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Micro Heat and Power Plants Working in Organic Rankine Cycle

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Abstract

Renewable heat sources are rarely suited to the temperature requirements of modern thermal power plants. Thus, our unique opportunity is to deliver to the market power plants optimized for these unused and overlooked thermal resources utilizing Organic Rankine Cycle (ORC). The ORC is similar to the cycle of a conventional steam turbine, except for the fluid that drives the turbine – a high molecular mass organic fluid, usually Freon or another low-boiling fluid. This paper analyzes micro combined heat and power plants (micro CHP) operating on ORC, which aims to replace conventional boilers in homes. The heat power of micro CHP is in the range of from ten to a hundred kilowatts, and electric power in the range of from a few to tens of kilowatts. Analysis concerns selection of a cycle, calculation of thermodynamic parameters, and determination of basic dimensions of heat exchangers: condenser and evaporator.

Keywords: thermodynamic cycle, organic Rankine cycle, micro CHP

Introduction

In recent years we have observed the tendencies arround the world and in the European Union of the growth of meaning of the energy dispersed generation sector based on local resources and technologies using both conventional fuels and renewable energy. Existing development of the conventional power sector is based on building more and larger power stations. The larger the power of a unit producing electricity, the smaller the cost of production of the unit of electricity. Large power units, however, have led to many operational problems.

The versatile processes of production of useful energy in energy dispersed generation sector has become attractive. Processes include cogeneration, trigeneration, and poligeneration, especially in reference to small and average-scale plants. The most popular is a cogenerative process that is a simultaneous production of heat and electricity. The aim of the EU is to achieve 18% in the market share in cogeneration realized in small and average-scale production units of electric energy up to 2010. The idea of dispersed cogeneration is particularly attractive in the 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 EU up to 2010. For Poland this requirement is set at 7.5% of total energy balance, which is still large and difficult to obtain (we had a share of RES at approximately 1.6% in 2000).

The best and most economically sound method of utilizing renewable sources of energy are the so-called agro energetic complexes, in which one technological process uses the accessible supplies of renewable energy to make its conversion to useful energy.

This means that such complexes from one side will realize processes connected with the production and processing of biomass, including esterification or gasification of waste that will produce heat and electric current in small co-generative plants - CHP. They utilize electricity generated from: wind, solar photovoltaic (PV), fuel cells, hydro, and heat generation from biomass, solar thermal collectors, and

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heat pumps as well as micro CHP, which produces heat and power from renewable or fossil fuels. Some of the mentioned technologies are mature and have demonstrated their potential performance and production costs, whereas others are at their earliest stages of market entry and are still relatively expensive and may not perform to their expected potential. This latter technology is currently produced in a small amount, but is likely to become significantly cheaper over the next few years as it enters mass production. At the same time, electrical efficiency levels (currently at around the 10% mark) are likely to rise as other products with fuel cell-based systems become available in the next decade or so with expected electrical conversion efficiencies of over 40%.

The power of expanding CHP devices ranges from a few to tens of kilowatts. This work considers the application of CHP in small households. Primary energy in such CHPs is better utilized than in units producing electricity only. Almost 90% of CHP units utilize the energy of fuel. About 70 to 80% of energy is delivered as heat, and about 10 to 20% is additional production of electricity (Fig. 1). Conventional units producing electricity achieve a rate of conversion of about 40%. Such units use energy contained in conventional fuels. The better utilization of energy in CHP fuel leads to lowering the harmful emissions released in the process of burning the fuel, because less fuel is utilized.

Biomass is only economically viable where there is no natural gas supply, and where a local biomass fuel supply is available. It is an alternative to fuel oil or LPG where no direct network of gas is available and, like them, requires fuel storage that can be in any dry building near the boiler. The economics of such venture depend on the cost of the local fuel supply, but the costs are generally competitive with other fuels. More importantly, biofuels are less susceptible to the highly volatile price variations in oil and gas prices and should become increasingly competitive. In environmental terms, biomass (provided the fuel is locally available or from a sustainably managed forest) can make a significant reduction in household CO_2 emissions, typically by 4-10 tons a year depending on the fuel considered. As wood absorbs carbon during its growth and releases CO_2 when it is burnt, it is considered a “carbon neutral” fuel (if the fuel source is managed sustainably to make sure it can be continuously harvested). However, some CO_2 is also released by processing and transporting the fuel so it is not entirely carbon neutral in practice. Although it is advantageous to source local biomass to minimize transport emissions, the urban myth that biomass loses its environmental benefits if it has to be transported more than 25 km is entirely unfounded. Even if shipped to Poland from Siberia, it is still lower carbon content than natural gas. A biomass boiler is installed with a conventional (radiator) central heating system. It burns biomass, usually wood pellets, in place of gas, oil, or LPG. It is somewhat larger than a gas- or oil-fired boiler and requires a substantial fuel store. The wood pellet fuel is stored in a bulk container from which a vacuum tube draws the fuel to a small store next to the boiler itself. The boiler then draws the pellets as required to the boiler, where it is first heated to produce combustible gas; this is then burnt to heat water as in a conventional boiler to provide space and water heating.

An examination of the market in Great Britain shows that demand for such micro CHP units is huge, and comparable to the market for home refrigerators [1]. The power of such units ranges from several to tens of kilowatts. It is easy to note that, in the case of Poland, we would have the potential to replace construction of large conventional power stations. As far as exectuting 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 each with different timeframes. Based on our own experience and knowledge we proposed at the Institute of Fluid Flow Machinery PAN in Gdańsk a concept for a new CHP based on a vapor turbine working in ORC cycle. This CHP would work on low-boiling fluid in range of considerably lower temperatures compared to the combustion engine and gas turbine. This kind of CHP requires less valuable materials and an easier technology for its production. With the help of such an approach in the design of the postulated CHP, the production of electricity would take place with the prices nearing those found in traditional large power plants.

Heat from a micro CHP unit can be used to warm water for home use as well as for spatial heating. Electricity can be used for our own needs or can be supplied to the electrical grid. The source of energy for CHP depends on local possibilities. It can be from renewable sources or from conventional fuel.

In the near future micro CHP will replace conventional boilers for heating such objects as single-family small houses, large houses, and settlements, etc. The sizes of boilers containing the micro CHP unit will differ a little from existing boilers but, apart from heat producion, will have additional electricity production. Typical micro CHP unit costs are estimated to be about 1,000 € more expensive than a boiler itself, but will offer economic and environmental benefits to the household. An average home with annual heat demand of 18,000 kWh will generate around 3,000 kWh of electricity; about 2,000 kWh will be consumed at home, with 1,000 kWh sold to the network. This electricity is typically worth around 150-200 €, depending on how much is consumed by the household and how much is sold back to the supplier and at what price. Although it consumes slightly more gas than a modern high-efficiency boiler, the net savings is still more than 150 € for a family home per year. In this example, the unit will therefore pay for itself in around 6 years [1]. However, as most commentators have noted, the target market for micro CHP is not the average home, but homes with at least average consumption, hence 12 million homes in the case of UK. For the average home, carbon savings of 1 ton CO_2 per year can be achieved; a larger family home could expect to generate 4,000 kWh or more, providing an economic benefit of up to 500 € – a carbon savings in excess of 1.5 tons annually.

The Concept of Domestic CHP

The author proposes the use of concept of a new co-generative vapor power plant-CHP for domestic use. It would be a vapor power plant operating with the vapor of a low boiling point fluid, such as that used in refrigeration applications. It would operate under much lower temperatures than a combustion engine or gas turbine. Such a design would require significantly fewer precious materials and be easier to manufacture. It is also quite feasible that the unit price of produced electricity would be comparable to prices found in professional power plants. 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 vapor. The general feature of the micro power plant is its small dimensions as well as the possibility for full automation of the operation of such a plant. The small dimensions of CHPs 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, lead to low costs of electricity production. In the future, the co-generative micro CHP will replace conventional boilers for heating of such objects as single households, multi-flat apartments, housing estates, etc. By size it will not be much different from present boilers, but it will also produce electricity. Intense activities can be found across the world, but in the author's opinion, domestic experience and knowledge should place us as leaders in the countries involved in that topic.

The Organic Rankine Cycle (ORC) is similar to the cycle of a conventional steam turbine, except for the fluid that drives the turbine, which is a high molecular mass organic fluid. The selected working fluids allow the exploitation of efficiently low temperature heat sources to produce electricity across a wide range of power outputs. The organic chemicals possible for utilization by an ORC include freons and other traditional refrigerants such as pentane, CFCs, HFCs, butane, propane, and ammonia.

A simplified routine for calculations of a cycle taking account of pressure drops in heat exchangers, as well as a comparison of different models for calculating of two-phase pressure drop in small diameter tubes is presented. Analysis of the cycle has been conducted for two sample cases of input data: without vapor superheating and low superheating. Results obtained acceptable thermal efficiencies for both cases. In parallel to the production of 20 kW of thermal energy (a typical value for larger dwellings), 1.2 kW or 1.6 kW, respectively, is produced for cases without superheating and with small superheating. Greater values of turbine power (assuming the same production of heat by application of different working fluids or optimization of the cycle parameters) are needed.

Selection of Appropriate Working Fluid for ORC

The selection of working fluid is an important aspect of attaining possible high-cycle efficiencies, enabling optimal utilization of available energy sources. There is a wide selection of organic fluids that can be used in ORC systems. Maizza et al. [2] 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.

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 a T-s diagram depends on the type of applied fluid. We discern here three possible cases presented in Fig. 2. The fluids considered in the present study have been carefully selected from among 24 [3]. They feature in general all possibilities of the slope of saturated steam line, namely the positive slope of saturated steam line (SES 36), a negative slope (ethanol, R134a), and almost isentropic distribution of temperature versus entropy (R141b) (Fig. 1). That has a bearing on the course of the 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, and in the third one partially in the wet region, and finally terminating in the superheated steam region.

A simple analysis was conducted that resulted in the development of a criterion for fluid selection. The analysis commences with the expression for cycle efficiency (Fig. 3):

$$\eta = \frac{l_{cycle}}{q_{in}} = \frac{h_1 - h_2}{h_1 - h_3} \tag{1}$$

In relation (1) enthalpy change due to the presence of the pump have been neglected. Enthalpies present in (1) can de written in terms of a corresponding liquid saturation state, and the enthalpy prior to expansion can be written in terms of h_3 (end of condensation):

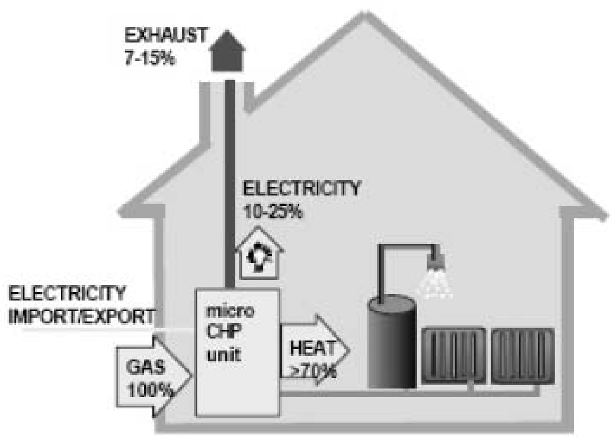

Fig. 1. General schematic of a micro CHP.

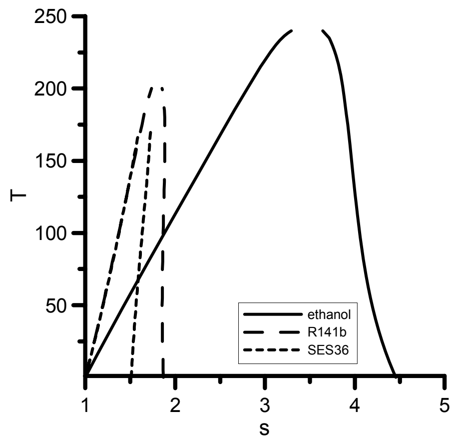

Fig. 2. T-s diagram for considered fluids calculated using Refprop 8.0.

$$h_1 = h_3 + c_p\left(T_1 - T_2\right) + h_{lv_1} \tag{2}$$

And after expansion:

$$h_2 = h_3 + x_2 h_{lv_2} + \Delta h_{superheat} \tag{3}$$

Relation (3) is a general formula describing the state after expansion in turbine. In the case of dry fluids x_2=1 and in the case of wet fluids $\Delta h_{superheat}$=0. In general, two latter terms in relation (3) can be combined to yield:

$$h_2 = h_3 + \Delta H\left(T_2\right) \tag{4}$$

Substituting this information into (1) we obtain the cycle efficiency

$$\eta = \frac{h_2 + c_p\left(T_1 - T_2\right) + h_{lv_1} - h_3 - \Delta H\left(T_2\right)}{h_3 + c_p\left(T_1 - T_2\right) + h_{lv_1} - h_3} = 1 - \frac{\Delta H\left(T_2\right)}{c_p\left(T_1 - T_2\right) + h_{lv_1}} \tag{5}$$

Temperature difference between condensation and evaporation levels can be expressed in terms of Carnot cycle efficiency and then:

$$\eta = 1 - \frac{\dfrac{\Delta H\left(T_2\right)}{h_{lv_1}}}{\dfrac{c_p T_1}{h_{lv_1}}\eta_c + 1} = 1 - \frac{\dfrac{\Delta H\left(T_2\right)}{h_{lv_1}}}{Ja\left(T_1\right)\eta_c + 1} \tag{6}$$

Analysis of relation (6) enables us to conclude that overall cycle efficiency is a function of a ratio $\Delta H(T_2)/h_{lv_1}$ and the Jakob number. It stems directly from (6) that we should consider the ratios of $\Delta H(T_2)/h_{lv_1}$ and c_p/h_{lv_1} when we want to consider a substance as a working fluid. In other words, it is not only saying that the fluid should feature a high value of specific heat and a low value of latent heat of evaporation, but the ratio of these values should assume high values for specified values of temperatures of upper and lower heat sources. Similar analysis can be performed for the denominator in equation (6), from which it results that the ratio $\Delta H(T_2)/h_{lv_1}$ should assume the smallest possible values in order to attain high overall efficiency values. $\Delta H(T_2)/h_{lv_1}$ and $Ja(T_1)$ values have been calculated for all fluids considered in the present study and presented in Table 1.

Thermodynamic Analysis of a Micro CHP

For the sake of calculating the efficiency of the thermodynamic cycle of the considered micro CHP, a special code has been developed to enable consideration of various working fluids [4]. The code also calculates 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 turbine size was neglected in calculations. It has only been assumed that the turbine is to be of the single-stage radial type and does not influence the volume occupied by the whole arrangement [5].

According to Fig. 3, the following parameters have been assumed in the present analysis (Table 1): heat demand of 20 kW; temperature of heated water at inlet and outlet from the condenser 20ºC and 50ºC, respectively; inlet oil temperature in boiler 320ºC; 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 250ºC and then it removes its heat in the evaporator, converting in such a way that the liquid becomes vapour. The vapour of working fluid has a temperature of 200ºC in case of such fluids as R141b and C_2H_5OH, 170ºC for SES36 and 95ºC for R134a.

The temperature of oil leaving the heat exchanger was assumed to be 230ºC. Physical properties of working fluids have been taken from the software Refprop 8.0 [6]. The following parameters have been determined: total cycle efficiency, η_R; Carnot efficiency, η_c; and exergy efficiency, η_b. The results of calculations have been presented in Table 2.

It results from Tables 1 and 2 that in the case of calculations without account of pressure losses in the cycle, (SES36, R141b, C_2H_5OH) are the best fluid for micro CHP application.

It should be mentioned that the above fluids have yet to be tested for applications involving production of heat and power in available worldwide literature.

Pressure Drop and Heat Transfer in Two-Phase Flow in Boiler and Condenser

In literature there are a number of both experimental and theoretical correlations describing pressure drop in two-phase flow, but their applicability is limited to restricted working fluids or vapor qualities. The pressure drop consists of three components, namely pressure drop due to friction, acceleration, and gravity.

Table 1. Characteristics of considered fluids.

Fluid	$\dot{Q}_s$	N_T	p_{cr}	t_{cr}	M	p_o	p_k	p_o/p_k	t_{max}
	kW	kW	bar	ºC	kg/kmol	bar	bar	-	ºC
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 2. Characteristics of cycle efficiencies, Jakob numbers and $\Delta H(T_2)/h_{lv_1}$.

Fluid	η_R	η_c	$\eta_b=\eta_R/\eta_c$	$Ja(T_1)$	$\Delta H(T_2)/h_{lv_1}$
	-	-	-	-	-
SES36	0.205	0.271	0.755	9.467	3.181
R141b	0.178	0.317	0.563	42.055	4.042
C_2H_5OH	0.201	0.271	0,742	4.51	1.669
R134a	0.070	0.122	0.574	21.723	2.002

$$\Delta p = \Delta p_{TP} + \Delta p_G + \Delta p_A \tag{7}$$

In the present work the author has assumed the pressure drop due to friction in two-phase flow in the following form:

$$\Delta p_{TP} = \Delta p_{LO} \cdot \overline{R} \tag{8}$$

$$\Delta p_{LO} = \frac{l}{d} C_{fLO} \frac{1}{2} \frac{G^2}{\rho_L} \tag{9}$$

$$\overline{R} = \frac{1}{l} \int_0^l R dl \tag{10}$$

In the above relations the friction factor is calculated from the Blasius equation $C_{fLO} = 0.316\,\mathrm{Re}^{-\frac{1}{4}}$ relevant to turbulent flow, where $\mathrm{Re} = \frac{Gd}{\mu_L}$.

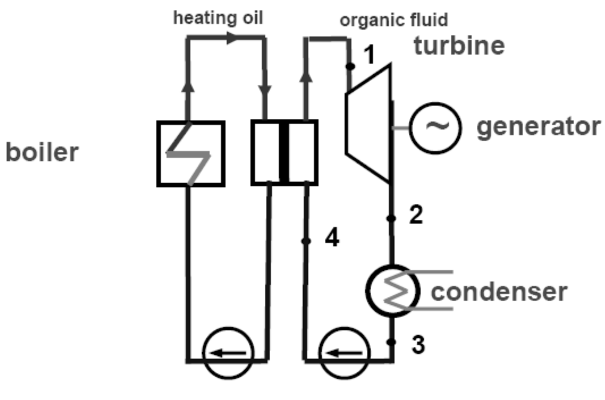

Fig. 3. Schematic of thermodynamic Rankine cycle of micro CHP.

The local two-phase flow multiplier, R, can be evaluated in accordance with the assumed model of two-phase flow. From among correlations existing in literature, the models of Lockhart-Martinelli [7], Chisholm [8], and Friedel [9], and a homogeneous model, have been selected for further calculations.

Acceleration and gravitation pressure drop can be derived from a momentum balance equation. The results are presented, respectively, in the form:

$$\Delta p_A = \Delta \left[\frac{x^2}{\rho_v \varphi} + \frac{(1-x)^2}{\rho_l (1-\varphi)} \right] \tag{11}$$

$$\Delta p_G = [\rho_l (1-\varphi) + \rho_v \varphi] g \tag{12}$$

Studies on the sensitivity of selection of pressure drop correlation [5] examine correlations describing the homogeneous two-phase flow model, correlations due to Chisholm, Friedel, and Lockhart-Martinelli with respect to their ability to predict the two-phase flow pressure drop. The results obtained were in close agreement. For that reason it was decided to use the homogeneous flow model for calculations of pressure drop inside evaporator and condenser. The local two-phase flow multiplier, R, can be evaluated in accordance with that model.

The pressure drop relationship presented above consists of two important parameters describing two-phase flow in channels: flow resistance R and void fraction φ. Literature presents various models for determination of these parameters. Heat exchange in flow boiling in the evaporator and condensation in condensor has been determined from the author's own two-phase flow model, Mikielewicz et al. [10], in the form:

Table 3. Characteristics of heat exchanger dimensions in the case of calculations incorporating pressure losses.

Fluid	Δp_K	Δp_{CH}	D_K	D_S	L_{rK}	L_{rCH}	$\dot{m}$	η_{th}	N_{nett}
	kPa	kPa	m	m	m	m	kg/s	%	kW
R141b	0.33	8.51	0.1	0.1	1.44	1.23	0.09	15	3.76
R123	0.35	8.91	0.1	0.1	1.13	1.25	0.11	14	3.43
R134a	0.44	2.82	0.1	0.1	0.48	1.31	0.14	5	1.17
Ethanol	0.34	1.23	0.1	0.1	0.99	0.25	0.024	10	2.42

$$\frac{\alpha_{TPB}}{\alpha_{LO}} = \sqrt{R_{M-S}^{0.76} + \frac{1}{1+2.53\times10^{-3}\,\mathrm{Re}^{1.17}\,Bo^{0.6}\left(R_{M-S}-1\right)^{-0.65}}\left(\frac{\alpha_{PB}}{\alpha_{LO}}\right)^2} \tag{13}$$

In relation (13) Re denotes the Reynolds number, Bo – Boiling number, R_{M-S} – two-phase flow multiplier utilizing the Muller-Steinhagen and Heck [9] model, α_{LO} – heat transfer coefficient for liquid-only flow, α_{PB} – heat transfer coefficient for pool boiling determined from the relation due to Cooper [8]. In paper [8] the two-phase flow multiplier R_{M-S} has been modified in order to obtain relevant asymptotic consistency, i.e. that the model indicates values of heat transfer coefficient for the liquid only flow, if the quality assumes a value of zero, and approximately that for vapour if x=1. A modified form of relation R_{M-S} is now expressed as:

$$R_{M-S} = \left[1+2\left(\frac{1}{f_1}-1\right)x\right]\cdot(1-x)^{1/3} + x^3\,\frac{1}{f_{1z}} \tag{14}$$

...where function $f_{1z} = \dfrac{\mu_G}{\mu_L}\cdot\dfrac{C_L}{C_G}\cdot\left(\dfrac{\lambda_L}{\lambda_G}\right)^{1.5}$

Function f_1 was developed from the ratio of pressure drop in the flow of liquid only to the pressure of vapour-only flow, whereas f_{1z} denotes the ratio of heat transfer coefficient for liquid only in a channel to the heat transfer coefficient in gas-only flow. In the case of small diameter channels (smaller than d=3 mm) it is recommended to introduce to relation (14) of a constraint number $Con=[\sigma/g/(\rho_L-\rho_G)/d]^{0.5}$.

The modified form of Muller-Steinhagen and Heck correlation assumes the form:

$$R_{M-S} = \left[1+2\left(\frac{1}{f_1}-1\right)x\,Con^{-1}\right]\cdot(1-x)^{1/3} + x^3\,\frac{1}{f_{1z}} \tag{15}$$

Boiler and Condensator Dimensions

In calculations of dimensions of boiler and condenser incorporating pressure losses (R141b, R123, R134a and C_2H_5OH) it has been assumed that the pressure drop inside tubes with diameter of 3 mm results from a fraction of a required length of tubes. Calculated heat exchanger dimensions relate to the construction of the shell-and-tube heat exchanger with the same number of tubes for all fluids. Assumed has been a number of tubes equal to 100, which enables determination of the evaporator diameter D_K as well as the condenser diameter, D_S equal about 0.1 m. The heat transfer coefficients for working fluids have been determined using the author's own correlation [10]. In Table 3, Δp_K denoted the pressure drop in the evaporator, whereas Δp_{CH} in a condenser. L_{rK} and L_{rCH} correspond to the lengths of heat exchangers.

Slightly different parameters have been assumed in calculations presented in Table 3, i.e. maximum temperature in the cycle is 15 K lower than the critical one, expansion in turbine starts at the saturations line x=1, and condensation temperature is 55°C. Parameters at turbine inlet are: P=3,351 kPa, T=189.2°C (for R141b), P=2,858 kPa, T=168.7°C (for R123), P=2,961 kPa, T=86.1°C (for R134a) and P=967.8 kPa, T=150°C for ethanol, [5].

Experimental Investigations

Apart from theoretical research, experimental activities aimed at developing the prototype for the ORC cycle. The working fluid initially has been selected as R123, which has relatively good heat transfer characteristics, is inexpensive, and fills the majority of requirements to the perspective working fluids. Once the stand is fully commissioned, SES36 will become the tested working fluid as recommended by the manufacturer (Solvay) for use in ORC [6]. Experimental evaporator and condenser are the plate heat exchangers manufactured by Secespol with heat transfer surfaces of 1.8 m^2 (LB47-40 PCE) and 0.9 m^2 (LB47-20 PCC), respectively, corresponding to 15 kW and 11 kW capacities. Thus far only preliminary experiments have been accomplished with the inverted scroll compressor LG ELECTRONICS model HQ028P for operation with the refrigerant R407C and capacity 6.974 kW determined by the producer to operational parameters T_{evap}/T_{cond}=7.2/54.4°C. The dimensions of the scroll were: width/depth/height = 235/235/374 mm.

Obtained results are quite encouraging. The thermal efficiencies of the cycle are 5-12% at Carnot efficiencies of 26-30% and the expander internal efficiency ranging from

30 to 50%. In subsequent experimental investigations the scroll expander is planned to be replaced by a micro turbine of in-house design.

Conclusions

It seems that further development of the micro CHP is attractive and feasible in Poland. Investigations on the development of such a micro CHP should focus on finding a relevant working fluid, which should fulfill all the requirements or at least a majority of them. Another challenge is the micro turbine prototype. Many laboratories worldwide are involved in the development of such a device, but so far without a breakthrough.

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 cycle power and heat exchanger dimensions. Assuming that only subcritical cycles are analyzed, 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 the possibilities of applying supercritical cycles as well as wet subcritical ones need to be examined.

It stems 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 cycle efficiency and turbine power. A possible working fluid seems to be ethanol, which 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 a non-toxic organic fluid. A drawback is its low pressure in the condenser, much lower than the atmospheric one, which can lead to leakage of air and moisture into the system. It seems to be true that an increase of temperature of the upper source leads to an increase of cycle power, but these increases are not significant and come at the expense of supplied heat.

The prototype of the micro heat and power plant developed in the Heat Technology Department of Gdańsk University of Technology seems to prove the case that micro CHPs are the future of micro generation.

Nomenclature

c – specific heat
C – friction coefficient
D – tube diameter
g – gravity acceleration
G – mass flow rate
h – enthalpy
ΔH – enthalpy difference defined by (3) and (4) relations, temperature dependent
l – work, lenth of heat exchanger tubes
Δp – pressure drop
R – two phase multipier
T – temperature
x – quality of vapor
q – heat
φ – void fraction
μ – dynamic viscosity
η – cycle efficiency
ρ – density
$Ja = \frac{C_p T}{h_{lv}}$ – Jakob number temperature dependent
$Re=Gd/\mu_L$ – Reynolds number.

Subscripts

1,2,3 – number of thermodynamic state
A – acceleration
c – Carnot efficiency
G – gravity
L – liquid
v – vapour
lv – latent heat
TP – two phase flow
fLO – flow of liquid only in, amand of total mass flow.

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