Assessment of efficiency and prospects for the use of hybrid thermal low-capacity power plants in the Republic of Vietnam

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Abstract. The growing demand for energy and fossil fuels creates increased number of difficulties, while renewable energy sources are still rarely used worldwide, particularly in Vietnam. In this article hybrid thermal power plants based on gas turbine plants are discussed, the increased efficiency of which is achieved by air heating after the compressor in solar air heaters. The basic design equations and the results of evaluating the efficiency and fuel consumption are presented for two thermal power plants of 4.6 MW and 11.8 MW. The dependence of the results on the intensity of solar extraction for the climatic conditions of the Ninh Thuan province of the Republic of Vietnam is discussed.

1 Introduction

1.1 The hybrid solar thermal power plant

According to experts, the main causes of climate change and global warming are greenhouse gas (mainly CO_2) emissions. The EIA's International Energy Outlook 2017 Reference predicts that CO_2-related emissions from energy production will grow by 0.6% annually from 2015 to 2040 [1].

Today, humanity faces the risk of instability in providing energy and reducing energy security, as sources of fossil fuels are gradually being exhausted, while energy consumption is increasing. According to the EIA, the level of energy consumption in the world will increase by 28% in the period from 2015 to 2040 [2], which will lead to higher prices for fossil fuels in the next 20 years.

The world needs reliable and inexpensive alternative energy sources to reduce pollution and ensure sustainable development. Wind and solar energy are very promising and the unit costs for implementing these projects decrease every year, but wind and solar energy cannot satisfy the energy needs of large consumers. For example, in the United States, according to the data for 2018, 30.29 GW of energy was generated on the basis of solar photovoltaic stations, 1.76 GW is for solar thermal energy, and the total energy output was 215.24 GW [3].

Nowadays gas turbine installations are considered to be the most efficient energy-generating equipment, the efficiency of which is more than 40%.

Natural gas, which is a high-calorific and ecological fuel, is used in such installations, and it can become the main fuel during the transition to economy based on renewable energy, in the next several decades.

The hybrid thermal power plants that use both traditional fossil fuels and renewable energy resources can become the basis for such a transition.

Solar thermal power plants use standard power generators, and therefore can easily be combined with other fuel sources [4, 5]. These sources include gas, coal, biofuel, wind, nuclear energy, etc.

Combining solar energy with other fuels such as natural gas can also benefit conventional thermal power plants, reducing emissions to the atmosphere and fuel costs [6, 7].

Such objects can be developed with the prospect of using solar energy, the share of which can be increased, which will contribute to the growth of thermal energy reserves [8, 9].

It will allow the power plant to gradually switch to predominant use of solar energy, and to burn natural gas when it has low price and during certain time periods at peak loads. In addition, hybrid solar power plants operating on solid organic or biofuel can reduce the economic and technical barriers associated with pure solar energy [10]. Therefore, it is economically advantageous to create power plants that combine solar energy with fossil fuels [11, 12].

1.2 Concentrated Solar Power

The concentrated solar power (CSP) is an alternative and renewable energy technology focused on indirect converting of sunlight into electricity. CSP uses a variety of mirrors placed on a large area of earth to direct and concentrate sunlight into a single point known as a receiver. Similar to photovoltaic solar, wind, hydropower and other alternative energy technologies, solar concentrators have the advantage of being sources of renewable, stable or self-sustaining and

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environmentally friendly energy. CSP technology is particularly promising in regions with high direct sunlight (DNI) solar intensity. According to the expert estimates, CSP can be the source of energy used at peak and intermediate loads in the sunniest areas by 2020 and the main source in the period from 2025 to 2030 [13].

Currently there are four available CSP technologies: Parabolic through collector (PTC), solar power tower (SPT), linear Fresnel reflector (LFR) and parabolic dish system (PDS).

Although the PTC technology is the most mature CSP design, solar tower technology occupies the second place and gains growing importance due to its advantages. The SPT system benefits from high operating temperatures. In addition, SPT technology allows obtaining a higher site gradient than PTC technology does, which reduces the site improvement costs [14]. Designing the SPT system allows the entire pipeline system to be concentrated in the power plant, shortening the piping system and thus reducing energy losses and facilitating the diagnosis of incidents and maintenance [15]. SPT technology has been proven commercially and large SPT facilities are now in operation, for example, Ivanpah Solar Power Facility, which is the largest CSP plant.

1.3 System overview

In this paper we present the study of the solar thermal hybrid power plant, which uses SPT technology, gas turbine, a simple Brayton cycle and operates in the environmental conditions of Ninh Thuan Province (Vietnam), with 2 power levels of 4.6 MW and 11.86 MW. The first power plant of 4.6 MW is equipped with a Mercury TM-50 turbine. The plant includes a heliostat field consisting of 69 units, each with a reflective area of 121 m^2. At the second power station with a capacity of 11.86 MW, a PGT-10 turbine is used, and its heliostat area consists of 180 units, each with a reflective area of 121 m^2.

1.4 Climate conditions in Ninh Thuan province of Vietnam

Ninh Thuan is a coastal province in the South Central Coast of Vietnam. The annual average temperature is 27.7°C, humidity is 72%, and DNI is 173 W/m^2. This is the most suitable place in Vietnam to build a hybrid solar thermal power plant.

2 The thermodynamic power plant model

The scheme of a solar power station with a simple Brighton cycle is shown in Figure 1. Air with ambient temperature after compression in the compressor reaches temperature T2. A portion of air passes through the solar tower and is heated to temperature T3 in the air heater, and then mixed with the remaining air. Air mixture of temperature T4 is fed into the combustion chamber, where the fuel is burned. The mixture of combustion

products of temperature T5 is sent to a gas turbine to generate electricity. Exhaust gases are released directly to the environment.
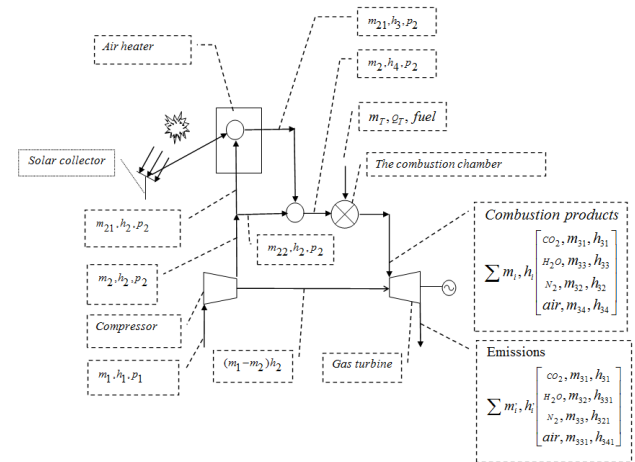

Fig. 1. The schematic diagram of a hybrid mini thermal power plant with solar air heater.

2.1 Solar collector efficiency

The solar subsystem receives heat input from the sun given by $D_a \cdot S$ where D_a is the direct solar irradiance and S is the aperture area of the solar field. The solar irradiance is a function of time because it depends on the sun position during the day, weather conditions, and seasonal fluctuations.

The efficiency of the solar subsystem is determined by the following dependencies:

$$E_2 = \varepsilon_{HC} \cdot \dot{E} \tag{1}$$

$$E_{HS} = (T_3 - T_2)/(T_{HS} - T_2) \tag{2}$$

$$\dot{E} = \eta_0 \cdot D_a \cdot S - \alpha \cdot \delta \cdot D_r \cdot \left(T_{HS}^4 - T_1^4\right) - U_L \cdot D_r \cdot \left(T_{HS} - T_1\right) \tag{3}$$

$$\eta_S = \frac{\dot{E}}{S \cdot D_a} = \eta_0 \cdot \left[1 - X_1 \cdot \left(T_{HS}^4 - T_1^4\right) - X_2 \cdot \left(T_{HS} - T_1\right)\right] \tag{4}$$

After discounting losses, the receiver releases useful energy *E* to a heat exchanger, which in turn releases a final heat rate *E₂* to the working fluid, T_{HS} is the working temperature of the solar collector, T_1 is the ambient temperature. D_a and D_r are, respectively, the aperture and absorber areas; $X_1 = \dfrac{\alpha \cdot \delta}{S \cdot C \cdot \eta_0}$ and $X_2 = \dfrac{U_L}{S \cdot C \cdot \eta_0}$ are the loss parameters, where U_L is the convective heat loss coefficient, α is the effective emissivity of the collector, $C = \dfrac{D_a}{D_r}$ is the

concentration ratio, $\delta = 5.67 \cdot 10^{-8}$ W·m^{-2}·K^{-4} is the Stefan–Boltzmann constant, η_0 is the solar collector efficiency, η_S is the solar subsystem efficiency.

2.2 The Brayton gas-turbine efficiency

The calculation was carried out according to the dependencies presented below:

$$m_1 h_1 + E_1 = m_2 h_2 \qquad (5)$$

Where m_1, m_2 are the mass flows of air at the entrance and at the outlet of the compressor, kg/s ($m_1 = m_2$); h_1, h_2 are the enthalpies of air at the entrance and at the outlet of the compressor, kJ/kg; E_1 is the energy received from the compressor.

$$m_{21} h_2 + E_2 = m_{21} h_3 \qquad (6)$$

where m_{21} is the mass flow of air at the entrance and at the outlet of the solar air heater, kg/s; h_2 is the enthalpy of air entering the solar air heater, kJ/kg; h_3 is the enthalpy of air leaving the solar air heater, kJ/kg; E_2 is the energy of solar radiation spent for heating of m_{21} air, kJ;

$$m_2 h_4 = m_{21} h_3 + m_{22} h_2 \qquad (7)$$

where m_{22} is the mass flow of the remaining air leaving the compressor, kg/s; h_4 is the enthalpy of the air mixture, kJ/kg; m_2 is the mass flow of the air mixture, kg/s;

$$m_2 h_4 + m_F Q_F = m_{31} h_{31} + m_{32} h_{32} + \\ + m_{33} h_{33} + m_{34} h_{34} = \sum m_i h_i \qquad (8)$$

where m_F is the mass fuel consumption, kg/s; m_{31}, m_{32}, m_{33}, m_{34} are masses of CO_2, N_2, H_2O and air, respectively; h_{31} is the enthalpy of CO_2 leaving the combustion chamber, kJ/kg; h_{32} is the enthalpy of N_2, leaving the combustion chamber, kJ/kg; h_{33} is the enthalpy of H_2O leaving the combustion chamber, kJ/kg; h_{34} is the enthalpy of excess air leaving the combustion

chamber, kJ/kg; Q_F is the net calorific value of fuel, kJ/kg.

$$m_{31} h_{31} + m_{32} h_{32} + m_{33} h_{33} + m_{34} h_{34} + (m_1 - m_2) h_2 = \\ = E_3 + m_{31} h_{311} + m_{32} h_{321} + m_{33} h_{331} + m_{341} h_{341} \qquad (9)$$

where m_{341} is the mass air flow after the turbine, kg/s; h_{311} is the enthalpy of CO_2 at the turbine outlet, kJ/kg; h_{321} is the enthalpy of N_2 at the turbine outlet, kJ/kg; h_{331} is the H_2O enthalpy at the turbine outlet, kJ/kg; h_{341} is the enthalpy of air at the turbine outlet, kJ/kg; E_3 is the energy produced by the system.

Isentropic compressor efficiency is:

$$\varepsilon_C = \frac{T_{2S} - T_1}{T_2 - T_1} \qquad (10)$$

where T_{2S} is the temperature of the working fluid after the compression process if it was adiabatic and T_2 is the actual temperature at the compressor outlet.

Thermal cycle efficiency is:

$$\varepsilon_{HC} = \frac{T_5 - T_4}{T_{HC} - T_4} \qquad (11)$$

$$E_{HC} = \varepsilon_{HC} \cdot \eta_C \cdot m_F \cdot Q_F = \\ = m_2 \cdot c_2 \cdot \varepsilon_{HC} \cdot (T_{HC} - T_4) \qquad (12)$$

where T_{HC} is the working temperature of the combustion chamber, E_{HC} is the heat input from the combustion chamber, η_C is the combustion chamber efficiency, c_2 is the specific heat of air.

The thermal efficiency of the whole system η is the ratio between the net mechanical power output P and the total heat input rate:

$$\eta = \frac{E_3}{m_F \cdot Q_F} \qquad (13)$$

The total heat input rate is:

Table 1. Assessment of efficiency of hybrid mini thermal power plants.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
	P =4.6MW											
η	0.374	0.39	0.389	0.391	0.378	0.369	0.367	0.363	0.375	0.363	0.368	0.375
f	0.075	0.114	0.11	0.115	0.085	0.06	0.057	0.044	0.076	0.047	0.059	0.075
m_F, kg/s	0.236	0.227	0.227	0.226	0.234	0.24	0.241	0.244	0.236	0.243	0.24	0.236
	P =11.86 MW											
η	0.342	0.356	0.354	0.356	0.345	0.337	0.336	0.332	0.342	0.333	0.337	0.342
f	0.071	0.108	0.105	0.109	0.081	0.058	0.055	0.043	0.073	0.045	0.057	0.072
m_F, kg/s	0.667	0.641	0.643	0.64	0.66	0.676	0.679	0.687	0.666	0.685	0.677	0.667

$$E_H = E_2 + E_{HC} \qquad (14)$$

Solar share:

$$f = \frac{E_2}{E_H} \qquad (15)$$

3 Results

When performing calculations for two hybrid stations with capacities of 4.6 and 11.86 MW, the expressions (1–15) were used to obtain results presented in Table 1. The basic version of the calculation assumed the absence of solar radiation ($D_a = 0$). The intensity of solar radiation over time varied in accordance with the climatic conditions of the Ninh Thuan province of the Republic of Vietnam.

4 Conclusions

According to the calculations performed for the basic case with no solar radiation, the efficiency of the 4.6 MW power station was η = 0.345, the fuel consumption mF = 0.056 kg/s, and for the 11.86 MW station η = 0.317 and mF =0.72 kg/s. During the year efficiency and fuel consumption are constant, since the ambient temperature is almost unchanged. The highest intensity of solar radiation is achieved in February and April. At that time the efficiency is increased by 13.3% compared to the basic case, while fuel consumption is reduced by 11.7% (for a 4.6 MW station). For a more powerful station, efficiency is increased by 12.3% and fuel is reduced by 11.1%. Thus, due to the use of solar energy, substantial fuel savings are achieved.

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