

Experiences from operation of different expansion devices for application in domestic micro CHP

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Abstract In the paper presented are experiences from operation of three different expansion devices for possible implementation in the domestic micro CHP. These were the modified scroll expander and two designs based on the variable working chamber volume pneumatic devices. Experiments showed the superiority of both "pneumatic devices" over the scroll expander, indicating the possible internal efficiencies in the range of 61–82%. Such efficiencies are very attractive, especially at the higher end of that range. The volume of these devices is much smaller than the scroll expander which makes it again more suitable for a domestic micro CHP. Small rotational velocities enable to conclude that connection to electricity grid will also be simpler in the case of "pneumatic devices". The "pneumatic devices" under scrutiny here could be an alternative to the typical vapour turbine in the ORC cycle, which is in the process of development at the IFFM.

Keywords: Micro CHP; Expansion devices; Central heating

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1 Introduction

Combined production of heat and power to cover the demand of the individual household or a small enterprise can be accomplished using the Organic Rankine Cycle (ORC) with the low boiling-point fluid as the working fluid.

In this light we can see significant perspectives for development of micro power plants, which should be capable of producing heat and electricity in so called dispersed scale. Bearing in mind a vast number of individual heat sources we can expect to be in line with the EU directive 2004/8/EU for decentralized production of heat and electricity. The heat from the power plant can be used for heating of utility hot water for domestic use and central heating, whereas electricity can be used on site or sold to the grid. 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 utilisation of chemical energy contained in the fuel.

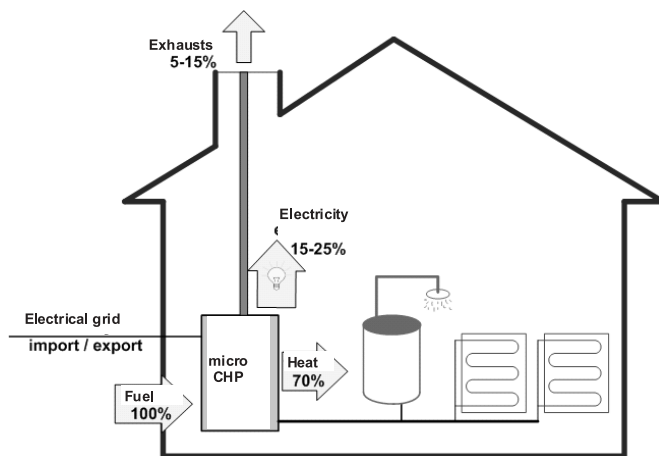


Figure 1. A general schematic of a micro CHP.

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 forty something percent. Better utilisation of

fuel energy in micro combined heat and power plant, (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.

Design and manufacture of a small size turbine (expansion machine) featuring the effective power of the order of few kilowatts, high internal efficiency and ensuring the full air-tightness in the presence of the low boiling-point fluid as a working fluid is a formidable engineering challenge. The smaller the turbine capacity the higher is the nominal speed of the turbine. Ensuring of a requested power implies the rotational velocities reaching 100 thousands rpm. For such range of velocities of significant importance are issues of manufacturing, dynamical balancing of rotating elements as well as coupling of turbine with the electricity generator.

The parallel path for searching of the expansion device to operate within the domestic micro heat and power plant encompasses activities related to adaptation and modification of existing devices available on the market to adjust them to operate as expansion machines. Authors accomplished already adjustments of three of such devices, namely the scroll compressor, pneumatic drill and pneumatic wrench. Some experiences from operation of these devices on the specially designed for that purpose experimental rig, see Fig. 2, will be presented in the paper.

2 ORC laboratory facility

The refrigerant R123 and ethanol have been selected as a working fluids, following earlier careful studies [1,2]. These fluids have a relatively good thermodynamic and heat transfer characteristics, are inexpensive and obey a majority of requirements facing the perspective working fluids [1,2].

Evaporator and condenser from the ORC cycle experimental facility

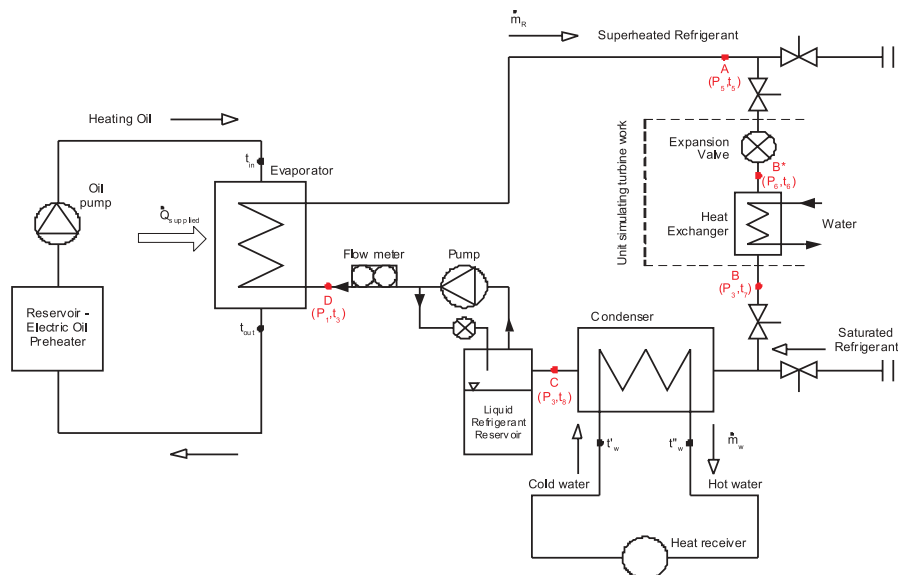


Figure 2. Schematic of laboratory ORC installation.

are the plate heat exchangers with heat transfer surfaces of 1.8 m^2 (LB47-40 PCE) and 0.9 m^2 (LB47-20 PCC) respectively, which corresponds to capacities of 15 kW and 11 kW. The heat exchangers were manufactured by a Polish heat exchanger company Secespol. Incorporation of the expansion machine into the facility is straightforward and easy, as it is mounted in a special bypass.

The flow of working fluid is accomplished in a closed loop of ORC and was forced by the positive-displacement pumps system. The electrically driven system of pumps (HP 0815004) was connected in series. The maximum flow rate available was 0.1 kg/s and attainable pumping pressure up to 24 bar. The flow rate was controlled by the voltage adjustments or by the manually operated throttle valve. The throttle valve was mounted in a bypass circuit.

The Coriolis mass flowmeter (SITRANS FCMASFFLO 2100 DI 6) together with signal converter is used to measure the flow rate. The accuracy of measurements readings is 0.1% of the whole scale, corresponding to 0.2 kg/h . The working fluid passing through the evaporator received the heat from the heating oil (MOBILTHERM 603) and evaporated. It was the intention of the experiment to reach the inlet parameters to the expanding

machine at the pressure level of 15 bar and temperature of 160 °C, preferably as close as possible to the vapor saturation line. Following expansion in the expander the working fluid is directed to the condenser. The condenser is cooled by the tap water having known parameters at inlet. In a practical implementation it will serve for preparation of hot water for central heating installation. Following the change of phase in the condenser the refrigerant is directed to the storage tank and then to repeat the cycle.

Applied heating oil features a large thermal durability, high resistance to oxidation, good properties in the indirect heating system at working temperature to about 315 °C and is non-toxic. Required oil temperature is obtained by application of resistance heaters placed in the oil tank of about 50 liters (total electric power of heaters switched sequentially is 20 kW). Oil flow was forced by an electrically driven gear pump. The weighting method was applied to measure its flow rate. An electrical balance (ITE 9010A) with the accuracy of $\pm 0.5\%$ was used to determine the operating parameters. The heating oil transferred heat in the evaporator and went back to the reservoir. Temperature in the characteristic nodes of the circuit was measured by temperature transducers PT100 and associated with them temperature controllers SRT-73-1321. The pressure was measured using the calibrated Bourdon pressure gauge and programmable pressure meters WW-11N. The pressure drop in the condenser (on the side of working fluid) is measured by a differential pressure transducer (Aplisens PR-28).

3 Testing of different expansion devices

Following our experience, J. Mikielwicz and D. Mikielwicz, [3], the simplest configuration of Rankine cycle was analysed. The simplicity of the cycle is believed to result in possibly small heat transfer surfaces of heat exchangers, which are a decisive factor in final dimensions of the micro CHP for domestic use. The similar applies to the size of the expansion device.

In calculations performed in earlier studies [3] it has been assumed that vapor parameters at turbine inlet are known, namely (p_1, T_1) , as well as temperatures of cooling liquid at inlet and outlet to the evaporator (T_{1w}, T_{2w}) . Similar conditions were attempted in the executed experiments. Thermal oil is heated in the boiler to the temperature of 260 °C which is subsequently giving its heat away in the heat exchanger to convert the organic fluid from liquid state to the vapour state (parameters (p_1, T_1)). These parameters correspond to temperature of 170 °C and related pressure. That tempera-

ture was not difficult to be obtained in the case of R123, whereas in case of ethanol the superheating of the working fluid was not possible due to insufficient heating capacity. In such case expansion starts at vapour saturation line and proceeds isentropically to the pressure corresponding to 50 °C. In theory the end of expansion should correspond to wet vapour region in case of ethanol and superheated state for R123.

3.1 The scroll expander

First experiments have been accomplished with the inverted scroll compressor LG ELECTRONICS model HQ028P featuring the nominal capacity of 6.974 kW. The optimum parameters determined by the producer for operation with R407C were $T_{evap}/T_{cond} = 7.2/54.4$ °C. The dimensions of the scroll were: width/depth/height equal to 235/235/374 mm.

The expander was connected through the belt transmission with a standard car alternator provided by Renault (digital symbol of alternator 8200 120286, voltage 12 V, electrical current 110 A). The rotational speed of expander's shaft was measured by tachometer (SENTRY ST 723) with the accuracy in the range of 0.01% of the whole scale, allowing measurements of rotational velocities ranging from 6 to 6000 rpm. The accumulator (battery) (12 V, 55 Ah) with variable resistor (resistance changes in the range 0.1-1 Ω) was connected to the alternator circuit. The electrical current in the system was measured in the shunt, allowing measurements up to 150 A/60 mV. The scroll expander connected with the alternator and the current loop are presented in Fig. 3.

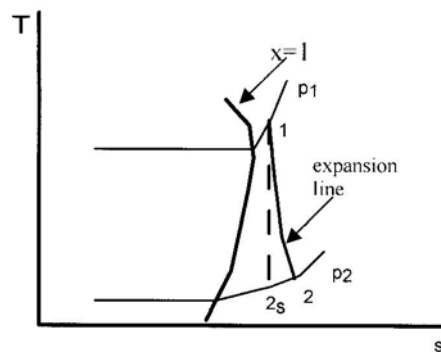


Figure 3. Determination of internal efficiency of expansion machine.

The results of experimental research performed using the scroll expander

LG HQ028P are presented in Tab. 1. Physical properties of R123 have been calculated using the Refprop8 software [4]. In the course of calculations determined have been values of evaporator capacity (\dot{Q}_{evap}), condenser capacity (\dot{Q}_{cond}), internal power of expander (N_i), cycle thermal efficiency (η_t), which was evaluated without account of work necessary for pumping operation, the maximum Carnot efficiency (η_C), internal efficiency of expansion machine (η_i) and finally the exergetical efficiency (η_b) of considered device. The internal efficiency of expansion machine was determined from the well known relation:

$$\eta_i = \frac{h_1 - h_2}{h_1 - h_{2s}}, \quad (1)$$

where:

h – enthalpy, [J/kg]

h_s – enthalpy after adiabatic expansion, [J/kg].

In Eq. (1) state 1 denoted inlet to expander, state 2 – the outlet and 2s – the state after isentropic expansion, Fig. 3. The Carnot efficiency, $\eta_C = 1 - T_2/T_1$, and exergetical efficiency, $\eta_b = \eta_t/\eta_C$.

In all cases the state 1 was found in superheated vapour region, however close to saturation line.

Table 1. The results of measurements and calculations for the scroll expander.

Run	\dot{m}_{R123}	\dot{Q}_{evap}	\dot{Q}_{cond}	N_i	$\eta_t = N_i/(\eta_i\dot{Q}_{evap})$	η_C	η_i	η_b
	kg/s	kW	kW	kW	%	%	%	%
1	0.034	8.14	7.69	0.473	5.8	29.2	29.0	19.9
2	0.040	9.12	8.70	0.456	5.0	26.2	25.2	19.1
3	0.041	9.01	8.12	1.050	11.7	28.1	52.1	41.6
4	0.043	10.13	9.22	0.959	9.5	30.6	42.9	31.0

3.2 Expander with variable volume expansion chamber

Second construction under scrutiny was a variable volume expansion chamber device featuring a “turbine mechanism” together with the planetary gear. Adapted device was a BOSCH pneumatic drill (series 180 W), appropriately modified to the specific requirements of the experiment. The basic parameters of the drill were: rotational velocity 1800/3000 rpm, power 180 W, feeding pressure 6.3 bar, weight 0.96 kg. The body of the drill was

made of steel. The modification relates to separation of mechanisms of change of rotational velocity of the spindle and a set of valves and followers. Additionally, the silencer was removed and the appropriate sockets were made. In experiments the machine was coupled with the hydraulic brake which enabled determination of the effective power. Results of experimental research performed using the rotating plate expander are presented in Table 2.

Table 2. The calculation results for the pneumatic device.

Run	\dot{m}_{R123}	\dot{Q}_{evap}	N_i	N_{eff}	η_t	η_C	η_i	η_b
	kg/s	kW	kW	kW	%	%	%	%
1	0.025	6.41	0.820	0.127	12.8	32.5	81.4	39.4
2	0.031	8.23	0.713	0.153	8.9	32.9	60.8	27.1
3	0.029	7.39	0.703	0.138	9.5	31.2	57.5	30.5
4	0.028	7.13	0.771	0.106	10.8	31.9	68.8	33.9
5	0.021	5.44	0.626	0.102	11.5	32.1	75.3	35.8

Schematic diagram of operation of that type of expansion device is presented in Fig. 4. Construction of the brake enabled direct measurement of force generated in purpose built manifold as a result of body deflection. The body deflection was a result of energy receiving from rotating impeller through oil filling the break. The anti-torque was generated by the friction forces. The force acting on the arm of 130 mm was estimated by a dynamometer. The calibration of dynamometer was accomplished with standard weights.

The results show technical possibilities of the ORC experimental setup with the “pneumatic device” as an expander. They demonstrate the ability of the arrangement to prepare vapour of the working fluid with suitable properties to obtain the effective work from expansion. Noticed maximum temperature exceeds 170 °C, while the working pressure reached 14 bar. Measured rotational speed of the shaft with hydraulic break was in the range from 790 to 940 rpm. The force generated by the breaking system as a result of the frictional drag ranged from 0.85 to 1.3 kG.

3.3 The pneumatic engine

The third examined device was the specially adapted pneumatic wrench (model VS02YU1260T). Its basic technical parameters are: rotational speed

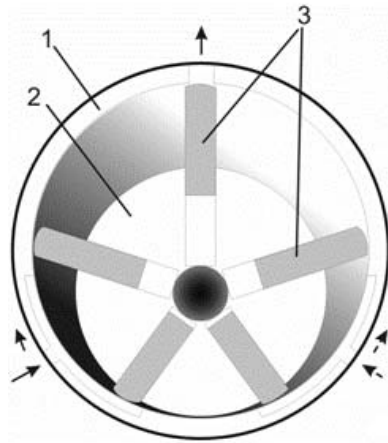


Figure 4. Schematic of the expander with variable volume working chamber; 1 – cylinder, 2 – rotor, 3 – blades.

6500 rpm, torque 813 Nm at the supply pressure of 7 bar (air), air flow rate demand of $0.36 \text{ m}^3/\text{min}$, weight 2.8 kg. Such nominal parameters of operation made it a very attractive option for implementation in the ORC. The body of the device was made of steel. In Fig. 4. presented is a schematic diagram of the device, which operates to the same principles as the pneumatic drill. In that case much higher power of the device was expected than in the case of the pneumatic drill.

The principle of operation of the pneumatic drill and pneumatic wrench is following. The bladed engine consists of the cylinder, where the rotor with blades is eccentrically fixed in the bearings. The blades are sliding in the longitudinal slots, leading in that way to variable volume of the expansion chamber. Compressed working fluid located between two plates experiences the pressure difference on both sides of the blades which induces its motion. The working fluid contained between the blades undergoes expansion due to the fact that the volume between the blades increases. The pressure in the chamber decreases but the force difference is constant as due to radial movement of the rotor the upstream blade surface is greater than the succeeding one. The expansion continues until the blade reaches the outlet window connected in the installation of receiving the working fluid. The blades are kept in a tight contact with the inner surface of the cylinder by means of the centrifugal force, sealing in that way the successive working chambers.

Table 3. The calculation results for the pneumatic wrench.

No.	$\dot{m}_{metanol}$	\dot{Q}_{par}	N_t	N_{el}	$\eta_t = N_t/\dot{Q}_{evap}$	η_C	η_i	η_b
	kg/s	kW	kW	kW	%	%	%	%
1	0.013	14.79	0.759	0.121	5.13	30.01	67.10	17.09
2	0.014	15.96	0.769	0.130	4.82	30.32	73.68	15.90
3	0.014	16.01	0.907	0,085	5.66	32.61	82.35	17.37
4	0.014	15.84	0.881	0.057	5.56	33.50	78.52	16.59

Obtained results showed that the examined expander was attaining the rotational speed in the range 1400-3680 rpm. The maximum electric power obtained was 130 W. That was again due to the poor design of the applied alternator and significant mechanical losses. The relatively small mechanical power was due to the fact that the outlet pipeline was somewhat throttling the flow of vapour, preventing in such way lower temperatures of exiting vapour, and hence not optimal enthalpy drop. In further studies the modification should proceed in the direction of modifying the working fluid outlet pipeline. That should definitely contribute to better thermal efficiencies, as the internal ones seem again to be very attractive.

4 Conclusions

Experiments showed the superiority of both “pneumatic devices” over the scroll expander, indicating the possible internal efficiencies in the range of 61-82%. Such efficiencies are very attractive, especially at the higher end of that range. The volume of these devices is much smaller than the scroll expander which makes it again more suitable for a domestic micro CHP. Small rotational velocities enable to conclude that connection to electricity grid will also be simpler in the case of “pneumatic devices”. The “pneumatic devices” under scrutiny here could be an alternative to the typical vapour turbine in the ORC cycle, which is in the process of development at the IFFM. Application of the pneumatic wrench remains still a positive option, however further modifications to that design must be carried out. For that purpose another prototype must be developed with a bigger vapour outlet socket to prevent from throttling.

Technical inspection of pneumatic device based on the drill done after the experimental investigations showed that seals were not resistant to

corrosion caused by the refrigerant R123. The seal (leak stopper) should be made of other materials, for example Teflon. The corrosion caused the drag increase of rotating centrifugal governor and automatic decrease of effective power estimated at the outlet shaft (adequately to the expression $N_i = N_m + N_{ef}$, where N_i – internal power of engine, N_m – mechanical power, N_{ef} – effective power). It should also be mentioned that the expansion devices were used without devoted lubrication, using only the working fluid for that purpose. Ethanol seems to be a very attractive working fluid for ORC applications and studies using that fluid will be further continued.

It should be mentioned that the nominal range of rotational speed of investigated pneumatic device (about 3000 rpm) is advantageous from the electricity generator point of view. These parameters can be obtained by the presented solution after increasing the steam pressure supplying the turbine.

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References

- [1] MIKIELEWICZ D., MIKIELEWICZ J.: *Cogenerative micro power plants – a new direction for development of power engineering?* Archives of Thermodynamics **29**(2008), 4, 109-132.
- [2] MIKIELEWICZ J., MIKIELEWICZ D.: *Comparative study of selected fluids for use in supercritical Organic Rankine Cycles.* Archives of Thermodynamics **30**(2009), 2, 3-14.
- [3] MIKIELEWICZ D., MIKIELEWICZ J.: *A new criterion for selection of working fluid for subcritical and supercritical micro CHP.* Applied Thermal Engineering, **30**(2010), 2357–2362.
- [4] *Refprop* v. 8.0. National Institute of Standards, 2007.