

# Design and testing of high temperature micro-ORC test stand using Siloxane as working fluid

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**Abstract.** Organic Rankine Cycle is a mature technology for many applications e.g. biomass power plants, waste heat recovery and geothermal power for larger power capacity. Recently more attention is paid on an ORC utilizing high temperature heat with relatively low power. One of the attractive applications of such ORCs would be utilization of waste heat of exhaust gas of combustion engines in stationary and mobile applications. In this paper, a design procedure of the ORC process is described and discussed. The analysis of the major components of the process, namely the evaporator, recuperator, and turbogenerator is done. Also preliminary experimental results of an ORC process utilizing high temperature exhaust gas heat and using siloxane MDM as a working fluid are presented and discussed. The turbine type utilized in the turbogenerator is a radial inflow turbine and the turbogenerator consists of the turbine, the electric motor and the feed pump. Based on the results, it was identified that the studied system is capable to generate electricity from the waste heat of exhaust gases and it is shown that high molecular weight and high critical temperature fluids as the working fluids can be utilized in high-temperature small-scale ORC applications. 5.1 kW of electric power was generated by the turbogenerator.

## 1. Introduction

There is an inherent demand for using sustainable and renewable energy systems for an energy production in the future. One of the promising alternatives to fulfil this demand are decentralized energy systems in which the power output is usually low [1]. These decentralized power systems could be used to utilize solar energy, biomass, and waste heat e.g. from combustion engines in mobile and stationary applications. Organic Rankine Cycle (ORC) is a promising technology to be used in these applications because it is a versatile and proven system for larger power.

The high temperature and relative small amount of heat requires the use of a turbine as an expander and an organic working fluid which tolerates high temperature without decomposition. Characteristic for the working fluids to be used in these circumstances is a high molecular complexity which leads to a low speed of sound and therefore, supersonic flow and non-ideal compressible flow phenomena in the turbine. There is a clear need for the research facilities where non-ideal compressible flow phenomena can be studied. In addition, the performance prediction of the turbines is usually relying on measured data of turbine test sections or existing turbines [2,3].

However, the measured data is not available for the high pressure ratio supersonic turbines and there are no test setups available where this kind of turbines and complete ORC utilizing highly complicated working fluid(s) could be tested. On the other hand, a blowdown wind tunnel for real gases is presented in [4] and the first preliminary results of an expansion of siloxane MDM are shown in [5]. In addition,



a hybrid test setup to conduct experimental campaigns in a nozzle and an expander is introduced in [6] and the test setup for an organic working fluid wind tunnel is presented in [7]. Both of these test setups are under construction. The test facility using ethanol as the working fluid and a partial admission axial turbine as an expander is described in [8]. To the best knowledge of the authors, the system is tested successfully.

The design procedure and the first test results of the micro-ORC power plant located in Lappeenranta University of Technology (LUT) are reported in the present article. The working fluid selection and process design are discussed in the next two sections. This is followed by a discussion of the first measured results of the ORC using the high pressure ratio supersonic turbine. The main focus of this article is to study the performance of a high temperature and high pressure ratio ORC suitable for small-scale applications. Especially, the use of a high-speed turbogenerator for producing electric power is discussed.

## 2. Working fluid selection

The selection of the working fluid is probably the most important step in designing ORC systems since it highly affects the component dimensioning, operational temperature and pressure, and cycle performance [9]. In general, the fluids having a high critical temperature can reach high thermodynamic efficiencies in cycles in which high turbine inlet temperatures can be adopted [10-12]. The relation of the molecular weight and the critical temperature as well as the relation of the critical temperature and the cycle efficiency for different kinds of working fluids are shown in Figure 1. Another advantage of using fluids with high molecular complexity is the relatively low enthalpy change when adopting high pressure ratios and high volumetric flow rates on the low pressure side of the process which enables the design of expanders with single or few stages. In addition, the optimal expander rotational speeds with this type of fluids are lower when compared to fluids with simple molecules which is beneficial especially in small-scale systems in which the expander rotational speeds tend to be impractically high [12]. The disadvantages of using fluids having a high critical temperature are the low speed of sound, which leads to highly supersonic flows when using high expansion ratios, and low condensing pressures, which introduces additional requirements for the system sealing during the operation. In addition, the low condensing pressure and high volumetric flow rate on the low pressure side leads to the use of relatively large sized heat exchangers, namely the recuperator and the condenser, when compared to fluids with lower critical temperature and simpler molecule.

The thermal and chemical stability of the fluid can be identified as important fluid selection criteria, particularly in the case of high-temperature ORCs. Erhart et al. [13] carried out an experimental study on siloxane decomposition in 7 large-scale high-temperature ORC plants. Their results indicated that decomposition of the fluid occurred over a long time period, and that the degradation of the fluid was caused by the use of petroleum-based lubricants in the system alongside the high working fluid temperatures in the process. Thus, the use of the working fluid lubricated bearings was evaluated to be beneficial in this type of systems.

After screening a large number of different fluid candidates, siloxane MDM was evaluated as the most suitable fluid for the high temperature experimental ORC system presented in this study. The selection of the working fluid was a compromise taking into account the cycle efficiency, process design, and turbine design. MDM was selected since it allows to design a turbine with a significantly lower rotational speed and larger turbine wheel when compared to the hydrocarbons and fluorocarbons and siloxane MM. In addition, the blade height at the rotor inlet is higher with MDM when compared to the hydrocarbons. The disadvantages of using MDM as a working fluid are the relatively low condensing pressure, high expansion ratio over the turbine which leads to a large variation on the rotor blade height between the rotor inlet and outlet and the low speed of sound. Also the high volumetric flow rate at the turbine outlet, which leads to the use of large sized heat exchangers and process piping on the low pressure side of the process and a moderate cycle efficiency when compared to more complex siloxanes such as D4 or hydrocarbons with a high critical temperature can be accounted as disadvantages. MDM

is classified as a linear siloxane having the molecular formula of  $C_8H_{24}O_2Si_3$ , the critical pressure of 14.2 bar and the critical temperature of 290.9 °C.

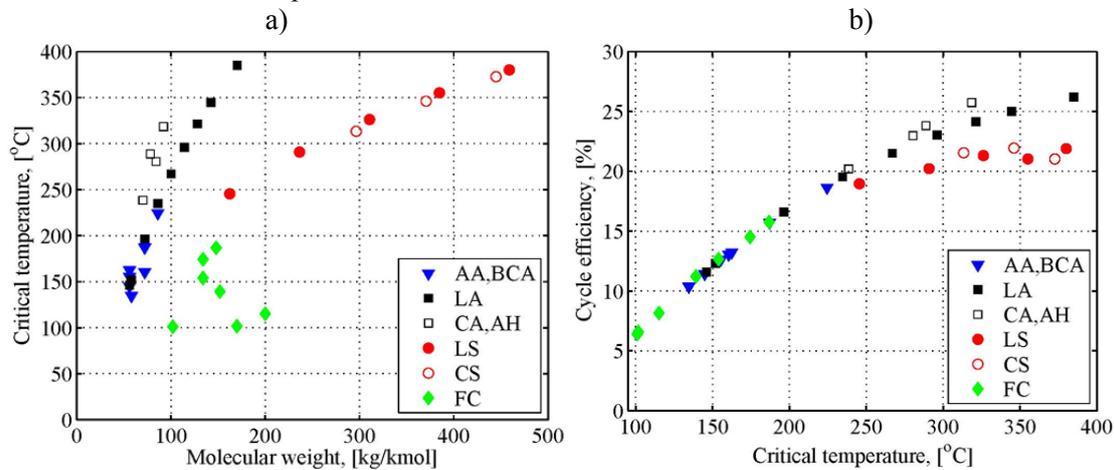


Figure 1. Relation between the molecular weight and critical temperature a) and relation between the critical temperature and cycle efficiency b). AA (acyclic alkane), LA (linear alkane), BCA (branched-chain alkane), CA (cyclic alkane), and AH (aromatic hydrocarbon) refer to different kinds of hydrocarbons, LS and CS refer to linear and cyclic siloxanes, and FC refers to fluorocarbons.

### 3. Process design

The aim of the process design was to achieve detailed information of the micro-ORC power plant using heat of the exhaust gases of a diesel generator as a heat source. Based on this process design, a power plant was built in Laboratory of Fluid Dynamics at LUT. The design objectives of the power plant were 10kW electric power output, hermetic construction, and the best possible conversion efficiency (>15%).

#### 3.1. Cycle design

The experimental setup is designed for studying the performance of a small-scale and high-temperature ORC. In this ORC test stand (see Figure 2), the high-speed turbogenerator contains the turbine, generator and main feed pump that are coupled on the common shaft as schematically presented in Figure 3a, and the liquid working fluid is used to lubricate the bearings of the high-speed turbogenerator. The process at the design point is shown on  $T,s$ -diagram in Figure 3b. As the rotational speed of the turbogenerator shaft is relatively high, in the order of 31 000 rpm, a frequency converter is used between the generator and the grid. The exhaust gas used as a heat source in the ORC test stand is obtained from the diesel generator. In the evaporator, the liquid MDM is preheated, vaporized and superheated using the heat transferred from the hot exhaust gas. The superheated vapor is directed to the turbine that runs the generator and the main feed pump coupled on the common shaft.

At the turbine outlet, vapor MDM is highly superheated, and hence a recuperator is used to desuperheat the vapor before being directed to the condenser in where the vapor is further desuperheated and finally condensed to liquid. In the recuperator, heat is transferred from the vapor MDM exiting the turbine to the liquid MDM entering the evaporator. The pre-feed pump is used to prevent cavitation in the main feed pump as well as to supply liquid to the bearings of the turbogenerator.

The process values used in the design of the ORC experimental setup are presented in Table 1. The electric power output of the ORC of 10 kW was used as a target value in the design of the generator as well as in the selection of the frequency converter. In the turbine design, the mass flow rate of vapor MDM of 0.2 kg/s, the turbine inlet temperature of 265 °C and the turbine inlet pressure of 7.9 bar (abs.) were used. The turbine outlet pressure of 0.07 bar (abs.) used in the design depends on the condenser pressure as well as on the pressure loss between the turbine outlet and the condenser. Under the conditions presented in Table 1, MDM has a low condensing pressure, 0.03 bar (abs.), which sets demands for the sealing of the process.



Figure 2. Test facility at the laboratory of fluid dynamics, LUT.

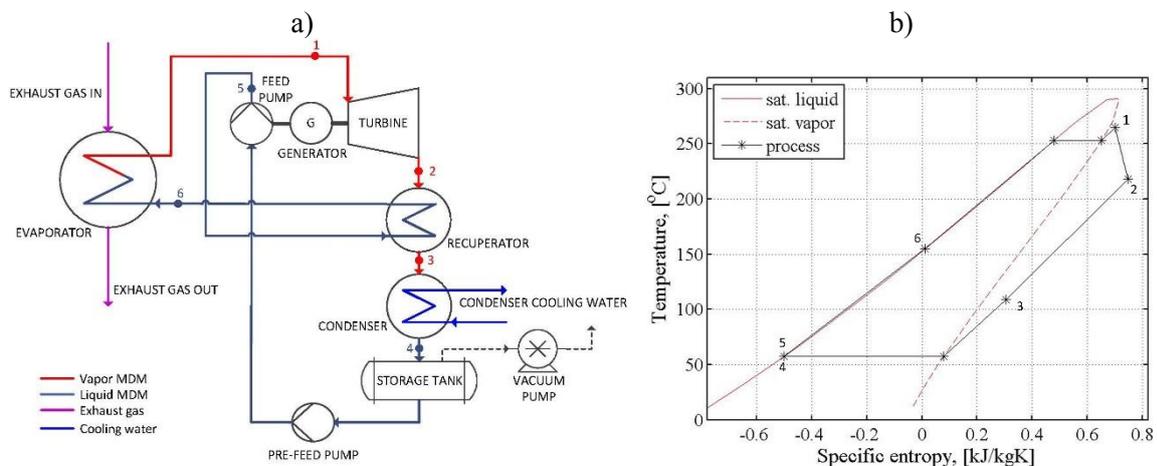


Figure 3. ORC process diagram a) and process at the design point on  $T,s$ -diagram b).

Table 1. Design values of the micro-ORC experimental facility. The pressures are absolute pressures.

<b>Working Fluid</b>	MDM	<b>Recuperator heat rate</b>	40 kW
<b>Working fluid mass flow rate</b>	0.2 kg/s	<b>Recuperator effectiveness</b>	0.68
<b>Electric power output</b>	10 kW	<b>Condensing temperature</b>	57 °C
<b>Evaporator heat rate</b>	67 kW	<b>Condensing pressure</b>	0.03 bar
<b>Turbine inlet pressure</b>	7.9 bar	<b>Turbine outlet pressure</b>	0.07 bar
<b>Turbine inlet temperature</b>	265 °C	<b>Conversion efficiency (heat-to-el.)</b>	15 %

### 3.2. Heat exchangers

The heat exchangers of the ORC experimental setup were designed to meet the desired process conditions of the turbine design. The temperature diagrams of the system heat exchangers are shown in Figure 4 and the main design values are presented in Table 2.

The evaporator is a plate-and-shell (PS) type once-through heat exchanger. The working fluid is preheated, evaporated and superheated directly by the exhaust gas. The exhaust gas temperature level is about 400 °C and the temperature of the superheated MDM is above 250 °C. The maximum working fluid temperature is limited to about 300 °C in order to avoid thermal decomposition of the fluid. One of the objectives for the evaporator design was the pressure drop on the exhaust gas side. The pressure drop cannot be large because it effects on the operation of the diesel engine. Therefore, the maximum accepted pressure drop on the exhaust gas side was set to 2 kPa.

The recuperator preheats the liquid working fluid by using the high-temperature and low-pressure MDM vapor exiting the turbine. The recuperator is a plate-and-shell type and the recuperation effectiveness of 0.68 was used in the design as presented in Table 1.

The condenser is a plate-and-shell type and uses cold water as a coolant for removing the heat from the system.

Table 2. Design values of the heat exchangers.

Heat exchanger	Type	$A_{tot}$ [m <sup>2</sup> ]	$U_{avg}$ [W/m <sup>2</sup> K]	$m$ [kg]	$\Delta T_{LM}$ [K]
Evaporator	PS	30.4	31	515	72
Recuperator	PS	16.0	43	218	59
Condenser	PS	7.0	201	168	40

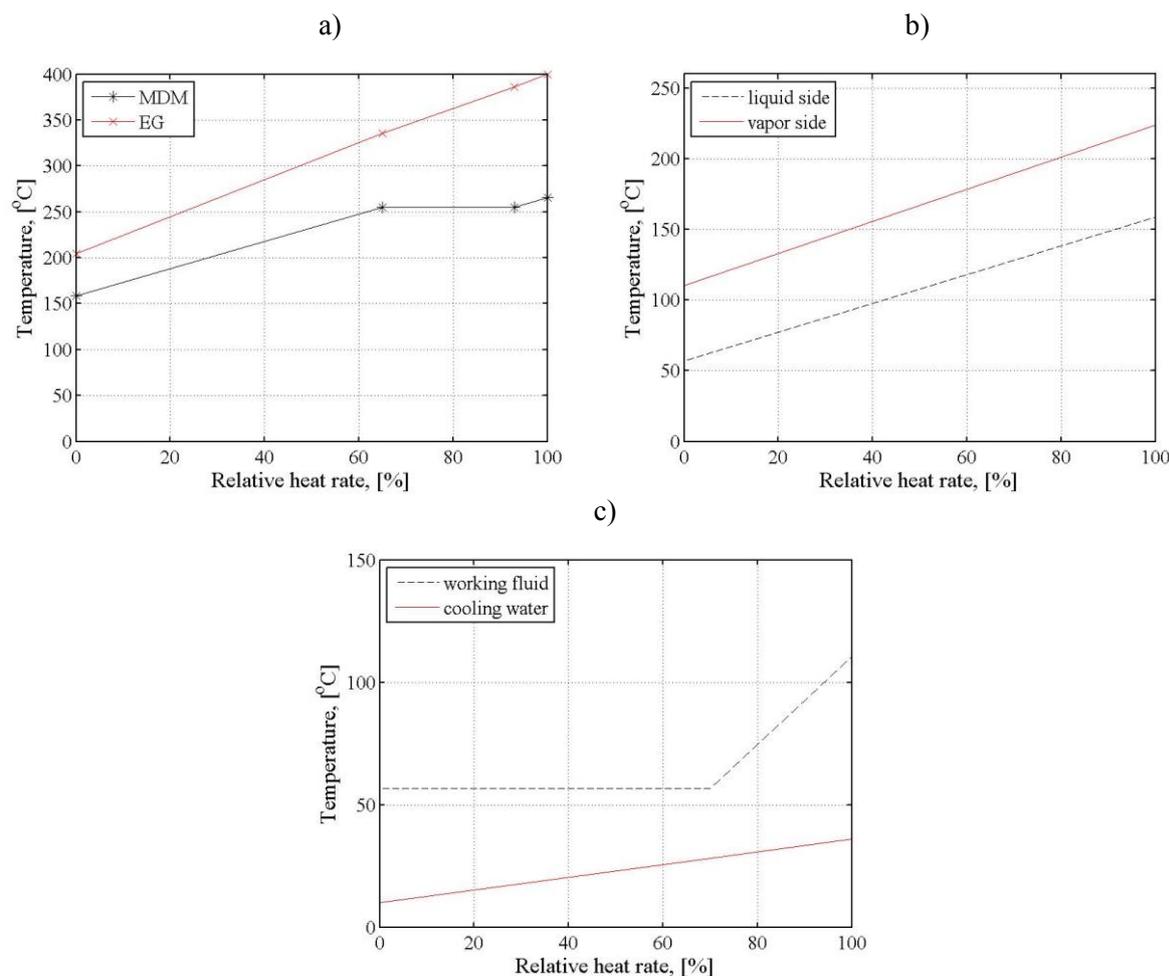


Figure 4. Temperature diagram of a) the evaporator, b) recuperator, and c) condenser at the design operating condition.

### 3.3. Turbogenerator

The most challenging task of the design process was the design of the turbogenerator. One of the design objectives was to have hermetic design and that can be achieved only if there are no shaft seals that are exposed to the environment. The turbogenerator consists of the turbine, generator and pump which are placed on the same shaft and using this kind of design shaft seals exposed to the environment are avoided. The designed turbogenerator is shown in Figure 5. The turbine is located on the left, the generator at the middle and the main feed pump on the right. In addition, the turbogenerator is equipped with the bearings lubricated with the working fluid in order to avoid the use of oil or other additional fluid in the system which could cause decomposition of the working fluid and/or deduction of the cycle efficiency. The bearings are located at the both sides of the generator.

The turbogenerator design consists of fluid dynamic design of the turbine and pump, electrical design of the generator, dynamic design of the bearings, mechanical design of the turbogenerator including dynamic and stress calculations, and design of cooling of the generator. All these design procedures are coupled to each other in order to achieve a functional and best possible outcome. Many iterations of the design procedure were needed to meet the goal. In addition, the process calculation of the ORC is coupled to the turbogenerator design because many details such as the turbine and generator efficiency, bearing losses and flows, mechanical losses, and generator cooling affect the process calculation.

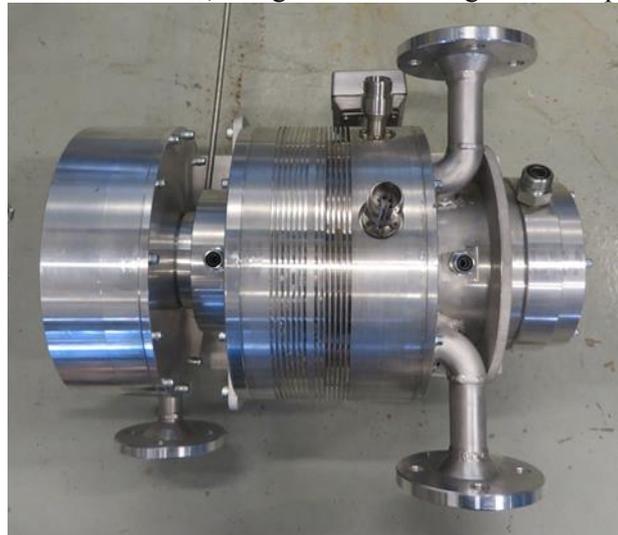


Figure 5. Turbogenerator. The turbine is located on the left, generator in the middle and pump on the right.

#### 3.3.1. Turbine

The designed turbine is a radial inflow turbine having a high pressure ratio over the turbine stator and rotor. The degree of reaction of less than 0.5 was adopted and thus, the fluid expands more in the stator vanes than in the rotor. The flow is supersonic at the stator outlet reaching the Mach number of about 2.1. The relative velocity at the rotor inlet was designed to be subsonic at the rotor inlet. The turbine design values are shown in Table 3 and the stator and the rotor of the turbine are shown in Figure 6. The turbine design methods are presented more in detail in [12].

Numerical modelling was carried out for five different supersonic stator geometries and three turbine rotors. The CFD calculations were carried out by using both  $k-\varepsilon$  and  $k-\omega$  SST turbulence models and the gas properties of MDM were implemented to the solver by using polynomial fittings for the superheated gas region. The numerical methods are described more in detail in [12]. The final stator and rotor geometries were selected based on the results of predicted flow angles and velocities, efficiencies, and pressure ratios obtained with the different designs. In addition, the mechanical design, vibrations and forces acting on the turbogenerator shaft were taken into account when selecting the final geometries for the setup.

Table 3. Turbine design values.

<b>Evaluated turbine efficiency</b>	76 %
<b>Specific speed</b>	0.49
<b>Specific diameter</b>	3.45
<b>Rotational speed</b>	31000 rpm
<b>Rotor diameter</b>	144 mm
<b>Stator blade height</b>	2 mm
<b>Mach number at the rotor inlet</b>	2.18
<b>Mach number at the rotor outlet</b>	0.7



Figure 6. ORC turbine stator and rotor.

#### 4. Measurements in the test facility

For evaluating the performance of the cycle, the temperatures and pressures at the turbine inlet and outlet (points 1 and 2 in Figure 3a) as well as the mass flow rate of liquid working fluid in the outlet pipe of the turbogenerator feed pump were measured (point 5 in Figure 3a). The mass flow rate of liquid working fluid was measured by oval gear flow meter the accuracy of which is  $\pm 0.5$  % of reading. The temperatures were measured by Pt100 temperature sensors and the pressures by pressure sensors including transmitters that have been connected to the data acquisition system. The electric power output of the ORC system was obtained from the frequency converter. The performance of the heat exchangers could be evaluated by temperature and pressure measurements in the process (points 1, 2, 4, 5 and 6 in Figure 3a) and by using the measured working fluid mass flow rate as well as by the temperature and pressure measurements of the exhaust gas and the temperature measurements of the cooling water to and from the condenser. In addition, the flow rate of the condenser cooling water was also measured by an oval gear flow meter.

#### 5. Experiences and results of the first test runs

The test runs with the ORC experimental setup were performed in two phases. First, the test runs were made without the turbogenerator and second, with the turbogenerator installed in the ORC experimental setup.

In the first test runs, the turbogenerator was removed from the process and the vapor coming from the evaporator was expanded through an expansion valve instead of the turbine. Thus, a separate electric-driven main feed pump in series with the pre-feed pump was used. The main idea was first to clean the process by circulating the working fluid through the filter and to adjust the process parameters. Also, the process components were tested and suitable test run procedures were developed to ensure a safe operation of the process before the use of the turbogenerator. In addition, the performance and operation of the heat exchangers at different heat loads were investigated and these results are reported in [15].

In the second phase, the turbogenerator was installed in the process. The test run with the turbogenerator showed that electric power can be extracted with this ORC system, and the maximum electric power output of about 5.1 kW was measured. The turbogenerator was first rotated using the

generator as an electric motor in order to achieve sufficient mass flow through the process and sufficient pressure at the evaporator and turbine inlet. The minimum rotational speed (12400 rpm) was used at the beginning and the evaporation of the working fluid was started at the corresponding rotational speed. The turbine was quickly accelerated to this rotational speed in order to avoid a wear of the bearings. At this point, the flow was bypassing the turbine and it was entering the turbine outlet pipe before being directed to the recuperator. As soon as there was working fluid vapor available the bypass valve was gradually closed and the turbine valve was gradually opened. There can be seen the peak of used power at this point (at 3000 s in Figure 7). This might be associated to the evaporation of the working fluid. When the bypass valve is open the flow was entering at the outlet of the turbine. It might generate forces to the turbine and therefore, there is a need for more power. However, the peak is quickly passed by when the turbine valve was opened (right after 3000 s in Figure 7).

At the minimum rotational speed the pressure at the turbine inlet and the mass flow were rather low and no power was yet produced. The consumed power was decreased quickly when the turbine valve was opened and bypass valve was closed. Gradually, the rotational speed of the turbogenerator was increased and correspondingly mass flow and pressure was increased which increased the power produced by the turbine.

At the rotational speed 14500 rpm, the turbine was generating more power than the pump and mechanical losses were consuming and the generator started to supply power to the grid (after 3500 s in Figure 7). The generated power increased as the rotational speed increased until maximum produced power of 5.1 kW was reached. At this operation point, the turbine was still operating at the off-design point in which the rotational speed and pressure ratio are lower than at the design point. Therefore, the electric power output reached in this operation point was lower than the design value of 10 kW used in the ORC experimental setup design. This can be partly explained by the elevated condenser pressure (see Figure 8) when compared to the design value of 0.03 bar (Table 1).

The rotational speed was further increased towards the design rotational speed (31 000 rpm) but the power generated by the turbogenerator decreased (Figure 7). However, at the same time condenser pressure (Figure 8) and mechanical losses increased which partly explains the lower rate of power.

In the test run, the condenser pressure was observed to be higher than the saturated pressure of MDM corresponding to the measured condenser temperature. This was estimated to be caused by the presence of small amounts of air in the process. There is also a possibility that the working fluid used in the test runs was not pure MDM but contained small amounts of other substances possibly having an influence on the condensing pressure.

In the future, the sealing of the process should be improved to prevent possible air leakages into the process and chemical analysis of the working fluid used in the ORC experimental setup should also be made. After improving the sealing of the process, further test runs will be carried out and a detailed analysis on the turbogenerator losses including the turbine, generator, main feed pump as well as the bearings will be done in order to estimate the chances to increase the electric power output of the ORC system.

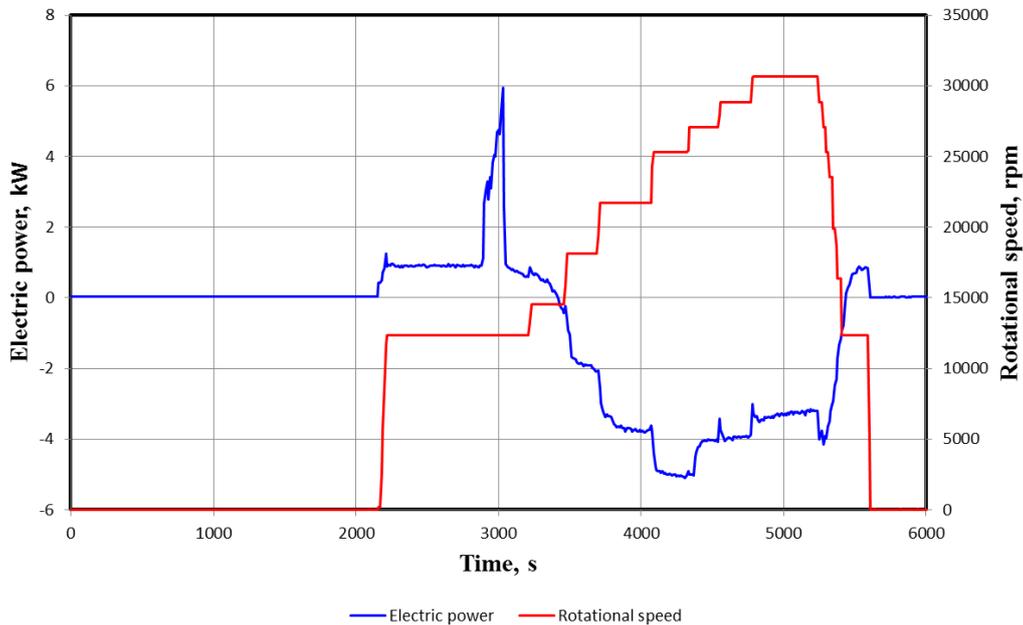


Figure 7. Measured rotational speed and electric power. Positive power means power consumed and negative power means generated power.

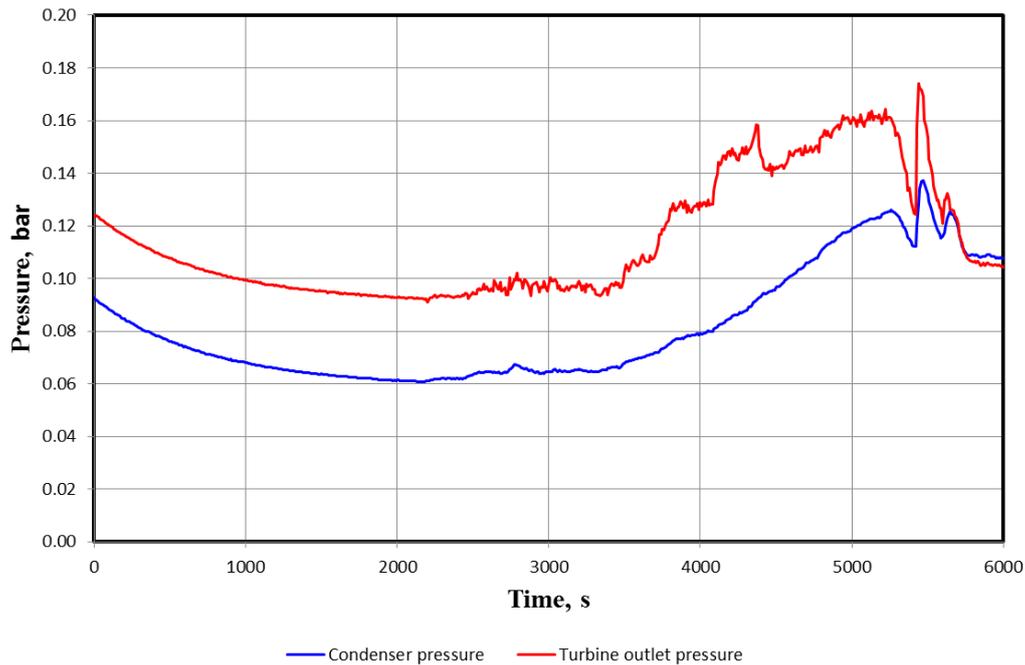


Figure 8. Measured pressure at the turbine outlet and in the condenser

## 6. Conclusions and discussion

Preliminary results from the micro-ORC test facility located in the Laboratory of Fluid Dynamics at Lappeenranta University of Technology are presented. The test facility is utilizing the exhaust gas of the diesel generator. The working fluid used in the test facility is siloxane MDM and the turbogenerator consisting of the radial inflow turbine, generator and pump is designed, manufactured and tested. The process was identified to be able to produce electricity from the diesel generator exhaust gas and the turbogenerator was confirmed to be mechanically functional.

The design operation point of the ORC was not reached because the condenser pressure low enough was not achieved. This was estimated to be due to small leakages of air into the process and the presence of non-condensable gases in the condenser. The maximum electrical power 5.1 kW was achieved.

The leakages of air into the process will be minimized in the future and the design point of the process will be measured. In addition, more detailed measurements in the turbine will be made in order to study flow phenomena occurring in the stator and rotor. The detailed study of losses in the turbogenerator will also be done.

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