

Article

Domestic Organic Rankine Cycle-Based Cogeneration Systems as a Way to Reduce Dust Emissions in Municipal Heating

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Abstract: Environmental issues are nowadays of great importance. In particular air and water quality should be kept at as high levels as possible. Energy conversion systems and devices which are applied for converting the chemical energy contained in different fuels into heat, electricity and cold in the industry and housing are sources of different gases and solid particle emissions. Medical data show $PM_{2.5}$ dust in particular is highly dangerous for human health. Therefore, limiting the number of low-quality fuel combustion processes is a key issue of modern energy policy. Statistical data show that domestic heating systems account for a large share of the total emissions of $PM_{2.5}$ and PM_{10} dust. For example in Poland in 2017, the share of households in the total annual emissions of $PM_{2.5}$ dust was equal to ca. 35.8%, while the share of $PM_{2.5}$ emission in industry (i.e., power generating plants, industrial power plants and technologies) was equal to only 23.6%. A possible way of solving this problem is by the successful replacement of old domestic furnaces by combined heat and power (CHP) or multigeneration boilers which can be used for heating the rooms and sanitary water and generating electricity and cold. Such systems can possibly contribute in the future to significant reductions of dust emissions and air pollution in urban and rural areas by limiting the number of low-quality fuel combustion processes. This article presents design considerations and experimental results related to a domestic micro-CHP unit which is based on organic Rankine cycle (ORC) technology. The main aim of the design works and experiments was therefore the analysis of the possibility of integrating the ORC system with a standard domestic central heating gas-fired boiler. The specially designed micro-ORC system was implemented in the laboratory and experiments were performed using this test stand. The main design aims of the test-stand were: low operating pressure, small working fluid flow, low price and compact dimensions. To meet these aims, volumetric machines were chosen as the expander and working fluid pump. The experimental results were positive and show that it is possible to integrate an ORC system with a standard domestic central heating gas boiler. For different heat source temperatures, the obtained expander power ranged from 109 W to 241 W and the thermodynamic cycle efficiency ranged from 4.3% to 8.8%. These positive research results were achieved partly thanks to the positive features of the different system subassemblies.

Keywords: air pollution; $PM_{2.5}$ dust; ORC; working fluid; selection method; volumetric expander; thermodynamic analysis

1. Introduction

One of the most important issues of present times is preventing excessive air pollution. Different industrial and domestic energy conversion systems and devices which are applied for converting the chemical energy contained in different fuels into heat, electricity and cold are sources of different gases and solid particle emissions. Especially in the countries (such as e.g., China, India, USA, RPA,

Japan, Poland) whose power sector and industry are mainly based on the use of different fossil fuels (i.e., coal, coke, furnace oil or biomass) for the heat and electricity generation limiting these emissions is of great importance.

As the result of solid fuels combustion, in addition to the emission of gaseous pollutants (i.e., carbon, sulfur and nitrogen oxides), solid particles of different sizes are contained in the flue gases and emitted into the atmosphere in the form of dust. This problem became especially visible since many new measuring stations were installed around the world. Recently, the issue of micron particle sized (i.e., PM₁₀ and PM_{2.5}) dust emissions has become particularly widely discussed in the media and the professional articles by many officials and researchers due to significant, negative impact of these dust on human health. PM_{2.5} dust (i.e., dust featuring a particle size smaller than 2.5 µm) is especially hazardous to human health as its microparticles are able to penetrate the lungs, where they are dissolved in biological fluids and carried via the bloodstream through the whole body, causing different health issues (e.g., respiratory reactions). The importance of this issue is raised in numerous scientific studies on dust emissions into the atmosphere [1–12] and their impact on human health and the environment [13–17].

Statistical data on dust emission is published by many governmental and private agencies, for example by the statistical office of the European Union (Eurostat) and Statistics Poland (GUS). Table 1 summarizes statistical data on PM_{2.5} dust emission in the EU-28 countries in 2017 [18].

Table 1. Statistical data on PM_{2.5} dust emissions in EU-28 countries in 2017 [18].

Source of Dust Emission	Emission Tonnes
Energy production and distribution	49,716
Energy use in industry	108,887
Road transport	142,621
Non-road transport	32,085
Commercial, institutional and households	732,863
Industrial processes and product use	142,801
Agriculture	44,137
Waste	50,773
Other	485
Total sectors of emissions for the national territory	1,304,368

Table 2 summarizes the data on dust emissions in 2017 from various branches of the economy in Poland [19].

Table 2. The data on PM_{2.5} dust emissions in 2017 from various branches of the economy in Poland [19].

Source of Dust Emission	Emission Tonnes
Power generating plants	10,400
Industrial power plants	35,500
Industrial technologies	34,700
Households	122,000
Other stationary sources (local boiler plants, trade, workshops, agriculture and others)	102,600
Mobile sources	35,500
Total	340,700

The data on the PM_{2.5} emissions reported in Table 1 and in Table 2 show that in EU-28 countries and in Poland households have the largest share of PM_{2.5} dust emissions. The households share of PM_{2.5} dust emissions in Poland reached ca. 45% of the total emissions in 2017. Statistical data [18,19] show that other stationary installations are the second and transport is the third main source of PM₁₀ and PM_{2.5} dust emissions. For example in Poland, it was reported in 2017, [19] that emissions of

these dusts from road transport, other vehicles and equipment (including rail transport) was equal to ca. 10% of total national emissions. It was also reported that a significant part of these emissions came from abrasion of tires and brakes and abrasion of road surfaces rather than fuel combustion [19]. The same data sources [18,19], show that the share of dust emission from the professional commercial and industrial installations which are commonly equipped with modern and efficient flue gas cleaning systems (such as for example electrostatic precipitators) is much smaller than in the case of other sectors of the economy.

Significant air pollution with dust is clearly visible in the autumn, winter and spring in rural and urban areas in which furnaces and boilers fed by low-quality fuels, such as wood, humid coal and other solid fuels are often used for domestic heating. The efficiency of these furnaces is low and usually ranges from 56% up to 70% [20], which additionally results in the unfavorable ratio of the amount of burned fuel to the heating effect, thereby increasing the amount of emitted flue gases. In large cities, the problem of dust emission is visible mostly in the areas of old, large housing estates consisting of buildings, which in most cases have not been properly insulated and whose residents use low-quality furnaces for heating the flats. Such settlements are, for example, Biskupin and Sępólno in Wrocław (Poland), which were built in 1930s and are composed of one floor brick blocks and single-family buildings.

Solid fuels which are usually combusted in domestic furnaces are featuring significant share of volatile matter. Table 3 reports the share of volatile matter (V^{daf}), and calorific values (Q_{wr}) for the solid fuels which are commonly combusted in the domestic furnaces [21]. Natural gas and heating oil were added to this Table for comparison.

Table 3. The share of volatile matter (V^{daf}), and calorific values (Q_{wr}) for the solid fuels which are commonly combusted in the domestic furnaces [21].

Fuel Type	V^{daf} %	Q_{wr} MJ/kg
Flame coal	30	25.0
Wheat	63–78	14.3–15.2
Waste wood	55–81	13.0
Natural gas	100	43.0
Heating oil	0	39.7–42.7

It was reported in [19] that in Poland, the average concentration of $\text{PM}_{2.5}$ dust in air was in 2018 equal to $22 \mu\text{g}/\text{m}^3$. This concentration was 10% above the $20 \mu\text{g}/\text{m}^3$ concentration which Poland should meet in 2015 and $4 \mu\text{g}/\text{m}^3$ above the national $\text{PM}_{2.5}$ reduction target which was set by the government to $18 \mu\text{g}/\text{m}^3$ and should be achieved by 2020. The highest values of the average concentration of $\text{PM}_{2.5}$ dust were observed in 2018 in Upper Silesian and Kraków agglomerations ($31 \mu\text{g}/\text{m}^3$), Rybnik-Jastrzębska agglomeration and Bielsko-Biała ($30 \mu\text{g}/\text{m}^3$) as well as in Częstochowa ($27 \mu\text{g}/\text{m}^3$) and Legnica ($25 \mu\text{g}/\text{m}^3$). These are agglomerations and cities having above 100,000 inhabitants. The highest values of emissions (for both PM_{10} and $\text{PM}_{2.5}$ dust) were recorded in 2017 in Central Europe (i.e., in Poland, Bulgaria, Croatia, Slovenia, Romania and Hungary) as well as in Italy and Cyprus.

In large agglomerations of cities, settlements and villages reducing the dust emissions from heating devices is not an easy task. One possible solution is connecting the houses, settlements and villages to the municipal heating network. However, sometimes it is not possible for various reasons (e.g., field restrictions, lack of space for heat exchangers in buildings, social issues, etc.). For the companies that are producers and distributors of network heat, such an investment is often not economically justified due to the above-mentioned problems, installation costs and a limited number (density) of potential heat recipients in a given area.

Currently, different municipal authorities in Poland and other countries are implementing co-financing programs focused on modernization of domestic heating systems by replacement of the

old furnaces by modern heating devices (such as e.g., heat pumps or gas boilers). There are many devices of this type available on the market. These devices are featuring different technical parameters depending on the manufacturer and the design. For example, domestic gas boilers can be classified by the design of the combustion chamber [22]. Boilers with a closed combustion, boilers with an open combustion chamber and condensing boilers are applied in the domestic heating. Condensing boilers are featuring the highest energy conversion efficiency.

One of the possible methods of reducing the dust emission from domestic heating into the environment may be the application of combined heat and power (CHP) or multigeneration systems specially designed for domestic application. These systems can be installed as replacements of the currently applied old low-efficient boilers. This solution can in effect result in limiting the number of combustion processes and emissions of harmful substances into the atmosphere.

Domestic CHP system can be successfully designed and implemented with the application of technologies that are currently available on the market such as photovoltaic (PV) cells combined with heat pumps or heating systems based on electric heaters [23]. However, the power output of PV systems is usually limited by the availability of space required for their installation (e.g., the roof surface). The multigeneration system can be implemented for example by a coupling set of PV cells, heat pump and air conditioning unit [24]. Domestic boilers which are currently available on the market are fed by gas, oil or ecological solid fuels and are manufactured as heating devices, not CHPs. To the knowledge of the author there are currently no cogeneration or multi-generation domestic boilers commercially available. Domestic CHP or multigeneration boilers are promising alternative to other domestic power technologies, especially if the system power output is taken into account as the assessment criterion. Thanks to the high temperature of the heat source obtained by the fuel combustion it is possible to implement the vapour power plant cycle (i.e., Clausius Rankine cycle) in micro scale using specially designed machines and devices. Therefore, the system power per unit size can be higher than in the case of alternative technologies (e.g., PV cells) and the investment costs can be lower.

The successful implementation of domestic CHP systems or multigeneration boilers which can be used for heating the rooms and sanitary water and generating electricity and cold can possibly contribute in the future to significant reduction of the amount of low-quality furnaces which are currently used for heating purposes, and therefore significant reduction of the dust emission and air pollution in urban and rural areas. Moreover, CHP or multigeneration boilers can operate in combination with other devices (i.e., PV cells, heat pumps, air conditioners and others).

Research and development studies on this type of domestic power systems are carried out in different research and development units around the world [25], including the Department of Thermodynamics and Renewable Energy Sources of Wrocław University of Science and Technology. The integration of a gas central heating boiler with a power generating unit is usually done by redesigning the boiler into a home CHP micro power plant. This way a CHP unit featuring the electrical power output of ca. 2 kW_e and thermal power output of 24 kW_t can be implemented. Depending on the size of the system and consumer's energy needs such systems can be sufficient to cover the energy needs of smaller properties or can be treated as additional energy systems which can be used in combination with other technologies such as for example fuel cells or PV cells. The operating principle of these power generation systems is the same as in the case of classic steam power plants which are operating according to Clausius-Rankine (C-R) cycle. Different working fluids (i.e., water and low boiling substances) can be applied in these systems. Often, organic Rankine cycle (ORC) technology (i.e., a power system which uses low-boiling working fluid instead of water and operates according to the modified cycle of a classic steam power plant) is selected to be applied in power generating unit. The ORC system must be properly designed and miniaturized for this purpose, which is a significant scientific, design and construction challenge because the thermal phenomena and issues related to large and micro scale power systems are different. Proper selection of the working fluid and expander to such a system is of great importance [26]. Due to the low power of the domestic system, low flow

rates of working fluid and low range of operating pressures are expected in the system. Therefore, in addition to axial or radial micro-turbines, it is possible to use much simpler and cheaper volumetric expanders (such as multi-vane, scroll, screw, lobe or piston expanders). These issues are treated in more detail in the following part of this paper together with design considerations and results of experiments which were proceeded at the Wrocław University of Science and Technology using the in-house designed and made prototype of domestic CHP ORC plant.

2. Design Considerations and Comparison of Different Domestic CHP Systems

Domestic CHP systems can be based on different technologies. Gas microturbines, fuel cells, photovoltaic cells, Stirling engines, small gas engines, steam micro power plants and ORC micro power plants are currently considered as possible technologies that can be applied in domestic energy conversion systems. Research on these types of cogeneration systems is carried out by various researchers around the world and these technologies are currently at different stages of development. PV cells are for example successfully applied in domestic conditions, however they are further being intensively developed. Present works are aimed at maximizing the power obtained from the PV panel per unit surface and energy conversion efficiency. New materials (such as e.g., perovskites) are considered as promising for application in PV panels [23]. Issues related to research and development works related to the application of fuel cells in domestic conditions are presented in [27,28]. Natural gas is considered as fuel for domestic fuel cells. Recently, research and development works have been carried out on the domestic CHP systems based on gas microturbines. The design of the domestic CHP system employing microturbine is similar to the design of the domestic gas boilers. Such systems are prototyped and presented in [29]. Small Stirling engines [30] are also developed to meet domestic operational conditions. Since some time domestic vapour power plants implementing Clausius-Rankine cycle and using different working fluids are developed and tested [31]. They are treated in more details in other sections of this article.

Domestic CHP systems should be simple in the design, easy to use and install, fully automated and their investment and service costs should be low. Not in the case of all of the above described systems these features can be satisfied. Table 4 shows a comparison of investment costs (per kW of power output) for different energy conversion technologies that can be applied in/as domestic CHP systems.

Table 4. A comparison of investment costs (per kW of power output) for different energy conversion technologies that can be applied in/as domestic CHP systems.

Technology Type	Investment Costs EUR/kW	Overall Efficiency %	Electrical Efficiency %	Electricity/Heat Supply Ratio kW_e/kW_t	Part-Load Flexibility	Ref.
PV cell	1700	-	14–16	5/-	Yes	[32]
Fuel cell	890	70–92	37–55	0.75/0.9–30.8	No	[33]
Heat pump	350	4–5	-	-/30	Yes	[34]
ORC system	1000	90	10	2.5–10/11–44	Yes	[25]
Micro CHP gas turbine	360	90	26	30/30	No	[35]
Stirling engine	2200	90	15	2–10/4–35	Yes	[36]

It can be seen from Table 4 that among the mentioned technologies ORC systems are characterized by positively low investment costs in comparison to competitive technologies.

ORC systems are modified classical steam power plants. Unlike classical steam power plants, they are driven by heat sources featuring lower thermal parameters, power and output therefore they are featuring lower power, lower efficiency and smaller external dimensions. Moreover, they are using a low-boiling working fluid instead of water. However, the set of thermodynamic processes proceeding in the ORC system and operating principle are the same as in the case of the classical Clausius-Rankine system. The main components of the ORC system are two heat exchangers (an evaporator and a

condenser), a liquid working fluid reservoir, a liquid working fluid pump and an expander coupled to a electric generator. In order to improve the efficiency of the system an additional heat exchanger (i.e., regenerator) is often applied. Figure 1 shows the scheme of the ORC system with regenerator. Figure 2 visualizes the thermodynamic processes of the simple organic Rankine cycle in the T-s plane.

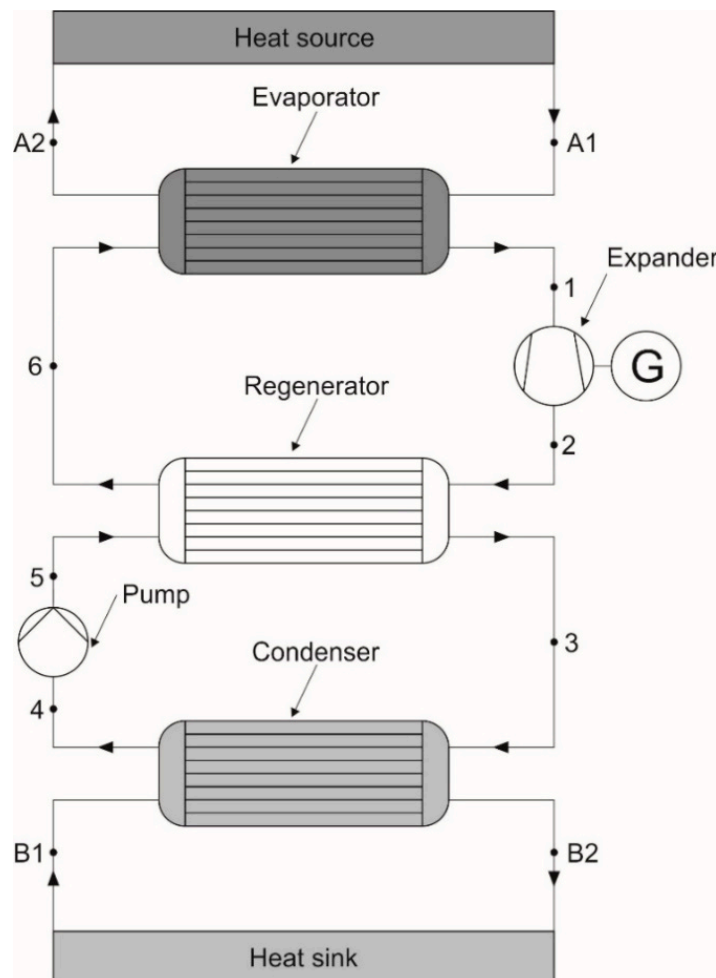


Figure 1. Scheme of the ORC system with regenerator.

The most important design choices of the ORC systems are the working fluid and expander selection. Different low-boiling substances are applied as working fluids in ORC systems. These can be for example refrigerants or silicone oils. By the shape and course of the saturation curve, the working fluids are classified as dry, wet and isentropic [26,37,38]. The selection of the working fluid for the ORC system is often challenging as environmental, thermodynamic and design considerations should be taken into consideration. Different methods for working fluid selection are reported in literature [39–50]. The working fluid should be non-toxic and featuring low values of global warming potential (GWP) and ozone depletion potential (ODP) indicators. In terms of thermodynamic properties, the working fluid should be selected to fit the thermal characteristics of the heat source (i.e., its temperature, thermal power, heat capacity and mass flow rate of the heating medium). The heat source thermal parameters and the type of the applied working fluid have the influence on the ORC system design and configuration. For example, ORCs with direct evaporation of the low-boiling working fluid (whose layout is presented in Figure 3) or using an intermediate (e.g., thermal oil) heat transfer working fluid (which layout is presented in Figure 4) are implemented depending on the heat source temperature and the type of the applied low-boiling working fluid. In the case of domestic scale power

systems, the thermal power of the heat source is usually low or medium (typically ranging from 10 up to 50 kW_t). Therefore, if the ORC system is considered to be powered by a heat source featuring such a range of thermal power the mass flow rate of the low-boiling working fluid should be kept at low value. In addition, to provide operational safety of the ORC system, the applied working fluid should feature low values of pressure within the temperature range of the heat source. The list of the selected working fluids which can be used in domestic ORCs is presented in Table 5.

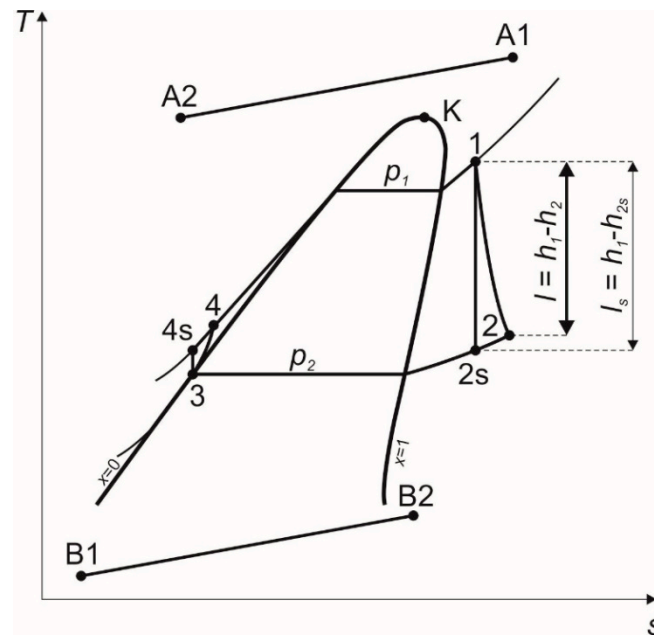


Figure 2. Thermodynamic processes of the simple organic Rankine cycle in the T-s plane.

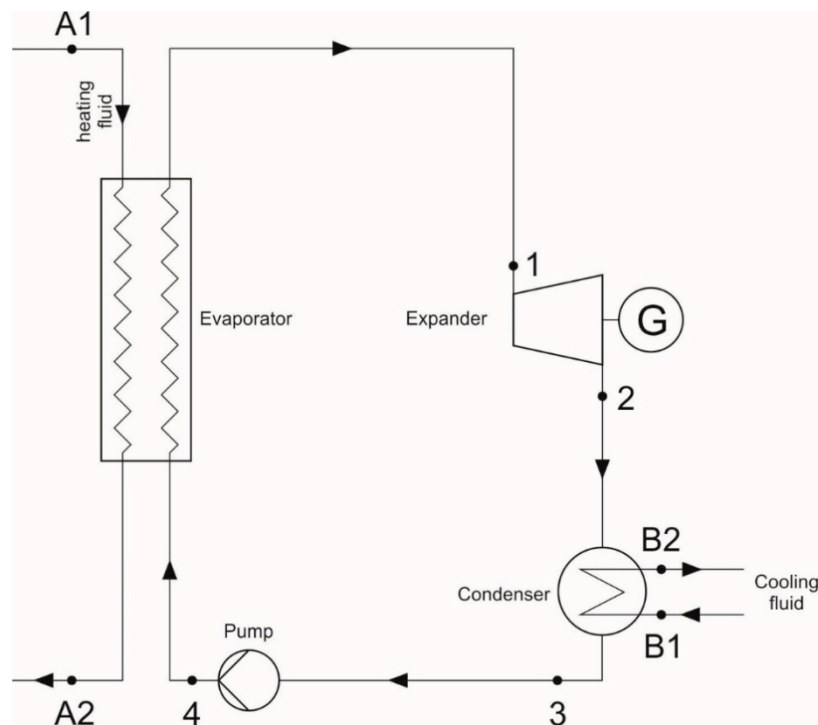


Figure 3. Scheme of the ORC system with direct evaporation of the working fluid.

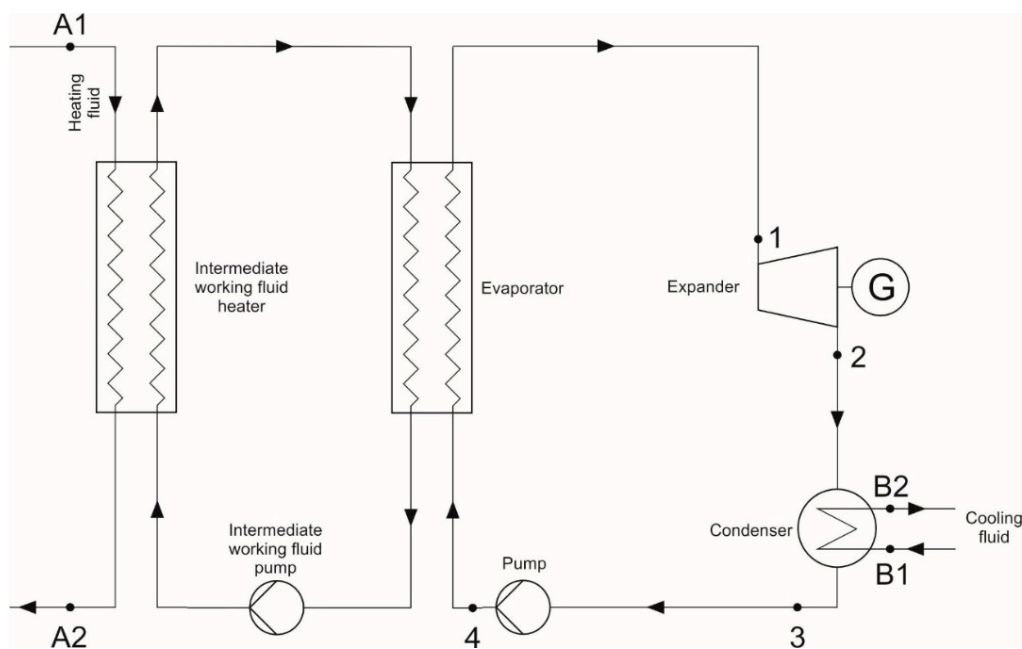


Figure 4. Scheme of the ORC system with intermediate working fluid loop.

Table 5. The working fluids suitable for domestic Organic Rankine Cycle (ORC) systems [37–50].

No.	Working Fluid	Triple Point Temperature	Normal Boiling Point Temperature	Critical Point Parameters			ODP	GWP	Working Fluid Class [39]
		t_{trp} °C	t_{nbp} °C	t_{cr} °C	p_{cr} MPa	ρ_{cr} kg/m ³			
1	R113	−36.22	47.59	214.06	3.39	560.00	0.8	4800	ANZCM
2	R114	−92.52	3.59	145.68	3.25	579.97	1.0	3.9	AZCM
3	R123	−107.15	27.82	183.68	3.66	550.00	0.02	77	ACNMZ
4	R124	−199.15	−11.96	122.28	3.62	560.00	0.02	620	ACNZM
5	R1234ze	−104.53	−18.95	109.37	3.63	489.24	0	6	ACNZM
6	R134a	−103.30	−26.07	101.06	4.06	512.00	0	1300	ACZ
7	R152a	−118.59	−24.02	113.26	4.51	368.00	0	120	ACZ
8	R227ea	−128.60	−16.34	101.75	2.93	594.25	0	3220	ACNMZ
9	R236fa	−93.63	−1.44	124.92	3.20	551.30	0	9810	ACNMZ
10	R365mfc	−34.15	40.15	186.85	3.22	473.84	0	825	ANZCM
11	R245ca	−81.65	25.13	174.42	3.39	523.59	0.12	N/A	ACNMZ
12	R245fa	−102.10	15.14	154.01	3.65	516.08	0	1030	ACNMZ
13	R601a	−160.50	27.83	187.2	3.38	236.00	0	20	ACNMZ
14	R141b	−103.47	32.05	204.35	4.21	458.56	0.12	725	ACNMZ
15	R142b	−130.43	−9.12	137.11	4.05	446.00	0.07	2310	ACNMZ
16	R236ea	−103.15	6.19	139.29	3.5	563.00	0	1330	ANZCM
17	R600a	−159.42	−11.75	134.66	3.63	225.5	0	3	ACNMZ
18	RC318	−39.80	−5.97	115.23	2.78	620.00	0	10,300	AZCM
19	R1234yf	−53.15	−29.45	94.7	3.38	475.55	0	4	ACNZM
20	R290	−187.63	−42.11	96.7	4.25	220.48	0	20	ACZ

R113 is banned by the Montréal Protocol. R143a and R123 will be phased out soon. However, the author decided to include these working fluids in the comparison as the reference substances which were considered to be applied or were applied in ORCs. The literature shows [51] that the electric power output of different domestic CHP systems usually ranges between 1 and 10 kW_e. Considering the advisable features of domestic CHP ORC systems they should be assembled of components featuring simple design and low investment and service costs. Braze or welded plate heat exchangers meet these requirements. These heat exchangers are simple in design and low weight thanks to the application of specially formed plates that are welded together. The specially formed plate increases the heat exchange surface and leads to the intensification of the heat exchange phenomena. This way it is possible to achieve high thermal power in relation to the heat exchanger dimensions. Plate heat exchangers can be applied for clean gases and liquids. In the case of dusty working fluids,

the intermediate heat transfer substance have to be applied. Plate heat exchangers feature small external dimensions in relation to their power and are cheap compared to the other types of the heat exchangers. What is more, the heat transfer rate in plate heat exchanger is high. Shell-and-tube heat exchangers can be also applied in domestic ORCs. Compared to plate heat exchangers, shell-and-tube heat exchangers are larger, heavier and more complex to manufacture as they require the use of many welded joints. The heat transfer rate in shell-and-tube heat exchangers is lower compared to plate heat exchangers and a larger heat transfer area is needed. Therefore these heat exchangers feature significantly larger external dimension, mass and investment costs compared to plate heat exchangers. In general, the ORC systems can utilize two types of expansion machines. First type are axial or radial (inflow or outflow) turbines [52]. The others are volumetric expanders (e.g., piston, screw, scroll, lobe and multi-vane machines) [53]. Only turbines which are featuring small external dimensions and a power output of few kW (i.e., microturbines) can be applied in domestic ORC systems [52]. However, microturbines are featuring very high rotational speeds and clearance losses become large for small scale turbines, which leads to the need of using complicated and high-precision manufacturing technologies and tools for their fabrication. Therefore, their investment costs are high. For these reasons, in the opinion of the author, their applicability to domestic systems is currently limited. Compared to turbines, volumetric expanders are having positive features as they are operating at much lower rotational speeds, lower working fluid flow rates, and higher pressure drops that can be achieved in one stage [54]. Therefore, they fit better to small scale ORCs. Piston [55–61], screw [62–66], scroll [67–73], lobe [74,75] and multi-vane [25] expanders can be applied in domestic ORCs. These expanders are at different stage of development. Scroll expanders are applied in many prototypes of the domestic ORC systems. Their mechanical power output ranges from 1 to 10 kW [53], while the pressure expansion ratio of a scroll expander typically ranges between 2 and 4.5. Piston and linear piston expanders feature the highest pressure expansion ratios among the volumetric expanders applied in ORCs. Therefore, these expanders are a good option for high-pressure ORC systems. Their power output ranges from 0.5 to 15 kW [52,53]. The value of pressure expansion ratio of a piston expander can reach 200. Screw expanders are insensitive to wet gas operating conditions, therefore these machines can be applied in systems utilizing floating heat sources. The pressure expansion ratio of a screw expander typically ranges between 10 and 15 and power output ranges from 1 to 8 kW [72]. Rotary lobe expanders are currently under research and development [74,75]. Thanks to their positive features (insensitivity to wet gas operating conditions, compact and simple design, high power output in relation to external dimensions) they seem to be good option for different industrial and domestic ORCs. Multi-vane expanders can operate in wet gas conditions, their design is simple and compact, they feature small gas consumption and low rotational speed and are relatively cheap. Therefore these expanders are particularly promising for application in domestic ORCs. Their possible application in such systems is currently being investigated by many researchers. The pressure expansion ratio of a multi-vane expander typically ranges between 3 and 5 and power output ranges from 1 to 10 kW [25]. Multi-vane expanders combine in one design the best features of turbines and volumetric machines. Working fluid pumps that can be applied in domestic ORC systems should be characterized by low energy consumption, high reliability and simplicity of design. Centrifugal and volumetric pumps are the options that can be applied in these systems. Centrifugal pumps are featuring high mass flow rates, therefore their application is better justified in the case of high-power industrial ORC systems utilizing turbines as the expanders rather than domestic systems. In micro-power ORCs, volumetric pumps are a better choice, as they are featuring smaller energy consumption, smaller mass flow rates, dimensions, lower power and rotational speed. Positive displacement pumps, such as for example multi-vane pumps [76] are currently applied in prototypes. ORC system subassemblies should be hermetically sealed and connected using similar techniques as are used in refrigeration and air conditioning units, (e.g., soldering with silver). The automatic control system should provide the safe operation of the system. Additionally it should give the opportunity of measurement and observation of system's operating parameters as well as signaling errors, emergency conditions and necessity of inspections.

Additionally, in order to efficiently use the generated heat in the summer, the micro-absorption cooling unit may be integrated with a CHP ORC system. This way, the multi-generation system can be implemented. According to the author's knowledge, no research is currently proceeding on the integration of absorption cooling unit with domestic ORC systems, therefore this opportunity should be considered as one of the possible ways of further research on domestic ORCs.

3. Description of the Experimental Test-Stand and the Experiment Results

At the Laboratory of Thermodynamics and Thermal Properties of Materials of the Department of Thermodynamics and Renewable Energy Sources at the Wrocław University of Science and Technology an experimental test-system composed of ORC test-stand and domestic gas boiler was designed and commissioned. Thanks to the special design of this test-setup and its flexibility to different experimental conditions it is possible to conduct a wide range of experiments related to micro CHP ORC systems. For example, it is possible to quickly change the expander or working fluid. The key assumptions for the design of this test stand were defined in order to meet the advisable features of domestic ORCs described in the first part of this article. In particular, the following design requirements were set: small dimensions and compact design, safety, low investment costs, low weight, low power needs, low operating pressure and small mass flow rates of the working fluids. These requirements were achieved thanks to the application of specially selected and designed subassemblies in the test-stand, i.e., plate heat exchangers (evaporator and condenser) and multi-vane volumetric machines (pump and expander). In addition, the working fluid featuring low pressure in the range of temperatures of domestic heat sources (such as e.g., solar heat, hot water, etc.) was applied in the test-stand.

3.1. Description of the Test-Stand

Central heating boilers fed by different fuels, e.g., natural gas, are commonly used for individual heating in many properties. The possibility of integration of a electricity generating system in the boiler seems to be interesting option. This way boiler can be transformed into a CHP system, which, in addition to heat, can generate electricity that can be consumed by the boiler user or fed to the grid. This solution can contribute to increase of energy generation efficiency and share of distributed energy systems and therefore to increased energy safety of end energy consumers. The ORC technology can be used for this purpose. In the following part of this paper in order to check the possibility of integrating the ORC system with the domestic gas boiler a standard, commercially available gas central heating boiler was selected and used to power the ORC test-stand. This is a closed combustion chamber boiler featuring the thermal power of 24 kW. The automatic controller which is applied in the boiler enables smooth regulation (per 1 °C) of the central heating water temperature in the range of 40–85 °C and temperature of the sanitary water in the range of 30–60 °C. Central heating water was used to supply the ORC test-stand as it is visualized in the scheme of the test-system (see, Figure 5). Figure 6 shows the general view of the test-system, while Figure 7 shows a detailed view of the ORC test-stand. The test-system operating principle is described in the following.

The central heating water from the gas boiler is pumped by a water pump (which is installed in the boiler) to the inlet of a plate heat exchanger i.e., evaporator of the ORC system (featuring the height of 622 mm, length of 165 mm and width of 197 mm). The direction of water and low-boiling working fluid flow in the evaporator is countercurrent. Thanks to the possibility of smooth regulation of the temperature of the central heating water, it is possible to conduct experimental tests for different temperatures of the heat source. Therefore, it is possible to simulate in laboratory conditions various domestic heat sources which can be applied for powering the ORC system. The implementation of the Clausius-Rankine cycle in the range of the heat source temperature provided by the applied gas boiler was possible thanks to the application of R123 (2,2-dichloro-1,1,1-trifluoroethane) as the working fluid in the ORC test-setup. This working fluid features a low boiling temperature (ca. 27 °C) and low pressures in the analyzed range of the heat source temperature.

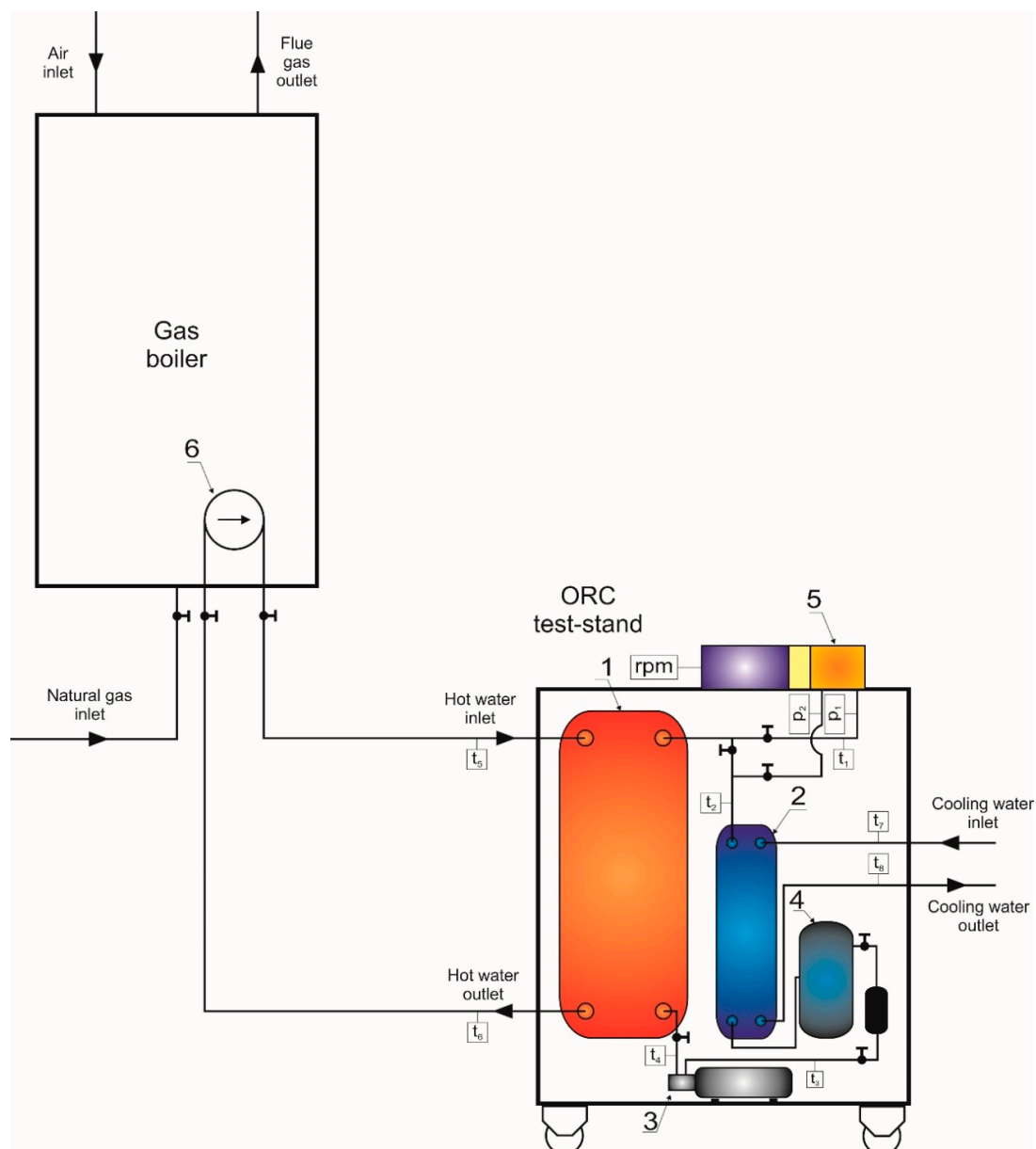


Figure 5. Scheme of the test-system.

A positive displacement multi-vane pump (featuring the power of 185 W) was applied in the ORC test-stand to force the flow of liquid low-boiling working fluid. By using this type of working fluid pump, it was possible to conduct the experiments under small and very small flow rates of low-boiling working fluid (typically during experiments the working fluid flow rate is regulated in the range of 50–250 dm³/h). In addition, this type of pump gives the ability to easily adjust operating parameters (i.e., rotational speed, working fluid flow rate and pressure) and is characterized by low rotational speed, simple and easy-to-service design and a low price. What is more it is insensitive to operation in two-phase flow conditions. The working fluid pump forces the flow of the working fluid from the liquid reservoir (featuring a volume of 3.9 dm³) through pipelines to all of the test-stand components. On the outlet pipeline of the liquid reservoir the working fluid drier, filter and a sight-glass are installed.

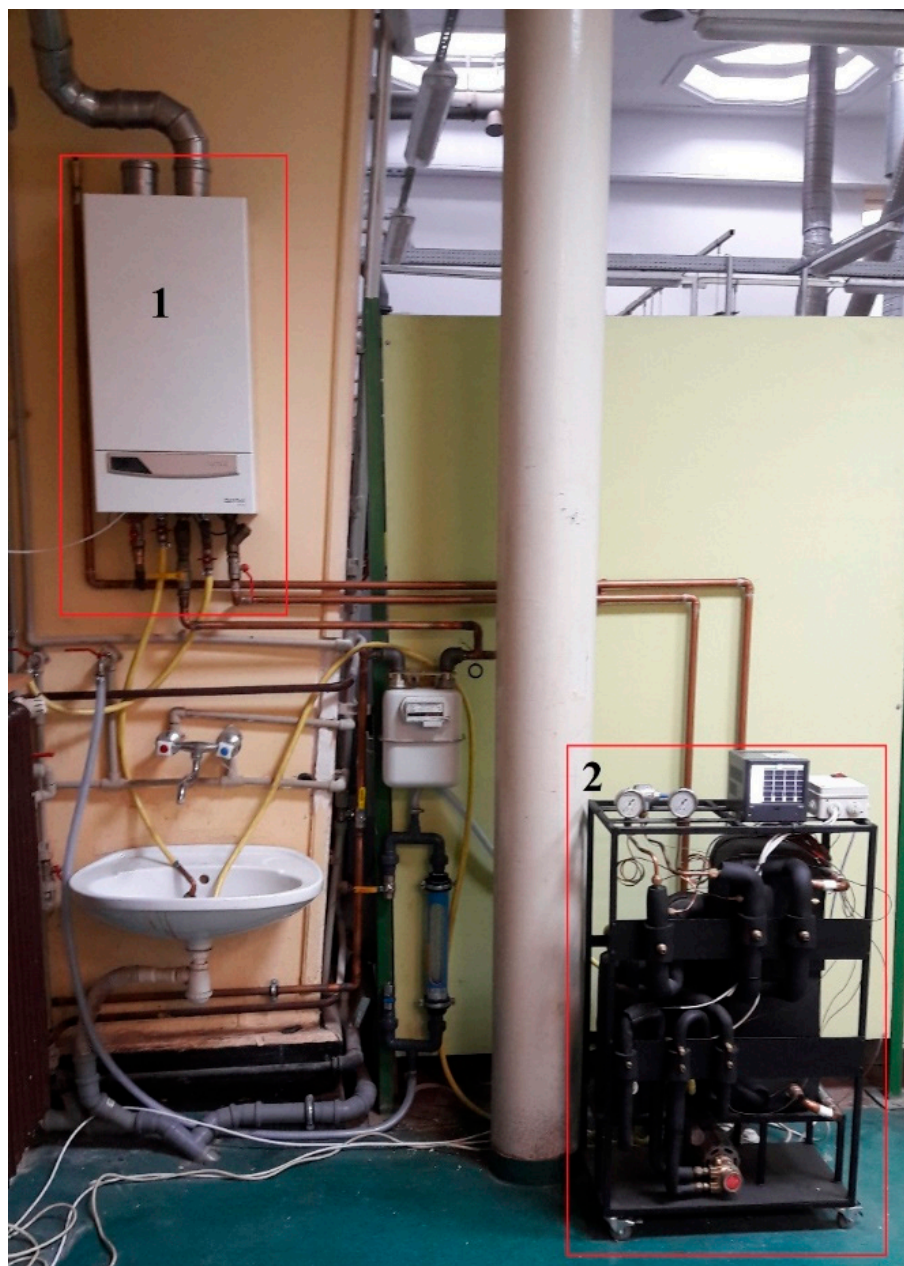


Figure 6. General view of the test-system (1—gas boiler; 2—ORC test-stand).

The working fluid vapour which is obtained at the outlet of the evaporator features high values of thermodynamic parameters (i.e., pressure and temperature) and it flows through the pipelines and control valves to the inlet of the expander. In the ORC test-stand a specially modified multi-vane air motor was applied as the expander (featuring the power of 330 W). The modifications of the expander design were necessary to adapt the machine to low-boiling working fluid operating conditions. These modifications were described in more details in earlier publications of the author [77,78]. This type of the expander (similarly to the applied multi-vane pump), features a simple design, low rotational speed and low price and is insensitive to two-phase operating conditions. It can be easily hermetically sealed and coupled to the generator. Detailed results of a comprehensive experimental and numerical research related to this expander and possibilities of its optimization are reported in [79].

After expansion in the expander, the thermodynamic parameters of the working fluid (i.e., pressure and temperature) decrease and the working fluid flows through the pipeline to the inlet of a plate

condenser. A bypass pipeline with a shut-off valve is installed between the outlet of the evaporator inlet of the condenser and the expander inlet. Thanks to its opening it is possible to direct the gas flow from the outlet of the evaporator directly to inlet of the condenser bypassing the expander. The bypass pipeline is used during starting (heating up) and stopping (cooling down) the test-stand. The plate heat exchanger (featuring the height of 456 mm, length of 47 mm and width of 80 mm) was applied as the condenser in the ORC test-setup. Condenser is cooled by the cold water from the municipal pipelines. The flow rate of the cooling water is regulated by means of set of control valves. Therefore it is possible to simulate different domestic heat sinks in laboratory conditions. The direction of the cooling water and low-boiling working fluid flow in the condenser is countercurrent. After condensation in the condenser, the working fluid is forced back by the pump to the liquid reservoir. The Clausius-Rankine cycle is therefore completed with this point.

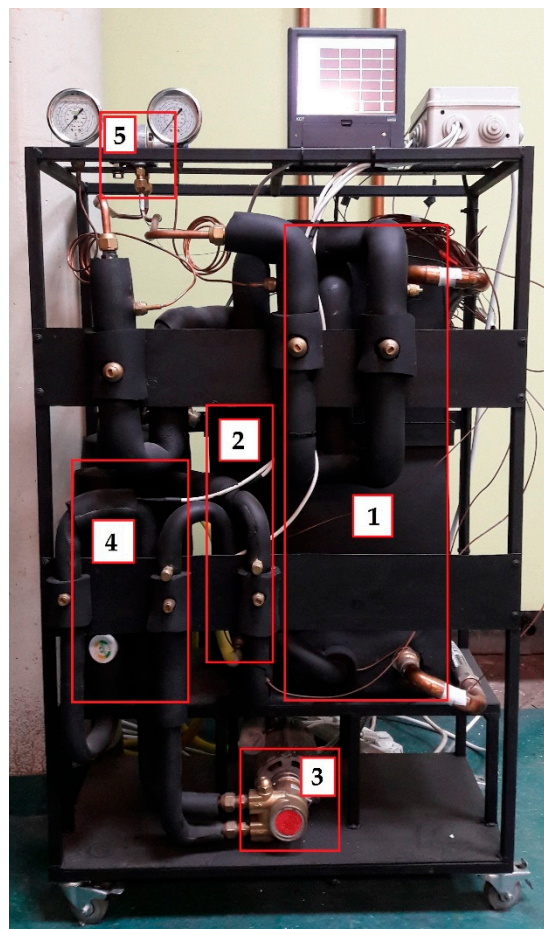


Figure 7. Detailed view of the ORC test-stand (1—evaporator; 2—condenser; 3—pump; 4—working fluid reservoir; 5—expander and generator).

The test-stand is equipped with different measuring sensors. Temperature measurements are carried out in 10 measuring points using T-type thermocouples (temperature measuring points are marked with t letters in Figure 5). Thermocouples are connected to a microprocessor recorder with LCD screen which enables continuous observation and recording of the measured values. Pressure is measured using pressure gauges (pressure measuring points are marked with p letters in Figure 5) while the rotational speed of the expander is measured using a laser tachometer.

An automatic control system is not applied in the ORC test-stand due to its limitations (automatic controller is however applied in the gas boiler). All operating parameters of the ORC system are manually controlled, therefore different experimental conditions (including simulation of failures and emergency situations) can be tested.

3.2. The Experimental Description

The aim of the experiment presented in this article was observation of the associated operation of a gas boiler and the ORC test-stand under various experimental conditions and recording the operating parameters.

Before starting the experiments, the test-setup was prepared for tests according to the following procedure. Initially, all of the shut-off valves that are installed in ORC test-stand were opened except the shut-off valves installed at the inlet and at the outlet of the expander. Then, the low-boiling working fluid pump was started and the flow of low-boiling working fluid was forced from the liquid reservoir tank through the pipelines to the inlet of the evaporator. Subsequently, using the control valve on the low-boiling medium pump, the flow of the low-boiling working fluid was set (constant flow of the working fluid was kept for each experimental series). Then, the condenser cooling water shut-off valve was opened and the flow of the cooling water was set constant using the control valve.

After preparing the ORC test-stand, using the boiler controller, the temperature of the central heating water (i.e., the temperature of the heat source) was set and the boiler was switched on.

After reaching the set temperature of the heat source, the bypass between evaporator and condenser was closed and the flow of the working medium through the expander was opened using shut-off valves installed on the inlet and outlet ports of the expander. When the gas flow through the expander was opened, the operation of the expander and variation of working fluid parameters in the system were observed. At the time when operating parameters stabilized (i.e., ORC system reached steady state operation) operating parameters were measured.

After the experimental series was completed for one setting of the heat source temperature and measurements were recorded, the shut-off valves at the inlet and the outlet of the expander were closed again, the boiler was switched off and the stand was cooled to ambient parameters using cooling water. The same experimental procedure was then carried out for different settings of the heat source temperature.

4. Experimental Results and Discussion

The experiments were carried out for variable temperature of the heat source, which was controlled between 45 °C and 85 °C with a step of 10 °C. During the experiments thermodynamic parameters of the working fluids (i.e., R123, heating water and cooling water) were recorded and operation of the test-system was put under the observation. In this experimental series the low-boiling working fluid flow rate was set to 35 dm³/h, the heating water flow rate was set to 150 dm³/h and cooling water flow rate was set to 70 dm³/h.

The collected experimental results are summarized in Table 6, while Table 7 reports the values of indicated and real power output of the expander (calculated from the relation $N_{\text{ix}} = m_{\text{WF}} \times (h_1 - h_{2s})$ and $N_{\text{rex}} = m_{\text{WF}} \times (h_1 - h_2)$ correspondingly, where m_{WF} is the mass flow rate of the working fluid), expander internal efficiency (calculated from the relation $\eta_{\text{ix}} = (h_1 - h_2)/(h_1 - h_{2s})$), power of the evaporator (calculated from the relation $Q_{\text{EV}} = m_{\text{WF}} \times (h_1 - h_4)$), power of the condenser (calculated from the relation $Q_{\text{CN}} = m_{\text{WF}} \times (h_2 - h_3)$) and thermodynamic cycle efficiency (calculated from the relation $\eta_{\text{ix}} = N_{\text{rex}}/Q_{\text{EV}}$). These operational parameters were calculated using the values of the enthalpy (h) of the working fluid in measuring points which were obtained using CoolProp computer software [80].

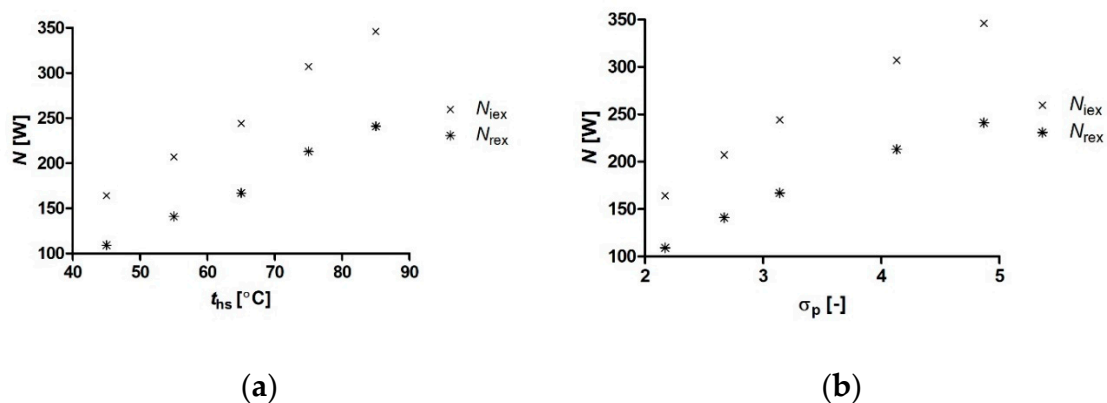
Table 6. Thermodynamic parameters of R123 and heating and cooling water in measuring points and rotational speed of the expander.

No.	t_{hs} (°C)	p_1 (bara)	t_1 (°C)	p_2 (bara)	t_2 (°C)	t_3 (°C)	t_4 (°C)	t_5 (°C)	t_6 (°C)	t_7 (°C)	t_8 (°C)	n_{ex} (rev/min)
1	45	1.5	38.9	1.6	24.1	17.4	17.9	39.5	38.4	14.7	15.6	700
2	55	1.85	45.2	1.6	26.0	17.6	18.0	46.9	46.4	14.8	16.9	3280
3	65	2.2	50.9	1.6	28.1	17.9	18.3	52.7	51.6	15.0	19.1	3300
4	75	2.9	60.7	1.7	31.4	18.1	18.4	60.9	59.8	15.1	17.5	2900
5	85	3.4	66.7	1.7	33.4	18.0	18.6	71.5	65.9	15.1	18.3	2950

Table 7. The values of indicated and real power output of the expander, expander internal efficiency, power of the evaporator, power of the condenser and thermodynamic cycle efficiency.

No.	σ_p (-)	N_{iex} (W)	N_{rex} (W)	η_{iex} (%)	Q_{EV} (W)	Q_{CN} (W)	η_{ORC} (%)
1	2.17	164	109	66.4	2525	2417	4.3
2	2.67	207	141	67.8	2572	2433	5.5
3	3.14	244	167	68.5	2613	2447	6.4
4	4.13	307	213	69.3	2687	2476	7.9
5	4.87	346	241	69.6	2735	2496	8.8

The experimental results are visualized in Figures 8–10. Figure 8a shows the variation of the expander isentropic power (N_{iex}) and expander power output (N_{rex}) vs. the heat source temperature (t_{hs}) while Figure 8b shows the variation of the expander isentropic power (N_{iex}) and expander power output (N_{rex}) vs. the pressure expansion ratio (σ_p). Figure 9a shows the variation of the expander isentropic efficiency (η_{iex}) vs. the heat source temperature (t_{hs}) while Figure 9b shows the variation of the expander isentropic efficiency (η_{iex}) vs. the pressure expansion ratio (σ_p). Figure 10a shows the variation of the thermodynamic cycle efficiency (η_{ORC}) vs. the heat source temperature (t_{hs}) while Figure 10b shows the variation of the thermodynamic cycle efficiency (η_{ORC}) vs. the pressure expansion ratio (σ_p).

**Figure 8.** The variation of expander isentropic power (N_{iex}) and expander power output (N_{rex}): (a) $N = f(t_{hs})$; (b) $N = f(\sigma_p)$.

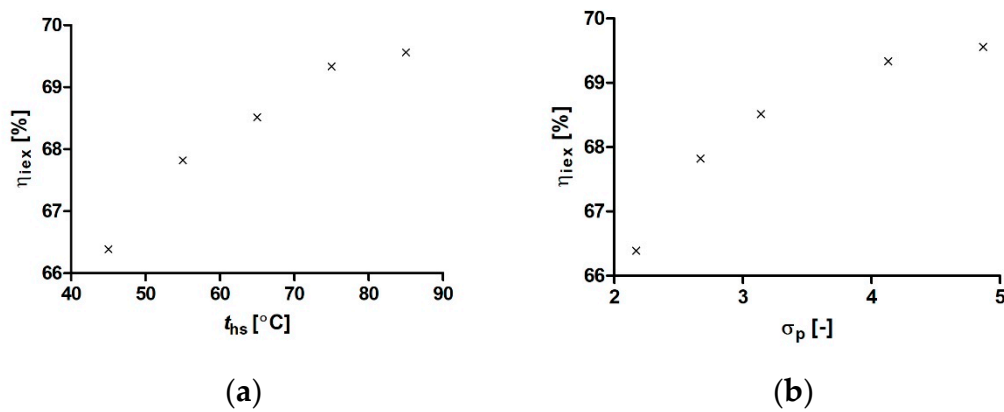


Figure 9. The variation of expander isentropic efficiency (η_{iex}): (a) $\eta_{iex} = f(t_{hs})$; (b) $\eta_{iex} = f(\sigma_p)$.

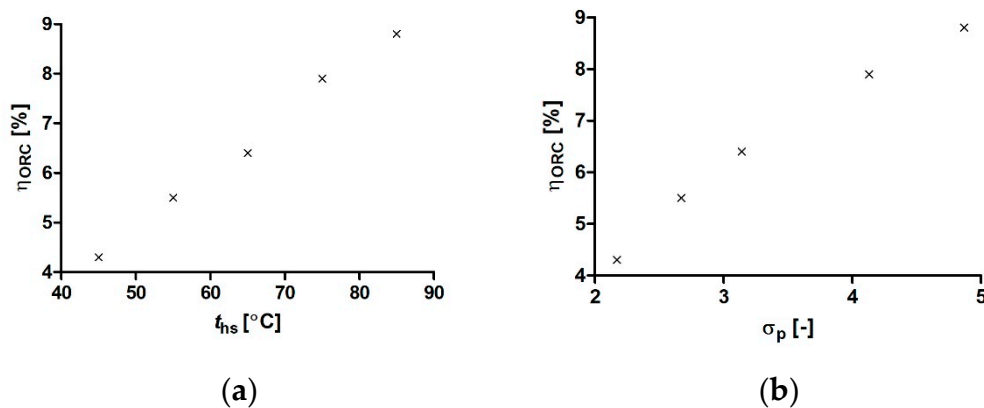


Figure 10. The variation of thermodynamic cycle efficiency (η_{ORC}): (a) $\eta_{ORC} = f(t_{hs})$; (b) $\eta_{ORC} = f(\sigma_p)$.

The experiment was conducted in the heat source temperature range of 45–85 °C. Figures 8–10 show the increasing trend of all of the ORC test-setup operational indicators which is caused by the growing heat source temperature (t_{hs}). Results visualized in Figure 8 show that the multi-vane expander power output was varying during the experiment in the range of 109–241 W. The increasing expander power output which is observed for increasing temperature of the heat source is caused by increasing temperature and pressure of vapour at the inlet to the expander. The experiment therefore proves, that the multi-vane expander power output depends mostly on the pressure expansion ratio (not the working fluid flow rate which was kept constant during the experiments). Results visualized in Figure 9 show that the multi-vane expander internal efficiency was varying during the experiment in the range of 66.4–69.6%, while the thermodynamic cycle efficiency was increasing with increasing heat source temperature (t_{hs}), increasing expander power output (N_{rex}) and pressure expansion ratio (σ_p) and was varying in the range of 4.3–8.8% as it is visualized in Figure 10. Experimental data which are reported in Table 6 show, that the decrease of heating water temperature between the inlet and the outlet of the ORC evaporator ranges between $\Delta t = 0.5$ – 5.6 °C depending on the experimental conditions. These low drops of water temperature results from the difference in heat capacity of heating water and low-boiling fluid and were observed by the author during his earlier experiments on ORC systems [54,77–79]. The value of the heating water temperature drop is particularly important in case of domestic CHP ORC systems in which the water flowing out from the evaporator will be then transferred to the inlet of the central heating system. Therefore, lower the temperature drops are higher is the thermal comfort and better are the operating conditions of central heating installation. Low drops of water temperature which were observed during described experiments are promising from the point of view of the possibility of integrating the ORC system with a gas boiler as such small drops of temperature they will be practically imperceptible to the user.

Literature reports [81] that the heat demand in standard houses ranges between 70 and 100 kWh/m², and in older buildings exceeds 120 kWh/m² per year. Table 8 reports the comparison of the PM_{2.5} and PM₁₀ dust emission of domestic boilers fed by different fuels [82]. Therefore, if technologies listed in Table 8 are considered as the heating boilers, the annual emission of dust per 1 m² of property area will reach the values reported in Table 8 as E_s (for standard house) and E_o (for old house). The analysis of the data reported in Table 8 shows that coal-fired boilers are the largest source of dust emission. For example in Wrocław (Poland) the literature reports [19,83] that the number of low-quality coal-fired boilers and furnaces is equal to ca. 20,000. These furnaces are used mostly for heating old houses and old flats. Application of gas-fired CHP ORC systems as the replacements of the coal-fired boilers in these properties can therefore lead to clean heating and power generation and significant reduction of the dust emitted into the environment. The average area of the flat in Poland is 70.4 m² [84]. If such an area is considered to be heated by coal-fired boiler, then for the earlier mentioned number of coal-fired boilers the avoided dust emission in Wrocław thanks to the application of the CHP ORC units can be as high as ca. 47.3 t/a. Additionally ca. 40,000 kW_e of electricity might be generated if single CHP ORC unit electric power is equal to 2 kW_e.

Table 8. The comparison of the PM_{2.5} and PM₁₀ dust emission from domestic boilers fed by different fuels [82].

Bolier Type	E mg/kWh _t	E_s mg/m ² ·a	E_o mg/m ² ·a
Oil-fired bolier	40	2800–4000	4800
Gas-fired bolier	5	350–500	600
Coal-fired bolier	280	19,600–28,000	33,600
Wood-fired boiler (manually operated)	140	9800–14000	16,800
Wood-fired boiler (automatically controlled)	65	4550–6500	7800
Pellet-fired bolier	40	2800–4000	4800
Wood chips-fired boiler	52	3640–5200	6240
CHP unit	0	0	0

5. Summary and Conclusions

This paper presents the results of experimental study on possible integration of domestic gas boilers with micro-ORC systems and conversion of such a boiler into a domestic CHP unit. Such systems are promising options for application as replacements for standard low-efficient boilers fed by low-quality solid fuels which are commonly applied for domestic heating. The design issues of such systems were discussed and the selection of heat exchangers, expander and working fluid were found to be most important issues that need to be solved during the implementation of such systems. The experimental test-setup was designed in order to fit to the stated design requirements (i.e., simple design, safety and low costs). The multi-vane expander was applied in the experimental ORC test-stand. The experimental series were performed using the test-system in order to record the operational conditions. The experiments succeeded. The power output of the expander was ranging from 109 to 241 W, and the thermodynamic cycle efficiency was ranging from 4.3 to 8.8%. Operating parameters were found to be dependent on the heat source temperature and working fluid pressure at the inlet to the expander. The decrease of heating water temperature between the inlet and the outlet of the ORC evaporator was found to range between $\Delta t = 0.5$ – 5.6 °C depending on the experimental conditions. Therefore, the experimental results show that there is a possibility of integrating the ORC system with a standard domestic central heating gas boiler. This positive research results were achieved partly thanks to the positive features of the applied multi-vane pump and multi-vane expander as well as plate heat exchangers, which are characterized by positively high values of the heat transfer coefficients, high thermal power in relation to the dimensions and high efficiency. The simple design of the multi-vane expander and its additional positive features

including the lack of the need for additional complicated and high-cost automatic adjustment systems, lubrication systems and rotational speed reduction devices.

During the tests, the operation of the test-system was observed and there was no emergency or safety hazards noticed. Further development of the design supported by the results of further experimental research is needed and can successfully lead to the implementation of these systems on a large scale. In effect, this may lead to the increased energy generation efficiency (energy will be generated and consumed at the same place without the necessity of transmission), decentralization of energy generation, and what is the most important, significant reduction of the number of low-quality and low-efficient furnaces used in domestic heating. This way, a reduction in the number of low-quality solid fuel combustion processes and therefore a reduction of solid pollutant emissions (such as PM_{2.5} and PM₁₀ dust) into the atmosphere can be achieved. In the case of large cities, like Wrocław (Poland) the avoided dust emission can be as high as 47.3 t/a.

Experiments were performed using a micro-power multi-vane expander. Further works should be focused on increasing the power output and efficiency of the expander and CHP system. Then an expander featuring bigger power output will be applied in the test-stand by the author in his further works. Another further research topic is the integration of the micro-absorption cooling unit in a CHP ORC system in order to efficiently use the heat generated during the summer.

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