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Domestic Combined Micro Heat and Power Plant

Jarosław Mikielewicz
Institute of Fluid-Flow Machinery PAS, Gdańsk

1. Introduction

Nowadays the main direction of developments in energy sector is production of electricity in big power plants. The bigger capacity of the plant the lower is cost of the unit of electricity. This direction of development reaches many barriers. So, in energy conversion field a new direction is established called dispersed cogeneration of heat and power in small power plants. In paper the micro combined heat and power unit (CHP) is presented. Produced heat by CHP can be used for preparation of hot water for domestic use, swimming-pools, heating purposes or production of ice water. The source of prime energy in the micro CHP can be gas from combustion of natural resources or biomass, geothermal resource or the solar collectors. Also the waste heat from technological processes can be used for that purpose. Electricity is produced by the generator driven by the micro turbine operating on vapor of low-boiling point liquid. The power of such turbine ranges from several to tens of kilowatts. The advantage of the micro CHP is its compactness and small dimensions as well possibility for full automation of the operation of such plant. Small dimensions of the CHP are obtained through implementation of modern materials and up to date micro- or even nanotechnologies. Small dimensions of turbine and heat exchangers, simple materials and simple fabrications of parts of the plants, working in low temperatures range, lead to low costs of electricity production.

The source of heat for the micro power plant, in relation to the local capabilities, can be fossil fuel or renewable sources of energy. Such heat in the micro power plants is better used than in professional power plants producing only electricity. A micro cogenerative power plant utilizes energy in fuel up to

90%, see fig. 1. In authors opinion that is by far the best utilization of chemical energy contained in the fuel.

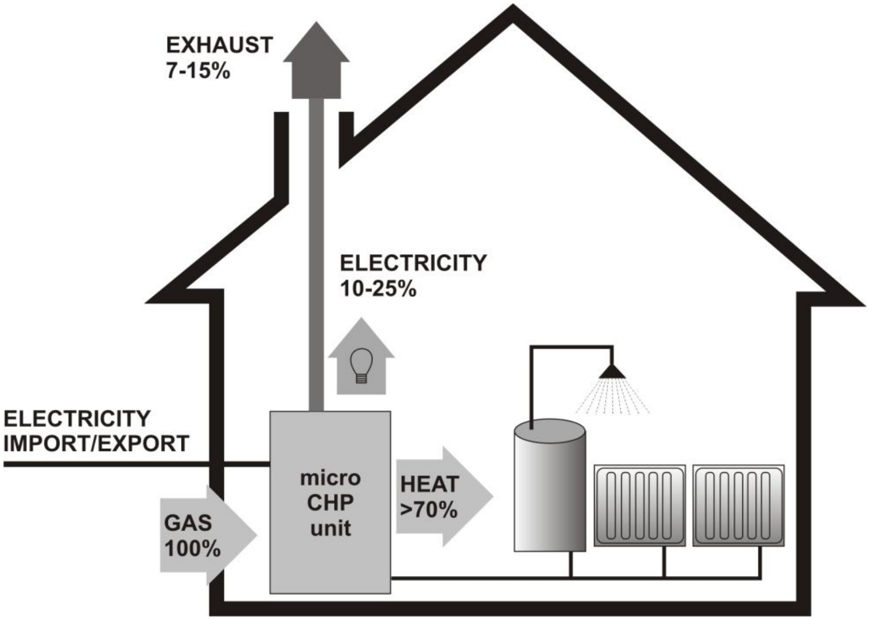

Fig. 1. A general schematic of a micro CHP
Rys. 1. Schemat działania domowej mikrosiłowni

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. A small cogenerative power plant can be fully automated and does not require operational staff. In such a way the energy users, i.e. owners of boilers, may become electricity producers. The market research accomplished in the United Kingdom showed that the demand for such cogenerative units may be quite large, comparable to the refrigerators market. Assuming that capacities of power plants range from a few to tens of kilowatts it can be easily calculated that these could replace construction of a series of conventional large professional power plants. There are different concepts of development of such micro power plants based for example on the Stirling engine, fuel cells, etc. The implementation perspectives of these concepts are different but most of them will be seen on the market in a not too distant future.

In the present paper the author, relying on own experience and knowledge, propose a concept of a new cogenerative vapour power plant for domestic usage. That would be a vapour power plant operating with the vapour of low boiling point fluid, i.e. such fluid which could be used in refrigeration applications. It 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. It is also quite feasible that the unit price of produced electricity would be not far off the prices found in professional power plants. In the vapour micro power plant the electricity is produced by the generator driven by the micro turbine operating with steam or low boiling point fluid vapour. Considered here are applications in small households, where the turbine capacity will be of the order of 2-3kW. The advantages of the micro power plant are its small dimensions, which are attained by using modern materials and modern micro and even nanotechnology.

As mentioned earlier, in future the cogenerative micro power plant will replace the 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 author opinion, domestic experience and knowledge should place us as leaders in the countries involved in that topic.

2. 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. [1] 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 thermo dynamical 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 on fig. 2 the fluids which were considered in the present study have been carefully selected from amongst 24 other considered [2]. 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. 1 .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.

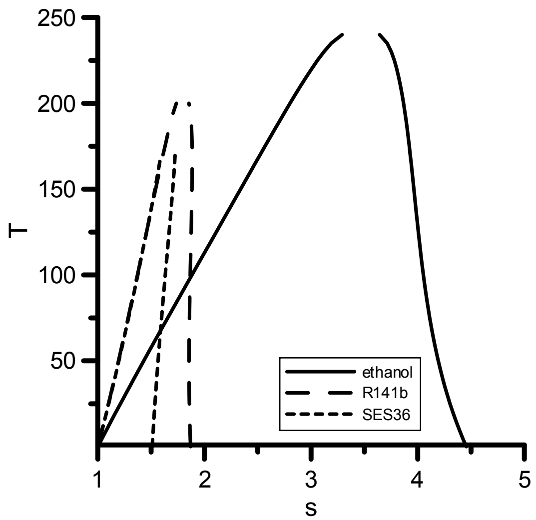

Fig. 2. T-s diagram for considered fluids calculated using Refprop 8.0
Rys. 2. Wykres T-s linii granicznych rozważanych czynników (Refrop 8.0)

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. The analysis commences with the expression for the cycle efficiency:

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

In relation (1) enthalpy change due to 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 the expansion can be written in terms of h_3 (end of condensation):

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

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 case of dry fluids x_2=1 whereas in case of wet fluids $\Delta h_{superheat}$=0. In general two latter terms in relation (3) can be combined to yield:

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

Substituting all these information into (1) we obtain the cycle efficiency

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

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

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

Analysis of relation (6) enables to conclude that the overall cycle efficiency is a function of a ratio $\Delta H(T_2)/h_{lv1}$ and the Jakob number. It stems directly from (6) that we should consider the ratios of $\Delta H(T_2)/h_{lv1}$ and c_p/h_{lv1} 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 nominator in equation (6) from which it results that the ratio $\Delta H(T_2)/h_{lv1}$ should assume smallest possible values in order to attain high values of overall efficiency. Values of $\Delta H(T_2)/h_{lv1}$ and $Ja(T_1)$ have been calculated for all fluids.

3. Thermodynamical analysis of a micro CHP

For the sake of calculation of the efficiency of thermo dynamical cycle of considered micro CHP a special code has been developed enabling consideration of various working fluids [3]. 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 is to be of the single stage radial type and does not influence the volume occupied by the whole arrangement [4].

According to the fig. 3 the following parameters have been assumed in the present analysis: heat demand of 20kW, 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=3mm. 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.

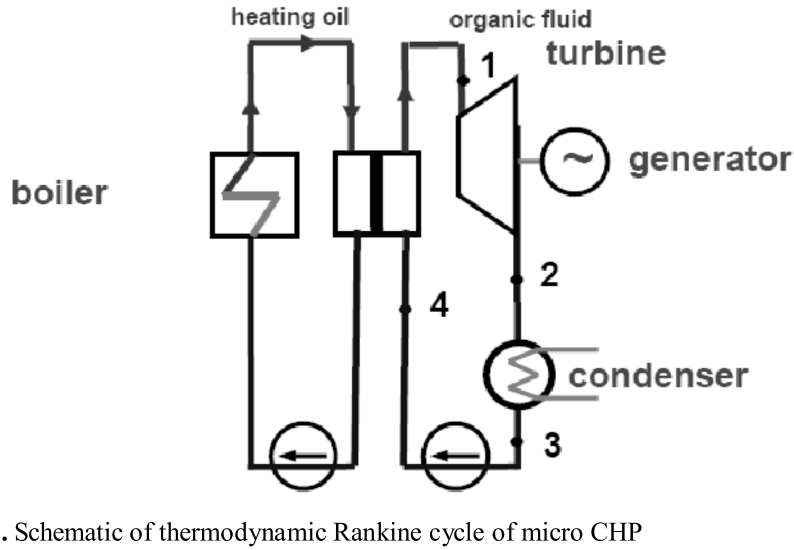

Fig. 3. Schematic of thermodynamic Rankine cycle of micro CHP
Rys. 3. Schemat termodynamicznego obiegu Rankine'a w mikrosiłowni

Temperature of oil leaving the heat exchanger was assumed 230°C. Physical properties of working fluids have been takes from the software Refprop8 [5]. 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 1 and 2.

Table 1. Characteristics of considered fluids
Tabela 1. Charakterystyki rozważanych czynników

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 2. Characteristics of cycle efficiencies, Jakob numbers and $\Delta H(T_2)/h_{lv1}$
Tabela 2. Charakterystyki efektywności obiegów, liczby Jakoba i $\Delta H(T_2)/h_{lv1}$

Fluid	η_R	η_c	$\eta_b=\eta_R/\eta_c$	$Ja(T_1)$	$\Delta H(T_2)/h_{lv1}$
	-	-	-	-	-
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

It results from table 1 and 2 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. 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, however their applicability is limited to a restricted working fluids or steam qualities. The pressure drop consists of three components namely pressure drop due to: friction, acceleration and gravity.

$$\Delta p = \Delta p_{TP} + \Delta p_{G} + \Delta p_{G} . \qquad (7)$$

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

$$\Delta p_{TP} = \Delta p_{LO} \cdot \overline{R} , \qquad (8)$$

$$\Delta p_{LO} = \frac{l}{d} C_{fLO} \frac{1}{2} \frac{G^2}{\rho_L} , \qquad (9)$$

$$\overline{R} = \frac{1}{l} \int_0^l R dl . \qquad (10)$$

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 $\mathrm{Re} = \frac{Gd}{\mu_L}$.

The local two-phase flow multiplier, R, can be evaluated in accordance to the assumed model. From amongst existing in literature correlations the models due to Martinelli-Nelson, Chisholm, Friedel and homogeneous model have been selected for further calculations.

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

$$\Delta p_{A} = \Delta [\frac{x^2}{\rho_v \varphi} + \frac{(1-x)^2}{\rho_l (1-\varphi)}] , \qquad (11)$$

$$\Delta p_{G} = [\rho_l (1-\varphi) + \rho_v \varphi] g . \qquad (12)$$

Studies on the sensitivity of selection of pressure drop correlation was done earlier by the authors [4] where correlations describing the homogeneous two-phase flow model, correlations due to Chisholm, Friedel and Martinelli-Nelson were examined with respect to their ability to predict the two-phase flow pressure drop and the results obtained were in a close agreement to each other. 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 to that model.

Presented above relation concerning pressure drops consist of two important parameters describing two phase flow in channels, i.e. flow resistance and void fraction. In literature presented are various models for determination of these parameters. Some of them are presented below.

Channel averaged void fraction

Precise determination of void fraction in two-phase zone is crucial for evaluation of pressure drop in boiler and condenser. There exists the relation between quality and void fraction:

$$\frac{x}{1-x}=\frac{\varphi}{1-\varphi}\frac{w_v}{w_l}\frac{\rho_v}{\rho_l}, \tag{13}$$

where s denotes the slip between the phases:

$$s=\frac{w_v}{w_l} \qquad .$$

Solving (3.7) with respect to void fraction one can obtain:

$$\varphi=\frac{x\dfrac{\rho_l}{\rho_v}}{x\dfrac{\rho_l}{\rho_v}+s-sx}. \tag{14}$$

Slip s can be determined from the Zivi formula:

$$s=\sqrt[3]{\frac{\rho_l}{\rho_v}}, \qquad \eta=\frac{\rho_l}{\rho_v}. \tag{15}$$

In literature exist a number of relations describing void fraction. In the case of homogenous model s=1. For separated phase model e.g. Chisholm relation for slip yields:

$$s1(x)=[1-x\cdot(1-\eta)]^{\frac{1}{2}}. \tag{16}$$

Coefficient for Two-phase flow fraction resistant

Most often used relations for fraction coefficient for two-phase flow are listed below:

a. Homogeneous model

$$R(x)=1+(\eta-1)\cdot x. \tag{17}$$

b. Lockhart-Martinelli model

$$R3(x)=\left(1+\frac{C}{X(x)}+\frac{1}{X(x)^2}\right)\cdot(1-x)^{1.75}, \tag{18}$$

where: $X(x)=\left[\left[\frac{(1-x}{x}\right]^{0.9}\cdot\eta^{-0.5}\mu^{0.1}\right], \qquad C=20$.

Heat exchange in flow boiling in the evaporator has been determined from the authors own two-phase flow model, Mikielewicz et al. [7], in the form:

$$\frac{\alpha_{TPB}}{\alpha_{LO}}=\sqrt{R_{M-S}^{0.76}+\frac{1}{1+2.53\times10^{-3}\,\mathrm{Re}^{1.17}\,Bo^{0.6}\left(R_{M-S}-1\right)^{-0.65}}\left(\frac{\alpha_{PB}}{\alpha_{LO}}\right)^2}. \tag{19}$$

In relation (19) Re denotes the Reynolds number, Bo- Boiling number, $R_{M\text{-}S}$ – two-phase flow multiplier utilizing the Muller-Steinhagen and Heck 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 [7]. In the paper [7] the two-phase flow multiplier $R_{M\text{-}S}$ 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 value of zero, and approximately that for vapour if x=1. A modified form of relation $R_{M\text{-}S}$ is now expressed as:

$$R_{M-S}=\left[1+2\left(\frac{1}{f_1}-1\right)x\right]\cdot(1-x)^{1/3}+x^3\frac{1}{f_{1z}}, \tag{20}$$

where function $f_{1z}=\frac{\mu_G}{\mu_L}\cdot\frac{C_L}{C_G}\cdot\left(\frac{\lambda_L}{\lambda_G}\right)^{1,5}$. The 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 the 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 case of small diameter channels, i.e. channels smaller than d=3mm it is recommended to introduce to relation (20) 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:

$$R_{M-S}=\left[1+2\left(\frac{1}{f_1}-1\right)xCon^{-1}\right]\cdot(1-x)^{1/3}+x^3\frac{1}{f_{1z}}. \tag{21}$$

5. Dimantions of boiler and condensator

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 3mm results from 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 100, which enables determination of the evaporator diameter D_K as well as the condenser diameter, D_S equal about 0.1m. The heat transfer coefficients for working fluids have been determined using authors own correlation [7]. In table 3 Δp_K denoted the pressure drop in evaporator whereas Δp_{CH} in a condenser. L_{rK} and L_{rCH} correspond to the lengths of heat exchangers.

Table 3. Characteristics of heat exchanger dimensions in case of calculations incorporating pressure losses.
Tabela 3. Charakterystyki wymiennika ciepła w przypadku obliczeń uwzględniających straty ciśnienia

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

In calculations presented in table 3 slightly different parameters have been assumed, i.e. maximum temperature in the cycle is lower by 15K from 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=3351kPa, T=189.2°C (for R141b), P=2858kPa, T=168.7°C (for R123), P=2961kPa, T=86.1°C (for R134a) and P=967.8kPa, T=150°C for ethanol.

6. Experimental investigations

Apart from theoretical research into the problem the experimental activities also started aimed at development of the prototype realizing the C-R cycle. The working fluid initially has been selected as R123. This is the fluid having relatively good heat transfer characteristics, is inexpensive and obeys a majority of requirements to the perspective working fluids. Once the stand is fully commissioned the SES36 will become the tested working fluid. Evaporator and condenser are the plate heat exchangers manufactured by Secespol with heat transfer surfaces of 1.8m^2 (LB47-40 PCE) and 0.9m^2 (LB47-20 PCC) respectively, corresponding to 15kW 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 compressor is planned to be replaced with a micro turbine of in-house design.

7. Conclusions

It seems that further development of the micro CHP is very attractive and can become the „Polish hit". The investigations on the development of such 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 world wide are involved in the development of such device, however without a breakthrough success thus far. In the present paper authors focused their attention one the selection of appropriate fluid.

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 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 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 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 statement seems to be true that increase of temperature of the upper source leads to the increase of the cycle power but these increases are not significant and at the expense of supplied heat.

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.

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Domowe mikrosiłownie kogeneracyjne

Streszczenie

Jednym z nowych obiecujących kierunków współczesnej energetyki uzupełniającym scentralizowany sektor energetyki jest sektor energetyki rozproszonej, w którym wytwarzana jest energia elektryczna w kogeneracji z ciepłem. Istnieje szereg technologii energetyki rozproszonej o małej mocy wytwarzania energii elektrycznej i ciepła. Najkrótszy horyzont czasowy związany jest z zastosowanie parowych obiegów Rankine’a na czynnik niskowrzący (Organic Rankine Cycle-ORC) w mikrosiłowni. Na tej bazie powstała w Instytucie Maszyn Przepływowych PAN koncepcja Domowej Mikrosiłowni Kogeneracyjnej. Mikrosiłownia ta o obiegu ORC ma ona służyć do produkcji energii elektrycznej i ciepła do użytku domowego. W przyszłości Mikrosiłownia Kogeneracyjna zastąpi konwencjonalne kotły do ogrzewania obiektów

takich jak: domki jednorodzinne, domy wielorodzinne, osiedla itp. Gabarytowo kocioł z Mikrosiłownią będzie niewiele różnić się od dotychczasowego kotła grzewczego ale będzie oprócz funkcji ogrzewania wytwarzać dodatkowo energię elektryczną. Mikrosiłownia parowa na czynnik niskowrzący pracująca w zakresie znacznie niższych temperatur niż silnik spalinowy i turbina gazowa wymaga mniej cennych materiałów, łatwiejsza też jest technologia jej wytworzenia. Przy jej pomocy staje się możliwe generowanie energii elektrycznej przy cenach zbliżonych do cen energii wytwarzanej w tradycyjnych siłowniach dużej mocy. Lepsze wykorzystanie energii paliwa w Mikrosiłowniach Kogeneracyjnych prowadzi do obniżenia szkodliwych emisji towarzyszących procesowi spalania paliwa. Mała siłownia kogeneracyjna może być w pełni zautomatyzowana i nie wymagać obsługi Podstawowymi elementami składowymi mikrosiłowni są: kocioł (parownik), turbina parowa, skraplacz (kondensator), generator elektryczny i pompa zasilająca. Nowa koncepcja mikrosiłowni wymaga rozwiązania szeregu nowych problemów. Jednym z nich jest wybór odpowiedniego czynnika roboczego Aby zastosować kompaktne wymienniki ciepła o intensywnej wymianie ciepła w mikrokanałach, analiza obiegu uwzględnia spadki ciśnienia w wymiennikach, które z kolei wpływają na różnice temperatur w wymiennikach a tym samym i na wymiary tych wymienników ciepła. Należało więc w obliczeniach koncepcyjnych uwzględnić spadek ciśnienia przy przepływie dwufazowym przez parownik i skraplacz. Wymianę ciepła podczas wrzenia w parowniku wyznaczono z własnego modelu przepływu dwufazowego. Oprócz prac teoretycznych prowadzone są w Instytucie prace eksperymentalne. Zbudowano stanowisko eksperymentalne symulujące pracę mikrosiłowni, na którym przeprowadzono wstępne pomiary parametrów obiegu, współczynników wymiany ciepła w wymiennikach metodą Wilsona oraz sprawności ekspandera spiralnego. Otrzymane wyniki są zachęcające. W dalszych badaniach eksperymentalnych ekspander (odwrócona chłodnicza sprężarka spiralna) będzie zastąpiony mikroturbiną własnej konstrukcji.