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# Analysis of hot spots in boilers of organic Rankine cycle units during transient operation

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## Abstract

This paper is devoted to the investigation of critical dynamic events causing thermochemical decomposition of the working fluid in organic Rankine cycle power systems. The case study is the plant of an oil and gas platform where one of the three gas turbines is combined with an organic Rankine cycle unit to increase the overall energy conversion efficiency.

The dynamic model of the plant is coupled with a one-dimensional model of the once-through boiler fed by the exhaust thermal power of the gas turbine. The heat exchanger model uses a distributed cross-flow physical topology and local correlations for single- and two-phase heat transfer coefficients.

The results indicate that severe load changes ( $0.4\text{--}1.0\text{ MW} \cdot \text{s}^{-1}$ ) can lead to exceedance of the temperature limit of fluid decomposition for a period of 10 min. Ramp rates lower than  $0.3\text{ MW} \cdot \text{s}^{-1}$  are acceptable considering the stability of the electric grid and fluid decomposition. It is demonstrated that the use of a spray attenuator can mitigate the problems of local overheating of the organic compound.

As a practical consequence, this paper provides guidelines for safe and reliable operation of organic Rankine cycle power modules on offshore installations.

**Keywords:** Hot spot, Organic Rankine cycles, Fluid thermochemical decomposition, Waste heat recovery unit.

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## Nomenclature

$A$	area [m <sup>2</sup> ]	$HPC$	high pressure compressor
$C_o$	dimensionless parameter eq. A.3	$HPT$	high pressure turbine
$D$	outer tube diameter [m]	$HRS\!G$	heat recovery steam generator
$G$	mass flux [kg · m <sup>-2</sup> · s <sup>-1</sup> ]	$LPC$	low pressure compressor
$N$	dimensionless parameter eq. A.3	$LPT$	low pressure turbine
$T$	temperature [K]	$ORC$	organic Rankine cycle
$X$	operand eq. A.22 [m <sup>-1</sup> ]	$OTB$	once-through boiler
$\dot{m}$	mass flow rate [kg · s <sup>-1</sup> ]	$PT$	power turbine
Bo	boiling number	$TUR$	organic Rankine cycle expander
Fr	Froude number	<i>Greek letters</i>	
Nu	Nusselt number	$\delta$	fin thickness [m]
Pr	Prandtl number	$\eta$	fin efficiency
Re	Reynolds number	$\mu$	viscosity [kg · m <sup>-1</sup> · s <sup>-1</sup> ]
$c$	speed of sound [m · s <sup>-1</sup> ]	$\rho$	density [kg · m <sup>-3</sup> ]
$c_p$	specific heat capacity [J · kg <sup>-1</sup> · K <sup>-1</sup> ]	$\varphi$	operand eq. A.22
$d$	inner tube diameter [m]	<i>Subscripts</i>	
$f_D$	Darcy friction factor	$c$	cold fluid
$g$	gravity acceleration [m · s <sup>-2</sup> ]	$f$	fin
$h$	heat transfer coefficient [W · m <sup>-2</sup> · K <sup>-1</sup> ] or enthalpy [J · kg <sup>-1</sup> ]	$G$	gas
$h_{LG}$	heat of evaporation [J · kg <sup>-1</sup> ]	$h$	hot fluid
$k$	thermal conductivity [W · m <sup>-1</sup> · K <sup>-1</sup> ]	$in$	inlet
$p$	pressure [Pa]	$L$	liquid
$q$	heat flow rate [W]	$o$	overall
$q''$	heat flux [W · m <sup>-2</sup> ]	$S$	static
$s$	entropy [J · kg <sup>-1</sup> · K <sup>-1</sup> ]	$T$	total
$x$	vapour quality	$t_0$	bare tube surface
<i>Abbreviations</i>		$th$	throat
$CC$	combustion chamber	$wi$	inner wall
$GEN$	electric generator	$wo$	outer wall
$GTA, GTB, GTC$	gas turbine A, B and C	$cb$	convective boiling
		$nb$	nucleate boiling

## 1. Introduction

Although investigated since the 1880s, Organic Rankine Cycles (ORC) have never been popular until today's growing interest in medium and low grade energy recovery systems where cycles using water as working fluid fail for technical and economic reasons [1, 2].

Organic fluids, i.e., refrigerants and hydrocarbons [3], can mitigate the technical problems associated with the use of steam. These compounds feature higher molecular mass and lower critical temperature than water. These aspects can make small or medium scale power plants technologically and economically feasible.

Their cycle architecture is similar to that of conventional steam Rankine cycles. The high pressure liquid is first evaporated, then expanded to a lower pressure, thus producing mechanical power. The cycle is closed by condensing the low pressure vapour (coming from the turbine outlet) and pumping the liquid to the high pressure side. Hence, an ORC unit has the same devices as a conventional steam power module: an evaporator, an expander, a condenser and a pump.

An organic Rankine cycle has several advantages over steam power plants, as pointed out by Tchanche et al. [3]. The evaporation process, usually taking place at lower temperature and pressure, requires less heat. Superheating is not required, and the risk of turbine blades erosion is avoided as the expansion process ends in the vapour region. Moreover, the relatively low pressure ratio of the expander enables the use of simple single stage turbines.

The ORC technology is suitable for recovering heat from solar radiation [4–12], ocean warm layers [13–17], hydro-thermal and engineered geothermal systems [18–21], abandoned oil fields [22–24], biomass [25–29], and industrial processes [1, 30, 31].

The choice of the working fluid tightly relates to the characteristics of the heat reservoir, as it determines the configuration, performance and economics of the plant [32]. These aspects justify the abundant literature dedicated to the fluid selection (see for example [33, 34]) and plant configurations [35].

As pointed out by Pasetti et al. [36], another key parameter is the thermal stability of the organic fluid. It is defined as the maximum temperature at which the fluid can be used in power plants without risk of decomposition. Fluid overheating or hot spot and the consequent fluid

1 decomposition is more likely to occur in the vapour film in contact with the tube metal walls of  
2 the terminal part of the primary heat exchanger. As the system performance strongly relates to  
3 transport and physical properties of the working fluid, hot spots can severely reduce the net power  
4 output, the fluid stability and the components' integrity [37].

5 Fluid thermochemical decomposition depends on the breakage of chemical bonds between the  
6 molecules and the formation of smaller compounds. These species can then react to create other  
7 hydrocarbons. Although studies on the thermal stability of organic compounds date back to the  
8 early 60's [36], the data available in literature are scarce and often contradictory [2, 37–39]. These  
9 research efforts paid attention to the development of testing techniques to quantify the maximum  
10 operating temperatures of the organic fluid.

11 The hot spot phenomenon is in some way analogous to that observed in the materials of boiler  
12 tubes, core of nuclear reactors and heat exchangers. Tanzer [40] described the effect of long-term  
13 material overheating on the lifetime of steam boilers. The overheating of the tube metal wall  
14 induces a reaction between the steam and the tube material itself. The result is an adhesive oxide  
15 layer. This additional resistance induces the deterioration of the metal walls as the temperature  
16 raises to the maximum tolerable limit. As surveyed by French [41], hot spot corrosion on the steam  
17 side of operating boiler tubes of fossil fuel-fired power plants is imputable to the departure from  
18 nucleate boiling. This phenomenon leads to acid or caustic attack, and deteriorates the protective  
19 magnetite film of the tube walls.

20 Occurrence of hot spots is a well-known problem in the core of nuclear reactors. This chemical  
21 process occurs if the ratio between the power density insisting on the fuel and its average value at  
22 design conditions exceeds the prescribed threshold. Statistical analysis and probabilistic evalua-  
23 tions were performed by Amendola [42] and Zhang et al. [43], respectively. Measurement tech-  
24 niques for hot spot identification in nuclear reactors were proposed by Gandini [44]. As regarding  
25 the hot spot formation in heat transfer devices, Francis [45] analysed the conditions inducing cor-  
26 rosion in copper alloys of condenser tubes. Prasher et al. [46] conducted similar investigations for  
27 micro heat exchangers utilized in electronic devices.

28 To the authors' knowledge, the fluid overheating (hot spot) and consequent decomposition dur-  
29 ing the transient operation of ORC power systems have not been analysed before. As underlined

1 by Benato et al. [47], dynamic analysis is a powerful tool to evaluate the effects of temperature  
2 fluctuations and component overheating during load cycling. The objectives of this paper are: i)  
3 to identify the dynamic events causing the thermochemical decomposition of the working fluid of  
4 an ORC unit, and ii) to suggest practical measures to tackle this issue.

5 The case study is the gas turbine-based power plant installed on an offshore oil and gas platform  
6 located in the Norwegian Sea. The use of an ORC turbogenerator is proposed to increase the  
7 overall energy conversion efficiency (see Pierobon et al. [48, 49]). Reliable operation for this  
8 plant is a priority. In fact, the economic revenue of a platform depends on stable production rates  
9 over the entire lifetime of the oil and gas field. A dynamic model of the plant based on first  
10 principles is developed using the Modelica language. This is then integrated with a discretized  
11 model of the once-through boiler (OTB) of the ORC unit. Geometric parameters and state-of-the-  
12 art correlations for heat transfer in single- and two-phase flow are adopted. This allows to estimate  
13 the temperature distribution across the OTB. Different dynamic simulations help identifying the  
14 largest possible ramp-rates of the plant. Moreover, a control system to tackle local overheating of  
15 the organic compound is proposed.

16 The case study selected for this paper is presented in Section 2. Subsequently, Section 3  
17 describes the dynamic models, and Section 4 presents their validation. The results are reported  
18 and discussed in Section 5. Concluding remarks are given in Section 6.

## 19 **2. Case study**

20 The case study is the gas turbine based-power system installed on the Draugen oil and gas  
21 offshore platform. The oil and gas field, located 150 km from Kristiansund in the Norwegian Sea,  
22 was discovered in 1984 and started production in 1993. The facility exports natural gas via the  
23 Åsgard gas pipeline to Kårstø (Norway) and oil via a shuttle tanker once every 1-2 weeks. Three  
24 Siemens SGT-500 gas turbines cover the electric power demand on board. The normal load is  
25 around 19 MW, but increases to 25 MW during oil export activities. High reliability and low risk  
26 of failure are obtained by sharing the load equally between two turbines. The third one is kept on  
27 stand-by. Despite the low energy conversion efficiency, this strategy ensures suitable reserve power

1 for peak loads and safe operation of the engines. Table 1 reports the design-point specifications of  
2 the gas turbines as provided by the manufacturer [50].

Table 1: Design-point specifications for the twin-spool gas turbine installed on the Draugen offshore oil and gas platform.

Model	Siemens SGT500
Turbine inlet temperature	850 °C
Exhaust gas temperature $t_{10}$	379.2 °C
Exhaust gas mass flow rate $\dot{m}_{10}$	91.5 kg · s <sup>-1</sup>
Electric power output	16.5 MW
Thermal efficiency	31.3 %

3 Figure 1 shows the layout of the power system with the organic Rankine cycle unit recuperating  
4 the thermal power produced by gas turbine A. The twin-spool engine employs two coaxial shafts  
5 coupling the low pressure compressor (LPC) with the low pressure turbine (LPT) and the high  
6 pressure compressor (HPC) with the high pressure turbine (HPT). The power turbine (PT) transfers  
7 mechanical power through a dedicated shaft to the electric generator (GEN). The fuel entering the  
8 the combustion chamber (CC) is natural gas.

9 Figure 2 illustrates the layout of the once-through boiler fed by the engine exhaust gases. The  
10 working fluid enters the first rows of tubes on the exhaust gas outlet end. It is then conducted  
11 by U-bends at each row in counter flow with the hot gas until it achieves the desired degree of  
12 superheating. A header collects the generated vapour which then proceeds to the turbine inlet.

13 The working fluid is preheated and evaporated continuously within each of the parallel circuits.  
14 Gravity is not used to create the head. A centrifugal pump produces forced flow in the tubes. In  
15 OTBs a thin-walled separator, not shown in Figure 2, replaces the function of the high-pressure  
16 drum of conventional heat recovery steam generators. Such component performs the function of  
17 water/steam separation during start-up and shut-down. At steady state operations, including low  
18 loads, the steam at the evaporator outlet is slightly superheated. Consequently, no separation is  
19 needed. The steam flow thus passes through this component as part of the interconnecting piping  
20 toward the superheater [51]. The superheated fluid expands in a single-stage axial turbine me-

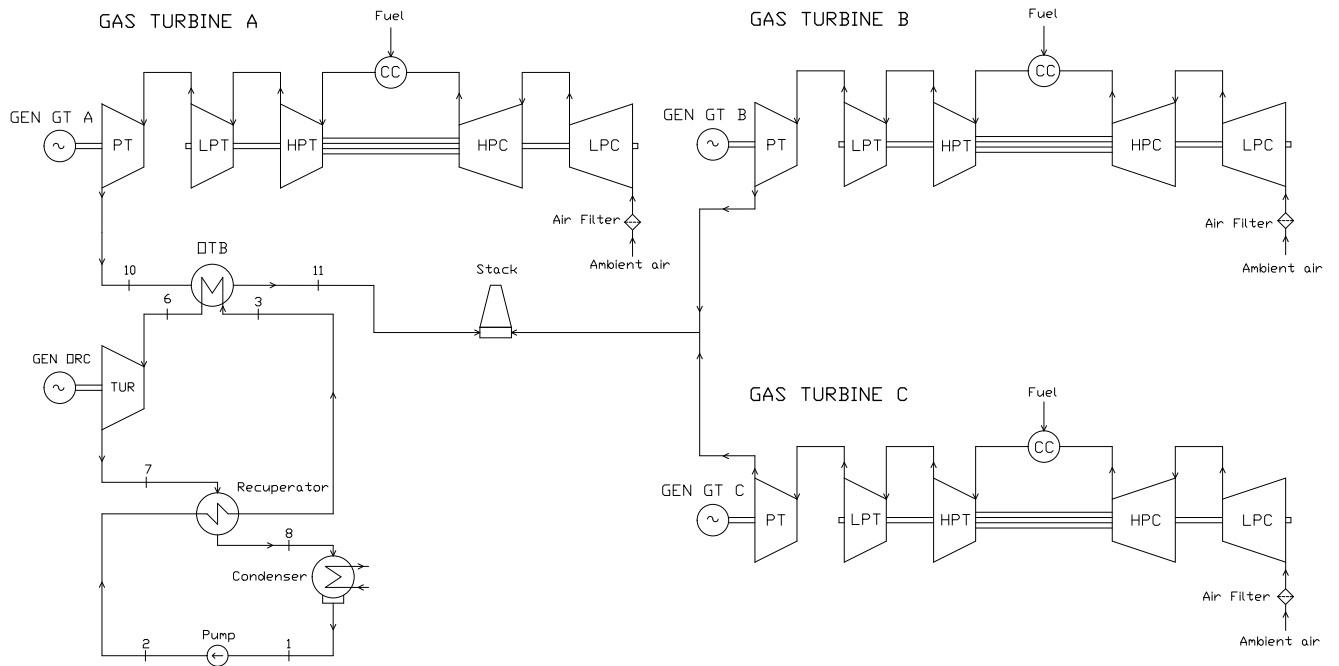


Figure 1: Layout of the power system on the Draugen oil and gas platform. The organic Rankine cycle unit recovers the thermal power of one engine (gas turbine A).

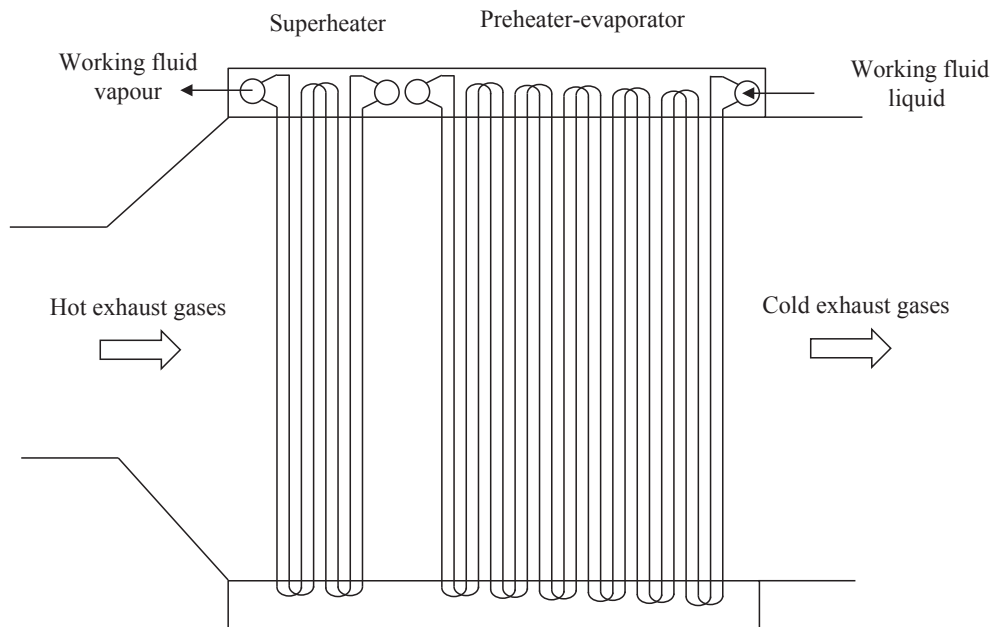


Figure 2: Layout of the once-through boiler serving the organic Rankine cycle power unit. The exhaust gases exiting the gas turbine heat up the working fluid which circulates first inside the preheater-evaporator, and, subsequently, in the superheater.



1 mechanically coupled with a dedicated electric generator. The recuperator then decreases the energy  
2 contained in the superheated vapour exiting the expander by pre-heating the liquid before entering  
3 the OTB. The variable speed electric-driven pump and the sea-water cooled condenser complete  
4 the cycle. The working fluid is cyclopentane. This organic compound is widely adopted for ORC  
5 turbogenerators with maximum source temperature between 250–350 °C, see, e.g., Del Turco et al.  
6 [52].

### 7 **3. Methods**

8 This part of the paper presents the adopted modelling language, see Section 3.1. Sections 3.2  
9 and 3.3 are dedicated to the mathematical description of the models of the gas turbines and the  
10 organic Rankine cycle.

#### 11 *3.1. The modelling language*

12 An effective way to build dynamic models is to use the fully modular approach of the equation-  
13 based, object-oriented modelling language Modelica [53]. Firstly, it allows to carry out the mod-  
14 elling task reliably and in a short time, as it leverages on existing libraries of reusable component  
15 models. Secondly, the equation-based approach of the language enables to easily customize the  
16 models for the specific requirements at hand.

17 The dynamic model of the combined cycle unit is developed using components from existing  
18 Modelica packages. The gas turbine sub-system model is built exploiting basic components in-  
19 cluded in the ThermoPower library [53]. The ORC system uses models from the Modelica ORC  
20 package [54], with suitable adaptations regarding the heat transfer coefficients and flow configu-  
21 ration inside the once-through boiler.

#### 22 *3.2. The gas turbine*

23 Figure 3 shows the Modelica object diagram of the gas turbine. Compressors and turbines are  
24 multi-stage machines. These are modelled as zero-dimensional components using steady-state off-  
25 design characteristics. The low and high pressure compressors are modelled employing the maps  
26 of axial compressors provided by Kurzke [55]. These maps, originally from Carchedi and Wood

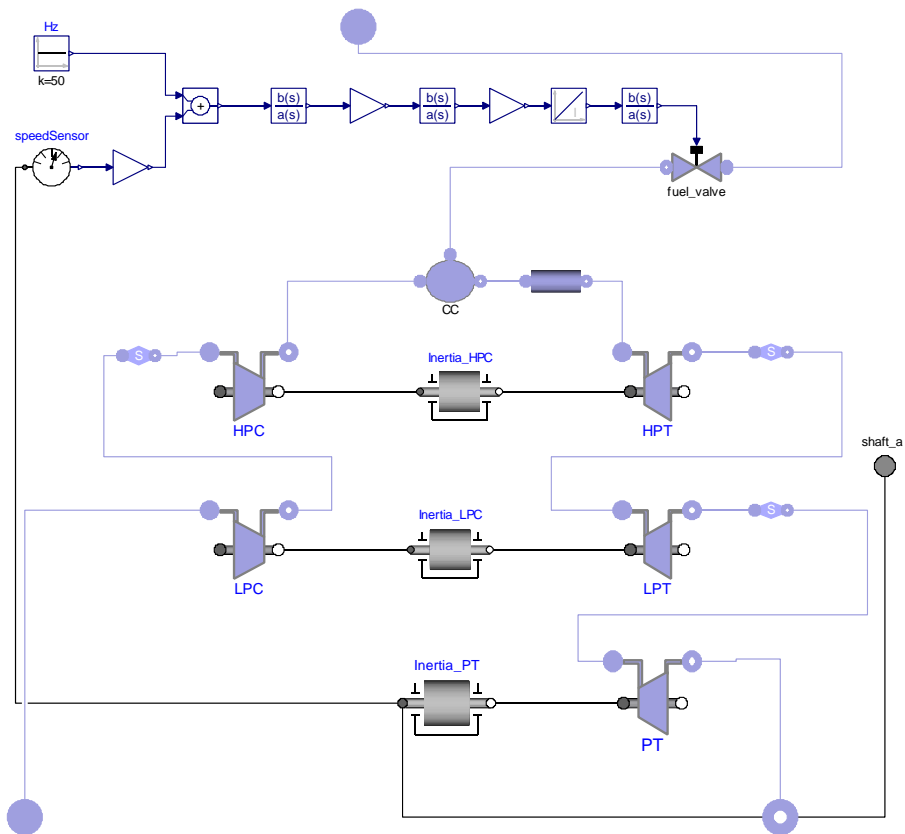


Figure 3: Object diagram of the gas turbine sub-system.

[56], include tables that state values for flow coefficient, pressure ratio, isentropic efficiency and rotational speed for the complete operating range. The maps are scaled to represent the part-load characteristics of the axial machines following the method proposed by Kurzke [57].

The equation proposed by Stodola [58] is employed for modelling the low pressure turbine, the high pressure turbine and the power turbine. This equation expresses the relation between the inlet and outlet pressure of the expander with the mass flow rate and the turbine inlet temperature in off-design conditions. The part-load isentropic efficiency of the expanders is predicted using the correlation proposed by Schobeiri [59].

The combustion chamber (CC) is built assuming a complete and adiabatic combustion process. In the component, mass and energy conservation are expressed including the dynamic terms. As suggested by Camporeale et al. [60], the mass and the internal energy are computed using the thermodynamic properties of the combustion products exiting the combustion chamber. Furthermore, it is assumed that the combustion process and the mixing of air and fuel take place within a constant volume. The pressure drops are lumped in an external device assuming a quadratic dependence on the volumetric flow rate.

The shaft dynamic balance is used to model the dynamics of each spool. The values of the inertia of the rotating masses (shaft, blades, generator) and the volume of the combustion chamber are set according to data provided by the gas turbine manufacturer. The part-load performance of the electric generator is predicted using the equation proposed by Haglind and Elmegaard [61].

The topside of Figure 3 shows the control system of the SGT-500 engine as provided by the manufacturer. The compressors are not equipped with variable inlet guide vanes. Therefore, the load of the engine can be adjusted only using the fuel valve. See Pierobon et al. [62] for a more comprehensive description of the control system blocks.

### 3.3. The organic Rankine cycle unit

Figure 4 shows the top-level interface of the organic Rankine cycle module implemented in the Modelica language. Compared to the layout given in Figure 1, the object diagram includes the inertia of the turbine shaft, the components accounting for the frictional losses in the heat exchangers and the blocks setting the thermodynamic states of the air and fuel. The proportional-

1 integral (PI) controller on the bottom-side of the diagram regulates the speed of the pump to keep  
2 the temperature of the exhaust gases constant. This operational strategy enables to avoid corrosion  
3 problems caused by the condensation of sulphuric acid vapour at any load condition and fuel  
4 composition. The input signal located on the top-right of Figure 4 sets the total power output of  
5 the combined cycle unit.

### 6 3.3.1. *The once-through boiler*

7 The model of the OTB is an extension of the generic evaporator model developed by Casella  
8 et al. [54]. Such models typically assume either co-current or counter-current flow configuration.  
9 Moreover, they do not resolve local fluid and wall temperatures inside tube bundles, which is a  
10 requirement for the current analysis. The flow configuration, i.e., the heat exchanger topology, has  
11 thus been extended and improved.

12 Figure 5(a) shows a top-view of the once-through boiler with a single longitudinal tube row.  
13 The OTB is recognized as a horizontal circular finned-tube bundle with counter-cross flow con-  
14 figuration. It is discretized in two dimensions, i.e., the exhaust gas flow direction (light purple  
15 arrows) and the organic fluid flow direction (blue arrows). The variables  $N_{\text{tube}}$  and  $N_{\text{pass}}$  are the  
16 number of cells per tube and longitudinal tubes, respectively. The temperature variations in the  
17 transverse direction of the finned-tube bundle are assumed negligible. Note that the total mass flow  
18 rate of the cold fluid is split in a series of circuits, which equal the number of transverse tubes,  $N_{\text{tr}}$ ,  
19 for the current tube circuitry. Similarly, the hot fluid is divided by the number of transverse tubes  
20 and the number of cells per tube.

21 Figure 5(b) shows the Modelica object diagram of the once-through boiler. The model uses a  
22 single one-dimensional organic fluid flow model (`coldFluid`) and  $N_{\text{tube}}$  one-dimensional models  
23 (`hotFluid`) for the exhaust gases. The cold fluid model is connected to its pipe wall capacitance  
24 (`tubeWalls`) and a heat exchanger topology model (`extCrossFlow`). The latter essentially con-  
25 nects the thermal heat ports (orange rectangles in Figure 5(b)) of each finite volumes (dashed red  
26 rectangles in Figure 5(a)), i.e., the hot fluid wall boundary with the external pipe boundary.

27 The tube wall model includes a one-dimensional dynamic heat balance equation in the radial  
28 direction for each finite volumes. The model neglects the small conductive thermal resistance. The

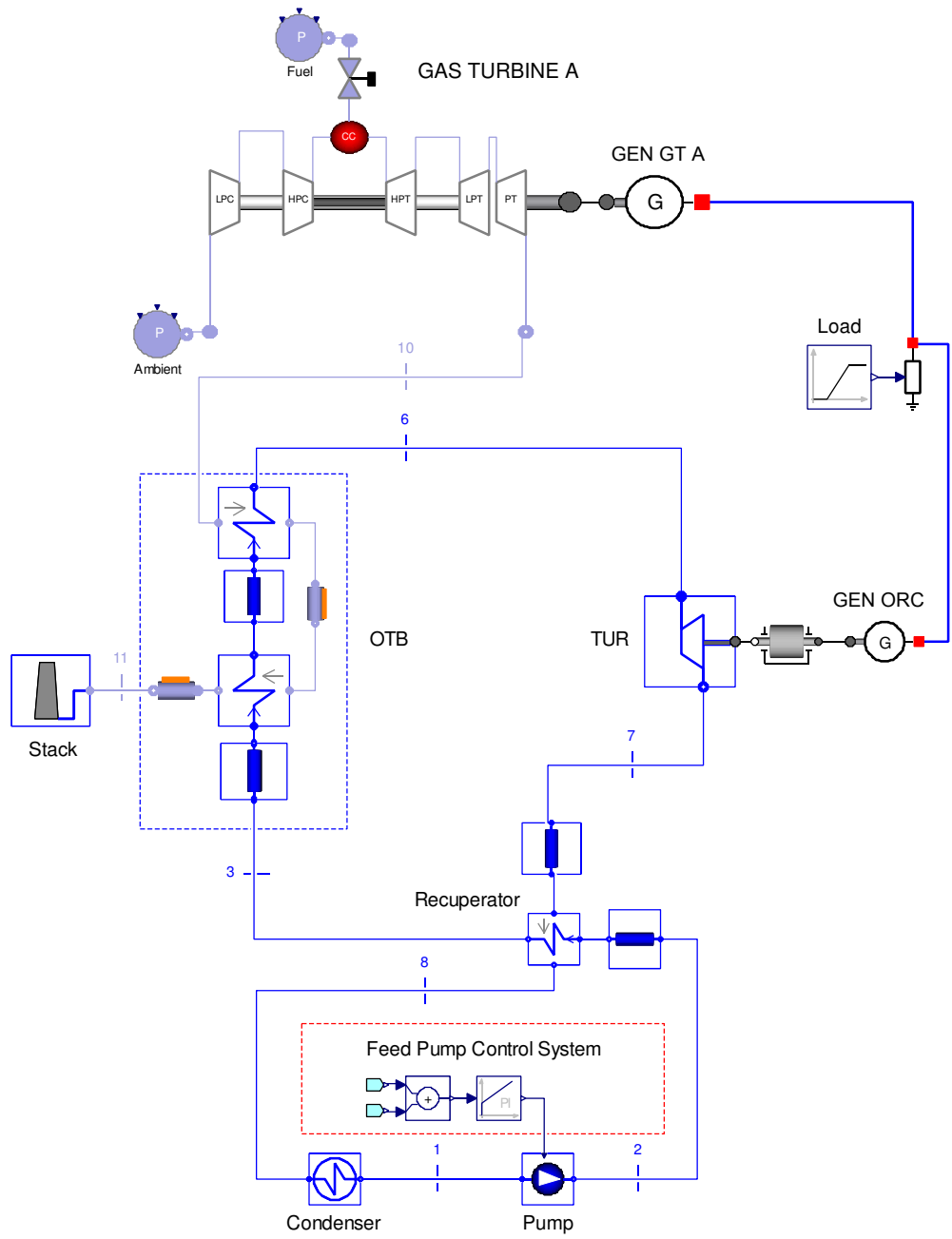


Figure 4: Object diagram of the organic Rankine cycle turbogenerator.

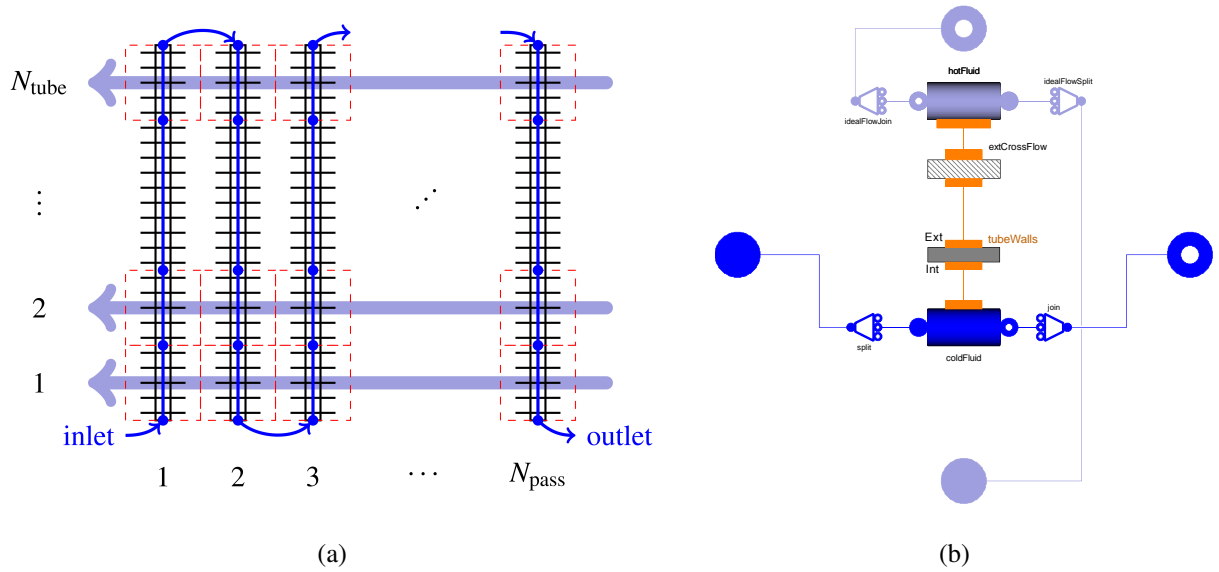


Figure 5: Extended ThermoPower heat exchanger model. 5(a) Top-view of the once-through boiler showing the discretization method. 5(b) Modelica object diagram.

1 flow models contain one-dimensional dynamic mass and energy balance equations, discretized by  
 2 the finite volume method, assuming a uniform pressure distribution. The relatively small friction  
 3 losses are lumped in an external component. As for the combustion chamber, the pressure drops  
 4 are estimated assuming a quadratic dependency on the volumetric flow rate. A more in-depth  
 5 description of the tube wall and flow models can be found in Casella et al. [54]. Appendix A  
 6 reports the equations used for the estimation of the heat transfer coefficients in the once-through  
 7 boiler.

### 8 3.3.2. The supersonic turbine

9 For MW-size ORC power plants, the expander is a one- or two-stage axial turbine. The pres-  
 10 sure ratio of each stage is relatively high (20). This implies that the flow at the outlet of the first  
 11 stage is usually supersonic. The turbine is modelled as an equivalent choked de Laval nozzle,  
 12 whose throat flow passage area is the sum of the throat areas of the nozzles of the first stator row.  
 13 An isentropic expansion is assumed from the inlet section to the throat, where sonic conditions are  
 14 attained. The corresponding system of equations is listed below.

$$\begin{cases} s_{in} = s(p_{T,in}, T_{T,in}) \\ h_{S,th} = h_{T,in}(p_{T,in}, T_{T,in}) - \frac{1}{2} \cdot c (h_{S,th}, s_{in})^2 \\ \dot{m} = \rho_{S,th}(h_{S,th}, s_{in}) \cdot c (h_{S,th}, s_{in}) \cdot A_{th} \end{cases} \quad (1)$$

1 where  $s_{in}$  is the specific entropy at the turbine inlet. The subscripts ‘‘S,th’’ and ‘‘T,in’’ indicate  
2 static conditions in the throat section and total conditions in the expander inlet section (i.e. total  
3 inlet pressure  $p_{T,in}$  and total temperature  $T_{T,in}$ ), respectively. The enthalpy and the speed of sound  
4 are represented with  $h$  and  $c$ , respectively. The variables  $\dot{m}$ ,  $\rho$  and  $A_{th}$  are the mass flow rate through  
5 the nozzle, the density and the flow passage area. The throat passage area  $A_{th}$  is obtained from the  
6 design calculation. During off-design conditions, the relation between the mass flow rate and the  
7 turbine inlet conditions is expressed by Equation system 1. The isentropic efficiency at part-load  
8 is predicted using the correlation proposed by Schobeiri [59].

### 9 3.3.3. The other components

10 Figure 6 shows the object diagram of the recuperator feeding the ORC power module. Start-  
11 ing from the topside of the figure, the models of the one-dimensional flow for the vapour side  
12 (hotFluid), the counter-current topology block (counterCurrent), the tube walls (tubeWalls  
13 and the one-dimensional flow representing the liquid side (coldFluid) are shown.

14 The counter-current establishes the topological correspondence between the control volumes  
15 of the tube metal walls and those of the working fluid on the hot and cold side. The flow models uti-  
16 lize one-dimensional dynamic mass and energy balance equations (discretized following the finite  
17 volume method, and assuming a uniform pressure distribution) and the static momentum balances  
18 (lumped at both ends of the component). The tube metal wall is modelled by a one-dimensional  
19 dynamic heat balance equation, also discretized in finite volumes, neglecting the conductive ther-  
20 mal resistance [53]. In the present case, the heat transfer coefficient is mainly controlled by the  
21 vapour side. The liquid side heat transfer coefficient is thus specified to be sufficiently large. The  
22 overall resistance is thus assumed equal to that of the vapour. At off-design conditions the heat  
23 transfer coefficient is computed by the correlation proposed by Incropera et al. [63].

24 The condenser is trivially modelled as a fixed pressure component. This is justified considering

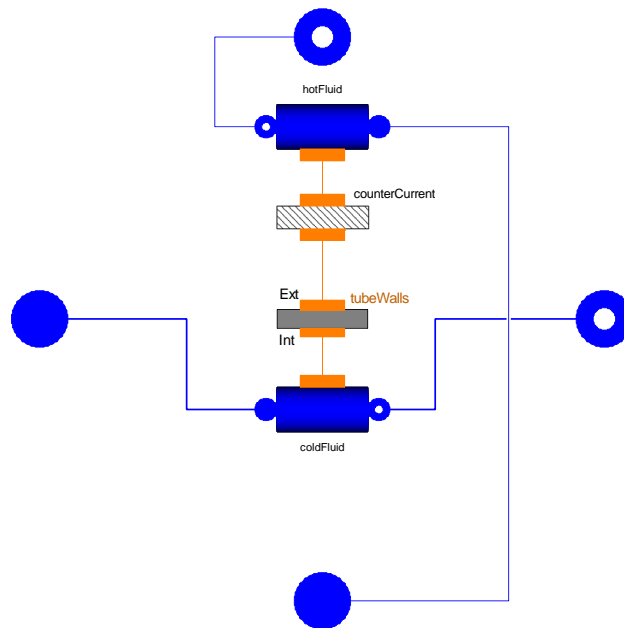


Figure 6: Object diagram of the organic Rankine cycle shell-and-tube recuperator model.

1 the large availability of cooling sea-water. The cooling circuit is thus controlled in such a way that  
 2 the condenser pressure is nearly constant. The pump model is based on a head-volume flow curve  
 3 derived by fitting the data of a centrifugal pump projected for similar design specifications.

4 The downside of Figure 4 shows the control system of the ORC unit. As mentioned, the  
 5 platform has a stand-alone system. Given that the topping units have the fastest load response, the  
 6 control of the network frequency is managed by the gas turbine itself. Conversely, the goal of the  
 7 control system of the ORC unit is to maximize the waste heat recovery.

8 This task can be fulfilled by varying the pump speed to control the exhaust gas temperature at  
 9 the once-through boiler outlet and operating in sliding pressure mode. This temperature should  
 10 be as low as possible, yet high enough to prevent acid condensation. The proportional-integral  
 11 controller is tuned to reach the minimum settling time of the controlled variable, to prevent speed  
 12 overshooting and obtain well-damped responses for all variables.

### 13 3.3.4. Model assumptions

14 An in-house simulation tool is used to design the ORC unit [48, 49]. Table B.3 in Appendix  
 15 B lists the organic Rankine cycle state points. Figure B.12 shows the  $T$ - $s$  diagram with the ther-



1 modynamic state points and the saturation dome of the ORC process. Table B.4 lists the main  
2 equipment parameters inserted in the model.

3 The thermodynamic and transport properties of the working fluid are computed according to  
4 the models implemented in the open-source software developed by Bell et al. [64].

5 It is assumed that the operational range of the gas turbine spans from 20 % to 100 %. The  
6 minimum load of the engine is thus 3 MW. This lower boundary gives a reasonable margin against  
7 chocking and surging of the compressors serving the gas turbine.

8 The time for the gas turbine trip, i.e., the period needed to pass from a certain load to zero,  
9 is set equal to 10 s. The system operates so that gas turbine A and the ORC unit share the load  
10 with the engine B. The third gas turbine is on stand-by. The test case implies that, at a given time,  
11 the engine B trips. The combined cycle unit counteracts by ramping up its load matching the total  
12 power request.

13 Considering the experimental measurements carried out by Pasetti et al.[36] and Ginosar et  
14 al. [65], a maximum temperature ( $T_{c,max}$ ) for the organic compound of 270 °C is assumed. The  
15 tests are performed at different ramp rates (0.3–1.0 MW · s<sup>-1</sup>) to estimate the frequency and the  
16 temperature trends of cyclopentane.

## 17 **4. Model validation**

18 The models of the gas turbine and of the ORC turbogenerator are validated in Section 4.1  
19 using proprietary experimental data. Section 4.2 presents the verification of the once-through  
20 boiler model, on the basis of information available in open literature.

### 21 *4.1. The gas turbine and ORC unit*

22 The steady-state part-load performance of the gas turbine is compared with the off-design  
23 characteristics given by the gas turbine manufacturer. The fuel and exhaust gas mass flow rate, the  
24 exhaust gas temperature and the pressure at the combustion chamber outlet are considered in the  
25 validation process. The largest mismatch is observed for the fuel mass flow rate. Its relative error  
26 is about 3 % from 60 % to 100 % load and 15 % from 10 % and 60 %. For the dynamic validation,  
27 the entire power system installed on Draugen platform is considered. The dynamic model of the

1 plant was validated using the operational data of the oil and gas facility. See Pierobon et al. [62]  
2 for a more comprehensive description of the validation process. Note that the model can predict  
3 the network frequency with a relative error lower than 1 % during the trip of one engine [62].  
4 The gas turbine dynamic model is thus capable to reproduce the steady-state and the dynamic  
5 characteristics of the engines with satisfactory accuracy over the entire range of loads.

6 The model of the ORC system is composed by software objects taken from a library developed  
7 to model a 150 kW ORC system using toluene as the working fluid. The model was successfully  
8 validated for transient operation against experimental data as discussed in Casella et al. [54]. The  
9 developed models are therefore deemed reliable, considering the similarity of the application at  
10 hand with the one presented in the cited reference. Furthermore, it was verified that the nameplate  
11 and off-design operating points predicted by the model are consistent with those computed by the  
12 simulation tool [66] utilized to design the system.

#### 13 *4.2. The once-through boiler*

14 The OTB model described in Section 3.3.1 was verified, for the design-point condition, with  
15 the results for the heat recovery steam generator (HRSG) designed in Dumont and Heyen [67]. The  
16 HRSG topology, geometry and operating conditions were used to parametrize the Modelica model.  
17 The deviation for the heat flux is less than 1 %. The outlet temperature of the steam and exhaust  
18 gases differ by 3 % and 7 %, respectively. The model discrepancies are due to the uncertainties  
19 on the fin thickness and efficiency assumed in the reference model and on the estimation of the  
20 physical properties of the fluids.

21 As further proof of the model accuracy, a comparison with a cross-flow heat exchanger model  
22 available in Thermal Power library [68] was performed at off-design conditions. The topology,  
23 geometry, correlations and number of finite volumes are equivalent in both models. The validation  
24 highlighted a difference smaller than 2 % in all process variables.

25 The model can thus reproduce the steady-state part-load characteristics of the once-through  
26 boiler with satisfactory accuracy, given the in-depth verification using commercial tools and data.

## 1 5. Results and discussion

2 Figure 7 shows the temperature trends of cyclopentane at the outlet section of the once-through  
3 boiler for different load changes. Table 2 reports the peak temperature reached by the metal wall  
4 and organic compound during each transient event.

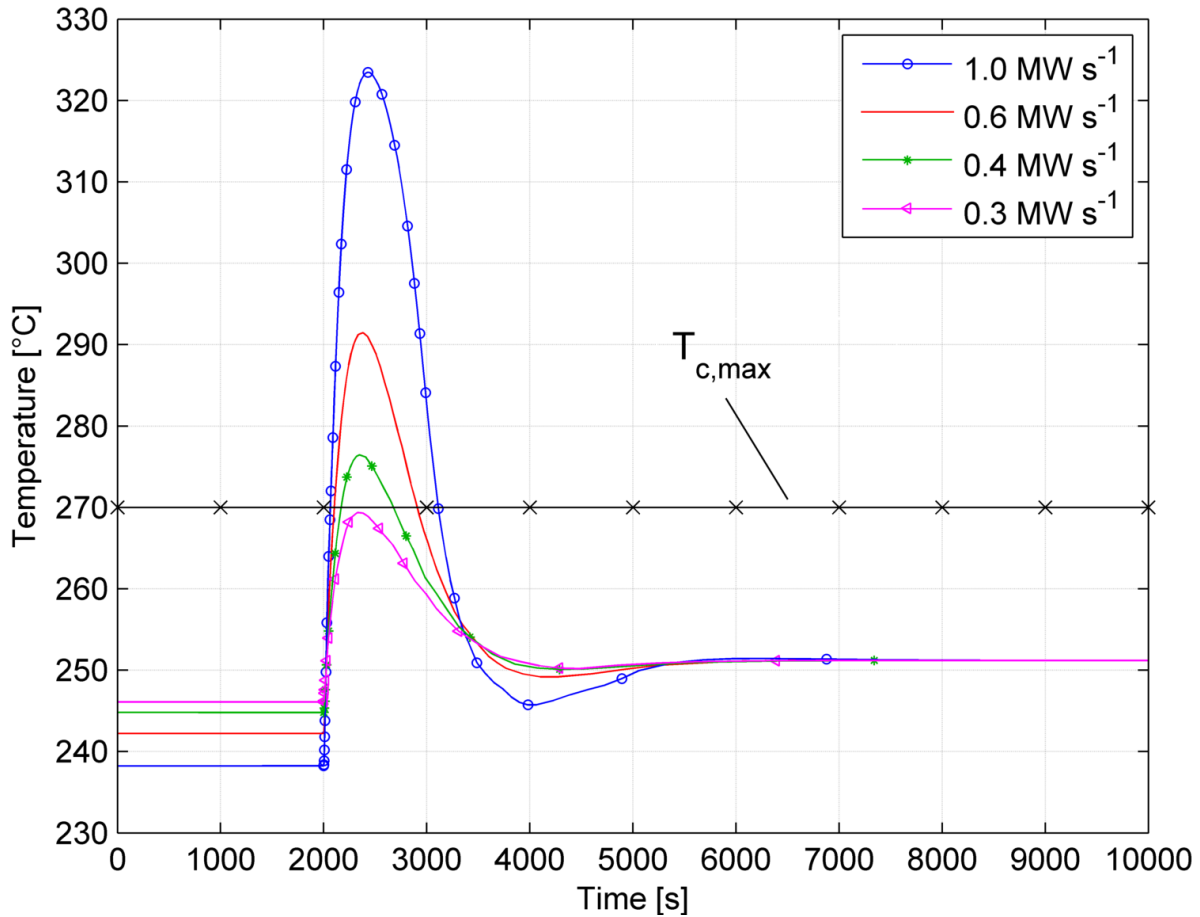


Figure 7: Dynamics of the temperature at the outlet section of the once-through boiler during load changes.

5 The results indicate that the temperature exceeds  $T_{c,max}$  for ramp rates higher than  $0.3 \text{ MW} \cdot \text{s}^{-1}$ .  
6 To estimate the long-term effects on the thermochemical stability of the fluid, the amount of time  
7 at which the fluid operates under this critical conditions has to be computed. The time is around 8  
8 minutes for a ramp rate of  $0.4 \text{ MW} \cdot \text{s}^{-1}$ ) and 18 minutes for  $1.0 \text{ MW} \cdot \text{s}^{-1}$ .

9 Offshore power systems connected to a standalone electric grid have strict constraints on the  
10 frequency tolerances and recovery time. The frequency undershooting (overshooting) is the mini-

Table 2: Power produced by the GTs and ORC unit before the failure of GT B. The metal wall and fluid temperatures in the hottest point of the heat exchanger are also listed.

Load GT A + ORC [MW]	Load GT B [MW]	Ramp rate [MW · s <sup>-1</sup> ]	T <sub>wall</sub> [°C]	T <sub>cyclopentane</sub> [°C]
9	10	1.00	331.3	323.4
10	9	0.90	324.9	316.0
11	8	0.80	318.3	308.0
12	7	0.70	310.8	299.2
13	6	0.60	304.2	291.4
14	5	0.50	297.4	283.5
15	4	0.40	291.2	276.4
16	3	0.30	284.9	269.3

1 mum (maximum) value reached by the frequency during a load change, expressed as a percentage  
 2 of the reference value. The second dynamic parameter is the rise time, defined as the time re-  
 3 quired for the frequency to return back to 99 % of the value at steady-state. Figure 8 shows the  
 4 two dynamic quantities as a function of the load change.

5 The grid specifications have a maximum undershooting of 5 %. Table 2 and Figure 8 show that  
 6 ramp rates higher than 1.0 MW · s<sup>-1</sup> are not acceptable as the frequency undershooting is higher  
 7 than 5 %. In the other cases, the dynamic parameters satisfy the requirements. The fastest load  
 8 change that fulfils that grid specification and ensures the thermochemical stability of the working  
 9 fluid is thus 0.3 MW · s<sup>-1</sup>.

10 In order to preserve the fluid stability, a spray attemperator system is added to the plant layout,  
 11 see Figure 9. This device is used to limit the temperature in the superheating section of the once-  
 12 through boiler. The saturated vapour exiting the preheater-evaporator section is collected into a  
 13 separator. The measurable temperature nearest to the temperature of the metal wall and the work-  
 14 ing fluid in the terminal part of the OTB is the turbine inlet temperature  $T_6$ . If this quantity exceeds  
 15 the maximum value imposed by the user (in this case 270 °C), the spray attemperator system in-  
 16 jects into the separator (see Figure 9) subcooled cyclopentane extracted from the pump outlet. A

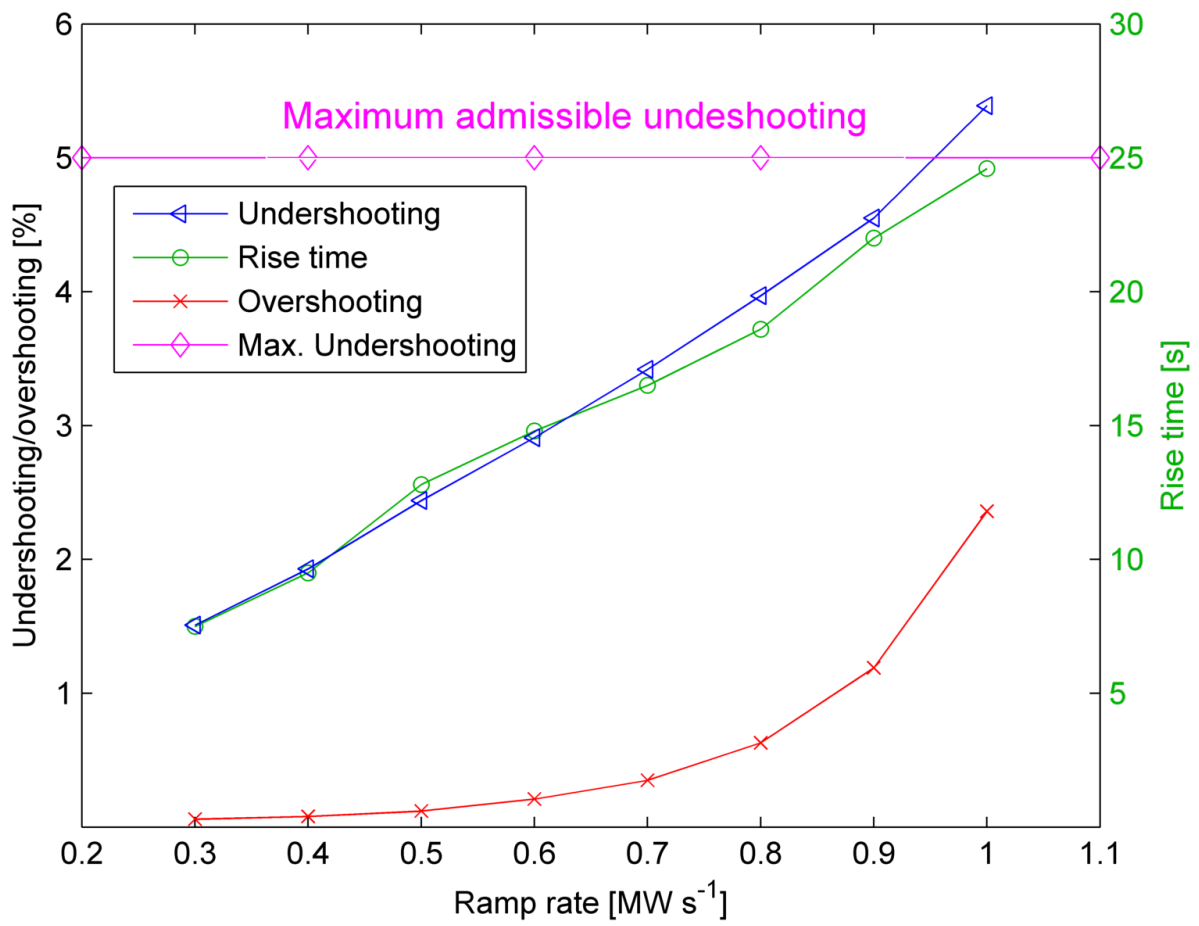


Figure 8: Dynamic metrics as a function of the load change.

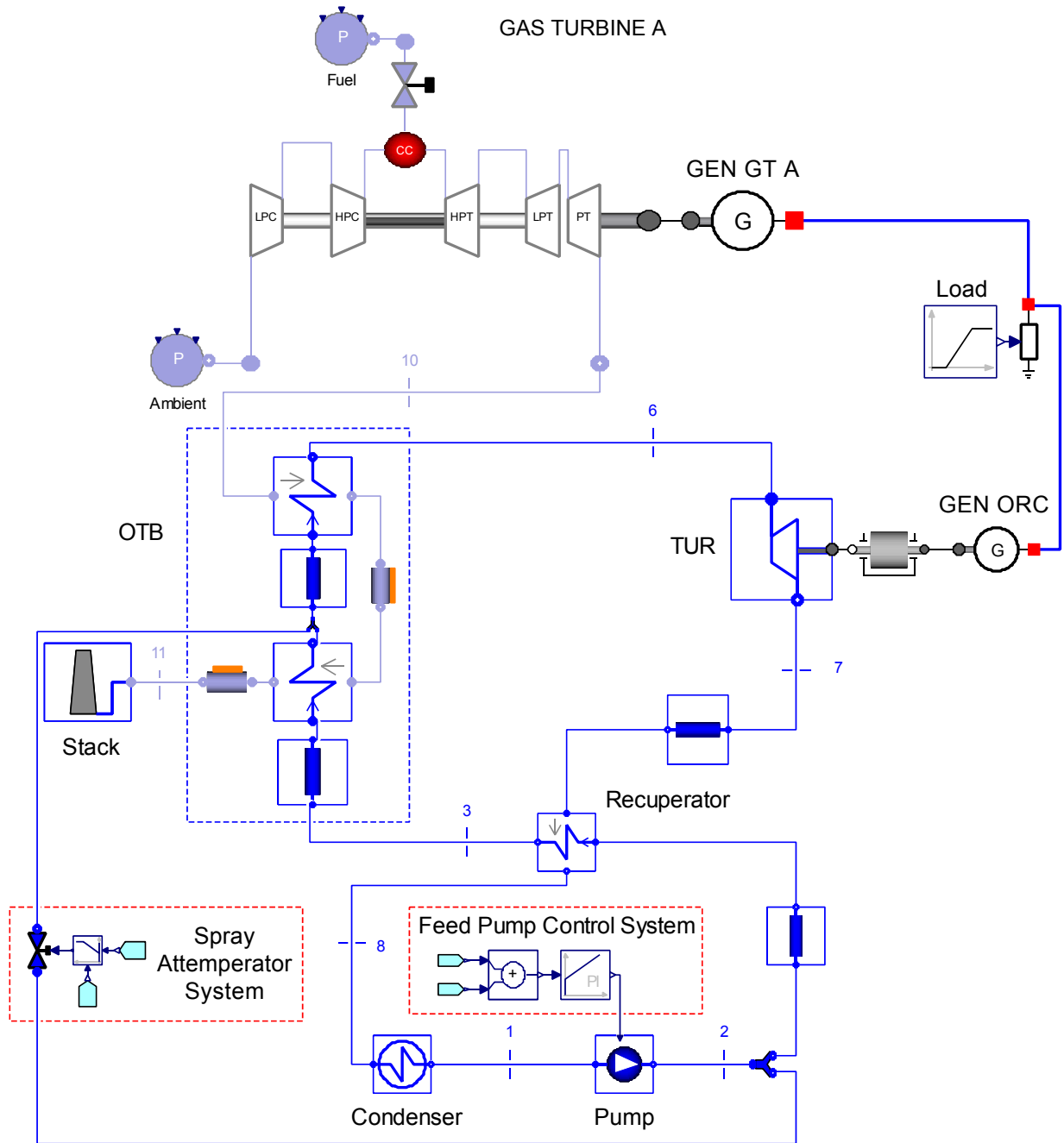


Figure 9: Object diagram of the combined cycle unit. The controller and the spray attemperator system are also shown.

1 properly tuned PI controller regulates the mass flow rate extracted using a dedicated valve. In the  
 2 PI controller, the measured temperature  $T_6$  is compared with the reference value  $T_{c,max}$ . The signal  
 3 is then transmitted to the attenuator valve. As for the previous plant configuration (Figure 4),  
 4 the pump speed is controlled to maintain the exhaust gas temperature exiting to the once-through  
 5 boiler at the design-point value.

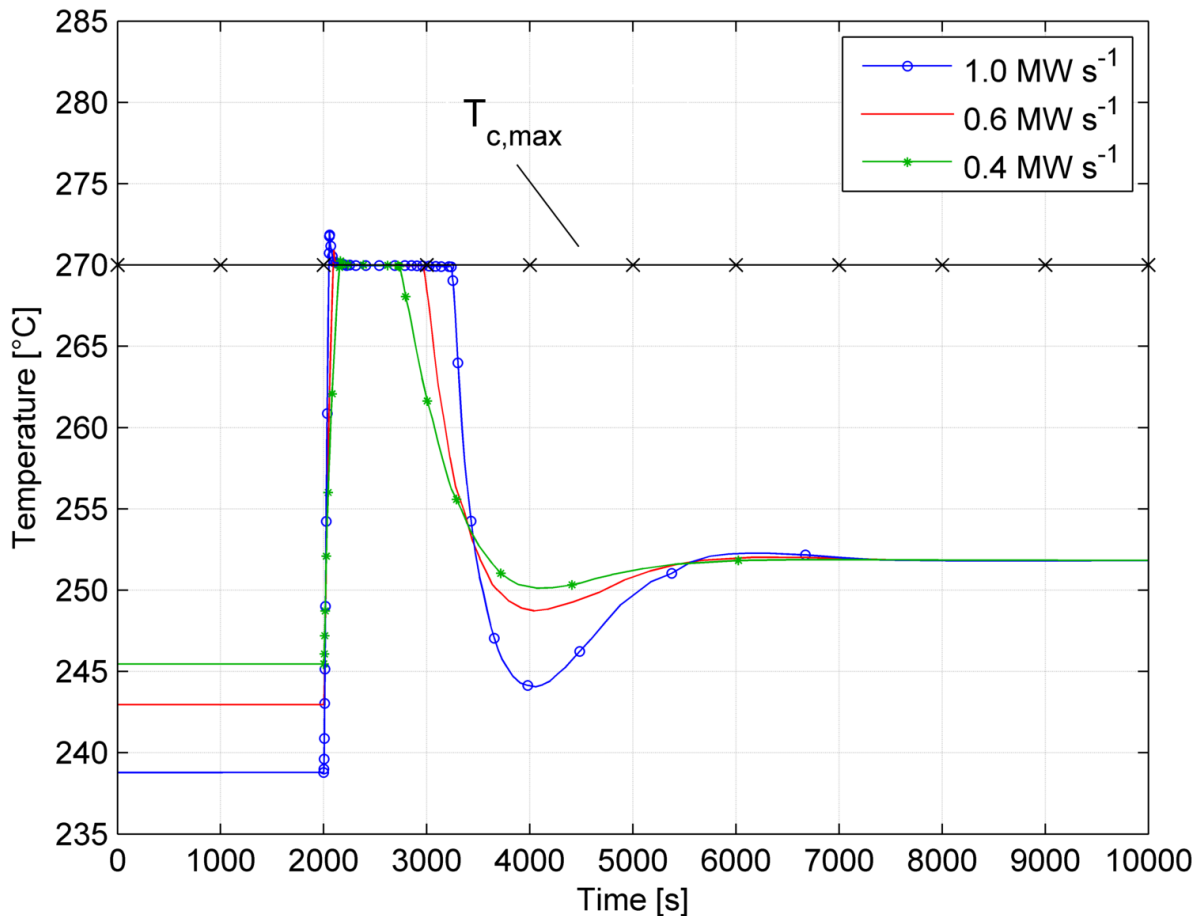


Figure 10: Dynamics of the temperature at the outlet of the once-through boiler using the spray attenuator system.

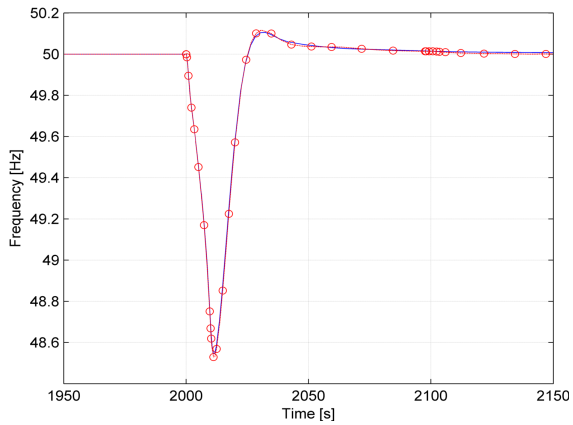
6 Figure 10 shows the trend of the working fluid temperature at the heat exchanger outlet with  
 7 the new control system. A ramp rate of  $0.3 \text{ MW} \cdot \text{s}^{-1}$  is not analysed as the activation of the spray  
 8 attenuator system is not needed, see Figure 7. The plots show that, with the exception of the  
 9 first instants of the transient, the controller can maintain the temperature at the reference value.  
 10 This prevents the hot spot formation and working fluid degradation.

11 For the sake of completeness, a comparison between the results obtained with and without the

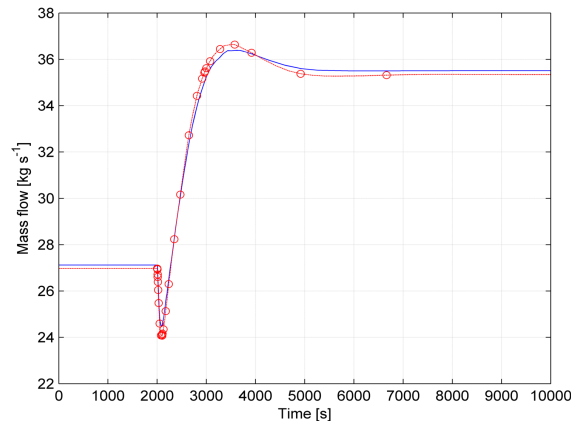
1 spray attemperator system is presented (see Figure 11). The purpose is to demonstrate that the  
2 introduction of the attemperator module does not modify significantly the dynamic response of  
3 the plant. The reference test case is a ramp rate of  $0.6 \text{ MW} \cdot \text{s}^{-1}$ .

4 Figure 11(a) demonstrates that the attemperator system does not affect the frequency trend.  
5 Figure 11(b) shows the variation of the mass flow rate entering the once-through boiler, while  
6 Figure 11(c) reports the pump speed. The spray attemperator entails a reduction of the mass flow  
7 rate. In fact, a fraction of the working fluid leaving the pump deviates towards the attemperator  
8 valve to be injected into the separator. Maximum differences of around  $1.0 \text{ }^\circ\text{C}$  are observed for the  
9 temperature of the exhaust gases exiting the OTB and for the outlet temperature on the cold-side  
10 of the recuperator, see Figure 11(d) and Figure 11(e). Finally, Figure 11(f) shows the dynamics  
11 of the turbine inlet pressure. The implementation of the attemperator system implies a pressure  
12 reduction of around 20 kPa.

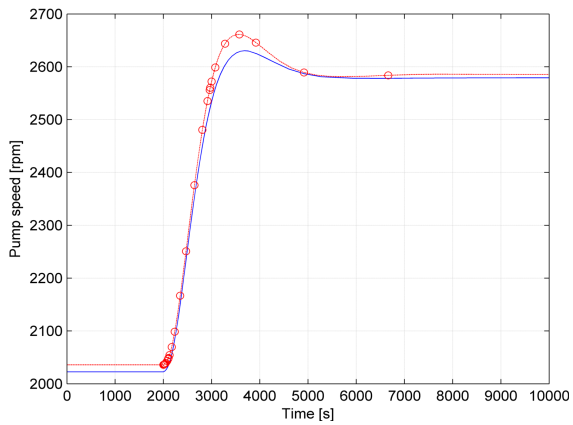




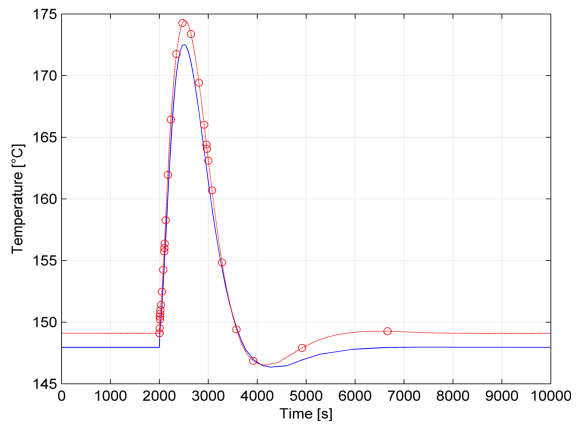
(a) Grid frequency



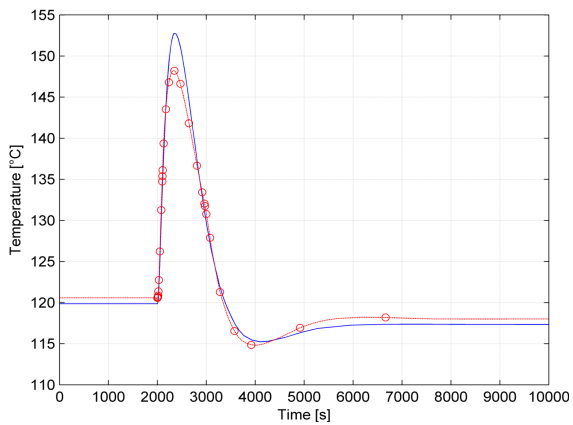
(b) Pump mass flow rate



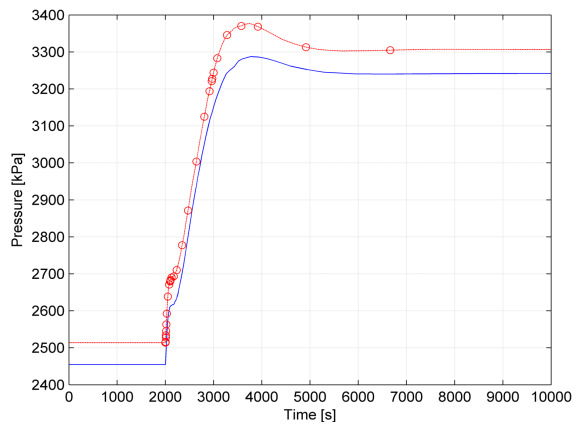
(c) Pump speed



(d) OTB exhaust gases outlet temperature



(e) OTB cyclopentane inlet temperature



(f) OTB cyclopentane inlet pressure

Figure 11: Dynamics of the combined cycle unit. The dotted line refers to the parameters of the plant with the spray attemperator system.

## 6. Conclusions

This paper presents a transient analysis that identifies critical dynamic events (hot spots) and a methodology to avoid fluid decomposition in ORC boilers.

Dynamic simulations performed at different ramp rates highlight that the most critical component is the once-through boiler. Simulation results suggest that the temperature of the working fluid exceeds the maximum admissible value for ramp rates higher than  $0.3 \text{ MW} \cdot \text{s}^{-1}$ . Such event becomes more critical during sharp load changes ( $> 1.0 \text{ MW} \cdot \text{s}^{-1}$ ), due to the longer periods ( $\approx 20$  min) of local overheating of the fluid.

This work demonstrates that the insertion of a spray attemperator module is a viable solution to tackle such operational issue. This device coupled with a properly tuned control system can maintain the temperature at the terminal section of the once-through boiler under a predefined threshold. This system does not affect significantly the dynamics of the process variables. Moreover, it eliminates the risk of fluid decomposition during aggressive load changes by injecting a fraction of the liquid exiting the pump in the superheating section.

The proposed approach and the relative solution are readily applicable without loss of generality to other power systems integrating organic Rankine cycle modules with gas turbines, boilers (fed by fossil and renewable fuels), fuel cells and solar units.

## 1 Appendix A. Heat transfer correlations

2 This appendix reports the single phase and two-phase boiling heat transfer correlations used  
3 to simulate the dynamics of the once-through boiler, see Section 3.3.1. Note that a continuous  
4 transition is required at the phase boundaries to ensure a smooth first derivative when entering  
5 the two-phase flow regime. In this work, we used the Stepsmoothing function provided by the  
6 Modelica.Fluid.Dissipation library [69], i.e., between the vapour qualities  $0 \leq x < 0.05$  and  
7  $0.95 < x \leq 1$ .

8 For single phase turbulent flow at  $Re > 3000$ , the correlation proposed by Gnielinski [70] is  
9 adopted

$$Nu = \frac{(f_D/8)(Re - 1000)Pr}{1 + 12.7(f_D/8)^{0.5}(Pr^{2/3} - 1)} \quad (A.1)$$

10 where the Nusselt number is  $Nu = h \cdot d/k$ . The Darcy-Weisbach friction factor is computed  
11 according to Petukhov [71].

$$f_D = (0.7904 \ln(Re) - 1.64)^{-2} \quad (A.2)$$

12 For laminar flow ( $Re < 2300$ ), the Nusselt number is equal to 3.66 assuming a constant wall  
13 temperature. A simple smooth transition function is used between the laminar and turbulent Nus-  
14 selt numbers.

15 For two-phase flow, the correlation proposed by Shah [72] is used to compute the heat transfer  
16 coefficient in local two-phase forced convective boiling. This correlation takes the largest value  
17 between the nucleate boiling heat transfer coefficient  $h_{nb}$  and the convective boiling heat transfer  
18 coefficient  $h_{cb}$ . The correlation is as follows:

19 For horizontal flow calculate the dimensionless parameter  $N$  by

$$N = C_o \quad Fr_L > 0.04 \quad (A.3)$$

$$N = 0.38 Fr_L^{-0.3} C_o \quad Fr_L \leq 0.04 \quad (A.4)$$

- 1 For the vertical flow, use Equation A.3 for all values of the liquid Froude number  $Fr_L$ . The  
 2 liquid Froude number and the dimensionless parameter  $C_o$  is given by

$$Fr_L = \frac{G^2}{\rho_L^2 g d} \quad (\text{A.5})$$

$$C_o = \left( \frac{1-x}{x} \right)^{0.8} \left( \frac{\rho_G}{\rho_L} \right)^{0.5} \quad (\text{A.6})$$

- 3 when  $N > 1$ , calculate  $h_{nb}$  from

$$h_{nb} = 230 h_L Bo^{0.5} \quad Bo > 0.0003 \quad (\text{A.7})$$

$$h_{nb} = h_L (1 + 46 Bo^{0.5}) \quad Bo \leq 0.0003 \quad (\text{A.8})$$

- 4 when  $1 > N \geq 0.1$ , calculate  $h_{nb}$  from

$$h_{nb} = h_L F Bo^{0.5} \exp(2.74 N^{-0.1}) \quad (\text{A.9})$$

- 5 when  $N < 0.1$ , calculate  $h_{nb}$  from

$$h_{nb} = h_L F Bo^{0.5} \exp(2.47 N^{-0.15}) \quad (\text{A.10})$$

- 6 where  $h_L$  is the liquid heat transfer coefficient calculated by the Dittus-Boelter correlation

$$Nu_L = 0.023 Re_L^{0.8} Pr_L^{0.4} \quad (\text{A.11})$$

$$h_L = Nu_L \frac{k_L}{d} \quad (\text{A.12})$$

$$Re_L = \frac{G(1-x)d}{\mu_L} \quad (\text{A.13})$$

$$Pr_L = \frac{\mu_L c_{p,L}}{k_L} \quad (\text{A.14})$$

- 7 The boiling number  $Bo$  is defined as

$$Bo = \frac{q''}{G h_{LG}} \quad (\text{A.15})$$

1 and the constant  $F$  is determined as follows

$$F = 14.7 \quad \text{Bo} > 0.0011 \quad (\text{A.16})$$

$$F = 15.43 \quad \text{Bo} \leq 0.0011 \quad (\text{A.17})$$

2 The convective boiling heat transfer coefficient  $h_{cb}$  is computed by

$$h_{cb} = h_L \frac{1.8}{N^{0.8}} \quad (\text{A.18})$$

3 Finally, the highest value of the two ( $h_{cb}$  and  $h_{nb}$ ) is chosen for the heat transfer coefficient  $h$ .

4 The heat flow rate in each cold fluid cells is then computed by the Newton's law of cooling as

$$q = h A (T_{wi} - T_c) \quad (\text{A.19})$$

5 where subscripts "wi" and "c" denote the inner wall and cold fluid, respectively. The variable  $A$  is  
6 the inner tube surface area of a single cell.

7 The gas-side heat transfer coefficient and the fin efficiency are computed with the correlations  
8 given for staggered circular finned-tubes in the Verein Deutscher Ingenieure Heat Atlas [73].

9 The Nusselt number based on the outer tube diameter  $D$  is computed by

$$\text{Nu} = 0.38 \text{Re}_D^{0.6} \left( \frac{A}{A_{t0}} \right)^{-0.15} \text{Pr}^{1/3} \quad (\text{A.20})$$

10 where  $A$  is the total heat transfer surface area including fins and  $A_{t0}$  is the bare tube surface area.

11 The Reynolds number is based on the outer tube diameter and the maximum gas velocity that  
12 may occur either transversely or diagonally in between the staggered tubes. The fin efficiency for  
13 circular fins is computed by

$$\eta_f = \frac{\tanh X}{X} \quad (\text{A.21})$$

$$X = \varphi \frac{D}{2} \sqrt{\frac{2h}{k_f \delta}} \quad (\text{A.22})$$

1 where  $h$  is the heat transfer coefficient,  $k_f$  is the thermal conductivity of the fins, and  $\delta$  is the fin  
2 thickness. For circular non-conic fins  $\varphi$  is computed by

$$\varphi = \left( \frac{D_f}{D} - 1 \right) \left[ 1 + 0.35 \ln \left( \frac{D_f}{D} \right) \right] \quad (\text{A.23})$$

3 where  $D_f$  denotes the fin diameter. Finally, the overall surface fin efficiency is calculated by

$$\eta_o = 1 - \frac{A_f}{A} (1 - \eta_f) \quad (\text{A.24})$$

4 where  $A_f = A - A_{t0}$ , i.e the finned surface area.

5 The heat flow rate in each hot fluid cell is computed by Newton's law of cooling similar to the  
6 cold fluid, but including the overall surface fin efficiency as

$$q = \eta_o h A (T_{wo} - T_h) \quad (\text{A.25})$$

7 where subscripts "wo" and "h" denote the outer wall and hot fluid, respectively. Note that  $T_c$  and  
8  $T_h$  in equation [A.19](#) and [A.25](#) are taken as the cell center average value.

## 1 Appendix B. Design-point analysis results

2 The organic Rankine cycle state points, the related  $T$ - $s$  diagram with the thermodynamic state  
3 points and the main equipment parameters included into the ORC model are reported.

Table B.3: Organic Rankine cycle state points

Point	T [°C]	p [kPa]	$\rho$ [kg · m <sup>-3</sup> ]	h [kJ · kg <sup>-1</sup> ]	s [kJ · kg <sup>-1</sup> · K <sup>-1</sup> ]
1	50.00	103.83	715.16	1.44	0.0044
2	51.53	3819.48	718.01	7.20	0.0062
3	118.00	3815.96	644.83	147.62	0.3985
4	224.48	3756.00	416.28	456.42	1.0892
5	224.48	3756.00	133.26	584.44	1.3465
6	258.72	3721.37	87.345	701.11	1.5746
7	156.19	110.29	2.2011	558.61	1.6477
8	70.31	103.83	2.6284	418.25	1.2917
9	50.00	103.83	2.8125	390.12	1.2073

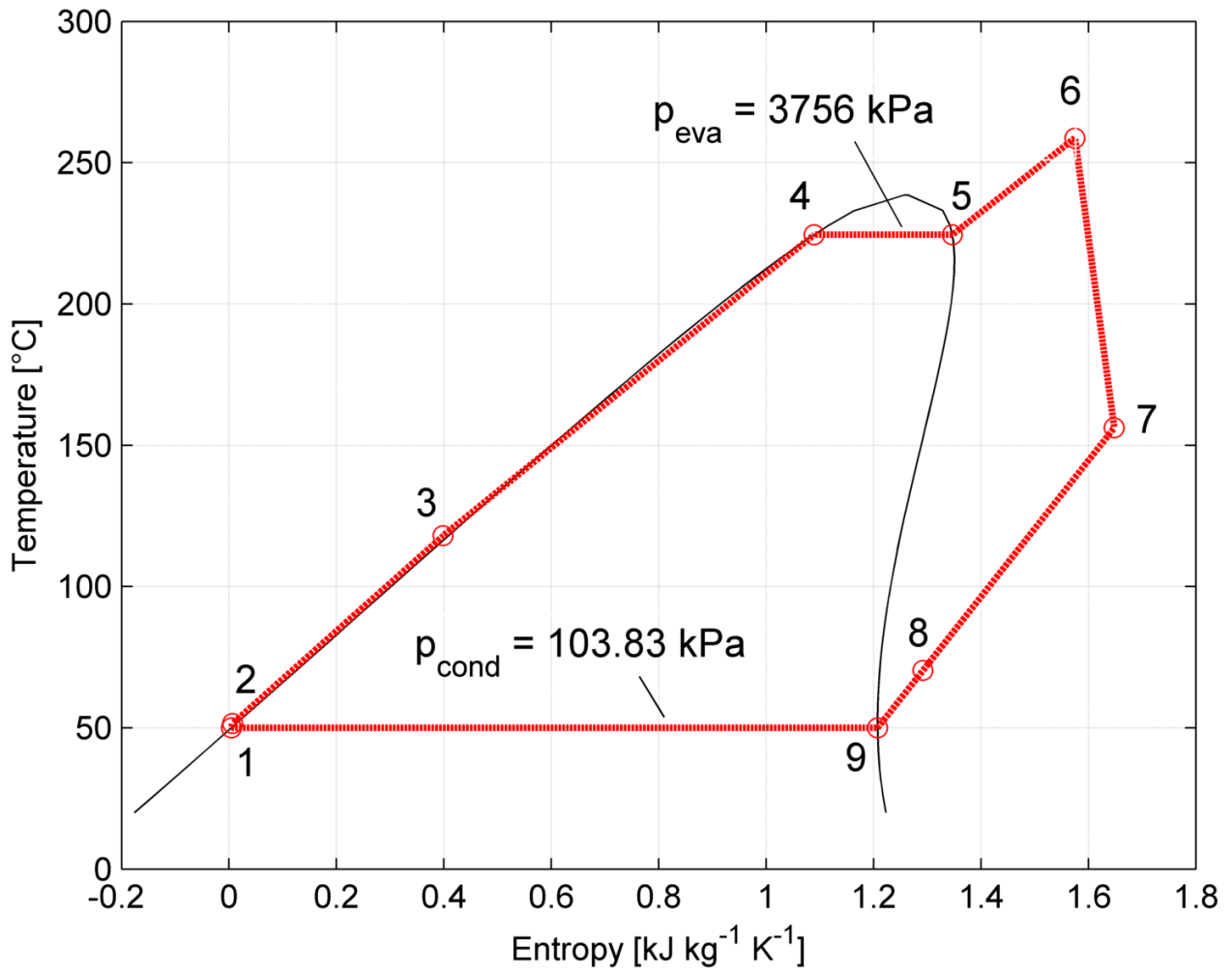


Figure B.12: Organic Rankine cycle  $T-s$  diagram.



Table B.4: Design-point variables utilized to parametrize the dynamic model of the organic Rankine cycle system.

Component	Parameters
<b>Once-through boiler</b>	
Number of tube rows	63
Number of tubes in parallel	64
Longitudinal and Transverse tube pitch	83 mm
Tube inner diameter	38 mm
Tube thickness	3 mm
Tube length	2.44 m
Number of fins	227 m <sup>-1</sup>
Fin height	15 mm
Fin thickness	1 mm
Tube wall density	7700 kg · m <sup>-3</sup>
Tube wall specific heat capacity	500 J · kg <sup>-1</sup> · K <sup>-1</sup>
Fin thermal conductivity	40 W · m <sup>-1</sup> · K <sup>-1</sup>
<b>Recuperator</b>	
Volume (cold side)	1.99 m <sup>3</sup>
Volume (hot side)	20.3 m <sup>3</sup>
Weight (metal walls)	16.6 ton
UA-value	202.3 kW · K <sup>-1</sup>
<b>Turbine</b>	
Throat flow passage area	0.040 m <sup>2</sup>
Isentropic efficiency	81 %
Electric generator efficiency	98 %
<b>Pump</b>	
Delivery pressure	3852.5 kPa
Inlet pressure	101.83 kPa
Isentropic efficiency	72 %

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