START-UP RESEARCH ON THE LABORATORY MICRO CHP ORC TEST STAND

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ABSTRACT

The paper presents the construction and the first results of analyzes performed on the newly built micro CHP ORC power plant test stand. The research covers initial start-up tests. This test stand has been built at the Institute of Fluid-Flow Machinery, Polish Academy of Sciences in Gdansk, Poland. It is a universal test stand where it is possible to examine among others micro turbines of various designs operating on a low boiling agent. The test stand is simulating Micro CHP based on ORC technology cogeneration unit producing electricity and heat as hot water up to 55°C (design conditions) applicable for low temperature heating (floor or wall heating systems). The supersonic turbine with a capacity of up to 3 [kW] was installed on the test stand. The HFE 7100 is a working medium in this system. This article briefly describes the construction of laboratory test stand, it is composed of turbine, pump, heat exchangers, regenerative heat exchanger and a set of sensors. The paper presents the results of experimental studies, for example graphs showing variations in temperature and in pressure at various measurement points on the test stand. The article also includes graphs of power generated by ORC turbine as a function of available pressure difference.

1. INTRODUCTION

There is a significant increase in the use of working fluids other than water in power plants and CHP systems, especially if the source temperature is lower than in traditional steam power plants. Different aspects of ORC systems including various heat sources (solar energy, geothermal fluids, waste heat), critical plant components (heat exchangers, expanders), thermodynamic cycle optimization and proper fluid selection were discussed in the literature (Angelino *et al.,* 1984). Different fluids for ORC systems are analyzed, some of them are pure fluids, mainly refrigerants (Maizza and Maizza, 2001, Borsukiewicz-Gozdur and Nowak, 2007) and some are mixtures of fluids (Angelino and Colonna, 1998, Bao and Zhao, 2013).

One can also observe the increasing number of papers dedicated to different aspects of turbines for ORC systems. Micro CHP systems requires small components. Literature indicates that the use of high-speed turbogenerator makes the ORC small, simple, hermetic and reduces significantly the maintenance expenses (Larjola , 1995). Different work summarized the findings of the marked research for the expanders, including turbines, and discussed the selection and choices of the expanders for the ORC-based micro-CHP systems (Qiu *et al.,* 2011). Numerical simulation of a radial turbine designed to work in a system based on a toluene cycle was presented by Harinck *et al.* (2011). According to the numerical model the turbine efficiency for the design point is equal to about 70%. Small ORC high-speed turbogenerators with siloxans as working fluids were also investigated (Uusitalo *et al.*2011). There are also some works dedicated to complete ORC CHP systems and its various types. Qiu *et al.* (2012) presented data from experiments with ORC for micro CHP and biomass boiler and HFE7000 working fluid in vane expander . Some construction, dynamic tests and experimental characterization of micro-scale ORC are reported by others (Pei *at al*, 2011, Peris *et al*, 2015). The performance potential of cycle modifications to the basic ORC was illustrated by Lecompte *et al.* (2015). Authors analyzed and discussed various types of cycles (transcritical, trilateral, cascade cycles and other) and identified potential future development, knowledge gaps and indicated the lack of experimental data in the subject.

The idea of combined production of heat and electricity on a small scale using ORC systems based on biomass is analyzed in the Institute for quite some time. Main works were focused on selection of a proper working fluid (Mikielewicz *et al.*, 2007), optimal working parameters (Mikielewicz *et al.*, 2006, Mikielewicz and Bykuć, 2006) and overall thermo dynamical cycle (Mikielewicz *et al.*, 2013). Some experimental works on first version of a test stand were carried out with ORC system and turbine measurements (Kaczmarczyk e*t al*., 2013). The dynamic aspects of microturbines were also analyzed (Kiciński and Żywica, 2014). The latest works relate to energy storage techniques for ORC CHP system (Bogucka-Bykuć et al., 2014).

This paper presents the results of initial measurements carried out at the newly built testing rig for investigation of analysis of ORC systems performance. The goal of the experiments was neither the optimization of the system performance, nor the achievement of highest efficiencies. It was the dynamics of the phenomena occurring in the system, with the special interest and consideration of the start-up phase of the high-speed rotating turbine which was investigated. The characteristics of installation of the operation variant of the ORC with regenerative heat exchanger were determined. The entire system operating with a low-boiling agent HFE 7100 with various service conditions was tested.

2. THE LABORATORY TEST STAND CHARACTERISTICS

The test stand is built as an integrated mobile ORC system equipped with set of sensors and automatic control system (Breńkacz *et al.*, 2012). All components are placed on a supporting structure. Biomass fuel is used as a source of heat.. The construction of test rig is allows for easy reconfiguration to connect different micro turbines, pumps, heat exchangers, valves, etc.

The 3D model was created in Autodesk Inventor Program. The view of this model as well as view of actual test rig are shown below on Fig. 1. In the described configuration a single-stage supersonic microturbine, developed (Kozanecki *et al.*, 2012a and 2012b) and tested (Bykuć *et al*, 2014, Kiciński *et al*, 2012) in IMP PAN was installed.

The laboratory test stand works like normal CHP system. Working medium (HFE 7100) is pumped through the circulation pump and regenerative heat exchanger to the evaporator, where evaporation and heating at specific and controlled parameters take place. In the next step it goes to a microturbine where its energy is converted into rotational motion of the rotor coupled to electric generator. The working medium expand in the turbine and then it is directed to regenerative heat exchanger, where it gives off some heat to preheat the liquid medium that is directed to an evaporator. Then the medium goes to the condenser where it changes phase. Water is used as a source of cooling. The working medium from the condenser flows into the tank, and next to the circulation pump, thereby closing the cycle. The laboratory test stand is equipped with sensors for measurements of temperature, mass flow rate, pressure, rotational speed and an appropriate set of safety sensors for correct work of the high-speed turbine.

Bearings in turbine use a vapor of the same working medium - HFE 7100 (HFE-7100 – Essential characteristics, 2006). The entire installation is a hermetically sealed structure.

Figure 1: 3D model and view of laboratory test stand

The basic elements of the supersonic microturbine are single-stage radial impeller, electric generator with rare-earth magnets and a set of aerostatic bearings. All components are sealed in the body with a water jacket that cools the generator.

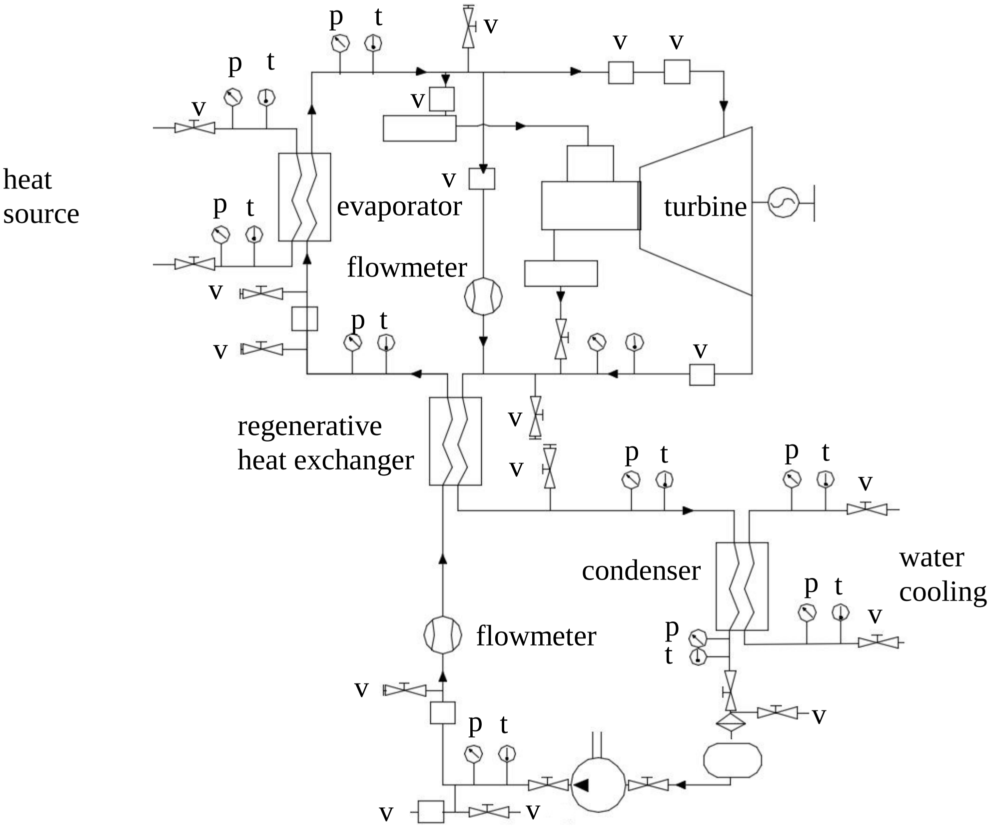

Figure 2: Schematic view of research test stand. v- valve, p – pressure measurement, t – temperature measurement

v

3. RESEARCH RESULTS

The testing rig is controlled from the level of one control panel. Fig. 3 presents a screenshot of control panel used to steer the testing rig. The most important points of regulation (i.e. EZP1, EZP2, EZP3, EZO1, EZO2 valves) as well as pressure, temperature and flow measurement points in the installation are marked. EZP1 and EZP2 valves regulating flow via bypass installation as well as via turbine play the most important role in terms of the regulation of the whole installation. Appropriate settings of those valves enable achievement of a proper pressure value before turbine, proper running of a process of heating up of gas bearings and turbine as well as enable nominal speed to be reached by the turbine.

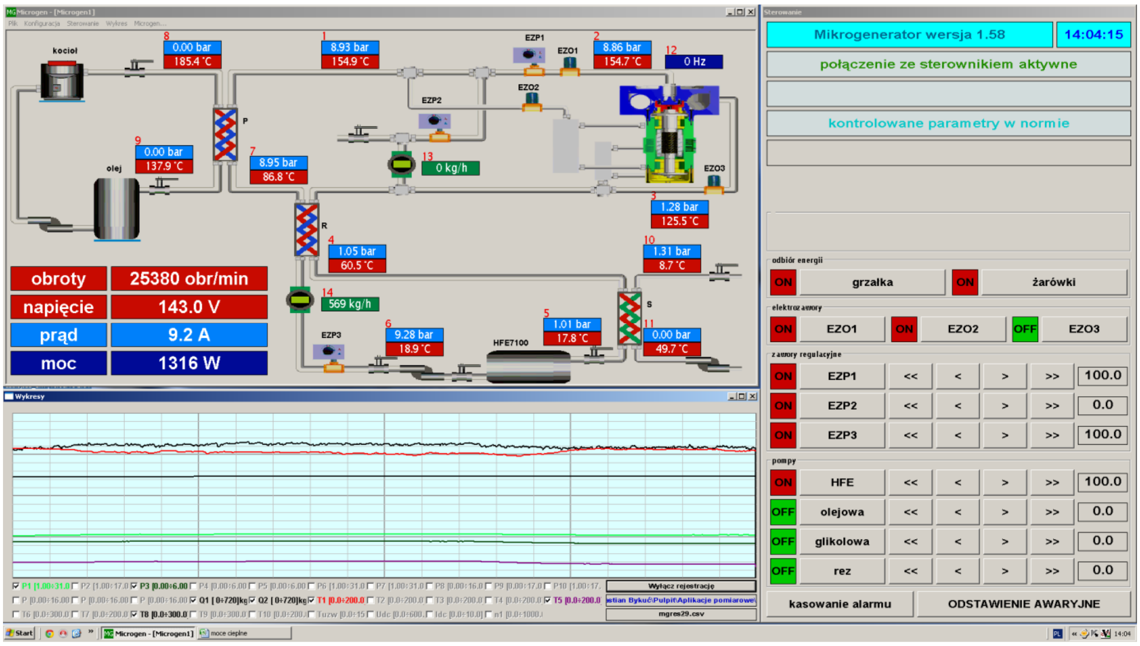

Figure 3: Control panel of the test stand

EZO1 and EZO2 valves are quick-closing valves and cut off inflow of working fluid into the turbine and bearings. Due to the application of slide bearings in the turbogenerator, working on the vapor of the working fluid, it is very important to prevent condensation of working agent in bearings and to maintain the proper flow during work performed with high speed. Therefore, the following is of great importance: a precise control of the temperature of the working agent at the inlet, assurance that work is performed in the area of superheated vapor and maintenance of the adequate pressure difference before and after turbine. An automatic system, set from the operator's level, cutting off inflow of fresh steam into the turbine - if any of the critical conditions are exceeded - is installed in the tested installation for safety reasons.

Graphs below exhibit the results of one of the series of measurements and start when turbine and its bearings are already fed (EZO2 valve is opened), turbine is heated up and rotates with speed of ca. 2000 rpm (EZP1 valve is partially opened). The process of acceleration of the turbine is divided into three stages depicted in the below figures. First stage (Phase I) refers to interval of 0÷30 sec., second one (Phase II) refers to 30÷240 sec. and the last selected stage (Phase III) refers to 240÷970 sec. Those stages reflect settings of regulation valves EZP1 and EZP2.

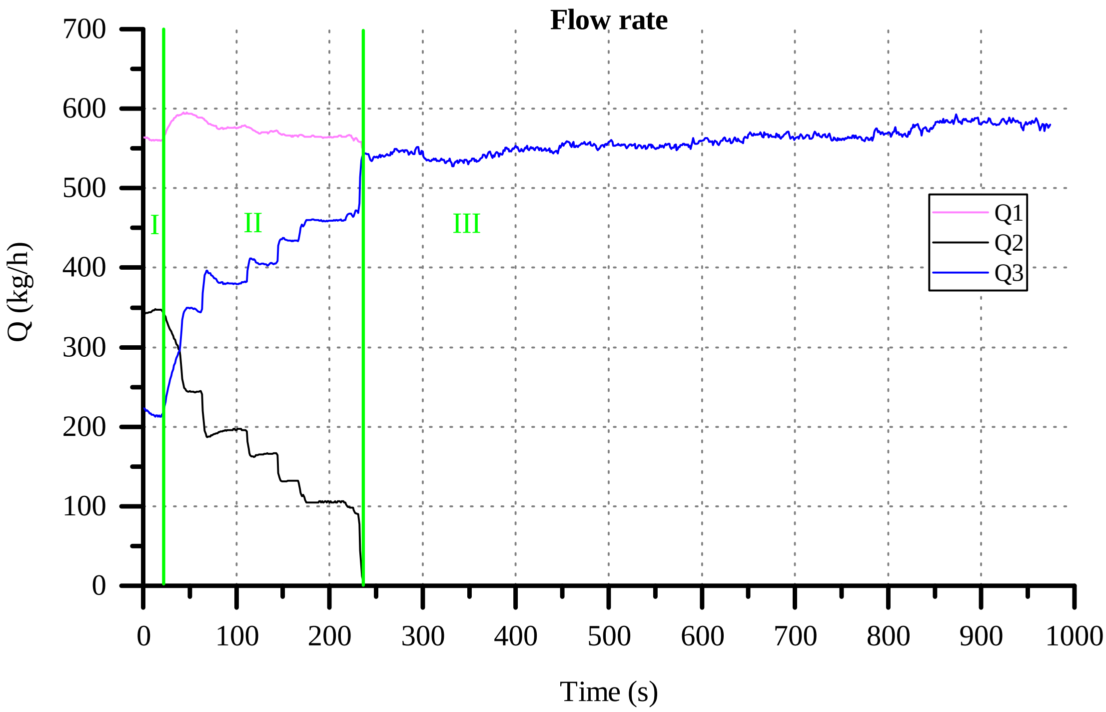

Figure 4: Mass flow rates (Q1- total flow rate, Q2-by-pass flow rate, Q3-turbine flow rate)

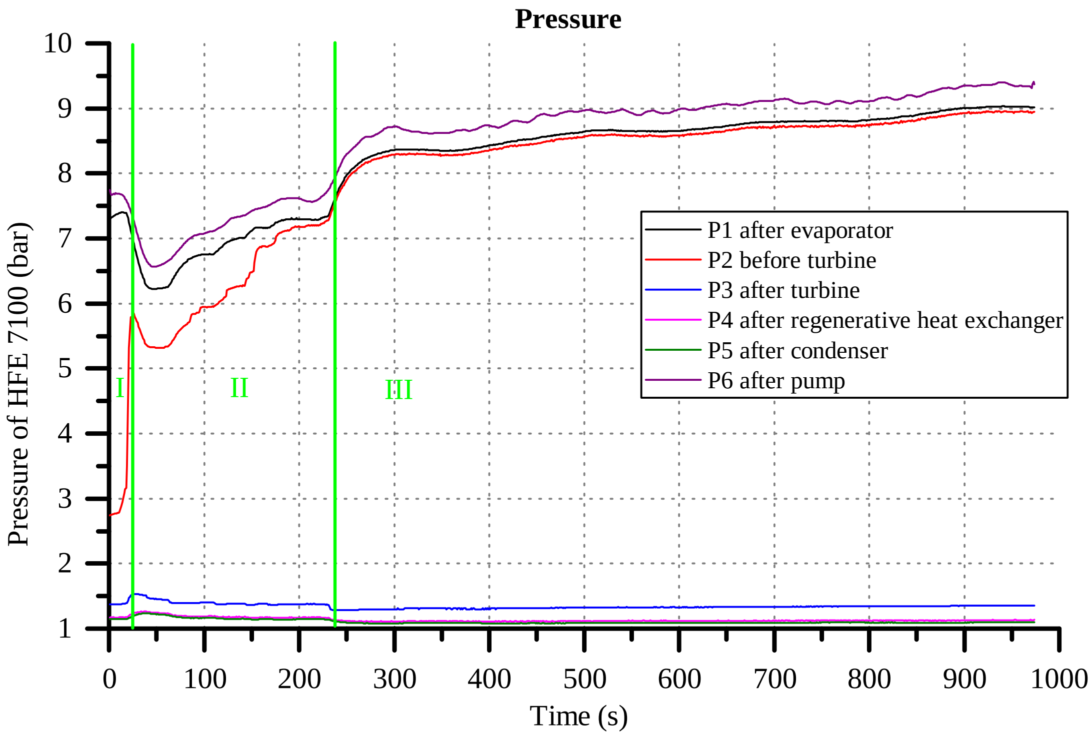

Figure 5: Pressure curves in the time domain obtained during startup

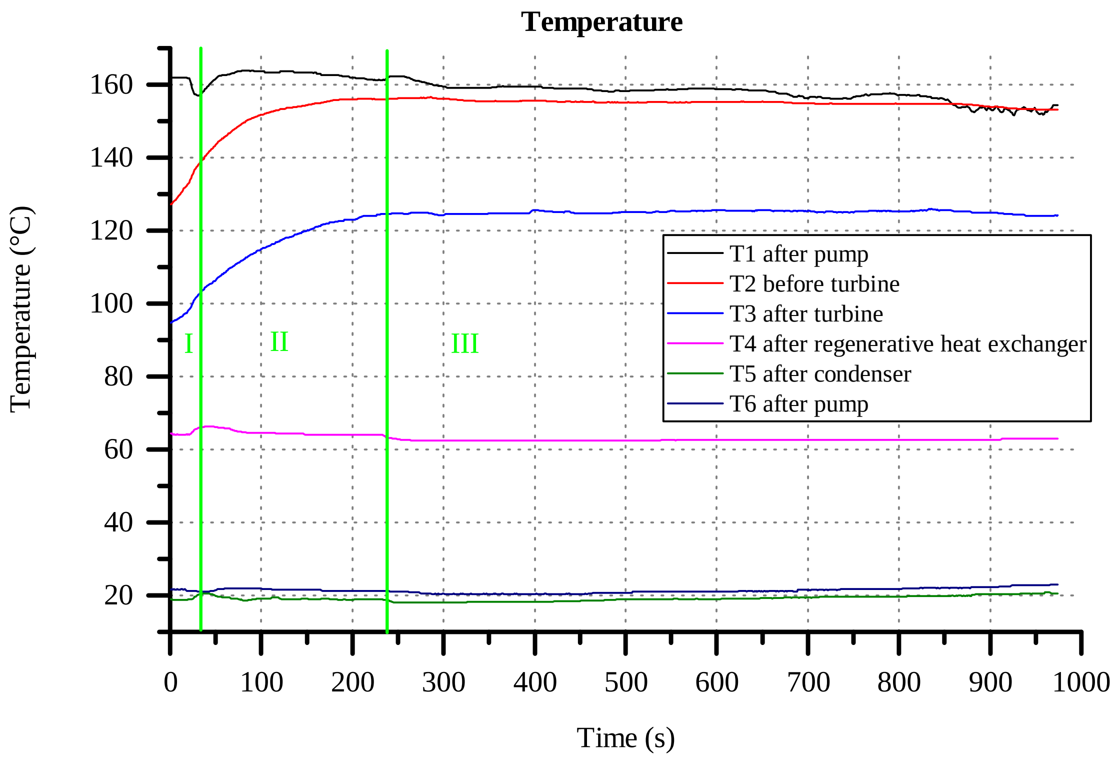

Figure 6: Temperature curves in the time domain obtained during startup

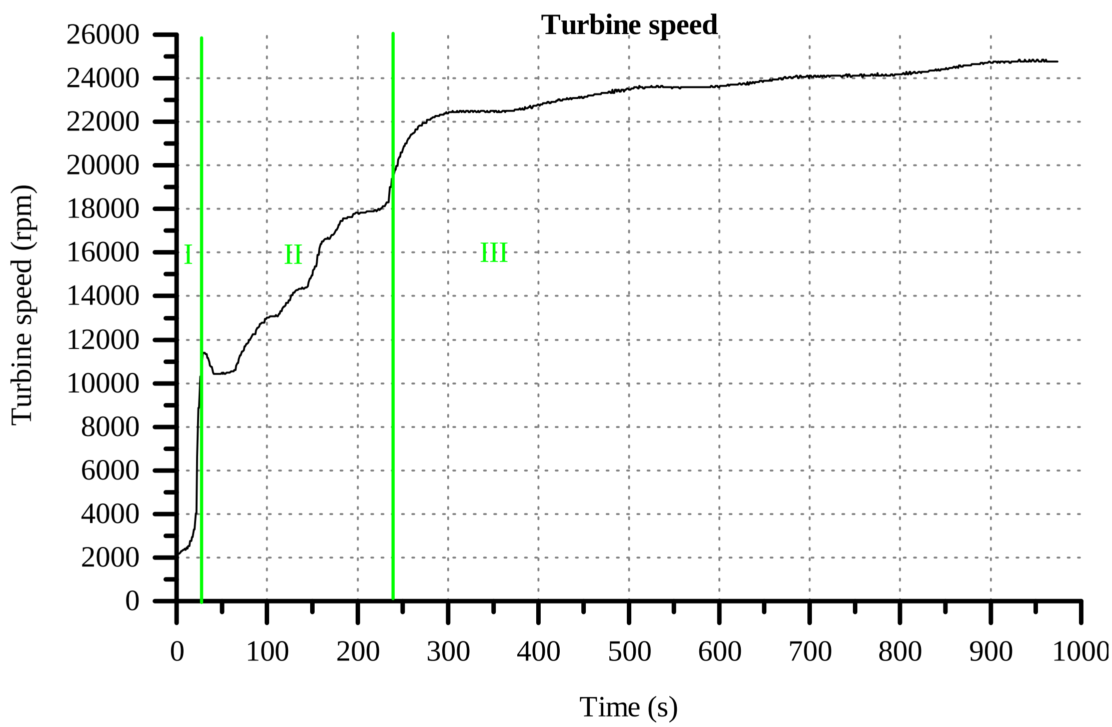

Figure 7: Rotational speed of turbogenerator in the time domain obtained during startup

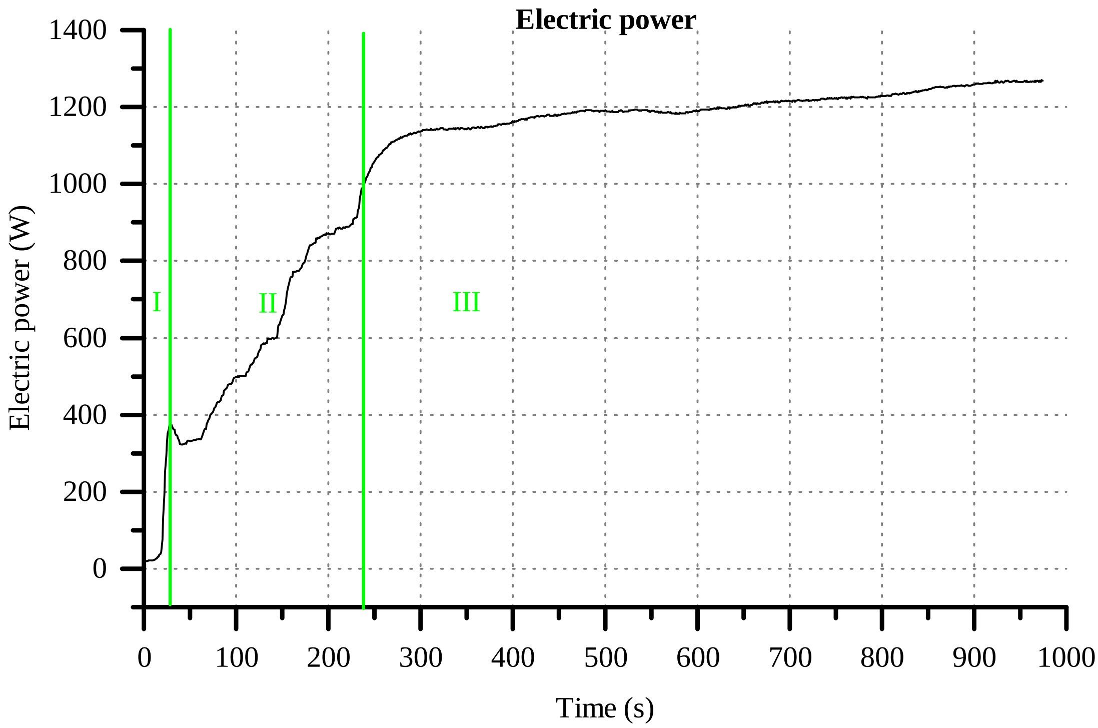

Figure 8: Power generated in the time domain obtained during startup

Fig. 4 presents the plots of the working fluid flows: total flow, bypass flow as well as calculated turbine flow. Placing the flow meters as presented it is possible to calculate the flow through the turbine without generation of additional pressure drops, decreasing the efficiency of the system. The unstable total flow of the working fluid stems from the settings of the inverter, the heating conditions, the cooling conditions as well as from the need of maintenance of the proper level of the working fluid in the tank.

As can be seen in case of Phase I, only 30 % of the total flow reaches turbine (EZP1 valve is only partially opened) and the remaining part flows via bypass. During this Phase, pressures are held at more or less constant level (Fig. 5). The pressure drop (ca. 0,5 bar) can be observed along the distance between pump and outflow from the evaporator. Due to throttling of the steam flow via EZP1 valve, pressure before turbine is ca. 4,5 bar smaller when compared to pressure in the outflow from evaporator.

The pressure drop (ca. 2,5 bar) on the regenerator after the turbine can be also observed. During this phase, the expansion in the turbine drops from ca. 2,7 bar and reaches 1,5 bar which makes it possible for the turbine to achieve speed of ca. 2000 rpm (Fig. 7) and generation of the trace amount of energy at the level of 20W (Fig. 8). During the Phase I, a small increase of temperature before and after turbine can be observed, which may be a sign of a turbine system not being enough heated up (Fig. 6).

During Phase II , the settings of the EZP1 and EZP2 valves were changed with the aim of gradual increase of vapor flow through the turbine, and simultaneous decrease of the flow via bypass. During this measurement series, Phase II lasted ca. 200 sec. and took place within few steps. This Phase is ended up with a total closure of the EZP2 valve at the bypass and opening of the EZP1 valve The gradual opening of the EZP1 valve and throttling of the EZP2 valve lead to the step increase of the flow via turbine flow system, whereas total mass flow was held almost constant (Fig. 4). At the beginning of Phase II, a rapid pressure drop of the steam (Fig. 5) in the system can be observed, due to the increase of the flow through the turbine. Simultaneously, the rotation speed of the turbine increases in steps from 2000 rpm up to ca. 11000 rpm during a few seconds (Fig. 7) and electric power generated increases to the level of ca. 380 W.

Next, the pressure starts to increase gradually up to the initial level at the end of the Phase II. Simultaneously a step increase of the rotation speed of the turbine is observed up to the level of

18000 rpm and generated power – up to ca. 1100W. The finalization of the Phase II due to the total closure of the bypass valve leads to the step increase of the flow through the turbine and this time also the increase of the pressure at the inflow to the turbine, and another increase of the rotation speed of the turbine as well as generated power.

Temperatures during this Phase are stable, however temperatures at the inflow and outflow from turbine are increasing in a uniform manner during the whole period, until they stabilize at the end of Phase II (Fig. 6). It is of major importance, from the safety point of view with respect to the performance of the high-rotation speed turbogenerator, to maintain the proper temperature and superheat of steam at the inflow to the bearings.

Phase III is initiated by the total cut off of the bypass and directing the total vapor flow through the turbine (Fig. 4). Since this moment, the increase of the pressure, the rotation speed and power of the turbogenerator result from the step increase of the rotation speed of the pump as well as slight increase of the temperature of the heat source.

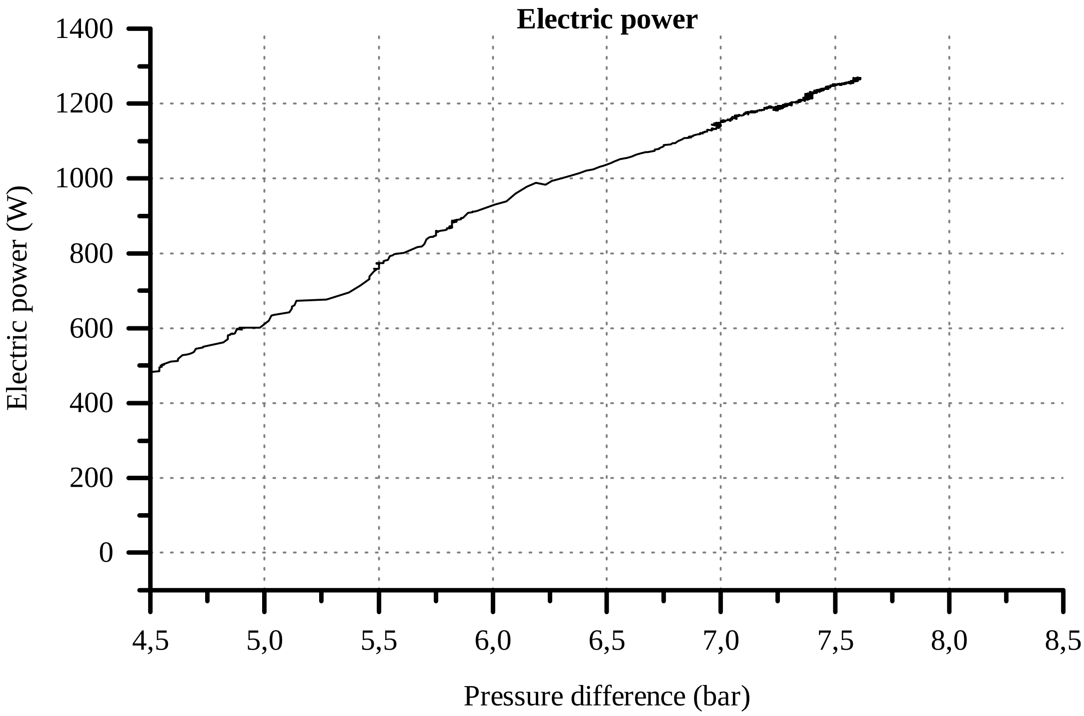

Figure 9: Power generated vs. available pressure difference between inlet and outlet of a turbine

During the work the linear increase of the power of the turbogenerator as a function of the increasing pressure difference was observed, as was expected (Fig. 9). The most important issue is the maintenance of the properly low level of the pressure in the condenser, i.e. assurance of the proper temperature of the cooling fluid, which may be a challenge.

4. SUMMARY AND CONCLUSION

The results of initial measurements carried out at the newly built testing rig for investigation of ORC systems performance were presented and analyzed. The goal of the experiments was the dynamics of the phenomena occurring in the system, with the special interest and consideration of the start-up phase of the high-speed rotating turbine. Curves of selected measures were obtained and it appeared that during the startup of turbine three characteristic phases may be shown. At the end of the measurement series under consideration the results shown below were obtained. The power of 1316 W was obtained on the terminals of the generator at the rotation speed of the turbine equal to 25380 rpm, pressure at the inflow to the turbine of 8,86 bar and temperature of ca. 155 °C. The expansion in the turbine occurred down to 1,28 bar and the flow of the working fluid was at the level of 569 kg/h. Optimization of the system performance was not the goal of executed works and values achieved are just exemplary. The results obtained will be used for formulating algorithms of automatic startup and working procedures of the laboratory test stand described.

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ACKNOWLEDGMENTS

The research work presented in this article was supported by the scientific project POIG.01.01.02-00-016/08 "Model agroenergy complexes as an example of distributed cogeneration based on local renewable energy sources".