Short Communication

Design Analysis of Combined Gas-Vapour Micro Power Plant with 30 kW Air Turbine

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Abstract

This article presents basic results of the design analysis of a combined gas-vapour micro power plant with a waste-heat boiler without exhaust reheat, working in the vapour section in the organic rankine cycle (ORC) system. The analysis concerned working media most frequently used in those types of power plants and that plan to be used in the future. The object of the analysis was a micro power plant with a 30 kW gas (air) turbine for which the low-boiling medium was selected in such a way as to obtain maximum possible power output of the vapour turbine and, consequently, the highest efficiency of the combined cycle. The analysis also included vapour micro power plants with heat regeneration for so-called dry media and cycles without heat regeneration for so-called wet media. The amount of thermal energy available for utilization in the cogeneration micro power plant at the assumed condenser temperatures equalling 95ºC, 55ºC, and 30ºC, respectively, was assessed. The most favourable medium in terms of the obtained power output and efficiency of the combined cycle was selected and the effect of the low-boiling medium on design parameters of the vapour micro turbine was assessed.

Keywords: combined gas-vapour cycles, organic rankine cycle (ORC), vapour turbines, turbine design, cogeneration

Introduction

In classical highly efficient combined large power systems working in the electric power generation system the use of water is profitable both in economic and thermodynamic terms. Gas turbines used in those systems generate exhaust gas of high temperature and are frequently designed to work in combined cycles. The time of full load operation of those power plants can reach as much as 8,000 hours per year, which economically justifies the use of technologically complicated and expensive three-pressure waste-heat boilers. For micro turbine systems with low temperature exhaust gas, a solution that will provide opportunities for effective utilization of the waste heat can be a combined cycle in which the steam-water section is replaced by a power plant with a low-boiling medium (ORC).

Gas Turbine Set

A basic criterion taken into account when evaluating the results of the design analysis was the efficiency level obtained for the assumed values of: the power output of the gas turbine set, the caloric value of the fuel, and the temperature of the working medium at gas turbine inlet. An important factor which was also taken into account was the scale of cycle complication (number of heat exchangers), which, generally, affects the total cost of the power plant investment project. Of certain importance was also the compressor pressure ratio, as it is easier to design a compressor and remaining turbine set components for lower

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compression. Moreover, the compressor designed for the latter case is smaller and its manufacturing is cheaper. A solution frequently used in technical applications of gas turbines is a gas turbine set with a regenerator, the structure of which is simple at relatively high efficiency. In the case of small power gas turbines with low temperatures behind the combustion chamber (especially those used for burning different kinds of fuel), an attractive solution can be a system with an external combustion chamber (Fig. 1a), although its efficiency is lower by a few percentage points [1-4] as a result of lower temperatures at the turbine inlet (max. 850ºC-900ºC, the limit forced by constructional materials of heat exchangers).

This system provides opportunities for much easier burning of different types of fuels and reveals lower levels of optimal compression. Another favorable feature of this system is the possibility to burn fuel at atmospheric pressure, without preliminary compression of the gas fuel, which is necessary in classical systems with the combustion chamber in front of the turbine. The analysis took into account changes of the average specific heat as a function of variation of both air and exhaust gas temperatures. The cycle calculations were performed after adopting the following assumptions: the power output at generator terminals is 30 kW, the air temperature at compressor inlet is 20ºC, the pressure at compressor inlet is 0.1 MPa, compressor efficiency is 82%, internal efficiency of the air turbine is 82%, efficiency of the external combustion chamber is 97%, efficiency of the generator is 88%, mechanical efficiency is 96%, temperature of the exhaust gas at turbine inlet is 850ºC, and the caloric value of the fuel is 16 MJ/kg. The pressure losses were assumed equal to: 0.5% at compressor inlet, 0.5% at turbine set outlet, 2% in the external combustion chamber, and 2% in the high-temperature heat exchanger.

The maximal efficiency of the electric energy generation obtained in the performed calculations was equal to 22.3% for the optimal compression equal to 2.7. The exhaust gas which left the high temperature heat exchanger had a mass flow rate equal to 0.3581kg/s and temperature equal to 215ºC. This exhaust gas can be used in the waste-heat boiler for producing the vapour of the working medium in the ORC system.

Low-Boiling Media Used in ORC Systems

Working media used in ORC micro power plants should meet certain requirements. First, they should not be toxic and should be environment friendly. They should also be relatively cheap and easily available, and they must not react with turbine flow system components. In practice, not

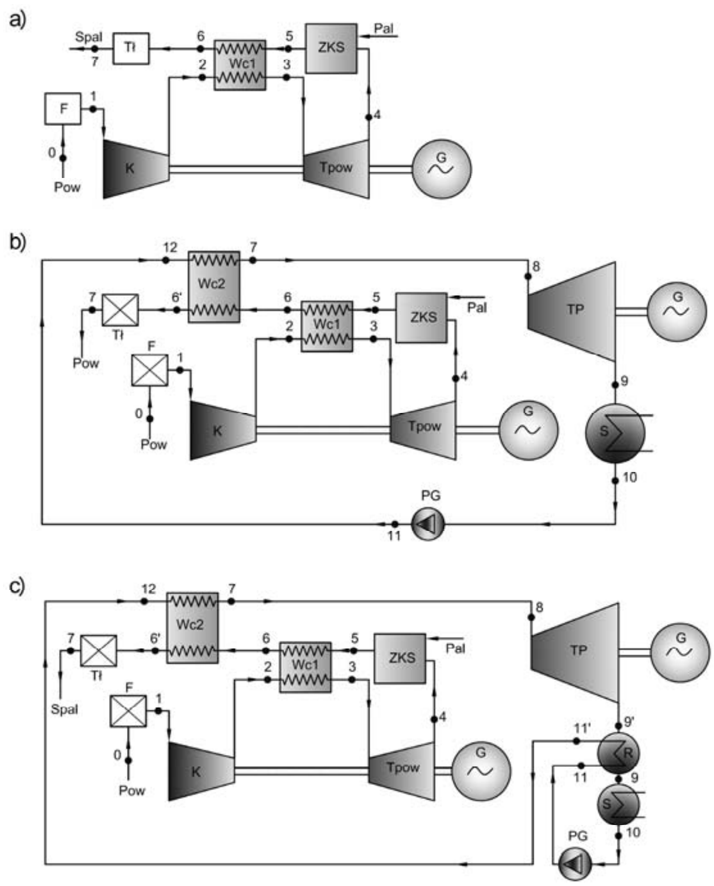

Fig. 1. Scheme of cycle with combustion chamber at the air turbine outlet (a) scheme of combined gas-vapour cycle with external combustion chamber at the air turbine outlet for wet (b) and dry (c) fluids.
Pow – air intake, F – filter, Tpow – air turbine, Wc1 – heat exchanger, ZKS – external combustion chamber, G – electric generator, Pal – fuel, Tł – silencer, Spal – exhaust outlet, Wc2 – boiler heat exchanger, TP – vapour turbine, R – regenerator, S – condenser, PG – main pump.

Table 1. Comparison of the results of calculations of the combined cycle variants.

Parameter/Fluid	Condenser 95ºC Temperature			Condenser 55ºC Temperature			Condenser 30ºC Temperature	
	N_{ORC}	η_{Kombi}	Q_{Kogen}	N_{ORC}	η_{Kombi}	Q_{Kogen}	N_{ORC}	η_{Kombi}
	[kW]	[%]	[kW]	[kW]	[%]	[kW]	[kW]	[%]
Water	2.74	24.33	25.56	5.34	26.27	34.4	7.53	27.90
Acetone	2.95	24.18	30.57	6.00	26.12	40.9	8.60	27.78
Ethanol	2.92	24.16	29.01	5.82	26.01	38.3	8.16	27.50
Methanol	2.81	24.39	27.54	5.57	26.44	35.1	7.69	28.02
Toluene	2.45	24.12	15.82	6.04	26.79	36.3	8.49	28.61
Pentane	3.13	24.62	31.30	6.65	27.24	40.2	9.59	29.43
Decane	3.08	24.59	25.89	6.27	26.96	36.0	8.93	28.94
Propylcyclohexane	3.02	24.54	23.42	6.19	26.90	35.9	8.81	28.85
R245fa	2.17	23.92	42.62	6.25	26.94	51.4	9.82	29.60
Octamethylcyclotetrasiloxane	3.16	24.65	27.50	6.49	27.12	38.1	8.96	28.96
Decamethylcyclopentasiloxane	3.13	24.62	26.29	6.41	27.06	37.2	8.89	28.91
Dodecamethylcyclohexasiloxane	3.13	24.62	26.36	6.37	27.04	33.1	8.76	28.81
Hexamethyldisiloxane	3.18	24.67	28.11	6.55	27.17	40.6	6.61	27.21
Octamethyltrisiloxane	3.16	24.65	28.10	6.49	27.12	39.5	8.94	28.95
Decamethyltetrasiloxane	3.13	24.63	25.98	6.44	27.08	35.5	8.93	28.94
Dodecamethylpentasiloxane	3.14	24.63	27.21	6.45	27.09	34.7	8.72	28.78
Ammonia	1.96	23.55	46.07	5.90	26.06	48.2	8.61	27.78
Dimethylether	1.62	23.33	44.22	5.56	25.84	49.2	8.23	27.54
Cyclopropane	1.84	23.47	45.40	5.45	25.77	51.1	8.88	27.96
Trifluoroiodomethane	2.45	23.86	46.06	5.37	25.72	52.5	8.92	27.99
R141B	2.93	24.47	30.87	6.22	26.92	41.0	8.92	28.93
R365mfc	3.53	24.92	34.00	6.82	27.37	40.6	9.85	29.62
Heptane	3.10	24.60	28.40	6.38	27.04	37.4	8.95	28.96

N_{ORC} – effective power of the vapour cycle, η_{Kombi} – combined cycle efficiency, Q_{Kogen} – thermal power

all of these conditions can be met if we want to have a medium revealing good thermodynamic characteristics. In the literature we can find various media that can be used in ORC systems [5-10]. The list of these media includes saturated, non-saturated, cyclic, heterocyclic, and aromatic hydrocarbons, along with various types of refrigerants, alcohols, siloxanes, etc. A well selected organic medium should allow the system to operate with maximum efficiency, and should provide opportunities for maximum possible utilization of the available source of heat. It is extremely important for the working medium to create a minimal possible threat to human beings and the environment. It should not interfere with the environment. The emission of refrigerants to the atmosphere is the source of very unfavorable changes in the environment [11]. The documented negative effect of commonly used halogen derivatives on degradation of the ecosystem was the motivation for introducing comparative indices, use of which affects a given medium on environment degradation described in a qualitative and quantitative way. Basic ecological indices include the ozone depletion potential (OZP) and the global warming potential (GWP).

Analysis of the Results of Combined Cycle Calculations

The object of the analysis was a combined cycle in which the exit air from the earlier analyzed air turbine cycle was delivered to the waste-heat boiler (Figs. 1b and 1c), producing the vapour of the low-boiling medium. The following assumptions were adopted in the calculations: the temperature and mass flow rate of the exhaust gas at the

Table 2. The main design parameters for multi-stage turbine for condenser temperatures of 95°C, 55°C, and 30°C (fluid R365mfc).

Temperature in Condenser 95ºC (fluid R365mfc)						
Turbine stage	[-]	1	2	3	4	
H_s	[kJ/kg]	7.348	7.348	7.348	7.348	
v	[-]	0.5	0.5	0.5	0.5	
ρ	[-]	0.35	0.35	0.35	0.35	
ε	[-]	0.089	0.165	0.270	0.415	
Ma_{c1}	[-]	0.93	0.80	0.73	0.69	
l_k	[m]	0.008	0.008	0.008	0.008	
D/l	[-]	6.0	6.0	6.0	6.0	
D_{sr}	[m]	0.05	0.05	0.05	0.05	
n	[rotation/min]	24,000	24,000	24,000	24,000	
Ma_{w2}	[-]	0.70	0.61	0.57	0.55	
N_u	[kW]	0.97	1.19	1.27	1.32	
N_{ut}	[kW]	4.73				
Temperature in Condenser 55ºC (fluid R365mfc)						
Turbine stage	[-]	1	2	3	4	
H_s	[kJ/kg]	10.761	10.761	10.761	10.761	
v	[-]	0.5	0.5	0.5	0.5	
ρ	[-]	0.28	0.28	0.28	0.28	
ε	[-]	0.100	0.189	0.335	0.580	
Ma_{c1}	[-]	0.94	0.88	0.81	0.80	
l_k	[m]	0.010	0.010	0.010	0.010	
D/l	[-]	7.1	7.1	7.1	7.1	
D_{sr}	[m]	0.07	0.07	0.07	0.07	
n	[rotation/min]	20,000	20,000	20,000	20,000	
Ma_{w2}	[-]	0.66	0.63	0.65	0.64	
N_u	[kW]	1.56	1.86	1.97	2.05	
N_{ut}	[kW]	7.44				
Temperature in Condenser 30ºC (fluid R365mfc)						
Turbine stage	[-]	1	2	3	4	5
H_s	[kJ/kg]	12.099	12.099	12.099	12.099	12.099
v	[-]	0.5	0.5	0.5	0.5	0.5
ρ	[-]	0.32	0.32	0.32	0.32	0.32
ε	[-]	0.112	0.225	0.425	0.780	1.000
Ma_{c1}	[-]	0.97	0.91	0.88	0.87	0.87
l_k	[m]	0.010	0.010	0.010	0.010	0.010
D/l	[-]	6.17	6.17	6.17	6.17	6.21
D_{sr}	[m]	0.062	0.062	0.062	0.062	0.062
n	[rotation/min]	24,000	24,000	24,000	24,000	24,000
Ma_{w2}	[-]	0.72	0.68	0.67	0.66	0.67
N_u	[kW]	1.81	2.14	2.27	2.33	2.29
N_{ut}	[kW]	10.84				

H_s – isentropic enthalpy drop in the turbine, ν – velocity coefficient, ρ – stage reaction, ε – partial admission arc, Ma_{c1} – Mach number at stage stator exit, l_k – stator blade (nozzle) length, D/l – mean diameter-to-blade length ratio, D_{sr} – mean stage diameter, n – rotational speed, Ma_{w2} – Mach number at stage rotor exit, N_u – power output of the turbine stage, N_{ut} – power output of the multi-stage turbine inlet to the heat exchanger Wc2 (waste-heat boiler) were equal to 215ºC and 0.3581kg/s, respectively (from gas cycle calculations). The temperature threshold in the heat exchanger Wc2 was assumed equal to 5ºC. The condenser temperature was assumed in variants as 95ºC, 55ºC, and 30ºC. The internal efficiency of the medium vapour turbine and pump efficiency were assumed equal to 0.8 and 0.4, respectively.

The pressure of the organic medium was determined in such a way that the maximal possible power output was obtained at vapour turbine shaft. It was also assumed that the waste-heat boiler produced dry saturated vapour (without superheating). The analysis took into account over 20 media most commonly used in ORC systems [5, 7, 9, 10].

The performed calculations have proved that obtaining 30 kW of electric power from the gas turbine and additionally 3.5 kW of electric power from the vapour turbine, along with 34kW of thermal power (for R365mfc, Table 1) at 95ºC, or, alternatively, 6.8kW of electric power and over 46kW of thermal power (also for R365mfc, Table 1) at a condenser temperature of 55ºC is possible. This result increases the efficiency of the analyzed gas cycle from 22.3% up to 24.9% and 27.4%, respectively (Table 1). When the condenser temperature is 30ºC, nearly 10kW of electric power can be additionally obtained, which increases the efficiency of the combined cycle to about 30% (Table 1). However, in this case we assumed that the heat in the condenser at temperature 30ºC is not used for heating and technological purposes. The performed cycle calculations have also proved that from the thermodynamic point of view, R365mfc works best as a medium in the vapour system for condenser temperatures equal to 55ºC and 95ºC, while comparable results can also be obtained using R245fa and pentane when this temperature drops down to 30ºC. That is why further part of the analysis, consisting in design calculations of micro turbines, will be performed for these media. Since in a single-stage turbine of this type large enthalpy drops result in the Mach numbers frequently exceeding 2, the analysis was performed for multi-stage turbines (four- and five-stage constructions) with subsonic flows. Low mass flow rates determine large rotational speeds exceeding 20,000 rpm, along with partial admission in practically all turbine stages. In construction of this type short blades (of about 10 mm in length) are mounted on small diameters, which results in slightly lower efficiency than that observed in large power steam turbines. At the same time the velocity coefficient, the reaction, and the ratio of disc diameter to blade length (D:L ratio) take values from the ranges typical for classical steam turbines. A collection of basic design parameters of multi-stage micro turbines for select parameters is given in Table 2 [12-14].

Conclusions

The performed analysis of the combined cycle with the air turbine having the external combustion chamber at the turbine outlet brought promising results. The gas turbine set of this type reveals numerous advantages. It works with pure air, as a result of which it is more durable and does not require as frequent inspections and overhauls as in case of other working media. It provides opportunities for burning, practically, an arbitrary fuel (even without additional overpressure).

Due to high air temperature (of about 600ºC) at air turbine outlet it is possible to combine the combustion chamber with the gasification process.

The next advantage of the presented system is its possible cooperation with the ORC system, which can improve the total efficiency to even as much as 30%. It was proved that designing an efficient multi-stage ORC micro turbine for the assumed thermodynamic parameters is possible.

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