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Development of ORC's for micro power generation

Abstract
In the paper, new trends in development of micro power generation of heat and electricity are presented. New type of CHP for domestic usage is developed in the Institute of Fluid-Flow Machinery PAS and methods of its design are presented. The most promising trends in equipment of ORC cycle for this purpose were discussed. Main attention was focused on micro-heat exchangers design based on micro-channels and micro-jets. In our opinion future development of high power heat exchangers will be based on nets micro-heat exchangers.

Keywords: CHP, ORC, heat exchangers, heat transfer intensification, thermodynamics.

1. Introduction

Last years clearly observe the tendencies both in the world and in the countries of European Union towards the growth of the meaning of **energy dispersed generation sector** based on local resources and technologies using both conventional fuels and renewable energy.

Attractive became the versatile processes of production of useful energy in **energy dispersed generation sector**. Processes belonging to them are such as: cogeneration, trigeneration and poligeneration, especially in reference to small and average scale plants. The most popular is a **cogenerative process** which is a simultaneous production of heat and electricity. The aim of EU is to achieve 18% in the market share in cogeneration realized in small and average scale production units of electric energy up to year 2020. The idea of dispersed cogeneration is particularly attractive in case of renewable resources of energy because in this case it is easy to apply new technologies. The EU directive requires 22% of "**the green electricity**" share in the scale of whole Union up to year 2020. For Poland this requirement is set on 15% of total energy balance, which is still large and difficult to obtain (we had a share of RES at approximately 1.6% in year 2000).

In recent years, organic Rankine cycle has become a field of intense research and appears as a promising technology for conversion of heat into useful work or electricity. The heat source can be of various origins: solar radiation, biomass combustion, ground heat source or waste heat from factories. Unlike in the steam power cycle, where vapor steam is the working fluid, organic Rankine cycles (ORC) employ refrigerants or hydrocarbons (Harrison, 2003; Mikielewicz, 2010; Borsukiewicz-Gozdur, 2010b).

For recovery technologies for gaseous waste heat, the method and its efficiency are mainly determined by the gaseous source temperature (Zhou, Wang, Chen, Wang, 2013). The waste heat sources are generally divided into three categories: low temperature (< 230°C), medium temperature (230–650°C) and high temperature (> 650°C) (Borsukiewicz-Gozdur, 2010b). The low-temperature gaseous waste heat accounts for about 50% of all the waste heat in an industry, as reported by statistic data. However, most waste heat recovery technologies aim for the heat resources with medium and high temperature, while the technology for low-temperature gaseous waste heat still needs to be developed.

Rankine cycle is widely used to convert thermal energy into work. However, the traditional Rankine cycle using water as the working fluid is adopted only for mid-and-high-temperature thermal resources, and it is not used for low-temperature thermal resources. To break through the temperature limitations, some new technologies are necessary. Organic Rankine Cycle (ORC) has been demonstrated to be advantageous to recover waste heat when the temperature is below 230°C. The working fluid in ORC should be the dry fluid so that the state point after the expansion lies in the superheat vapor region and has properties such as high molecular mass and low boiling point, which enables an improvement of the cycle efficiency.

Several theoretical and experimental investigations have also been reported in recent years (Zhou et al., 2013). For instance, Yari and Mahmoudi investigated utilization of waste heat from a gas turbine-modular helium reactor (GT-MHR) by using three types of ORC including of a simple ORC, an ORC with internal heat exchanger and a regenerative ORC. The authors proved that solar Organic Rankine Cycle (SORC) achieves the best performance and the cycle efficiency is about 10% higher than that of the GT-MHR alone. Dai et al. examined and optimized the thermodynamic parameters of the ORC for different working fluids. The results show that the cycle with R236ea has the highest exergetic efficiency and adding an internal heat exchanger into the ORC system is not helpful for the system performance. Zhang et al. investigated the parameter optimization and the performance comparison of the fluids in a subcritical ORC and a transcritical power cycle in low-temperature binary geothermal power system. The results indicate that adoption of R123 can help to achieve the highest thermal efficiency and exergetic efficiency of 11.1% and 54.1%, respectively in a subcritical ORC system, whereas R125 presents an excellent economic and environmental performance in a transcritical power cycle. Quoilin et al. studied the ORC system by using a scroll expander and demonstrated that the scroll expander is a good candidate for small-scale power generation. Yamamoto et al. investigated the ORC apparatus through numerical simulation and an experiment. In their studies, R123 and water were chosen as working fluids. The results show that a better Rankine Cycle performance is found in the case when R123 is used (Zhou et al., 2013).

The generation of power using industrial waste heat has been growing in the past years. Due to the increasing energy prices, it is becoming more and more economically profitable to recover even the low grade waste heat. The potential for exploiting waste heat sources from engine and power plant exhaust gases or industrial processes is particularly promising. An often used solution is the transformation of waste heat into electricity. Amongst the available technologies to achieve that task the conventional steam turbine is a typical option. Another widely used technology is application of the organic Rankine cycle (ORC) with the low boiling-point (organic) fluid (Borsukiewicz-Gozdur, 2010b; Zhou et al., 2013).

In ORC installation the working fluid is an organic substance which is better adopted than water to undergo phase changes at lower temperatures of the heat source. The waste/geothermal/process heat or heat producing in domestic boiler can be used to produce vapour in evaporator that is being subsequently expanded in the ORC turbine to generate electricity. The temperature of the heat source is a major determinant of the type of technology required to extract the heat and the uses to which it can be applied. The ORC the experimental facility was constructed using the waste heat recovery for low-temperature flue gas, which temperature is in the range of 90–220°C. R123 was selected as the working fluid because of its non-toxic, non-flammable, no threat to the environment, chemical stability and low cost. Moreover, it has many characteristics such as high cycle efficiency, high thermal conductivity and moderate working temperature and pressure so that it is suitable to apply in this experiment. The relationships were investigated between the system output performance and the evaporating pressure, the temperature of the heat source and the superheat degree of the working fluid. The aim is to study the system performance by applying flue gas waste heat re-covery to an Organic Rankine Cycle, thus providing basis for industrial utilization (Zhou et al., 2013).

A solar-geothermal integration for electricity generation was proposed by Lentz and Almanza for possible application in the Cerro Prieto field in Mexico (Astolfi, Xodo, Romano, Macchi, 2011). The proposed plant is based on a single or double flash cycle, where a steam turbine expands the steam produced in flash separators from the geothermal brine. Different configurations were suggested to enhance steam production with solar energy, all based on the Direct Steam Generation concept, where the geothermal brine is evaporated by flowing directly in the solar collectors.

Although almost 90% of the worldwide installed geothermal power depends on dry steam or single-flash and double-flash units, binary units based on ORCs are becoming increasingly common (Borsukiewicz-Gozdur, 2010a). About 45% of the existing units are binary plants which, however, produce less than 10% of the world geothermal electricity because of their lower average size (Astolfi et al., 2011).

The power of expanding device of CHP ranges from a few to tens of kilowatts. Considered here will be application of CHP in small households, where the turbine capacity will be of the order of 2–3 kW. The better utilization of energy in the fuel in CHP leads to lowering of harmful emissions released in the process of burning the fuel, because less fuel is utilized (Zhou et al., 2013).

Worlwide demand on such micro CHP units is huge, comparable with the market of home refrigerators (Harrison, 2003). It is easy to evaluate that, in case of our country, if the domestic micro CHP units were commonly implemented that would have a total potential to replace construction of several large conventional power stations. As far as the way of executing the CHP production there are also different concepts of their construction. Such units can be piston combustion engines, gas turbines, steam turbines, Stirling engines or fuel cells. Different are also time horizons of realizations of these conceptions. Basing on own experience and knowledge we propose a concept of a CHP based on vapor turbine working in ORC cycle (Mikielewicz, 2009; Mikielewicz, & Bykuć, 2006; Mikielewicz et al., 2007b; Mikielewicz, Bykuć, Mikielewicz, 2007; Mikielewicz, Bykuć, Mikielewicz, 2006). This CHP would be operating with a low-boiling fluid in the range of considerably lower temperatures comparing to the combustion engine and gas turbine. This kind of CHP requires less valuable materials and an easier technology of its production. With the help of such approach in the design of the postulated CHP the production of electricity will take place with the prices nearing those found in traditional large power plants. Heat from the micro CHP unit can be used to warm up water for home use as well as for space heating. Electricity can be used for own needs or can be supplied to the electric grid. Source of energy for CHP depends on local possibilities. It can be from renewable sources or from conventional fuel.

In our opinion in nearest perspective micro CHP will replace conventional boilers for heating objects such as: single-family small houses, large houses, settlements etc. Sizes of boilers containing the micro CHP unit will differ a little from existing boilers but will have a function, that apart from heat additional electricity production will be possible. Domestic CHP would operate in a range of much lower temperatures than the combustion engine or the gas turbine. Such type of design requires therefore significantly less of precious materials as well as the manufacturing technology is easier. In the micro CHP power plant the electricity is produced by the generator driven by the micro turbine operating with steam or low boiling point fluid vapour. The general feature of the micro power plant is its small dimensions as well as possibility for full automation of the operation of such plant. Small dimensions of CHP are attained by using modern materials and modern micro technology. Simple materials and simple fabrications of parts of the CHP plants, working in low temperatures range, lead to low costs of electricity production.

In a paper we concentrate on ORC in domestic CHP and specifically on micro heat exchangers working in domestic CHP.

2. Thermodynamical analysis of a micro CHP

The simple organic Rankine cycle is considered, consisting namely of the evaporator, turbine, condenser and circulation pump, Fig. 1. In case of dry fluid the internal regeneration could also be considered, Fig. 2. All modifications of the ORC's improving their efficiency lead to additional heat exchanging areas and small improvements of effective efficiencies of the cycles. It is assumed that the working fluid is in the state of saturated liquid at the exit of the condenser. Its pressure is then raised to reach evaporation pressure. Subsequently the fluid is heated to reach evaporation temperature and then it is converted to saturated vapour. Hot pressurized working fluid vapour expands in the turbine performing the useful work (Mikielewicz, Bykuć, Mikielewicz, 2007; Mikielewicz, Bykuć, Mikielewicz, 2006; Mikielewicz, & Mikielewicz, 2014; Mikielewicz, Wajs, Mikielewicz, 2011).

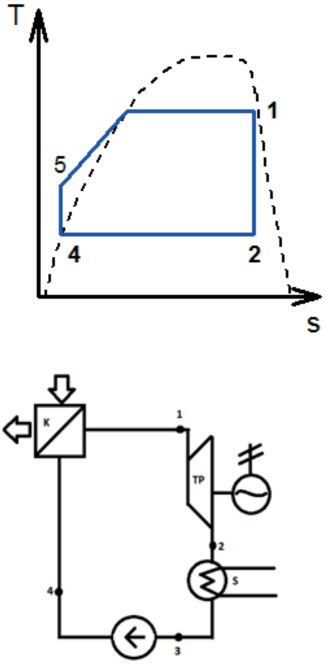

Fig. 1. ORC with negative slope of saturated vapor curve and saturated vapor at the turbine inlet

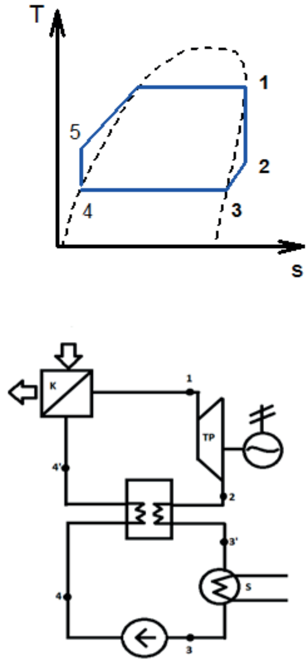

Fig. 2. ORC with positive slope of saturated vapour curve and superheated vapour at the turbine inlet

The sub-critical ORCs are different types according to the shape of the saturated vapor curve in the temperature versus entropy diagram. We distinguish two types of ORC processes with the negative or positive slope of the saturated vapor curve in the T-s diagram, as shown in Fig. 1 and 2 respectively. As shown in Fig. 1, the working fluid leaves the condenser as saturated liquid (state point 4). Then, it is compressed by the liquid pump to the sub-critical pressure (state point 5) desired in the heat addition process. The working fluid is heated in the evaporator until it becomes saturated vapour (state point 1). The saturated vapor flows into the turbine and is expanded to the condensing pressure (state point 2). At the condensing pressure, the working fluid lies in the wet (Fig. 1) or superheated vapour region (Fig. 2). The vapour passes through the condenser where heat is removed until it becomes a saturated and subsequently saturated liquid (state point 4). In the cycle, as described in Fig. 2, if the temperature t2 is markedly higher than the temperature t4, it may be rewarding to implement internal heat exchanger (IHE) into the cycles as shown in Fig. 3. In such case the turbine exhaust flows into the internal heat exchanger and cools in the heat exchanger in the process (2–3) by transferring heat to the compressed liquid that is heated in the process (5–6) Fig.3. In the present paper only the positive slope of the saturated vapour curve is considered with regeneration, Fig. 3. Next, an isentropic vapour saturation line is desirable as on one hand it prevents the turbine blades erosion and on the other hand the superheated vapour is not required to be cooled in the condenser and only pure condensation (phase change) process is required. Another feature of the working fluid is the requirement that the fluid has a higher pressure than atmospheric in condenser, which prevents the air penetration.

Up to date research was focused mainly on subcritical cycles. There is, however, an unexplored area of supercritical parameters which offers also great opportunities for exploitation. The analysis of supercritical fluid parameters may lead to higher efficiencies making the micro CHP even more attractive. The critical point of organic fluids is reached at lower pressures and temperatures compared with steam. Therefore supercritical fluid parameters are much easier to be realized in practical applications in these cases than in case of water. However, the objective for a domestic micro CHP is also designing of small sized heat exchangers and for that reason not all supercritical fluids could be suitable.

The algorithm of calculations is based on well known relations describing the Rankine cycle. It has been assumed that the maximum pressure in installation should not exceed 60 bar nor temperature of heating oil 300°C, respectively. Notation of state points for subcritical and supercritical cycles is presented in Fig. 3.

2.1. Heat supply to the ORC installation

The most common way of providing heat to the ORC installation is by means of the single-phase fluid. Such situation is schematically shown in Fig. 4. In the considered case it is required that throughout the entire heating period there is sustained a minimum temperature difference T_{min} between the source temperature and temperature of working fluid. It ought to be equal to at least 5 K, however it can be changed to the required level.

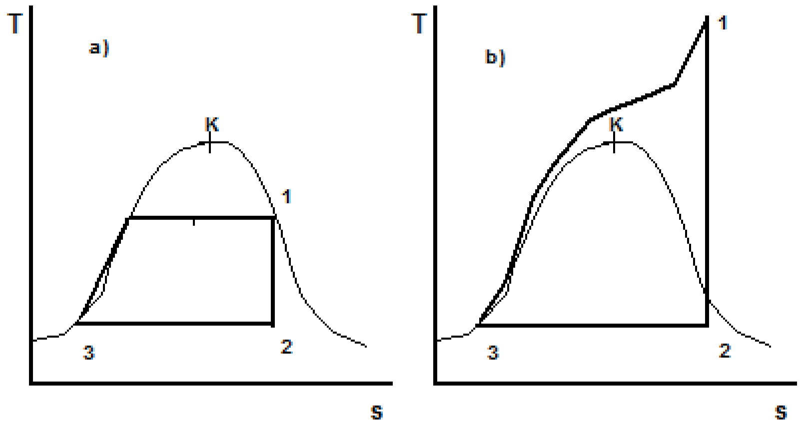

Fig. 3. Schematics of subcritical (a) and supercritical (b) cycles

A similar analysis can be carried out for the case of the heat source in the form of a fluid which can change its phase during transfer of heat to the ORC installation. A schematic of such situation has been presented in Fig. 4.

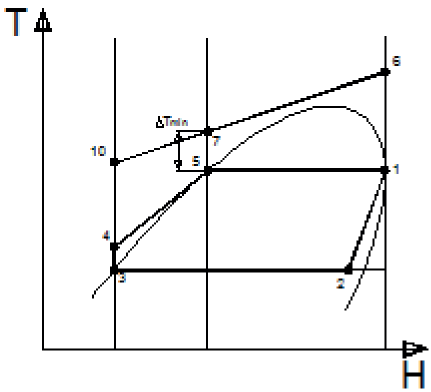

Fig. 4. Schematic of heat supply to the ORC installation by means of the single phase fluid

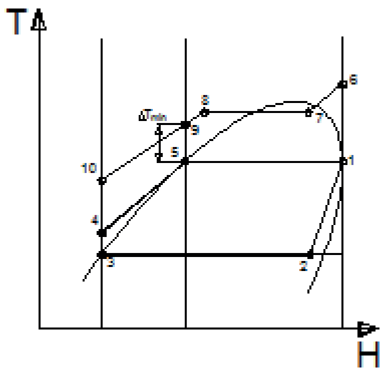

Fig. 5. Schematic of heat supply to the ORC cycle by means of the fluid changing phase, H = m·h

Properly selected evaporation temperature in the ORC installation should be determined from the optimization criterion at the condition that the logarithmic mean temperature difference should be minimised, LMTD. Variation of the logarithmic mean temperature difference as a function of the temperature before turbine for different temperatures of evaporation of ethanol and hot water inlet temperature T_6 = 90°C is shown in Fig. 6. The chart shows that the LMTD for the given evaporation temperature of the working fluid in the ORC system increases with the increase of pinch point temperature difference. On the other hand the gradient of the change of LMTD increases with lowering the evaporation temperature of the working fluid leading to excessive heat transfer surfaces in the evaporator. In Fig. 6 to 9 presented are the results of the use of waste heat, available power in ORC turbine and the ORC cycle efficiency, respectively.

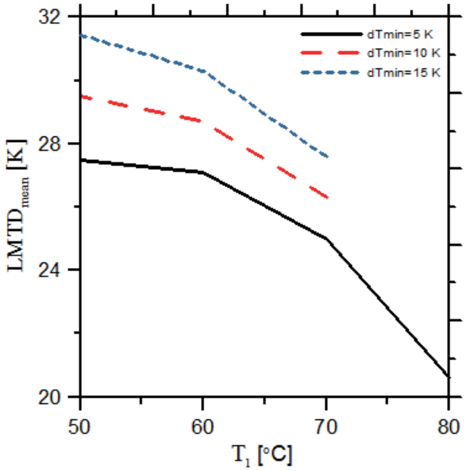

Fig. 6. Logarithmic mean temperature difference in function of temperature before the turbine for ethanol and inlet hot water temperature $T_6 = 90°C$

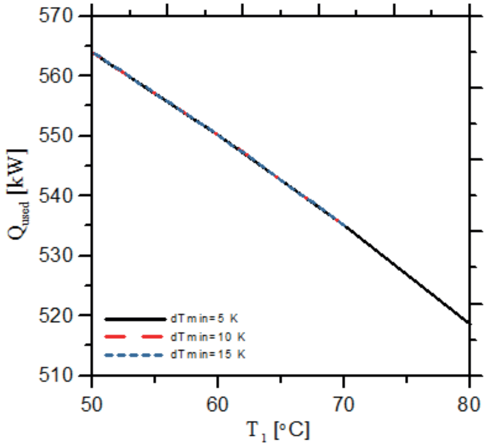

Fig. 7. Utilised waste heat supplied to the ORC installation in function of temperature before the turbine for ethanol and inlet hot water temperature $T_6 = 90°C$

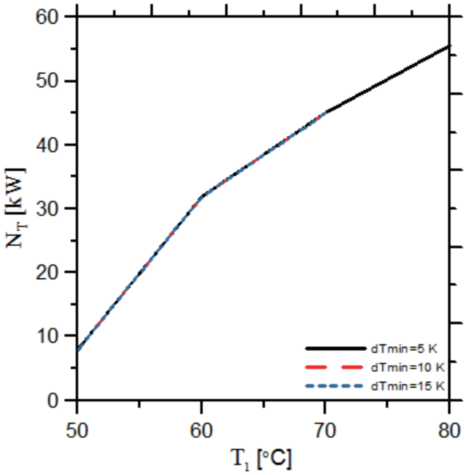

Fig. 8. ORC turbine power in function of temperature before the turbine for ethanol and inlet hot water temperature $T_6 = 90°C$

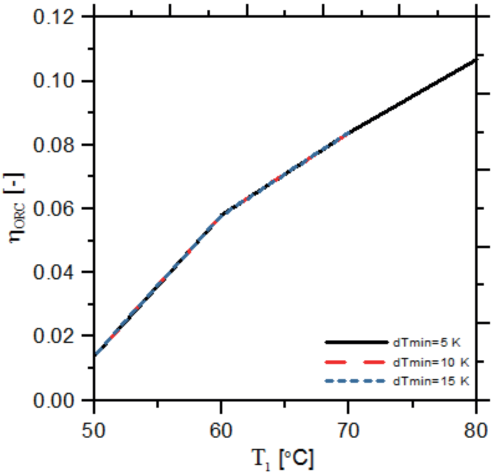

Fig. 9. Thermal efficiency of ORC installation in function of temperature before the turbine for ethanol and inlet hot water temperature $T_6 = 90°C$

In calculations it has been assumed that condensation temperature is 40°C and there is 1 kg/s of working fluid in ORC installation. Turbine power is calculated for the internal efficiency of turbine equal unity. We can see that reduction of evaporation temperature leads to increase of heat possible to be acquired from the flow rate of waste heat. However turbine power will also be increasing with increasing evaporation temperature. There is no effect of the minimum temperature difference.

To show the different configurations of ORC installation sample calculations have been done for the plant with evaporator capacity of 5 MW. For that configuration the heat source temperature was selected as 90°C and four different evaporation temperatures were considered. The results with corresponding values of thermal efficiency are presented in Table 1.

Table 1. Results of calculations for the stand alone ORC system heated from the single phase heat source having temperature 90°C. Heat added to evaporator equal 5 MW, condensation temperature 40°C

		Waste water inlet temperature/boiling temperature T_1			
		90/80	90/70	90/60	90/50
Efficiency of ORC system	—	0.107	0.084	0.058	0.014
Mass flow rate in waste heat water	kg/s	216.29	73.50	44.87	32.55
Electrical power of ORC	kW	552.66	425.9	292.11	66.87
Water outlet temperature T_{10}	°C	84.50	73.76	63.40	53.34

The installation generates power in function of the ratio of mass flow rate of hot water to the mass flow rate of working fluid in ORC installation. In case we assume the heat source is with the infinite flow rate, having temperature of 90°C, then in case of required evaporation temperatures of 80°C the production of electricity in the ORC installation is highest. In such case there will be however excessive power involved in circulation power required to drive the pump as such case requires a large mass flow rate of hot water to be circulated. In such utilization of heat the exergy losses are significant.

In the second considered instance the heat supply is from the hot thermal oil. It is assumed in calculations that oil inlet temperature, i.e. T_6 = 310°C, whereas the condensation temperature is 100°C. That is a model of the ORC plant, where the heat of condenser cooling can be used for other applications such as for example central heating. Considered are three possible working fluids, namely two silicone oils, MDM and D4 and dodecane. It must be also borne in mind that in case of a large temperature difference at pinch point there will be required a greater flow rate of thermal oil, which is directly related to the extra pumping power. That, on the other hand, is pretty large in case of silicone oils and dodecane. If that is too excessive then considered should be the decrease of the evaporation temperature, equivalent to reduction of temperature before the turbine. Obviously that leads to the reduction of the ORC cycle efficiency, but considering the extra power required for thermal oil circulation the net output of the installation as a whole could be greater. We should notice that by changing the evaporation temperature of working fluid in ORC installation we can influence the amount of required thermal oil in installation and its outlet temperature T_{10}. We can therefore influence the mean temperature difference in the evaporator and hence the efficiency. It is apparent that the greater the pinch point temperature difference and the lower the evaporation temperature the greater is the logarithmic mean temperature difference. LMTD exhibits also a decreasing tendency with increasing temperature of working fluid vapour before the turbine. In case of supplying the ORC system with a single phase heat source the LMTD for a given evaporation temperature of working fluid in ORC installation increases with the increase of temperature difference at pinch point as well as decreasing the evaporation temperature. The gradient of LMTD increase is raising with decreasing of evaporation temperature of working fluid. The greater the temperature difference at pinch point the greater is the mean temperature difference. It must also be noticed that LMTD decreases with decrease of temperature of working fluid before the turbine. That is of particular importance as the heat transfer area of heat exchangers increases and also the overall efficiency of ORC decreases.

Presented calculations should convince the reader to the fact that the issue of variable heat source temperature is a very important aspect of designing the ORC installation with external fluid supplying the system with heat.

3. Selection of appropriate working fluid for ORC

Selection of working fluid is an important aspect of attaining possibly high cycle efficiencies. That enables for optimal utilization of available energy sources. There is a wide selection of organic fluids, which can be used in ORC systems. Maizza et al. (2001) conducted investigations with different organic fluids for systems with heat recovery. The most important features of a good organic working fluid are:
- low toxicity,
- good compatibility and chemical stability in operation with other materials,
- low flammability, corrosives and small potential for decomposition.

A simple analysis has been carried out which resulted in development of a thermodynamical criterion for selection of an appropriate working fluid. Such criterion should be used with other criteria related to environmental impact, etc. It is difficult to find an ideal working fluid which exhibits high efficiencies, low turbine outlet volume flow rate, reasonable pressures, low ODP, low GWP and is non-flammable, non-toxic and non-corrosive.

In their opinion the refrigerants are most promising fluids for ORC cycles, especially with the view of their low toxicity. Another characteristic feature, important in selection of a fluid, is the boiling curve at a specified saturation temperature. That feature has a particular influence on the restrictions in application of a fluid in thermodynamic cycles (cycle efficiency, device sizes in the energy production systems). The slope of the saturated vapour curve in T-s diagram depends on the type of applied fluid. We discern here three possible cases presented in Fig. 10 the fluids which were considered in the present study have been carefully selected from amongst 24 other considered (Saleh, Koglbauer, Wendland, Fischer, 2007). They are featuring in general all possibilities of the slope of saturated steam line, namely a positive slope of saturated steam line (SES 36), a negative slope (ethanol, R134a) and almost isentropic distribution of temperature versus entropy (R141b), Fig. 10 .That has a bearing on the course of expansion line meaning that the expansion in the first case is all the way through the superheated steam region, in the second one in the wet steam region whereas in the third one partially in the wet region and finally terminating just in the superheated steam region.

In the course of selection of appropriate for use fluids a simple analysis has been carried out which resulted in development of a criterion for selection of a good fluid from thermodynamic point of view. Up to date research was focused mainly on subcritical cycles. However, there is some limitation in case of domestic micro CHPs. The objective for a domestic micro CHP is also to design small sized heat exchangers and for that reason not all supercritical fluids could be suitable. It is well known that heat transfer to single phase fluids is much less efficient than to two-phase fluids and that leads to excessive sizes of heat exchangers.

There are selection criteria leading to high thermodynamic effectiveness provided in literature but they are of no general character (Saleh et al., 2007; Tchanche, Papadakis, Lambrinos, Frangoudakis, 2009).

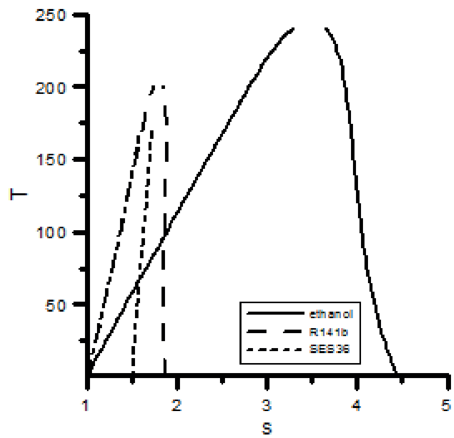

Fig. 10. T-s diagram for considered fluids calculated using Refprop 8.0

In a large installation the cost of working fluid is a very important element and hence it attracts so much attention. In some cases it can be proved that the less efficient thermodynamically working fluid is more cost efficient as it turns out to be cheaper in the overall economic balance of the system. Majority of modern fluids suitable for use in ORC installations is rather expensive. The criterion suggested in the paper is merely the preliminary indicator for initial selection. At the final selection other criteria must be also employed such as for example economic criterion or toxicity criterion.

Theoretically every fluid could be used as working fluid in the motor or refrigeration cycle, if only it is used in the appropriate temperature range. In practice there can be specified several characteristic features of the potential working fluid to be selected for use in the particular cycle. The selection of the working fluid in an important element of attaining of highest possible cycle efficiencies, which on the other hand allows for optimal utilization of available energy sources. The organic fluid as working fluid must be very carefully selected. There is a wide range of organic fluids which can be used in ORC installations (Mikielewicz, & Mikielewicz, 2010; Mikielewicz et al., 2007a).

4. The in-house project of domestic CHP

Analysis of the cycle has been conducted for two sample cases of input data, namely without vapor superheating and a small superheating. As a result of calculations obtained have been small dimensions of heat exchangers as well as acceptable thermal efficiencies for both cases. In parallel to production of 20 kW of thermal energy, typical value to larger dwellings, at the same time 1.2 kW or 1.6 kW respectively of electricity is produced for the cases without superheating and a small superheating. It is envisaged to obtain greater values of turbine power assuming same production of heat by application of different working fluids or optimization of the cycle parameters.

A micro cogenerative power plant utilizes energy in fuel up to 90%, see Fig. 11. In authors opinion that is by far the best utilization of chemical energy contained in the fuel (Harrison, 2003).

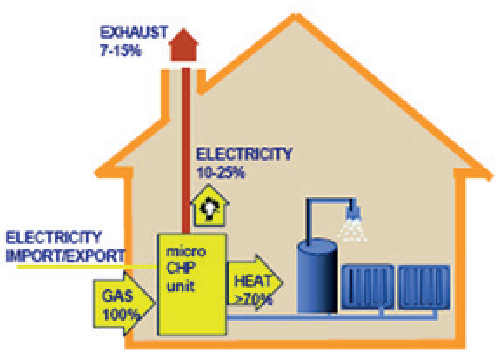

Fig. 11. A general schematic of a domestic micro CHP

About 70 to 80% of energy is produced to cover the demand for the heat, whereas about 10 to 20% is an additional production of electricity. Conventional power plant producing electricity only utilize the energy contained in fuel only and up to 40%. Better utilization of fuel energy in micro CHP leads to reduction of harmful emissions accompanying the combustion process.

4.1. Thermodynamic cycle

For the sake of calculation of the efficiency of thermodynamical cycle of considered micro CHP a special code has been developed enabling consideration of various working fluids (Zhou et al., 2013). The code calculates also the basic dimensions of such heat exchangers as condenser and boiler. Due to the fact that the dimension of the micro CHP is primarily determined by the size of heat exchangers, hence in the preliminary calculations the turbine size was neglected in calculations. It has only been assumed that the turbine does not influence the volume occupied by the whole arrangement (Harrison, 2003; Mikielewicz, 2010).

According to Fig. 12 the following parameters have been assumed in the present analysis: heat demand of 20 kW, temperature of heated water at inlet and outlet from the condenser respectively 20°C and 50°C, inlet oil temperature in boiler 320°C, turbine and pump efficiency 0.8 and 0.95, respectively, diameter of boiler and condenser tubes d = 3 mm. In all cases superheating in the boiler is present. The thermal oil is heated in the installation to temperature of 250°C and then it removes its heat in evaporator converting in such way the liquid of working fluid into vapour. The vapour of working fluid has a temperature of 200°C in case of such fluids as R141b and C_2H_5OH, 170°C for SES36 and 95°C for R134a.

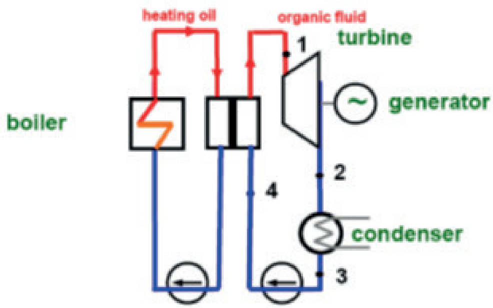

Fig. 12. Schematic of thermodynamic Rankine cycle of micro CHP

Temperature of oil leaving the heat exchanger was assumed 230°C. Physical properties of working fluids have been taken from the software Refprop 8 (2007). In the frame of calculations determined have been such parameters as: total cycle efficiency, η_R, Carnot efficiency, η_c, and the exergy efficiency η_b. The results of calculations have been presented in table 2 and 3.

Table 2. Characteristics of considered fluids and cycle efficiencies

Fluid	$\dot{Q}_S$	N_T	p_{cr}	t_{cr}	M	p_o	p_k	p_o/p_k	t_{max}
	kW	kW	bar	°C	kg/kmol	bar	bar	—	°C
SES36	20	5.36	28.49	177.55	184.50	25.07	1.62	15.475	170
R141b	20	4.51	42.12	204.35	116.95	39.48	1.83	21.578	200
C_2H_5OH	20	5.10	61.48	240.75	46.00	29.90	0.30	100.7	200
R134a	20	1.88	40.593	101.06	102.03	35.91	13.18	22.09	95

Table 3. Characteristics of cycle efficiencies

Fluid	η_R	η_c	$\eta_b = \eta_R / \eta_c$
SES36	0,205	0.271	0.755
R141b	0,178	0.317	0.563
C_2H_5OH	0.201	0.271	0,742
R134a	0.070	0.122	0.574

It results from table 2 and 3 that in case of calculations without account of pressure losses in cycle (SES36, R141b, C_2H_5OH) are the best fluid for application in micro CHP. It ought to be mentioned that the above fluids have yet to be tested for applications involving production of heat and power in available worldwide literature.

4.2. Some elements of domestic CHP

4.2.1. Turbine

The turbogenerator system which is used in a small power source, of power output ranging between 1 and 20 kW, can be based on the Brayton or Rankine cycle and consists of a turbine and a generator. Other working principles of expansion devices can be based on adaptation of scroll expanders or vane compressors. In case of turbines the system requires high revolutions to generate sufficient power at the small-size turbine. The existing conventional oil-lubricated sliding or roller bearings reveal performance limits under such conditions, especially when the lifetime or the stability of a bearing system is considered. Development of new technologies, along with the availability of modern machine design materials, provide opportunities for considering the idea to apply the working fluids used in small capacity high-speed turbomachines as a lubricating medium for its bearings very realistic. Application of such types of bearings leads to a design of the oil-free shaft support system. Elimination of the oil system and seals related to it makes it finally possible to maintain absolute purity of the working medium and to ensure operation of bearings at high and low temperatures.

Therefore, while studying possible applications of non-conventional lubricating media, one should analyze thoroughly the dynamics of the "rotor-bearing-casing" system within the whole range of machine operation. The presented here concept of the micro-turbo-generator is featuring the following assumptions:

– electrical power output 2–4 kW,
– working medium of the turbine — ethanol,
– oil-free technology for the bearing system design,
– turbine shaft integrated with a electrical generator.

As far as small high-speed turbo-generators are concerned, the application of bearings lubricated with a low-viscosity working medium (gas or liquid) makes it possible to built a "hermetic" machine without a rotating shaft end protruding outside the casing and increase total efficiency of the machine by several or more per cent via decreasing hydrodynamic friction losses. This concept is very useful for closed cycles of the ORC systems.

One of the basic problems connected with a practical application of above mentioned high-speed machine concept equipped with non-conventional gas (steam) lubricated bearings, is the machine operational reliability under various working conditions, which requires adjustment of the machine design at the early stage of the study. The main factors of this adjustment connected to the oil-free technology specificity are as follows:

– possible diminution in static thrust and lateral loads of the bearings by a correct design of the flow structure,
– correct selection of the gas bearing type.

Within the performed design analyses the following types of axial micro turbines were taken into account:
– single-stage axial-flow turbine,
– two-stage axial-flow turbine,
– four-stage axial-flow turbine,
– five-stage axial-flow turbine,
– seven-stage axial-flow turbine.

For all the variants preliminary optimisation of the main design parameters was performed and, as a result, a 7-stage axial turbine with partial admission of all the stages was chosen for further detailed design calculations. For this solution, compared with other variants, very high value of output was obtained (about 3 kW) and relatively low speed of rotor (about 8000 rpm). The particular data of the turbine geometry are as follows: (-) **stage diameter: 100 mm**; (-) **blade height: 10 mm**; (-) **partial admission arc for the first stage: 0.05**; (-) **partial admission arc for the fifth stage: 0.4**; (-) **rotor speed: 8000–12000 rpm**; (-) **turbine power: 3 kW**.

Due to partial admission in all stages it was possible to keep the values of the main design parameters within the range typical of steam turbines (velocity ratio, degree of reaction, blade height to chord ratio or Mach number). This allowed to apply typical nozzle and blade profiles. The turbine consists of 7 similar (almost identical) stages with constant section blades, the only difference being the increase of the admission arc (number of nozzles) in the successive stages. The general view of the turboset coupled with the electric generator is shown in Fig. 13. The turbine performance was calculated for ethanol as working fluid but also for air and nitrogen as a working medium for testing purposes. In the first stage of experiments the turbine behaviour was checked using these gases and the results were compared with the calculation data. Accuracy of the pressure measurement was estimated taking into account the data of the producers of the elements of the measuring system and appeared to be equal to less than 1% (accuracy of the pressure sensors — 0.5%, accuracy of the amplifier/converter — 0.16%).

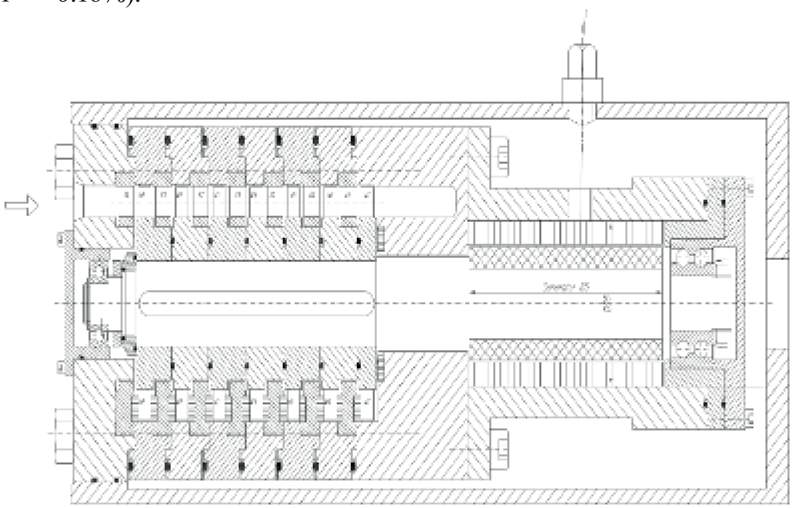

Fig. 13. General view of the turbogenerator

4.2.2. Heat exchangers

Nowadays, traditional cooling techniques are not suitable in many fields of technology, because of a large size of cooling systems and their low efficiency. Miniaturization such applications as computers, biomedicine automobile, aerospace and domestic CHP technology are required. Therefore, over the past few decades various micro heat exchangers were developed for example; shell and micro tubes exchangers, impinging jets, micro channels etc. Compared to other conventional heat exchangers micro channel systems seems to be most effective in applications where high efficiency and small size required. The development of micro heat exchangers began with solving the heat dissipation problem in integrated electronic circuits. The idea of using micro channel for the cooling of very large-scale integration circuit was first proposed by Tuckerman in 1984. He used etching and precision sawing techniques to manufacture 280 micron deep micro channel in 500 micron thick silicon wafers. He reported laminar water flow in the micro channels absorbing a heat flux of 150 W/cm^2 (Mikielewicz et al., 2007a). Friedrich and Kang fabricated micro channels on metal foil using diamond machining techniques. The metal foils are then bonded together using diffusion bonding. The total heat transfer rate can reach 300 MW/m^{3}K (Rybiński, 2013). Compact heat exchangers (CHE) are characterized by having a comparatively large area density. Area density is the ratio of heat transfer surface to heat exchanger volume. Their large area density, indicating small hydraulic diameter for fluid flow, results in a higher efficiency than conventional shell and tube heat exchanger in a significantly small volume. A CHE has an area density over 700 m^2/m^3 and hydraulic diameter less than 3 mm. Human lungs are one of the most compact heat exchangers example, having an area density of about 17500 m^2/m^3. Presently some micro-scale heat exchangers are as compact as human lung and even more compact (Rybiński, 2013).

Different types of heat exchangers can be used for above presented applications, the most common being shell & tube (mainly in larger-scale systems) and plate heat exchangers (mainly in small-scale systems, due to their compactness).

An idea of novel plate heat exchanger with mini channels is presented together with thermal-hydraulic characteristics of its prototype. This kind of heat exchangers could be prospectively applied in domestic ORC systems but other applications are also possible.

There has been presented a simplified routine for design of micro heat exchangers working in ORC cycle that is evaporator and condenser with account of pressure drops in heat exchangers as well as compared have been different models for calculation of two-phase pressure drop in small diameter tubes. As a result of calculations there have been obtained small dimensions of heat exchangers.

4.2.3. Impact of micro channels hydraulic diameter on heat exchangers size

For the laminar flow, such flow takes place in micro and mini channels, Nusselt number is constant. Area density which is the ratio of heat transfer surface to heat exchanger volume can be written as:

$$\frac{\dot{Q}}{V} = \frac{k \cdot \Delta T \cdot A}{V}, \tag{1}$$

where:

$$A = n \cdot \pi \cdot d_h \cdot L, \quad V = 2 \cdot n \cdot \pi \cdot \frac{d_h^2}{4} \cdot L, \quad k = \frac{1}{1/\alpha_1 + 1/\alpha_2} \approx \frac{\alpha}{2} = \frac{\lambda \cdot Nu}{2 \cdot d_h}. \tag{2}$$

Introducing (1) to equation (2) we have:

$$\frac{\dot{Q}}{V} = \frac{k \cdot \Delta T \cdot A}{V} = \frac{2 \cdot k \cdot \Delta T}{d_h} \approx \frac{\lambda \cdot Nu \cdot \Delta T}{d_h^2}, \tag{3}$$

The outcome of equation (4) means that:

$$\frac{\dot{Q}}{V} = \propto \frac{1}{d_h^2}. \tag{4}$$

The lower hydraulic diameter of the mini channels in heat exchanger the higher heat transfer rate density.

4.2.4. The methods of heat transfer enhancement in micro channels heat exchangers

There exist few methods of increasing heat transfer coefficient in micro channels. One of them was presented above; it is to reduce of hydraulic diameter of the micro channels of the micro heat exchangers. Another one is to take advantage of the fact that developing flow has higher heat transfer coefficients than fully developed flow, what is schematically presented in Fig. 14. So short micro channels of the length shorter then thermal entrance region should be applied for increasing heat transfer coefficient.

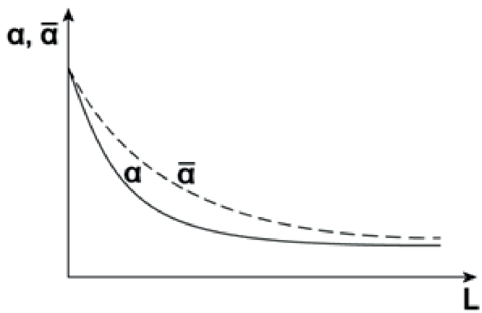

Fig. 14. Change of local and mean heat transfer coefficient along the pipe

Other methods base on use of corrugated or rough mini channels.

4.2.5. New concept of micro channel heat exchangers developed PG

There are conducted research of new concepts of micro heat exchangers:

1. Micro jet heat exchangers.
2. Plate micro channels heat exchangers with artificially roughed surfaces.
3. Shell & tube heat exchangers with porous layers supporting flow with the aim of capillary forces.
4. Works concerning nets of micro heat exchangers replacing high power heat exchangers.

4.2.6. Micro jet heat exchangers

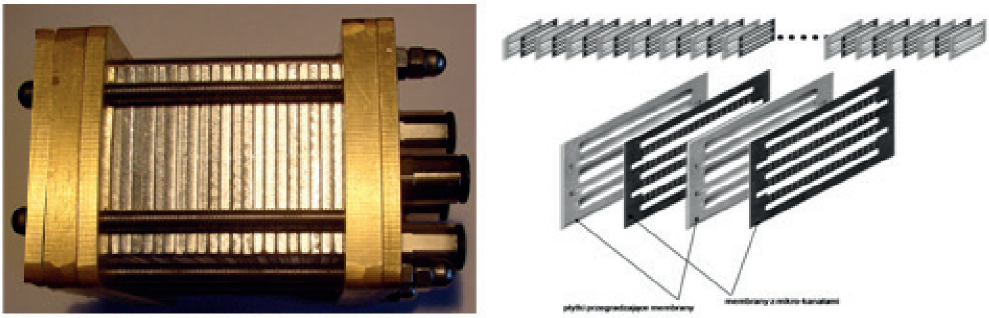

Fig. 15. Micro jet heat exchanger and its thin metal plates

4.2.7. Mini channel plate heat exchanger

The presented mini channel heat exchanger is a screwed unit and its main part consist of brass plates of 2 mm thickness with cut mini channels. Their length is 50 mm, width — 1 mm and depth — 0.7 mm, respectively. Each plate includes 51 mini channels and the distance between adjacent channels is equal to 1 mm. Additionally, the collectors of working fluid (inlet and outlet) are made on the plates. Their depth is 1 mm and width varies from 4 to 20 mm. The variable width results from a requirement of similar flow rate distribution in each mini channel. Modular construction allows for flexible increase of heat transfer area (Refprop 8.0, NIST, 2007).

Several types of heat transfer enhancement methods were tested:

1. Version I — heat exchanger with unmodified mini channels.
2. Version II — heat exchanger with mini channels cut perpendicularly.
3. Version III — heat exchanger with mini channels cut at 60 angle.
4. Version IV — investigations carried out on the roughness made by glass machining.

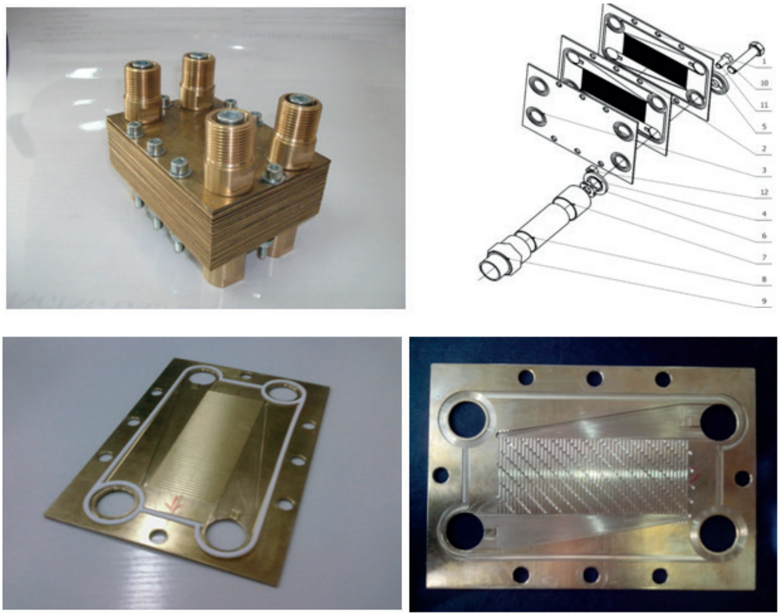

Fig. 16. Mini channel heat exchanger and view of plates with mini channels of different types. 6. Experimental investigations of prototype micro heat exchangers

4.2.8. Experimental facility

The analysis was conducted on model heat exchanger built-up of one hot and one cold passages. It was prepared in such a way for the purpose of better understanding of heat transfer in the novel construction. The hot water was circulating in the system, prepared by the electric heater, while the cold water was a tap water. The heat was transferred in the counter-current flow of working fluids.

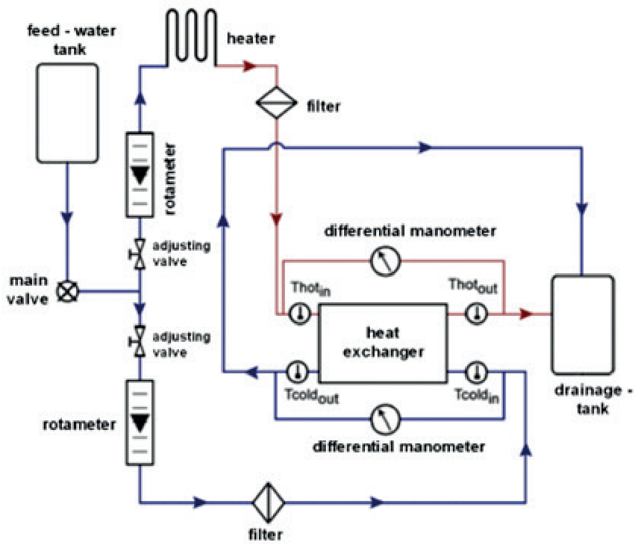

Fig. 17. Schematic view of experimental set-up

The fluid flow rates were measured by means of rotameters with accuracy of ±3 l/h. The heater was controlled by the power supply in the range 0–100% of heating power. Input temperature of hot fluid was taken as a variable parameter. The pressure drop was measured by means of differential pressure transducer with accuracy of 0.25% of full scale. Thermocouples of J-type were used to measure temperature in four points i.e. at the inlet/outlet of heat exchanger's cold side and at the inlet/outlet of heat exchanger's hot side (Refprop 8.0, NIST, 2007).

4.2.9. Modified Wilson plot method for measurement of micro heat exchangers (Rybiński, 2013)

Thermal-flow characteristics of micro and mini channel heat exchangers cannot be measured directly because it is very difficult or impossible to measure fluids temperature by use of thermocouples put into the narrow channels. It is however possible to measure total fluids mass rate, pressure and bulk fluids temperature at inlets and outlets flow. Variation of these parameters in a series of measurements allows separation of total thermal resistance between two fluids:

$$R = \Delta T_{\log} / \dot{Q}$$

into three partial resistances R_{oil}, R_{wall}, R_{fluid} by use of statistical methods. Since the separated thermal resistances are available, use of already known heat transfer area A allows derivation of heat transfer coefficients α_{oil}, α_{fluid} and their dependence on mass flow rates m_{oil}, m_{fluid} known from the used correlations.

$$\alpha_{oil} = \frac{1}{R_{oil} \cdot A}, \qquad \alpha_{fluid} = \frac{1}{R_{fluid} \cdot A}. \tag{5}$$

This way of statistical determination of heat exchanger characteristics is generally called a Wilson plot method. There are many versions suitable for different situations.

There are difficulties in the use of common versions of Wilson plot method in the case of an evaporator or condenser. For example in the evaporator there generally exist three zones: liquid fluid preheating, boiling, fluid vapour overheating (Fig. 18). These zones have very different heat transfer coefficients. It is also impossible to use one of the typical Wilson plot method for these zones because partial heat transfer areas of these zones A_1, A_2, A_3 are unknown.

To solve these problems, the new versions of Wilson plot method have been designed and successfully implemented (Rybiński, 2013; Mikielewicz et al., 2013).

Apart from theoretical research into the problem the experimental activities also started aiming at development of the prototype realizing the ORC cycle. Experimental evaporator and condenser are the plate heat exchangers. Thus the preliminary experiments have been carried out. The obtained results are quite encouraging. Experimental investigations of the system were carried out with a micro turbine of in-house design.

5. Conclusions

It seems that further development of the micro CHP is very attractive . Many laboratories worldwide are involved in the development of such device, however without a breakthrough success thus far. It can be concluded from the presented analysis that critical temperature of the working fluid has a significant influence on the effectiveness of operation of the ORC cycle, particularly on the cycle power and dimensions of heat exchangers. At the assumption that analyzed are only subcritical cycles it ought to be said that the working fluid should be evaporated as closed as possible to the critical point. That statement is of particular importance in the problems of utilization of low temperature waste heat or geothermal heat. In subsequent analyses there ought not to be exclude possibilities of application supercritical cycles as well as wet subcritical ones.

It seems from the conducted calculations, that in the case of neglecting pressure losses the best fluid for application in a micro heat and power plant is the synthetic fluid SES36, followed next by ethanol and R141b. The above conclusions have been arrived at by consideration of the cycle efficiency and turbine power. Very perspective working fluid seems to be ethanol. It belongs to the so called wet fluids and the quality at the end of expansion process are usually greater than x = 0.9, which should not lead to operational problems with the turbine. Ethanol is an organic fluid and first of all is non-toxic. A drawback is a low pressure in the condenser, much lower than the atmospheric one, which can lead to leakages of air and moisture into the system. The prototype of the micro heat and power plant developed at the Heat Technology Department of Gdansk University of Technology seems to prove the case that micro CHP are the future of micro generation.

Nomenclature

A — heat transfer area, m^2
c_p — specific heat at constant pressure, J/(kgK)
d_h — mini channel hydraulic and thermal diameter, m
h — specific enthalpy, J/kg
l — cycle work per unit mass, kJ/kg
k — overall heat transfer coefficient, W/(m^2K)
L — exchanger length, m
— mass flow rate, kg/s
Nu — Nusselt number
n — number of fluid mini channels
R — thermal resistors
Q — heat per second
q — heat per unit mass, kJ/kg
T — temperature, K
V — total mini channels volume, m^3
X — quality

Greek symbols

α — heat transfer coefficient, W/ (m^2K)
λ — thermal conductivity, W/ (mK)
η — cycle efficiency

Subscripts

1 — state before expansion machine
2 — state after expansion
3 — state at outlet from condenser
4 — state after leaving the pump
c — related to Carnot cycle
in — input heat
lv — related to latent heat of evaporation
l, v — saturated liquid or vapour

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