Transient modelling and simulation of a double-stage Organic Rankine Cycle

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Abstract

Geothermal energy is a renewable resource for power and heat production. For low enthalpy reservoirs the geothermal energy is usually converted to electricity by an Organic Rankine Cycle (ORC). The efficiency and profitability of these power plants can be increased by combined heat and power production. In this study, a dynamic model of a double-stage ORC power plant is developed to investigate and evaluate geothermal combined heat and power plant concepts. The simulation model is validated by operational data of a real geothermal power plant in the South of Germany. For the validation, the relative root mean squared error (RRMSE) is used. In addition, the coefficient of correlation is calculated to evaluate the dynamic behavior. The results show that the electrical power output of the power plant can be predicted by an RRMSE of 3.9 %. The coefficient of correlation is 0.99 and shows that the model is capable to predict the dynamic behavior of the power plant.

Keywords: transient simulation, Organic Rankine Cycle, geothermal heat and power production

1 Introduction

Geothermal energy is a renewable resource for low carbon heat and power production. In binary systems, the thermal power of the brine is usually converted to electricity by Organic Rankine Cycles (ORC). A previous study (Heberle et al., 2016) shows that the efficiency and profitability of these power plants can be increased by an additional heat supply. Because of the fluctuating heat demand, the power plant is driven more often in part load conditions. For that reason, a dynamic model of a double-stage ORC power plant is developed to evaluate different power plant concepts for combined geothermal heat and power production.

In the literature, several dynamic ORC models are presented for waste heat recovery (WHR) from engines. Huster et al. (2018) modelled a one-stage ORC for WHR in a diesel truck. For the simulation, the software gPROMS is used and the model is validated against measurement data. The results show that the initialization process of the model is a challenging task and that the dynamics in the heat exchangers are dominated by the pressure level. Jiaxin Ni et al. (2017) developed also an ORC for waste heat recovery (WHR) from diesel engines using the software Dymola. The study shows that the dynamics of the system can be damped by the integration of an intermediate oil cycle. Bin Xu et al. (2017) developed a model for WHR from diesel engines in MATLAB/Simulink and showed that the vapor temperature and the evaporation pressure can be predicted with 2 % and 3 %, respectively.

Next to WHR dynamic ORC models are also developed for solar and geothermal applications. Baccioli et al. (2017) built up a dynamic model for an ORC with compound parabolic solar collectors and developed a control strategy to drive the system without a thermal energy storage. Proctor et al. (2016) developed a one-stage ORC simulation model for geothermal power production and validated the model against measurement data with a standard deviation of 1.4 %. The model will be used to test potential improvements to the control system of the power plant. To sum up, so far only small scale dynamic ORC models are developed.

In this study, a dynamic model of a double-stage ORC is developed with the software Dymola to investigate potential plant concepts for geothermal combined heat and power production. The simulation model is described and in chapter 3 the results of the validation are presented.

2 Methodology

In this section the double-stage ORC concept is introduced and the dynamic modelling of the cycle components is presented. For modelling and simulation the software Dymola (Dassault Systèmes, 1992-2004) is combined with the Modelica based library ThermoCycle (Quoilin et al., 2014). For the calculation of the fluid properties, the software CoolProp (Bell et al., 2014) is used.

2.1 Double-stage ORC

In this study, a double-stage ORC is considered based on a real operating power plant in the German Molasse

Basin. A scheme of the power plant and its components is shown in Figure 1.

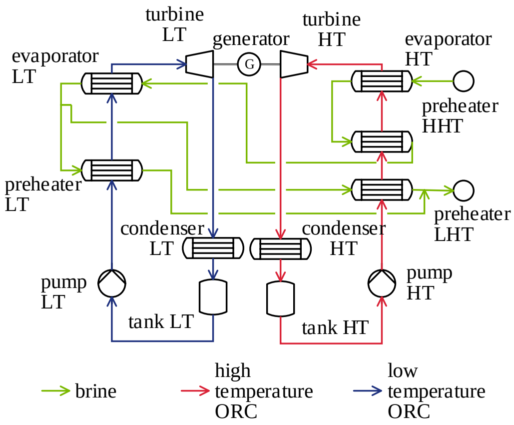

Figure 1. Scheme of the considered double-stage Organic Rankine Cycle.

The double-stage ORC consists of two modules, a high temperature (HT) and a low temperature (LT) ORC. In both modules, R245fa is used as working fluid. Regarding the cycle components, each ORC contains a pump, at least one preheater, an evaporator, a turbine and a condenser. The thermal water enters the HT ORC with a temperature of 138 °C und and a mass flow rate of 120 kg/s. Firstly, heat is supplied to the HT-ORC and then to the LT-ORC. A detailed *T*-$\dot{H}$-diagram of the power plant is shown in Figure 2. Table 1 summarizes some characteristic data for the heat exchangers and the rotating equipment at the design point.

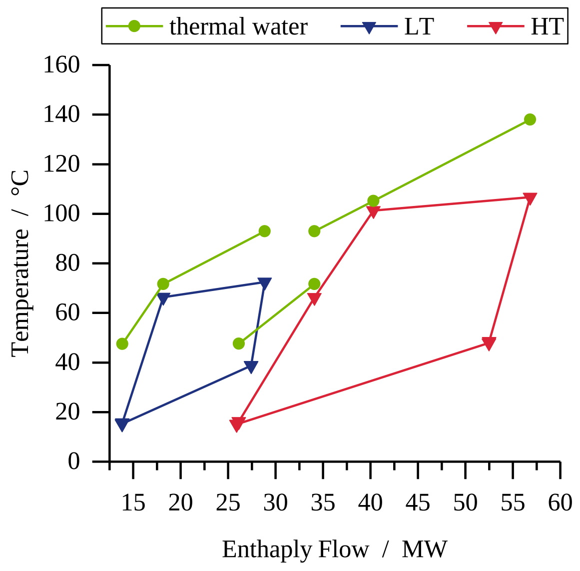

Figure 2. *T*-$\dot{H}$-diagram of the considered double-stage Organic Rankine Cycle.

Table 1. Characteristic data of the considered power plant. (Heberle et al., 2015)

parameter	value
rotating equipment	
LT pump isentropic efficiency	78.4 %
HT pump isentropic efficiency	76.7 %
HT turbine isentropic efficiency	82.7 %
HT turbine isentropic efficiency	88.3 %
generator efficiency	98.0 %
heat exchanger areas	
LT-preheater	201.0 m²
LT-evaporator	741.4 m²
LHT-preheater	270.4 m²
HHT-preheater	277.6 m²
HT-evaporator	741.4 m²
LT-condenser	7512.0 m²
HT-condenser	3756.0 m²

2.2 Modelling

In principle, two types of components have to be modelled to simulate a double-stage ORC system: turbomachines (pumps, turbines) and heat exchangers (preheaters, evaporators, condensers).

Turbomachines

Several previous investigations of dynamic ORC models show that the time constants in the turbomachines are relatively low compared to those of the heat exchangers. Therefore, the pump and the turbine can be modelled as quasi-stationary components (Quoilin, 2011; van Putten and Colonna, 2007; Wei et al., 2008).

For the pump the exhaust enthalpy is calculated according to the following equation:

$$h_{out} = h_{in} + \frac{p_{out} - p_{in}}{\eta_s \rho_{in}} \qquad (1)$$

The isentropic efficiency of the pump η_s depends on the pumped volume flow rate. For the simulation, the characteristic curve of the manufacturer data sheet is implemented for the LT- and the HT-ORC pump, respectively. The normalized curves are shown in Figure 3.

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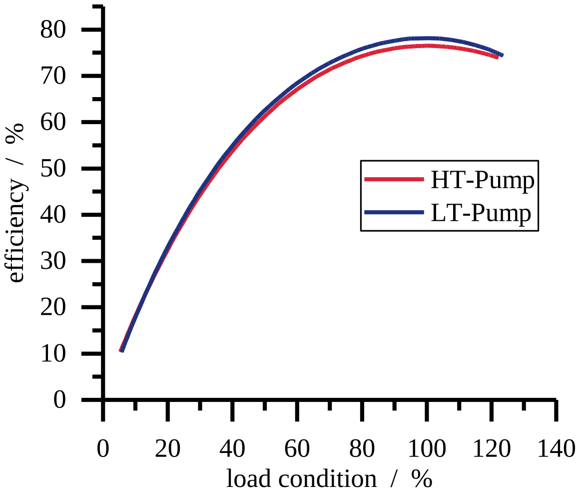

Figure 3. Characteristic curve of the pump isentropic efficiency for the HT- and LT-ORC.

The turbine is modelled based on Stodola's law:

$$\dot{m}_{in} = K\sqrt{\rho_{in}p_{in}\left[1-\left(\frac{p_{out}}{p_{in}}\right)^2\right]} \tag{2}$$

The coefficient K is calculated for the HT and the LT ORC by the respective nominal turbine inlet and outlet conditions.
The isentropic efficiency of the turbine depends on the volume flow rate at the turbine outlet and on the enthalpy difference utilized in the turbine (Milora and Tester, 1977). Both parameters are influenced by part load operation. In addition, in the considered power plant air-cooled condensers are used. Therefore, the condensing pressure varies during the day according to the ambient temperature. This affects the turbine outlet pressure and thereby the utilized enthalpy difference. For that reason, a quasi-stationary model of the isentropic efficiency is implemented in the turbine based on the approach of Ghasemi et al. (2014):

$$\eta_s = \eta_{s,nom} r_h r_v \tag{3}$$

The nominal isentropic efficiency $\eta_{s,nom}$ is corrected by two factors: r_h takes into account the off-design enthalpy difference and is calculated by

$$r_h = \left\{\left[\left(1.398 r_T - 5.425\right) r_T + 6.274\right] r_T - 1.866\right\} r_T + 0.619 \tag{4}$$

with

$$r_T = \sqrt{\frac{\left(h_{turbine,in} - h_{turbine,out}\right)}{\left(h_{turbine,in} - h_{turbine,out}\right)_{nom}}}\,. \tag{5}$$

r_v takes into account the off-design volume flow rate at the turbine outlet and is given by

$$r_v = \left\{\left[\left(-0.21 r_{VT} - 1.117\right) r_{VT} - 2.533\right] r_{VT} - 2.588\right\} r_{VT} + 0.038 \tag{6}$$

with

$$r_{VT} = \frac{\dot{V}}{\dot{V}_{nom}}\,. \tag{7}$$

Heat exchangers

The dynamic models for the heat exchangers are built up in three steps according to Quoilin et al. (2011):

1. At first, a stationary model is built up with detailed correlations for the heat transfer and the pressure drop depending on the geometry.
2. In the next step, the stationary model is simulated at the design point and nominal values for the heat transfer coefficient α_{nom} and the pressure drop Δp_{nom} are calculated.
3. In the third step, the detailed heat and pressure drop correlations are simplified and a dynamic heat transfer coefficient and pressure drop is calculated based on the nominal values. Exemplarily, the dynamic heat transfer coefficient α of the working fluid in the preheaters can be calculated according to equation (8):

$$\alpha = \alpha_{nom}\left(\frac{\dot{m}}{\dot{m}_{nom}}\right)^n \tag{8}$$

The preheaters are shell and tube heat exchangers with double-segmental baffles and two passes for the hot and the cold side, respectively. The thermal water flows in the tubes and the working fluid in the shell. For modelling of heat exchangers, the most common approaches are the finite volume approach and the moving boundary approach (Jensen, 2003). For the simulation of the preheaters, the finite volume approach is used.

The heat transfer coefficient of the thermal water in the tubes is calculated by Gnielinski (2013). For the working fluid on the shell side an adapted version of the correlation of Bell-Delaware is used (Milcheva et al., 2017). For the pressure drop on the hot side a correlation of Kast and Nirschl (2013) is implemented and for the working fluid the pressure drop is calculated by Taborek et al. (Hewitt, 2008).

The evaporators in the considered power plant are designed as kettle boilers. The working fluid on the shell side is heated by the thermal water in the tube bundle. On the tube side there are four passes.

Since the dynamics on the shell side can not be covered accurately by the finite volume or moving boundary approach, a two-volume model for the evaporator is developed in ThermoCycle according to Pili et al. (2017). For the tube side the finite volume approach is used.

The heat transfer coefficient for pool boiling on tube bundles is calculated by Macchi and Astolfi (2017)

$$\alpha_{bundle} = \alpha_{nb,1} F_b + \alpha_{nc} \ . \tag{9}$$

F_b takes into account the effect of convective boiling and is calculated by Taborek et al. (Hewitt, 2008):

$$F_b = 1.0 + 0.1\left[\frac{0.785 D_b}{0.866(p_t / D_o)^2 D_o} - 1.0\right]^{0.75} \tag{10}$$

According to Welzl et al. (2018) for the pool boiling heat transfer coefficient $\alpha_{nb,1}$ of R245fa the correlation of Cooper (1984) is used. The convective boiling heat transfer coefficient α_{nc} is assumed to be 250 W/m²K according to Macchi and Astolfi (2017). For the heat transfer in the vapor region the Bromley equation (Bromley, 1950) is implemented. The pressure drop is calculated by the static pressure drop since the pressure drop in kettle boilers is dominated by the static part (Thome, 2004). For the thermal water, the same correlations for the heat transfer coefficient and pressure drop are used as for the preheaters.

As mentioned above, in the considered power plant air-cooled condensers are implemented. The working fluid flows in tube bundles with two passes and the air in cross-flow over the tube bundles. On the working fluid side the heat transfer for one-phase regions (liquid and vapor) is calculated by Gnielinski (2013). For the two-phase region Cavallini et al. (2006) is used. For the pressure drop Kast and Nirschl (2013) is implemented for one phase regions and for the two-phase region Friedel is used. The heat transfer coefficient on the shell side is calculated by Haaf et al. (Steimle and Plank, 1988).

2.3 Validation parameters

The developed dynamic models are validated against operational data of a real geothermal heat plant. The validation takes place in two steps.
At first, the relative root mean squared error (RRMSE) (Despotovic et al., 2016) is calculated as an indicator for the quantitative quality of the simulation results by

$$RRMSE = \frac{\sqrt{\frac{1}{N}\sum_{i=1}^{N}(R_i)^2}}{\overline{x}} \ , \tag{11}$$

where N is the number of intervals and R the difference between the measured value x and the simulated value y.

The RRMSE, however, is not able to take into account the dynamic behavior of the model. Therefore, in a second step the coefficient of correlation ρ is calculated:

$$\rho = \frac{\sum_{i=1}^{N}(x_i - \overline{x})(y_i - \overline{y})}{\sqrt{\sum_{i=1}^{N}(x_i - \overline{x})^2 \sum_{i=1}^{N}(y_i - \overline{y})^2}} \tag{12}$$

The coefficient of correlation is used for time series analysis and can range from -1 to +1. A value of +1 means that both time histories are identical in shape. (Sarin et al., 2010) Therefore, the coefficient of correlation evaluates the dynamic behavior of the model compared to the operational data.

3 Results

For the validation of the double-stage ORC a period of 24 hours in steps of one minute is simulated. The validation results are presented in Table 2.

Table 2. Validation results of the cycle components.

parameter	**RRMSE [%]**
HT-ORC	
pump outlet pressure	3.5
pump outlet volume flow rate	17.0
turbine inlet pressure	4.8
turbine outlet pressure	3.6
HHT preheater inlet temperature	0.4
HHT preheater outlet temperature	0.9
HHT preheater temperature difference	6.2
evaporator outlet temperature	0.4
tank temperature	1.8
LT-ORC	
pump outlet pressure	4.2
pump outlet volume flow rate	5.3
turbine inlet pressure	1.4
turbine outlet pressure	4.7
LT preheater outlet temperature	0.5
evaporator outlet temperature	0.1
tank temperature	1.8
Thermal water	
HT evaporator temperature difference	11.0
HHT preheater inlet temperature	0.9
HHT preheater outlet temperature	0.7
HHT preheater temperature difference	5.1
LT evaporator inlet temperature	0.7
LT evaporator outlet temperature	0.5
LT evaporator temperature difference	5.0
LHT preheater inlet temperature	0.5
LHT preheater outlet temperature	1.2
LHT preheater temperature difference	12.8
LT preheater outlet temperature	0.8
LT preheater temperature difference	6.3
injection temperature	1.1

The volume flow rate and the temperature of the geothermal fluid, the electrical power consumption of the fans and the ambient temperature are used as inputs for the simulation.

The RRMSE is on average 3.6 %. For the pressure and the temperatures, the RRMSE is lower than 5 %. Except for the temperature differences and the volume flow rates in LT- and HT-ORC module the RRMSE is higher than 5 %. According to the manufacturer for the volume flow rates in the ORC-modules the uncertainties of the integrated flow rate sensors are responsible for the deviations (Heberle et al., 2015). Regarding the temperature differences, the reason for the deviation is the uncertainty in the measurement of the volume flow rate of the geothermal fluid.

In Figure 4 the results for the evaporating pressure of the HT-cycle are presented. The deviation between simulation and operational data is quantified by an RRMSE of 4.8 %. Regarding the coefficient of correlation is 0.97 and shows that the simulation model can reproduce the dynamic behavior.

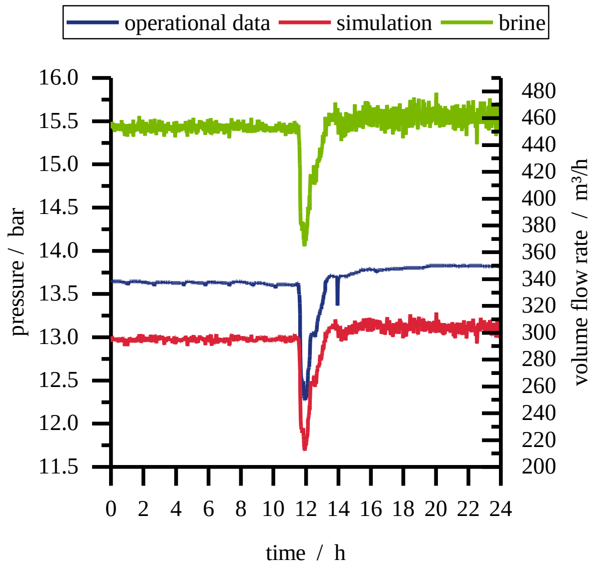

Figure 4. Validation results for the evaporating pressure of the HT-ORC.

Figure 5 shows the validation results for the reinjection temperature of the geothermal fluid, which is an indicator for the heat supplied to the ORC. The RRMSE is 1.1 %. In addition, the dynamic behavior is evaluated by a coefficient of correlation of 0.99 and shows that the simulation and the operational data are almost identical in shape.

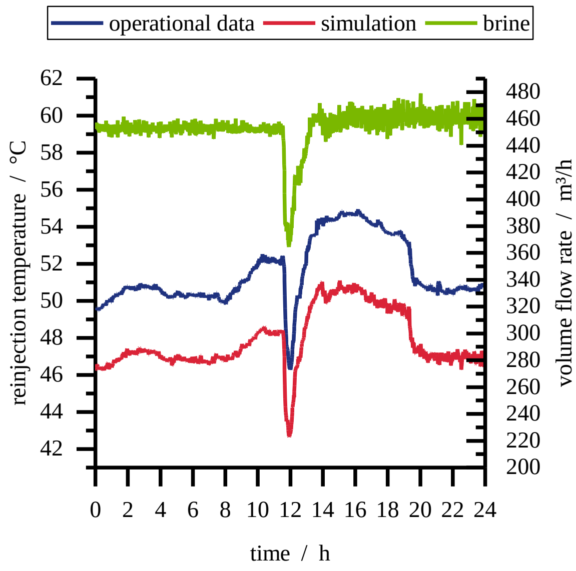

Figure 5. Validation results for the reinjection temperature of the geothermal fluid.

For the evaluation of the whole system, the electrical power output of the generator is used as the validation parameter. The results are shown in Figure 6. The RRMSE is 3.9 % and the coefficient of correlation is 0.99. Therefore, the simulation model can predict the dynamic behavior.

As mentioned, the measurement of the volume flow rate of the thermal water is connected to high uncertainties. For that reason, the simulated electrical power output of the generator is lower than the real power output even though no heat losses to the ambient are considered in the simulation model.

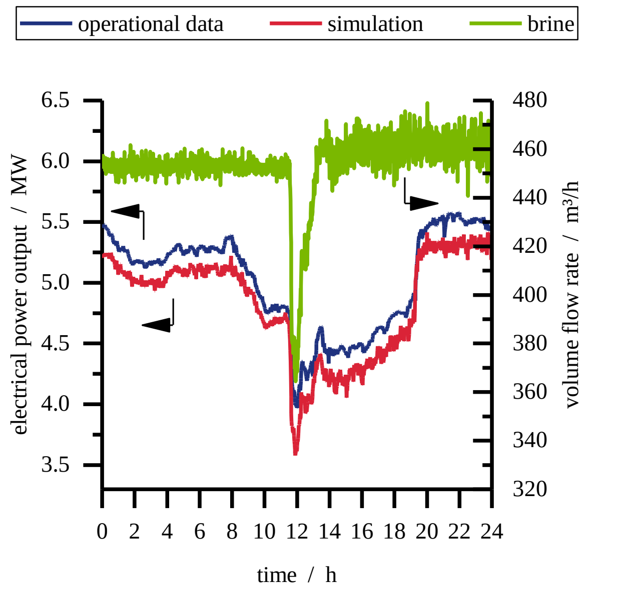

Figure 6. Validation results of the double-stage Organic Rankine Cycle.

The dynamic model is developed to investigate different concepts for geothermal heat and power production. Geothermal reservoirs usually provide a fixed volume flow rate and temperature. Therefore, the parameters of the thermal water can be defined in the model so that the results are not affected by measurement uncertainties of the thermal water volume flow rate. For that reason, the deviation regarding the generator output is acceptable. Due to the fluctuating heat demand of district heating networks, the power plant is driven more often in part load conditions. Therefore, it is necessary to predict the dynamic behavior of the power plant accurately. As the results show, the dynamics can be reproduced by the developed simulation model.

4 Conclusion

In this study, a transient simulation model of a double-stage ORC is developed and validated by operational data of a real power plant in the German Molasse Basin. The model is built up in Dymola based on the ThermoCycle library.

The results show a RRMSE of 3.6 % on average The pressure and temperatures can be predicted by an RRMSE lower than 5 %. For the whole double-stage ORC power plant the electrical output of the generator can be predicted by 3.9 %. The coefficient of correlation is 0.99 and shows that the simulation model can reproduce the dynamics of the real power plant.

In future work, based on the dynamic simulation model different geothermal combined heat and power plant concepts are investigated and evaluated by annual return simulations. For the district heating network heat demand profiles based on real heat plant data will be implemented. In addition, different peak loads as well as supply and return temperatures are investigated.

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Nomenclature

Symbols

D	*diameter*
h	specific enthalpy
m	mass flow rate
N	number of intervals
p	pressure
p_t	tube pitch
R	residual
x	measured value
y	simulated value

Greek symbols

α	heat transfer coefficient
η	efficiency
ρ	density

Subscripts

b	bundle
in	inlet
nom	nominal
o	outer
out	outlet
s	isentropic

Superscript

$\bar{\ }$	mean value

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