of the CORC into a temperature-entropy (*T*-*S*) diagram reveals the exergetic utilization of a given sensible waste heat source (cf. Fig. 2).

Fig. 2 shows a schematic example for a CORC where the HT cycle is operating with a zeotropic mixture working fluid in subcritical mode and the LT cycle is operating with a pure working fluid in supercritical mode [2, 7]. The goal is to minimize the area where the heat is brought into the cycle above ambient temperature, minimizing exergy losses [7]. In this work, the use of an internal heat exchanger (IHE) was not studied because it does not affect the exergetic utilization, but rather the thermal efficiency which is insignificant for rating the performance of a cycle.

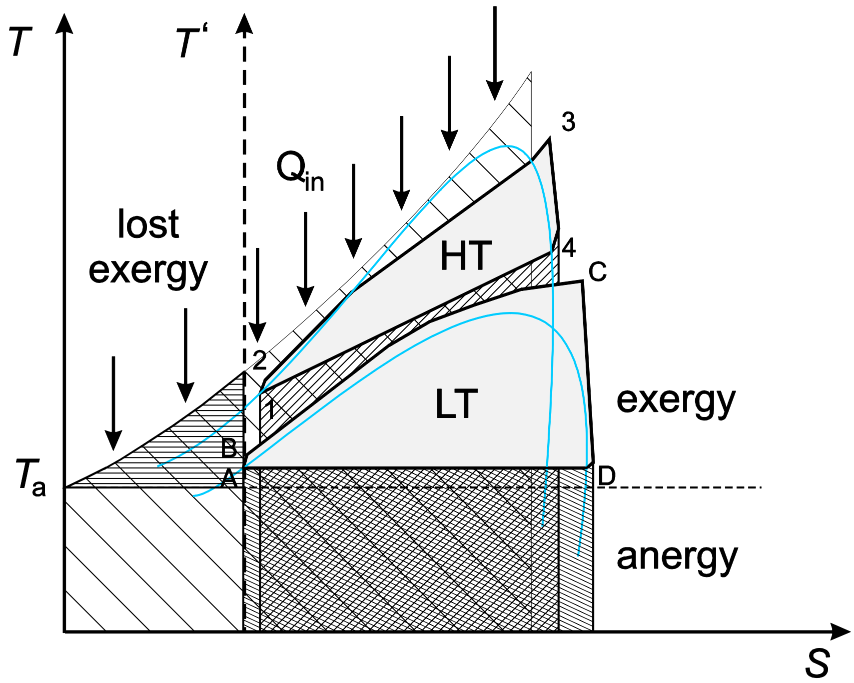


**Figure 2:** Exergetic analysis of the CORC process: The HT cycle cools the sensible waste heat source down to low temperatures; the LT cycle is linked to the HT cycle via a heat exchanger that acts as a condenser for the HT cycle and as an evaporator for the LT cycle.

## 2. CORC SETUP

For start-up and testing reasons of the CORC and its components, the HT cycle was filled with about 50 l of cyclopentane and the LT cycle with about 30 l of hexamethyldisiloxane (MM). Even though MM is not a classical low temperature working fluid and is therefore not very much suited for the LT cycle, it was chosen here for testing reasons in terms of its applicability for the installed pump and plate heat exchanger. In addition to its favorable safety in handling, MM might be suitable for trilateral ORC processes [4] and therefore its behavior in a throttle was of interest here.

Due to safety reasons for handling flammable or hazardous fluids, the CORC was placed outside of our laboratory building in a separate container. The CORC was designed in PN 100 for the HT cycle and PN 60 for the LT cycle according to AD 2000-Regelwerk B7. All flanges were sealed with meshed metal (chrome-nickel-molybdenum alloy, material no.: 1.4404) and strengthened graphite gaskets novaphit SSTCTA-L provided by Frenzelit [11], limiting the leakage rate to a tolerable amount.

The heating cycle (HC) was driven by a pump (Allweiler NTWH 25 200/01; cf. Fig. 3 (1)) which was capable of a mass flowrate of up to 450 g/s of therminol 66 [10] which was heated by four 50 kW electrical heating rods. The combined power could fully variably be adjusted from 0-200 kW up to a temperature of 350°C.

A progressive cavity pump (Netzsch Mohnopump: NM031SY12S; cf. Fig. 3 (4a,b)), based on a steel rotor, which is embedded into a chemically resistant synthetic stator with a helical geometry, worked up to a pressure of 4 MPa and was capable to feed a mass flowrate of up to 200 g/s. This model was used as a feed pump for both cycles.

The overall waste heat from the condensers was dissipated to the environment via an air-cooled heat exchanger outside of the container, which was connected by plate heat exchangers (GEA WTT PL 150; cf. Fig. 3 (5)) to the HT and the LT cycle. This separate cooling cycle worked on the basis of an ethylene-glycol/water mixture (1:1.25) and was fed by a pump (Grundfos TP 50-120/2) with a volume flowrate of 4.89 l/s.

The design of the CORC turbine (cf. Fig. 3 (6)) was based on an axially fed centrifugal pump that expanded the superheated vapor through its curved Laval nozzles that were embedded in a blade wheel (1.4305 / X8CrNiS) outwards to the radial expansion tube in the turbine casing (1.4006+QT / X12Cr13). Due to the impulse principle, the torque was mainly generated by the acceleration of the high velocity flow in the blades. The connection between turbine and generator, which was a six pole synchronous servomotor working at 50 Hz (type SK-190-1-30-560 T1), was done by a non-contacting magnetic coupling. For minimizing the gap losses in the turbine, the gap was designed as a labyrinth seal with 12 steps and a rotary shaft seal. Two angular ball bearings in o-arrangement and one deep groove ball bearing form the fixed bearing at the turbine side were used to minimize the slackness of the blade wheel. For cooling and lubrication of the rolling bearings, the casing was flooded with hydraulic oil (Mobil DTE 10 EXCEL 15) in a separate cooling cycle (BCC). The focus of this turbine lied on the application for testing different working fluids for a wide range of thermodynamic conditions and therefore its efficiency was initially subordinate.

The CORC control software framework is based on two systems: the graphical user interface (GUI), which was programmed in Agilent VEE, and a C program which provided functions for the GUI to access the programmable logic controller (PLC). The PLC worked for digital in- and output (DI/DO) and analog in- and output (AI/AO) which expect either a voltage from 0 to 10 V, a current from 4 to 20 mA or a resistance, such as in case of temperature (Pt1000) sensors. The AO connectors were used for setting the motor speed of the pumps by sending a signal from 0 to 10 V to the device. The DI could only detect an activated current (24 V) or deactivated current (0 V) and were used for receiving the error status from the devices. With the DO, variable frequency drives or non-controllable actors were turned on or off by sending a current to the device or their dedicated contactor. All these in- and output data were monitored and were accessible in the GUI. The alternative to a PLC is a hard-wired programmed logic controller (HPC), which was used for important security functions. A variable-frequency drive (VFD) frequency converter was applied in electro-mechanical systems to control pumps and generators.

The mass flowrate measurement worked with a differential pressure aperture according to DIN EN ISO 5167. To ensure that the maximum working temperature of the pressure sensors did not exceed 80°C, a flexible metal tube was used as an extension which was sealed with a copper gasket, respectively, PTFE to the piping system.

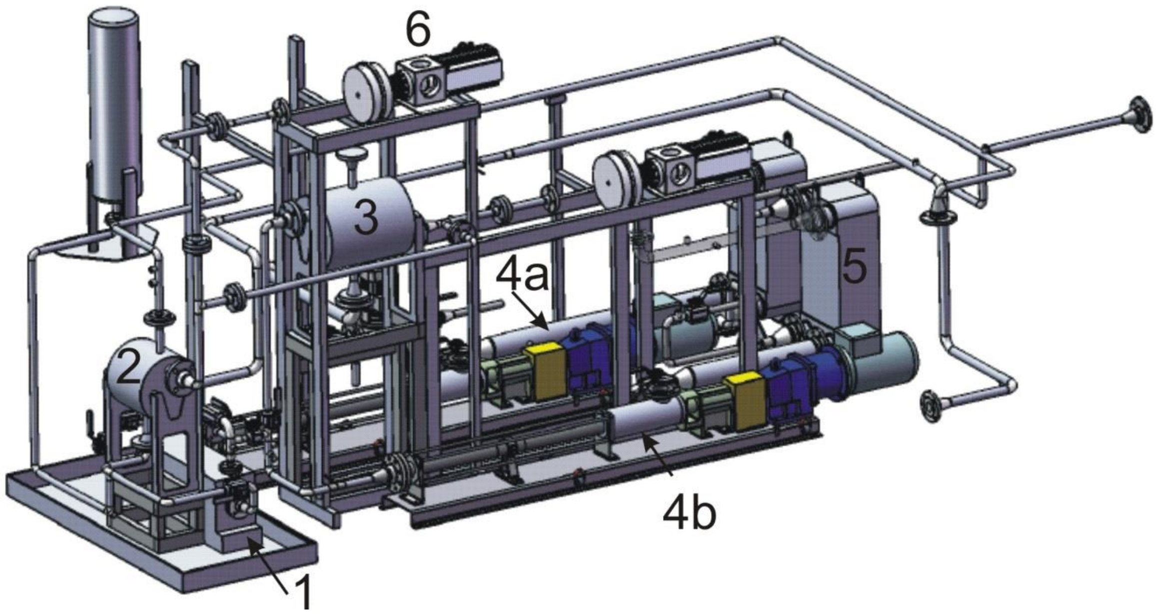


**Figure 3:** CORC design: 1: pump for heating cycle; 2: heat exchanger (HE1) of HT cycle; 3: heat exchanger (HE2) of LT cycle; 4a,b: feed pumps for HT and LT cycle; 5: condensers; 6: HT turbine and generator.

## 3. EXPERIMENTAL SETUP AND OPERATION

Before operating the CORC, a standardized safety procedure was carried out, such as checking sensors, valves and leakage in the piping and devices. First, the CORC cooling cycle with the air-cooled heat exchanger was started, before the heating cycle was fired up to a temperature of 180 °C with a thermo oil mass flowrate of about 400 g/s. After activating the cooling cycle (BCC) for the bearings of the turbine of the HT cycle, the turbine (ST1) was set to 3000 rpm. Subsequently, the mass flowrate of the HT cycle was increased by the feed pump FP1 until the inlet temperature of the heating cycle in heat exchanger HE1 stayed constant. This was done until the mass flowrate in the HT cycle and the temperature of the heating cycle was balanced at about 550 K and a mass flowrate of about 144 g/s with an overall transferred heat flow in HE1 of approximately 100 kW was reached.

After reaching a steady state in the HT cycle, the mass flowrate of the feed pump FP2 was set such that the maximum pressure of 0.8 MPa of the throttle TH2 in the LT cycle was reached. This was the case for a mass flowrate of 32 g/s of MM with a total heat flux of 10 kW in HE2.

## 4. MODEL DESIGN

The CORC structure was implemented in *EBSILON®Professional* [13], which is a simulation tool based on the thermodynamic equations of state and using the *REFPROP* database [12] for fluid properties. The first design calculation of the CORC builds the foundation for later off-design simulations and a system validation, which were not part of the present work. Therefore, the simulation model was generated with a strong association to the experimental setup. Component's efficiencies for the thermodynamic design were defined according to the manufacturer's instructions or in accordance with preceding experimental operations. The final flow diagram of the cycle is shown in Figure 4.

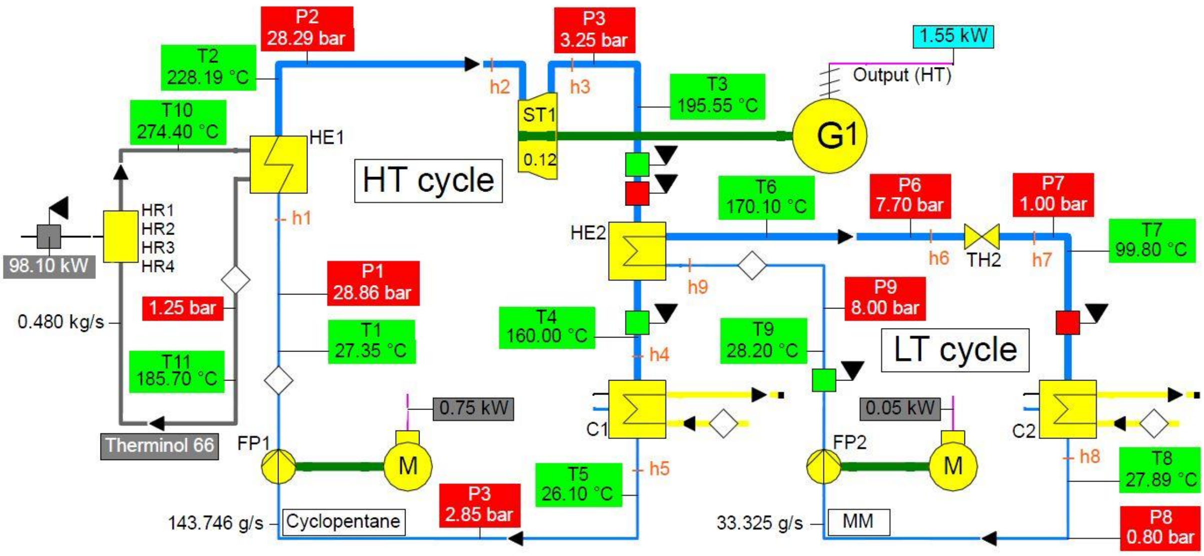


**Figure 4:** CORC design calculation with *EBSILON®Professional* for a steady state in case of an input heat flow of 98.1 kW.

### MODEL COMPOSITION ON THE BASIS OF THE EXPERIMENTAL SETUP

The modelling started with the thermo oil cycle, which supplied 98.1 kW of electrically generated heat to the HT by HE1. For the thermo oil cycle calculation the thermodynamic properties of Therminol 66 [10] were manually implemented into the EBSILON®*Professional* thermoliquid-database. The thermo oil mass flow was calculated with the experimentally specified temperature difference (T10-T11) and pressure. With the required input parameters of HE1 – working fluid mass flow, input temperature (T1) and main steam pressure (P1) – the thermodynamic start condition ($h_1$) for the HT calculation was defined. Pressure losses were taken into account by definition of a pressure drop inside a heat exchanger (e.g. 0.057 MPa – P1-P2). Here, the calculated turbine inlet condition ($h_2$) and the specification of the turbine outlet ($h_3$) enable for the calculation of the turbine characteristics. It was intended that the high temperature heat is transferred with a suitable high temperature difference (T3-T4), associated with the inlet temperature (T9) of the LT working fluid. However, the experimental system operation presented a high outlet temperature (T4) due to the low load in the LT. Condensing the HT working fluid against the ambient air leads to the condenser outlet condition ($h_5$).

With a given heat input into the LT and experimental definition of the HE2 inlet condition ($h_9$) the throttle inlet condition ($h_6$) was calculated with the given low temperature working fluid mass flow. The working fluid was throttled with a defined pressure drop known form the experiment before it passes through the condenser (C2), where it is also cooled down to ambient condition.

## HIGH TEMPERATURE CYCLE

During a steady state operation, the high temperature working fluid was compressed up to 2.8 MPa, (state $h_5$ to $h_1$), before the driving heat flow from the electrical heater (98.1 kW) was transferred in the ECO/evaporator/superheater (HE1,state $h_1$ to $h_2$). This leads to heatingof the working fluid mass flow (0.144 kg/s) from 27.4 °C up to its superheated steam state as shown in the *T*,*s* diagram in Figure 5. With the calculated turbine inlet temperature (228.2 °C) and given enthalpy slope, i.e. $h_2$ to $h_3$, the isentropic turbine efficiency (0.12) was calculated with

$$\eta_{\mathrm{s,turbine}} = \frac{|h_3 - h_2|}{|h_{3,s} - h_2|}. \tag{1}$$

After expansion, the working fluid was still far in the superheated vapor state, due to the poor turbine efficiency, which enables the transfer of the excess heat into the LT cycle, i.e. $h_3$ to $h_4$. Fig. 5 shows that the state $h_4$ is still superheated and that no condensation takes place in HE2 for the HT cycle. Therefore, the cyclopentane was condensed and supercooled only in the condenser (C1), i.e. $h_4$ to $h_5$ and a large part of the heat flow, which was assigned for the LT cycle, was dissipated to the environment.

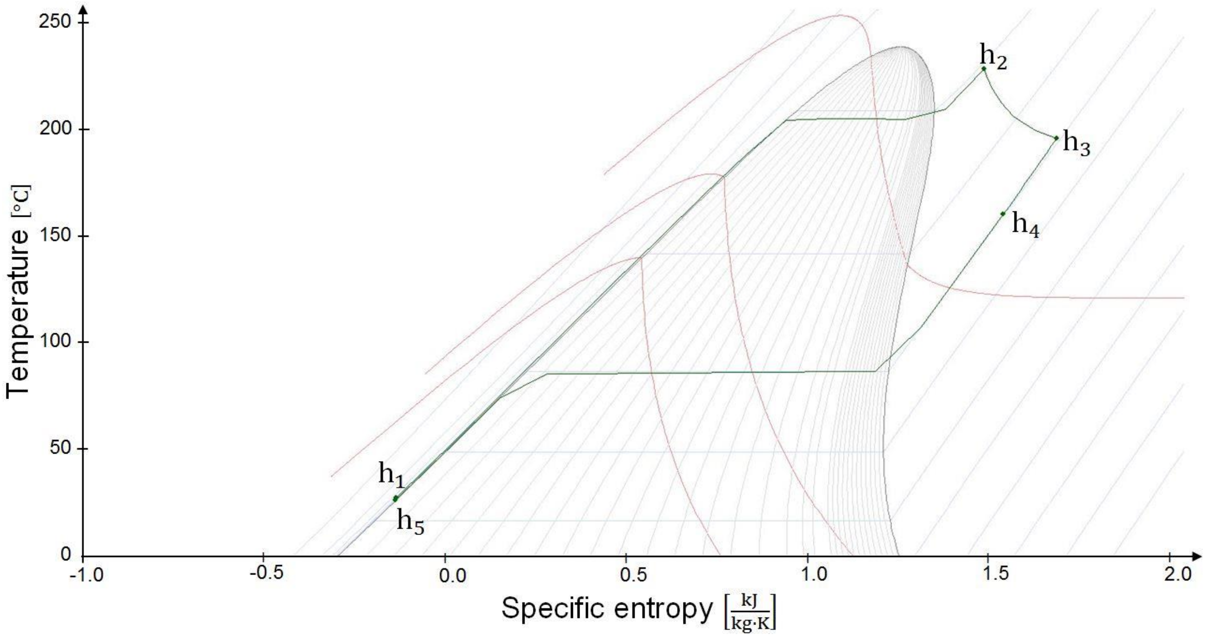


**Figure 5:** Temperature-entropy (*T-s*) diagram of the HT cycle.

### LOW TEMPERATURE CYCLE

The working fluid MM of the LT cycle was compressed to 0.77 MPa before it is heated up to 170 °C in the HE2 to $h_7$. Note that this state point was still in the liquid state, cf. Fig. 6. Subsequently, the fluid was throttled (TH1) through the two-phase region by a pressure drop of about 0.7 MPa, but did not pass the dew line ($h_7$ to $h_8$). As described above, the heat flow from the HT cycle to the LT cycle in HE2 was insufficient and rather poor so that most of the heat flow was condensed and supercooled from $h_8$ to $h_9$ in C2 to the ambient.

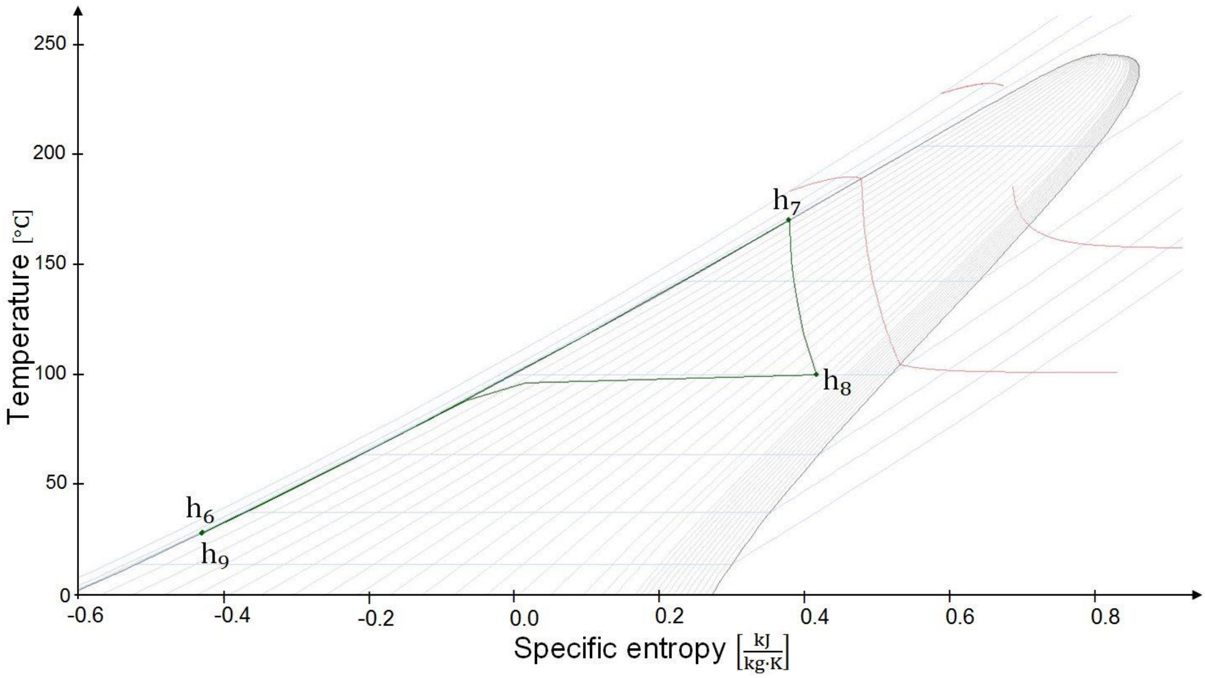


**Figure 6:** Temperature-entropy (*T*-*s*) diagram of the LT cycle.

## 5. EXPERIMENT VS. SIMULATION

In general, cycle simulations are performed using optimized operating parameters, with respect to a suitable electric cycle efficiency, to improve plant operation. Thus high cycle efficiencies are reached by adjusting the operating parameters gradually until the calculated ones are met. However, for the present vice versa approach, the simulation parameters were thoroughly deduced from the experimental data of the CORC power plant (cf. Table 1) in order to represent the design state of the current construction.

**Table 1:** Component efficiencies specified for the present CORC simulations.

| Cycle | Abbrev. | Component | Efficiency | Isentropic |
|---|---|---|---|---|
| | ST1 | Turbine | | 0.12 |
| | G1 | Generator | 0.90 | - |
| | FP1 | Feed pump | 0.85 | 0.80 |
| | TH2 | Throttle | - | - |
| | FP2 | Feed pump | 0.85 | 0.80 |

Hence, a suitable comparison between simulation and experiment, with a special focus on the temperature T2 of the HE1 as an essential component, was performed by a separate comparison of the cycle components (e.g. heat exchangers) in the following way: As listed in table 2 a steady state with a 98.1 kW heat flow brought into the HT was performed. For the simulation the temperatures HC in- and outlet (T10, T11) were implemented. T1 was calculated by the given isentropic efficiency of the feed pump FP1 and given pressure P1.

**Table 2:** Experimental data and simulation data – 98.1 kW

| | Experimental data | Simulation data |
|---|---|---|
| Transferred heat | 98.1 kW | 98.1 kW |
| HC inlet (T10) | 274.4 °C | 274.4 °C |
| HC outlet (T11) | 185.7 °C | 185.7 °C |
| | | |
| Mass flow HT | 144 g/s | 144 g/s |
| HE1 inlet (T1) | 27.3 °C | 27.4 °C |
| HE1 inlet (P1) | 2.886 MPa | 2.886 MPa |
| HE1 outlet (T2) | 226.3 °C | 228.2 °C |
| HE1 outlet (P2) | 2.829 MPa | 2.829 MPa |
| ST1 outlet (T3) | 194.6 °C | 195.6 °C |
| ST1 outlet (P3) | 0.325 MPa | 0.325 MPa |
| HE2 outlet (T4) | 160.1 °C | 160.0 °C |
| C1 outlet (T5) | 22.0 °C | 26.1 °C |
| C1 outlet (P5) | 0.286 MPa | 0.285 MPa |
| | | |
| k*A (HE1) | | 1077 W/K |

The simulation result for T2 (cf. Table 2) agrees well with the experimental result. Obviously, the implementation and description of the HE1 in the off design-simulation was well reproduced.
The temperature T2 for the full load (Table 2) was determined backwards via the isentropic efficiency of ST1 and therefore the simulation agreed well.

## 6. CONCLUSIONS

This work presented the initial operation of the CORC setup at the University of Paderborn. The objective was to test the functions of all components and their interactions, with a special focus on the performance of the plate heat exchangers and the coupling of the heat flow between the HT and LT cycle. For the first design calculation a steady state CORC operation was performed and simulated with *EBSILON®Professional* at the University of Applied Sciences in Düsseldorf. It was shown that the simulation was able to accurately describe the operating conditions of the CORC, thus delivering a model which can be used for further off-design calculations and validation of experimental data.

Experimental and simulation results also show that the heat flow that is transferred from the HT cycle to the LT cycle via the heat exchanger HE2 is at present with 10 kW very small compared to the enthalpy offered by the HT cycle after expansion in the turbine. This might be due to the fact that the mass flowrate of the LT cycle was small and therefore HE2 was operated far away from its optimal operating conditions because of a low heat transfer coefficient at low flow velocity.

The next step of our project is to adjust HE2 and to extend the pressure range of the LT cycle to allow for higher flowrates. In addition, it is aimed at identifying a zeotropic mixture as a working fluid for the HT cycle and to adapt turbine TH1 to the according thermodynamic conditions. Before designing a turbine for the LT cycle, the heat flow should be optimized with an adequate supercritical working fluid which fits to the mixture of the HT cycle.

## NOMENCLATURE

| | | |
|---|---|---|
| AI | Analog input | $(-)$ |
| AO | Analog output | $(-)$ |
| BCC | Cooling water cycle for bearings | $(-)$ |
| C | Cycle or condenser | $(-)$ |
| CORC | Cascaded two-stage organic Rankine cycle | $(-)$ |
| DI | Digital input | $(-)$ |
| DO | Digital output | $(-)$ |
| FP | Feed pump | $(-)$ |
| G | Generator | $(-)$ |
| GUI | Graphical user interface | $(-)$ |
| $h_{\mathrm{i}}$ | Specific enthalpy at cycle state *i* | $(\mathrm{kJ/kg})$ |
| HE | Electrical heat exchanger | $(-)$ |
| HC | Heating cycle | $(-)$ |
| HT | High temperature cycle | $(-)$ |
| LT | Low temperature cycle | $(-)$ |
| M | Motor | $(-)$ |
| MM | Hexamethyldisiloxane | $(-)$ |
| ORC | Organic Rankine cycle | $(-)$ |
| PLC | Programmable logic controller | $(-)$ |
| $Q$ | Heat flow | $(\mathrm{kJ})$ |
| $s$ | Specific entropy | $(\mathrm{kJ/(kg\,K)})$ |
| $S$ | Entropy | $(\mathrm{kJ/K})$ |
| $T$ | Temperature | $(°\mathrm{C})$ |
| ST1 | Turbine | $(-)$ |
| TH | Throttle | $(-)$ |
| VFD | Variable-frequency driver | $(-)$ |

## REFERENCES


[1] Angelino G., Colonna P., 1998, Multicomponent Working Fluids For Organic Rankine Cycles (ORCs), Energy, vol. no. 23: p. 449-63

[2] Chen H., Goswami D. Y., Rahman M. M., Stefanakos E. K., 2011, A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power, Energy, vol. no. 36: p. 549-555

[3] DiPippo R., 2004, Second Law assessment of binary plants generating power from low-temperature geothermal fluids, Geothermics, vol. no. 33: p. 565-586

[4] Lai N. A., Fischer J., 2012, Efficiencies of power flash cycles, Energy, vol. no. 44: p. 1-11

[5] Lai N. A., Wendland M., Fischer J., 2010, Working fluids for high-temperature organic Rankine cycles, Energy, vol. no. 39: p. 199-211

[6] Liu B., Rivière P., Coquelet C., Gicquel R., David F., 2012, Investigation of a two stage Rankine cycle for electric power plants, Applied Energy, vol. no. 100: p. 285–294.

[7] Rant Z., 1956, Exergie, ein neues Wort für ‘Technische Arbeitsfähigkeit, Forsch. Ing. Wes., vol. no. 22: p. 36-37

[8] Saleh B., Koglbauer G., Wendland M., Fischer J., 2007, Working fluids for low-temperature organic Rankine cycles, Energy, vol. no. 32: p. 1210-1221

[9] Schuster A., Karellas S., Aumann R., 2010, Efficiency optimization potential in supercritical

Organic Rankine Cycles, Energy, vol. no. 35: p. 1033-1039

[10] FRAGOL, Therminol Productinformation (FRAGOL SCHMIERSTOFF GMBH + CO. KG)

[11] Frenzelit, novaphit SSTC Technical Datasheet (Frenzelit Werke GmbH)

[12] Lemmon E. W., Huber M. L., McLinden M.O., 2013, "NIST Standard Reference Database 23, Fluid Thermodynamic and Transport Properties REFPROP, User's Guide," National Institute of Standards and Technology (NIST)

[13] Steag Energy Services, 2014, "*EBSILON®Professional* Program Documentation"

## ACKNOWLEDGEMENT

We gratefully acknowledge Deutsche Forschungsgemeinschaft (DFG) for funding the experimental setup of the CORC.