EXPERIMENTAL INVESTIGATION OF A LOW PRESSURE STEAM RANKINE CYCLE FOR WASTE HEAT UTILIZATION OF INTERNAL COMBUSTION ENGINES

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ABSTRACT

High heat losses via exhaust gas and coolant in internal combustion engines (ICE) are the basis for numerous investigations regarding downstream processes for power generation. The most promising concepts are Organic Rankine Cycles (ORC) and Steam Rankine Cycles (SRC).
In previous work of the Center of Innovative Energy Systems, Düsseldorf (Germany), the technical and economic feasibility of a low pressure SRC has been investigated and the advantages in comparison to Organic Rankine cycles are highlighted. A distinctive feature of the SRC cycle is the use of the ICE coolant heat for evaporation, which limits the cycle's maximum steam pressure to values below atmospheric pressure. This work presents first results of a test rig with data reconciliation according to DIN 2048 to validate the simulation results and design calculations.
As a basis for the experimental investigations, a gas fired CHP plant is selected. The design of the test rig is optimized for the operation in the laboratory, where the coolant heat is emulated by a tempering device and the exhaust heat of the CHP plant is emulated by a gas burner. With the designed test rig, it is possible to control the volume flow and the temperatures of the coolant and exhaust gas, so that different load conditions of the gas-fired CHP plant can be investigated. For initial tests, the turbine in the SRC is replaced by a throttle to achieve the pressure drop of the turbine.
The experimental results show, that the exhaust and coolant heat of a 38 kW_{el} CHP plant can be emulated and the performance expectations of the cycle can be met in stable steady-state conditions. Based on the measurement results and the turbine design calculations, an electric power output of the cycle of 4.5 kW will be possible, which results in a cycle efficiency of about 7.8 % and an increase of the electrical power output of the CHP plant of about 11.8 %. The results show that the plant concept is technically feasible and, with further optimization, also represents an alternative to ORC plants in terms of increasing the efficiency of a cogeneration plant.

1. INTRODUCTION

Small, compact combined heat and power plants (CHP plants) with less than 1 MW electrical power output, which simultaneously provide electrical energy and useful heat, have become increasingly widespread in recent years (Figure 1). Most CHP plants in this power range are motor-driven CHP plants (ASUE, 2014, 2015). Such cogeneration systems in a performance class up to 500 kW_{el} achieve an electrical efficiency of 38 % on average and an overall efficiency of about 87 % by default (BAFA, 2019). It follows, that about 49 % of the supplied fuel power is available as useful heat. The coolant heat is available at a low temperature level of about 90 °C to 120 °C (motor exit) and the exhaust gas at a high temperature level up to 650 °C. The coolant heat represents 53 % and the exhaust heat 47 % of the maximum usable waste heat. Due to the high exhaust gas temperature, the question arises whether the useful heat can be converted to electrical energy in a downstream process in order to increase the electrical efficiency of the entire system. Therefore, many different recovery systems have already been investigated.

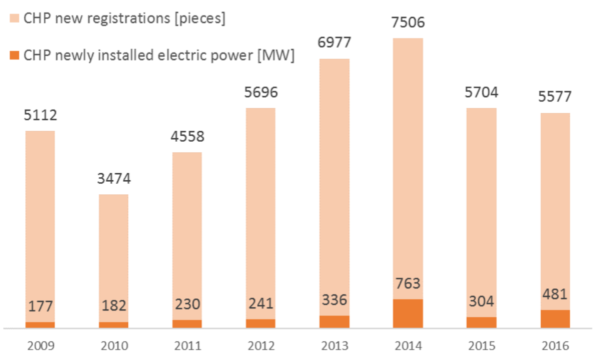

Figure 1: New registrations and newly installed electric power of CHP-plants < 1 MW_{el} from 2009 - 2016 in Germany (BAFA, 2019)

Panesar (2015) investigated different downstream processes for waste heat utilization of combustion engines and differentiates between thermoelectric generators (TEGs) and Rankine Cycles. Due to the low efficiency of the TEGs the most promising process is the Rankine Cycle, which is subdivided in the Clausius Rankine Cycle (CRC) and the Organic Rankine Cycle (ORC). Both system concepts offer different advantages and disadvantages depending on the boundary conditions of the heat source.

In recent years, a large number of investigation focus on the waste heat recovery of ICE in vehicles. Sprouse and Depcik (2013) present a literature review on waste heat recovery of ICE with the conclusion that ORC plants are the most common. However, ORC fluids have some disadvantages in terms of safety, environmental compatibility and temperature resistance. So Sprouse and Depcik (2013) conclude that for exhaust gas temperatures above 370 °C the CRC is advantageous over than the ORC. Ringler *et al.* (2009) arrives at a similar conclusion, but sets the temperature threshold at 300 °C. Overall, the temperature from which the CRC is superior to the ORC cannot be accurately defined. Based on the advantages of the CRC a large number of investigation focused on the waste heat recovery of ICEs. Due to the advantages of the CRC in the high temperature range most publications only consider the exhaust heat as a heat source, see Ringler *et al.* (2009) and Liming *et al.* (2010).

Dolz *et al.* (2012) investigated three different system concepts including a low pressure CRC, a high pressure CRC and a combination of high pressure CRC and ORC. The results show that the combination of the CRC for the waste heat utilization of the exhaust gases and the ORC for waste heat utilization of the coolant heat achieve the highest electrical power output. The low pressure CRC achieves the lowest electrical power output but system modifications like a higher superheating with recuperate or a flash evaporation to increase the evaporation pressure is not taken into account. For this reason the low pressure CRC has to be investigated with different system modifications and has to be compared with the ORC and high pressure CRC.

The authors of this paper published the results of a theoretical analysis of technical and economic feasibility of a low pressure CRC for the waste heat utilization of exhaust and coolant heat of a ICE (Laux *et al.*, 2015). The conclusion is, that a cycle efficiency of 9 % is possible and can by further increased by higher cooling temperatures. However, also with cooling temperatures up to 95 °C the cycle can be competitive to ORC and high pressure CRC due to the use of the complete waste heat of the engine.

All literature cited above is based on theoretical research. So far, few results are published of experimental research for waste heat recovery of ICEs with the CRC. Latz *et al.* (2015) presented the results of a first test for the exhaust gas recovery heat of a ICE with the use of a CRC. Latz uses a piston expander and reaches a cycle efficiency of about 10 %. Due to the fact the piston expander is not designed for the use in the applied boundary condition the efficiency is low.

In order to evaluate the efficiency of a downstream process for CHPs, the percentage electric power increase of the engine and the plant complexity of the Rankine Cycle are of special interest. In order to be able to compare the downstream processes presented in this paper, the exhaust gas and coolant temperatures have to be taken into account. In Table 1 several published ORC concepts for waste heat utilization of ICEs are summarized. The most important boundary conditions of the heat source and the plant concept are specified for each research.

Table 1: Electric power increase of different ORC concepts for ICE waste heat recovery

Theoretical studies based on thermodynamic simulations			
References	**Heat source**	**Electric power increase of ICE [%]**	**Plant concept**
Chatzopoulou and Markides (2018)	Exhaust gas with > 600 °C	15,0	Single loop ORC with recuperator
Wang et al. (2012)	Exhaust gas with > 600 °C	15,0	Dual loop ORC without recuperator
Yang et al. (2013)	Exhaust gas with > 600 °C	10,6	Single loop ORC without recuperator
Experimental results			
References	**Heat source**	**Electric power increase of CHP [%]**	**Plant concept**
Briggs et al. (2010)	Exhaust gas with > 700 °C	4,8	Single loop ORC with recuperator
Uusitalo et al. (2017)	Exhaust gas with < 400 °C	6,1	Single loop ORC with recuperator

The publications in Table 1 which are based on thermodynamic simulations show that by using an ORC, the electric power of a combined heat and power plant can be increased by up to 15%. The results of Chatzopoulou and Markides (2018) can be used as a benchmark for the low pressure SRC, since this research uses an ICE of a CHP with similar exhaust gas temperatures as in this study. The experimental investigation of Briggs *et al.* (2010) and Uusitalo *et al.* (2017) show that there is a significant difference between the experimental and predicted results for the maximum electric power increase of an ICE.
The survey shows that there is no experimental research on low pressure CRCs for the waste heat utilization of ICEs. Therefore, this work aims to validate the calculation methods reported previously, see Laux *et al.* (2015), and to use the validated tools to calculate the performance of the cycle at design and off-design conditions.

2. METHODOLOGY

The general approach of this research is shown in Figure 2. Based on the calculation methods laid out in Laux *et al.* (2015) a test rig for the validation of the calculation methods and performance analysis is designed. In a first step, the test rig is built without the turbine to validate the heat exchanger calculations and to define the design point of the cycle. Therefore, the maximum thermal heat of the heat source is measured. Based on the defined design point and the thermodynamic simulation of the cycle, the steam parameters for the turbine design can be calculated more accurately. Therefore, the measurements are checked for plausibility on the basis of the VDI 2048 standard (Verein Deutscher Ingenieure, 2017). The CRC is simulated with the software Ebsilon®Professional. Based on the validated heat exchanger calculation, characteristic curves for the heat exchangers are calculated and implemented in the thermodynamic simulation. Using an estimated efficiency of the turbine at the design point, the CRC with turbine is simulated. With the results of the simulation, the steam parameters at the inlet of the turbine are calculated, thus setting the turbine design. The turbine design and the calculation of characteristic curves at off-design are used to adjust the thermodynamic simulation. Based on the adjusted thermodynamic simulation, a power prediction of the cycle at design and off-design conditions is carried out.

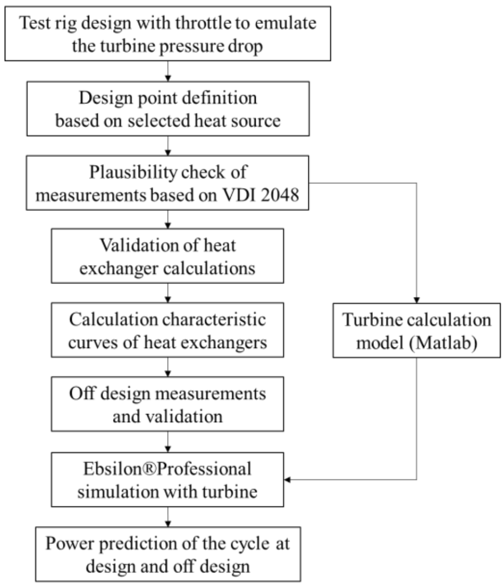

Figure 2: Cycle design approach

2.1 Description of the system concept

The steam cycle presented in this paper uses both the coolant and exhaust heat for evaporation and superheating. The according flow diagram of the system and the corresponding T-s-diagram are shown in Figure 3 and Figure 4.
The booster pump ensures the pressure increase of the water to the required boiling pressure of 0.65 bar. The evaporation takes place in the evaporator, which includes three heat sources, namely the exhaust gas after the superheater, the steam after the turbine and the cooling water. In the superheater, the steam is superheated to a maximum of 540 °C by the exhaust heat and then expanded in the turbine. In the condenser the steam is condensed and extracted as useful heat for domestic heating up to minimum 50 °C.

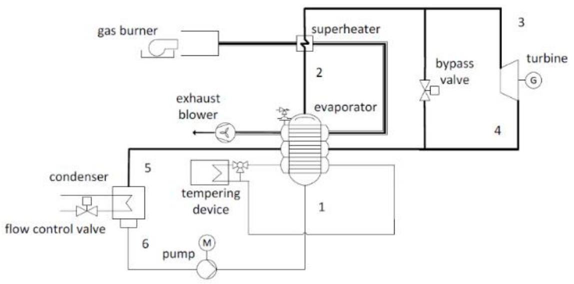

Figure 4: Flow diagram of the SRC plant

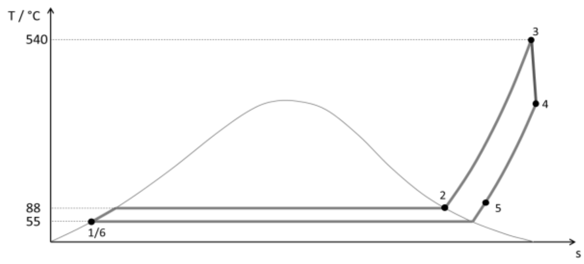

Figure 3: T-s-diagram of the SRC

2.2 Thermodynamic simulation and measurement data reconciliation

The steady-state thermodynamic simulation of the SRC is carried out using Ebsilon®Professional, which is a simulation software for power plant engineering processes. On the basis of preprogrammed blocks, a determined equation system is created, which is solved via an internal algorithm. The iterative calculation procedure utilizes fluid properties from the Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) developed by the National Institute of Standards and Technology (NIST). To calculate the off-design performance of the cycle, an initial design point is defined, where the efficiencies, kA-values and pressure losses of the components are specified. These values are calculated by characteristic curves at the off-design operation points. To calculate the cycle performance of the test rig the mass flow, the temperature and the pressure of the exhaust gas, the coolant and the cooling water of the condenser are required. In addition, the evaporation pressure have to be specified (Figure ***5***).
The toolbox EbsValidate allows to perform a data reconciliation based on the VDI 2048 standard which accounts for the measuring tolerance of all employed measuring devices. With the data reconciliation, gross errors in the measurement data can be detected and the most likely plant condition can be calculated. Therefore, it is necessary to add more measuring point than required in the simulation model to solve the over determined equation system.
In Figure 5, the flow diagram from the simulation model is shown together with the specification values to calculate the data reconciliation of the measurement points. The specification values with the black marker in Figure 5 are used to solve the determined equation system and the specification values with the colored flags are used as additional measurement points for the data reconciliation. Additional input are the heat losses, which are calculated based on the energy balances of the heat exchangers, as well as the efficiency of the booster/coolant pump and the fuel/air ratio of the combustion, which have to be assumed to solve the determined equation system.
It has to be noted that the measurement values obtained from the dynamic pressure probe showed a large fluctuation in steady-state operation, so it is assumed that the specified accuracy is not met. For this reason, the mass flow is additionally calculated by the rotational speed of the gear pump and the energy balance of the condenser. Based on the expected maximum heat losses in the condenser and the accuracies of the temperature, pressure and volume flow measurements the accuracy of the calculated mass flow can be calculated. The accuracy of the mass flow, calculated over the rotational speed, can be estimated over the accuracy of the characteristic curves of the pump from the data sheet.

Due to the current problems with the dynamic pressure probe, a data reconciliation based on VDI 2048 standard is carried out in advance for the steam mass flow in Figure 5. The expected accuracies are shown in Table 2.

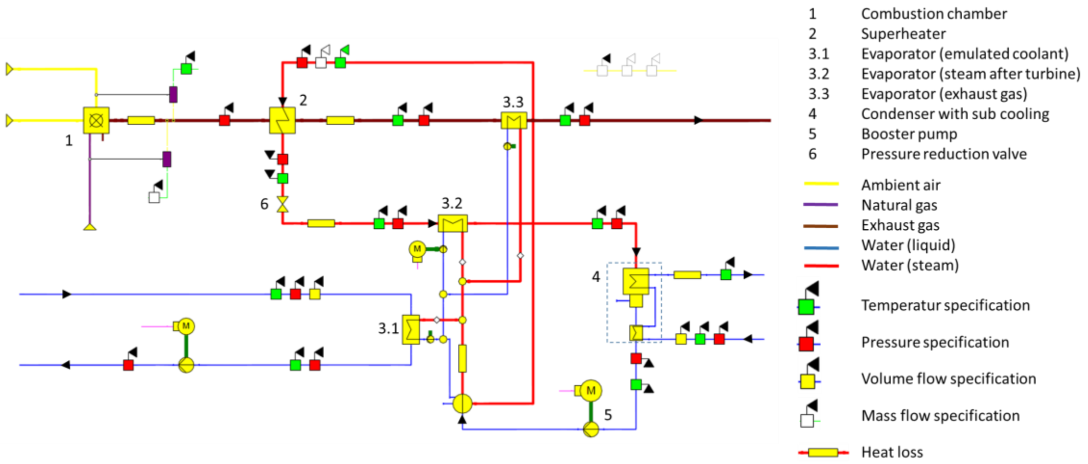

Figure 5: Ebsilon®Professional model for thermodynamic simulation and measuring data reconciliation

In Table 2 the assumptions, the specification values for the determined equation system and the additional specification values for the date reconciliation are summarized.

Table 2: Input parameters for the data reconciliation based on VDI 2048

Measured values for determined equation system	**Assumptions to solve the determined equation system**	**Additional measured values for data reconciliation**
11 temperatures	pump efficiencies = 80 % (constant value)	Mass flow calculated by: dynamic pressure probe (5 % accuracy) rotational speed of the pump (10 % accuracy) condenser heat balance (5 % accuracy)
12 pressures	fuel/air ratio of combustion = 1 (constant value)	Temperature in evaporator
3 mass flows		

Based on the validated simulation model the cycle performance with the turbine can be calculated. Therefore, the throttle in the validated simulation model is replaced by a turbine. To calculate the pressure drop in the turbine and the isentropic efficiency the Ebsilon®Professional simulation model is coupled with Matlab as follows (Figure 6). Ebsilon®Professional simulates the cycle with the expected pressure drop and efficiency of the turbine.

Ebsilon®Professional simulates the cycle with the expected pressure drop and efficiency of the turbine. Matlab takes the steam parameters of the Ebsilon®Professional simulation model and calculates the pressure drop and the isentropic efficiency of the turbine and passes this values to Ebsilon®Professional simulation model. Subsequently, Ebsilon®Professional simulates the SRC with the new values, calculated by Matlab. This iterative process is carried out until the predetermined termination criterion has been reached.#

Additionally, a CHP module is integrated in the simulation, which calculates the mass flow and temperature of the exhaust gas and the transferred heat of the coolant. The characteristic curves are calculated based on the data sheet of the chosen engine. To calculate the cycle performance of the SRC including the turbine and CHP module the electrical power of the CHP Plant, the Temperature of the coolant, the condensation pressure and cooling water Temperature are required.

2.3 Component calculations

The off-design performance of the SRC is mainly influenced by the evaporator, the superheater and the turbine. The evaporator in the test rig is a kettle with three integrated straight tube bundles. The superheater is a tube bundle heat exchanger with straight tubes and four deflectors. The turbine is a radial-inflow turbine. The calculation methods of the heat exchangers and the turbine are described by (Laux *et al.*, 2015). The turbine calculation method is supplemented by a one dimensional loss model. The loss model calculates the losses in stator and rotor separately. The stator loss is calculated according to the correlation of Rodgers (1987). The rotor loss is separated in the friction loss, incidence loss (calculated according to Wasserbauer and Glassman (1975)), trailing edge loss (calculated according to Bammert and Fiedler (1966)), gap leakage loss (calculated according to Deng *et al.* (2018)) and secondary loss (calculated according to Whitfield (1990)). The friction loss on the back surface of the Rotor is calculated according to the publication of Daily and Nece (1960) and is treated like a friction loss in the bearings.

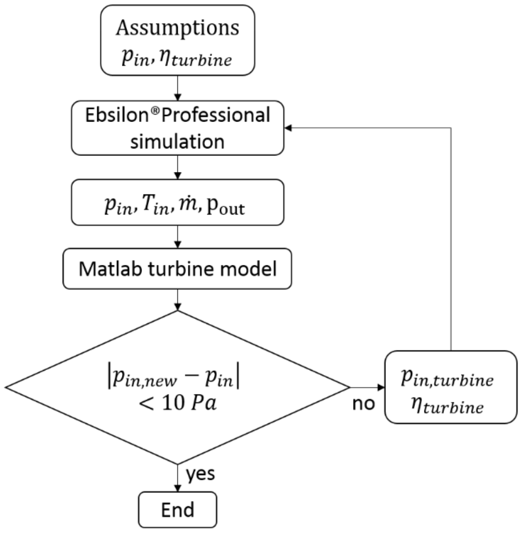

Figure 6*:* Calculation of the evaporation pressure and the turbine efficiency in the simulation

2.4 Design of the test rig

To design the test rig, the boundary conditions of the heat source have to be determined. Depending on the engine type, the exhaust aftertreatment, the boost pressure etc., different boundary conditions occur in the waste heat.

Generally the coolant temperature in ICEs varies in a relatively small temperature range of about 90 - 120 ° C at the exit and 85 - 115 ° C at the inlet of the engine. Overall, a distinction is made between standard cooling (coolant temperature at the exit of the ICE < 100 °C) and hot cooling (coolant temperature at the exit of the ICE > 100 °C). The exhaust temperature varies widely due to the reasons mentioned above. In most cases, the exhaust gas temperature is in a range between 250 – 900 °C. The range is limited by the used catalytic converter, which works best in a narrow temperature range only. The usable thermal power is divided between the cooling water and the exhaust gas heat. According to van Basshuysen and Schäfer (2017), between 42 – 58 % is accounted for by the coolant heat. To define the boundary conditions of the heat source, the engine E0834 E302 of the company MAN is chosen as an example, which is commonly used in different CHP plants like the “g-box 50” of the company G2. According to the data sheet, the mechanical efficiency of the motor amounts to 36.5 % and the thermal efficiency amounts to 52.6 %.

In order to install and operate the test rig in the research laboratory at the University of Applied Sciences in Düsseldorf, the following simplifications are made:

1. Emulation of the heat source (CHP-plant)
    a. Emulation of the coolant heat by a tempering device
    b. Emulation of the exhaust gas by a gas burner
2. Replacement of the turbine by a throttle (first step before adding the turbine)
3. Reducing the temperature of the steam after the superheater to 350 °C due to lower costs in the turbine manufacturing

As shown earlier, see Laux *et al.* (2015), the evaporation temperature of the SRC is set at about 88 °C to reach a cycle efficiency of 9 % competitive to ORCs. The test rig is designed for hot cooling with coolant temperatures up to 120 °C at the inlet of the evaporator. Lower temperatures of a standard cooling would require lower terminal temperature differences in the evaporator and thus larger heat exchanger surfaces, rendering the SRC less economical.

Based on these simplifications, the test rig is developed (Figure 7). Since the exhaust gas temperature of the gas burner is not adjustable, the control of the exhaust gas temperature is achieved by a fresh air inlet at the combustion chamber. An exhaust gas fan is installed in the exhaust gas pipe at the exit of the evaporator to control the volume flow of the exhaust gas and the fresh air inflow. Thus the temperature of the mixture of exhaust gas and fresh air can be controlled by the rotational speed of the exhaust gas van. The exhaust gas power can be controlled by the gas burner.
The tempering device emulates the coolant heat of the engine. As the tempering device provides a constant volume flow with a defined temperature, it is necessary to control the volume flow through the evaporator by means of a controlled three-way valve.
To obtain a stationary operation of the SRC, it is necessary to control the evaporation pressure and the condensation pressure. The pressure reduction valve is used to control the evaporation pressure. The

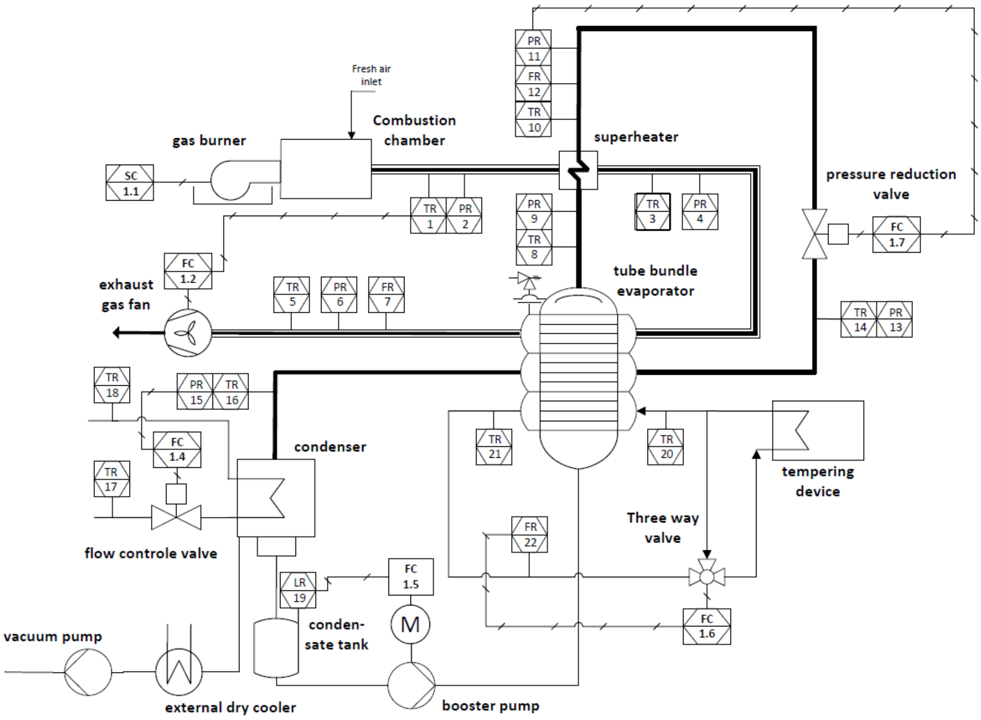

Figure 7: Flow diagram of the test rig with measuring devices

condensation pressure is controlled via the transferred heat inside the condenser. Therefore, the volume flow of the condenser coolant is controlled. If the volume flow is increased, the average temperature of the coolant inside the condenser is decreased. As a result, the temperature difference between coolant and steam increases and thus the transferred heat inside the condenser increases.
The booster pump has to control the fill level of the condensate inside the condensate canister. Therefore, the rotational speed of the booster pump is controlled by the signal of the fill level sensor.
To counteract possible air ingress, a vacuum pump is placed at the coolest point of the condenser. An external dry cooler is installed so that as little water as possible condenses inside the vacuum pump.
The recording of the measurement and the control of the SRC is implemented using Matlab Simulink. The temperature measurement is carried out using PT100 sensors and thermocouples for temperatures below and above 600 °C respectively. For the volume flow measurement a dynamic pressure probe is installed (steam and exhaust gas) as well as a magnetic-inductive flow meter (water). The precision of the measuring instruments is summarized in Table 3 In addition to the measurement error caused by the measuring sensors the measuring error caused by the 0-10V converters have to be taken into account. The measuring error caused by the measuring card can be neglected.

Table 3: Precisions of the installed measuring instruments

Measuring device	**precision**
Measuring card	0.00031 % FSO
0-10 V converter	0.1 % FSO
Pressure sensors	0.1 % FSO
PT100 (1/3 DIN)	± (0,1+0,0017 |t|)
Thermocouples (Typ K)	± 1,5°C
Dynamic pressure probe	1.0 % FSO
Magnetic-inductive flow meter	0.5 %

3. RESULTS

Initial performance tests of the SRC are based on emulation of the MAN engine type E0832 E302 as the heat source. According to the data sheet, the engine can be operated down to 50 % partial load. To validate the simulation results and calculation methods, the engine is scaled down to 38 kW electrical power at full load. The boundary conditions of the exhaust gas and coolant of the engine are calculated with Ebsilon®Professional at six load conditions from 50 % to 100 % (Table 4). According to Figure 7, the exhaust gas temperature (measuring point 1), the mass flow (measuring point 7), the coolant temperature inlet (measuring point 20) and the evaporating pressure (measuring point 9) are given.

Table 4: Operating points of validation tests (38 kWel CHP engine)

	Exhaust Gas				Coolant			SRC	
CHP electric power [%]	T [°C]	m_{pkt} [g/s]	$\dot{Q}$ [kW]	$\frac{P}{P_{design}}$	T_{IN} [°C]	$\dot{Q}$ [kW]	$\frac{P}{P_{design}}$	$p_{evap.}$ [bar]	$T_{evap.}$ [°C]
100	649,1	37,6	28,1	1,00	107,0	32,6	1,00	0,675	86,8
90	628,2	34,5	24,9	0,89	107,0	31,0	0,95	0,636	85,1
80	609,2	31,7	22,2	0,79	107,0	29,6	0,91	0,604	83,8
70	592,4	29,0	19,6	0,70	107,0	28,3	0,87	0,573	82,6
60	578,3	26,3	17,4	0,62	107,0	26,8	0,82	0,544	81,4
50	567,0	23,4	15,1	0,54	107,0	24,9	0,76	0,499	80,3

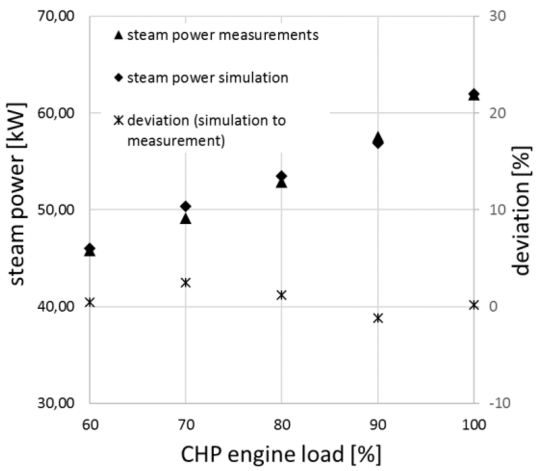

Figure 8: Validation of the simulation results

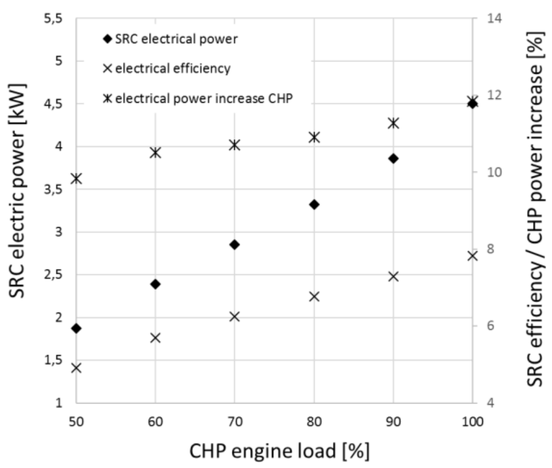

Figure 9: Electric power prediction of the SRC (calculated by the simulation model, extended by the turbine model)

The SRC, including CHP and turbine, is simulated with the Ebsilon®Professional simulation model (according to Figure 6). Since the turbine has already been designed, the dimensions of the turbine and the calculated rotational speed are the basis for the turbine simulation model. The turbine has no variable stator geometry, so the test rig will have to operate in sliding pressure mode. Based on the simulation model of the SRC and the assumption that the condenser pressure should be constant at 0.21 bar_{abs}, the evaporation pressure is calculated at each operating point. These operating points in Table 4 are approached experimentally with the test rig (without turbine). Based on the measurement results of these stationary operation points, the simulation results of the simulation model according to Figure 5 are validated.The results of the validation are shown in Figure 8. As a comparative value, the steam power after the supereater is used. The deviation between the simulation results and the measuremets are between -1,2 % to 2,5 %. The largest deviation in the simulation is attributed to the coolant heat exchanger. The heat transfer coefficient in the evaporator depends on the temperature difference between the primary and secondary side, especially in the area of small temperature differences.

Based on the validated simulation model of the test rig, the expected performance of the rig including turbine is now calculated as described in chapter 2.4 (according to Figure 6). The results of this simualtion are shown in Figure 9. It is shown that the low pressure SRC achieves an electric power output of about 4.5 kW at the design point. This results in an electric cycle efficiency of about 7.8 % and an increase of the total electrical output of the CHP plant of about 11.8 %. In off-design conditions the electric efficiency decreases from 7.8 % to 4.9 % and the CHP electrical power increase, caused by the SRC, decreases from 11.8 % to 9.8 %.

As shown in Figure 10, the calculated isentropic turbine efficiency achieves values over 80 % in all engine load conditions and a maximum isentropic efficiency of 87 % at 90 % engine load.

In comparison with the published ORC concepts of mentioned in chapter 1, the electric power increase of the low pressure SRC is significantly higher than the published experimental results of Briggs et al. (2010) and Uusitalo et al. (2017) but lower than the comparable ORC concept of Chatzopoulou and Markides (2018). Due to the fact, that the research of Chatzopoulou and Markides (2018) is based on simulation results only, the proposed low pressure SRC is still in a competitive range compared to ORC concepts.

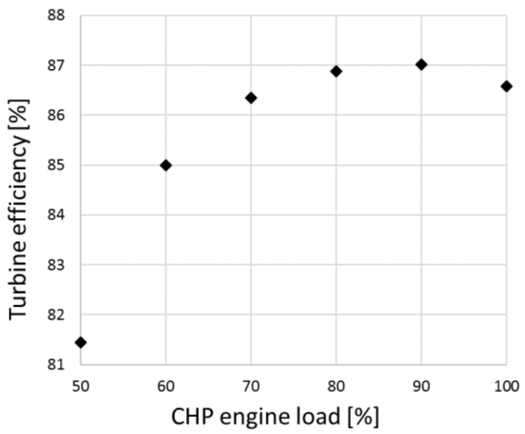

Figure 10: Calculated isentropic turbine efficiency (total to total) of the turbine (simulation results)

4. CONCLUSIONS

The validation of the Ebsilon®Professional simulation model has shown that it can predict the cycle performance of the proposed low pressure SRC test rig without the turbine with an accuracy of about $\pm$ 3 %.

The validated Ebsilon®Professional simulation model and the turbine model in Matlab are used to simulate the low pressure SRC with the turbine in design and off-design conditions. The results of this simulation show that it is possible to achieve an electrical efficiency of the SRC of about 7.8 % in design condition and that the electrical power output of the CHP plant can be increased by 11.8 % with the low pressure SRC. Since the heat of the exhaust gas and the coolant is used in the low pressure SRC, the cycle efficiency is competitive compared to high pressure SRCs or high temperature ORC for exhaust gas utilization with an electrical efficiency of about 18 %.

Next steps will be aimed at improving the cycle simulation by coupling the Matlab calculation of the coolant heat exchanger with the Ebsilon®Professional simulation and installation of the turbine, which is currently beeing installed.

NOMENCLATURE

CHP	Combined heat and power	ORC	Organic Rankine Cycle
CRC	Clausius Rankine Cycle	SRC	Steam Rankine Cycle
ICE	Internal combustion engine	TEG	Thermoelectric generator

REFERENCES

ASUE, 2014, BHKW-Kenndaten 2014/2015 - Module Anbieter Kosten, Berlin.

ASUE, 2015, Gasturbinen-Kenndaten und -Referenzen.

BAFA, 2019, Zulassung von KWK-Anlagen nach dem Kraft-Wärme-Kopplungsgesetz (KWKG).

Bammert, K. and Fiedler, K., 1966, Hinterkanten- und Reibungsverlust in Turbinenschaufelgittern, Forschung im Ingenieurwesen, Vol. 32 No. 5, p. 133–141.

Briggs, T.E., Wagner, R., Edwards, K.D., Curran, S. and Nafziger, E., 2010, A Waste Heat Recovery System for Light Duty Diesel Engines, SAE International.
Chatzopoulou, M.A. and Markides, C.N., 2018, Thermodynamic optimisation of a high-electrical efficiency integrated internal combustion engine – Organic Rankine cycle combined heat and power system, Applied Energy, Vol. 226, p. 1229–1251.
Daily, J.W. and Nece, R.E., 1960, Chamber Dimension Effects on Induced Flow and Frictional Resistance of Enclosed Rotating Disks, J. Basic Eng., Vol. 82 No. 1, 217-230.
Deng, Q., Shao, S., Fu, L., Luan, H. and Feng, Z., 2018, An Integrated Design and Optimization Approach for Radial Inflow Turbines—Part II: Multidisciplinary Optimization Design, Applied Sciences, Vol. 8 No. 11.
Dolz, V., Novella, R., García, A. and Sánchez, J., 2012, HD Diesel engine equipped with a bottoming Rankine cycle as a waste heat recovery system. Part 1: Study and analysis of the waste heat energy, Applied Thermal Engineering, Vol. 36, p. 269–278.
Latz, G., Erlandsson, O., Skara, T., Content, A., Andersson, S. and Munch, K., 2015, Water-based rankine-cycle waste heat recovery systems for engines: challenges and opportunities, 3rd International Seminar on ORC Power Systems.
Laux, C., Neef, M., Gotter, A. and Goebel, J., 2015, Investigation in the technical and economic feasibility of a low-pressure-steam cycle for the waste heat utilization of a cogeneration plant, Vol. 2015.
Liming, F., Wenzhi, G., Hao, Q. and Bixian, X., 2010, Heat Recovery from Internal Combustion Engine with Rankine Cycle, Power and Energy Engineering Conference.
Panesar, A., 2015, Waste heat recovery using fluid bottoming cycles for heavy duty diesel engines, PhD Thesis, University of Brighton, Brighton, 2015.
Ringler, J., Seifert, M., Guyotot, V. and Hübner, W., 2009, Rankine Cycle for Waste Heat Recovery of IC Engines, SAE Int. J. Engines Vol. 2, p. 67–76.
Rodgers, C., 1987, Mainline performance prediction for radial inflow turbines: VKI lecture series.
Sprouse, C. and Depcik, C., 2013, Review of organic Rankine cycles for internal combustion engine exhaust waste heat recovery, Applied Thermal Engineering, Vol. 51, p. 711–722.
Uusitalo, A., Honkatukia, J. and Turunen-Saaresti, T., 2017, Evaluation of a small-scale waste heat recovery organic Rankine cycle, Applied Energy, Vol. 192, p. 146–158.
van Basshuysen, R. and Schäfer, F., 2017, Handbuch Verbrennungsmotor, Vieweg+Teubner Verlag, Wiesbaden.
Verein Deutscher Ingenieure, 2017, Control and quality improvement of process data and their uncertainties by means of correction calculation for operation and acceptance tests - Corrigendum concerning standard: VDI 2048.
Wang, E.H., Zhang, H.G., Zhao, Y., Fan, B.Y., Wu, Y.T. and Mu, Q.H., 2012, Performance analysis of a novel system combining a dual loop organic Rankine cycle (ORC) with a gasoline engine, Energy, Vol. 43, p. 385–395.
Wasserbauer, C.A. and Glassman, A.J., 1975, Fortran program for predicting off-design performance of radial-inflow turbines, Washington, D.C.
Whitfield, A., 1990, The Preliminary Design of Radial Inflow Turbines, J. Turbomach., Vol. 112 No. 1, p. 50–57.
Yang, K., Zhang, H., Wang, Z., Zhang, J., Yang, F., Wang, E. and Yao, B., 2013, Study of zeotropic mixtures of ORC (organic Rankine cycle) under engine various operating conditions, Energy, Vol. 58, p. 494–510.

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