1. Introduction

Microcogeneration Unit with Nonconventional Engine. Combustion engines with gas as fuel, with the mostly used being natural gas, are most used as power unit of microcogeneration devices. Losses in electricity generation are mainly associated with imperfect energy transformation in burning fuel in an imperfect transformation of energy working medium in the turbine. There are also represented mechanical losses and loss of energy in transmission lines. The minimal losses have cogeneration plants. Cogeneration unit is a technical device, which is manufactured by electric and thermal energy simultaneously. As an example cogeneration unit may be mentioned with an internal combustion gas engine. The engine burns the gas, thereby gaining the mechanical power on the shaft to drive an electric generator. The engine has no classic cooler but has the heat exchanger from which we obtain thermal energy. Used heat exchangers are connected in series circuits, where the working medium, usually water, is preheated and heated in several stages. Overall efficiency of CHP unit can be increased with multistage heat recovery and this reduces also the total cost of fuel [1]. Unconventional engines are a possible alternative to the internal combustion engines. They work with external combustion, which allows, unlike conventional internal combustion engines, controlling the course of combustion and therefore its quality, which is reflected in the composition of air pollutants emitted to the atmosphere. The most known hot-air engines are Stirling and Ericsson engines. Ericsson engine is also an external combustion engine. In contrast to Stirling engine, it has two possible alternatives—open and closed [2]. In the case of Stirling engine dual function of regenerator is immediately apparent. Regenerator works as heater and cooler while in Ericsson engine cooler and heater are separated. Figure 1 presents Ericsson-Brayton engine with open cycle.

The air is compressed in the compressor and flows through the heat exchanger, where at constant pressure it is receiving heat. Consequently, it is led to the expansion cylinder, which expands adiabatically. Part of this work will be used to drive the compressor and part is used as mechanical work to drive an electric generator. As the heat source can be used almost any fuel for burning, as it is an external combustion engine. Fuel is burned in a separate combustion chamber and heat energy is transformed through
a heat exchanger to the working media. The working medium in open cycle, mostly dry air, after passing the cycle, is discharged into the atmosphere. In a closed cycle the medium after each cycle cools in refrigerant heat exchanger, where it gives heat energy and is fed back into the cycle. With the use of closed cycle we can improve the efficiency of heating equipment [3, 4].

The proposed microcogeneration unit uses two heat exchangers: cooler and heater (see Figure 2). A different highest temperatures from 500°C of the operating conditions we have preliminary set the should supply energy for household. In the determination heatexchanger, but also on the whole system. The whole unit engine sets a wide range of specifications, not only on the

2. Heat Exchanger Design

As the first step, the working conditions of the CHP were set. The experimental application with Ericsson-Brayton hot-air engine sets a wide range of specifications, not only on the heat exchanger, but also on the whole system. The whole unit should supply energy for household. In the determination of the operating conditions we have preliminary set the highest temperatures from 500°C up to 620°C, according to [3, 5]. In this paper, the authors presented the highest temperature of 600°C. Mr. Creyx [1] has presented systems with different working fluids and also different hot-air engine configurations. The system presented in this paper should work with closed cycle, with dry air as working fluid [4]. The closed cycle enables heat recovery from working fluid, so the regenerated heating power is bigger than that in opened cycle, where the most part of heat energy is used to preheat the air after compression. It is assumed that temperature of the working fluid after expansion is in the range of 240°C–320°C [4, 5]. For each working fluid, the dry air in the tubes and the exhaust gases outside the tubes were set as the characteristic temperatures and physical properties.

There are many ways for how to calculate the properties of flowing media. In order to determine the heat transfer, it was necessary to know the thermodynamic properties of flowing gas. It is important to determine the dynamic and kinematic viscosity. For heat transfer the thermal conductivity of the gas is also needed to be known. The following equations were used for calculation [6, 7].

Dynamic viscosity is

$$\eta_{TP} = 1.0607 \cdot 10^{-6} \cdot T^{0.5} \cdot k_T \cdot k_p.$$  \hspace{1cm} (1)

Kinematic viscosity is

$$\nu_{TP} = 304.52344 \cdot 10^{-6} \frac{T^{1.5}}{p} k_T k_p,$$ \hspace{1cm} (2)

and the thermal conductivity is

$$\lambda = 1513.8151 \cdot 10^{-6} T^{0.5} (k_T k_p)^{1.5}.$$ \hspace{1cm} (3)

The coefficients $k_T$ and $k_p$ are based on temperatures from 0°C up to 1000°C. The main difference to real values of parameters is up to 3%, so it is possible to say that the computation is accurate. Table 1 shows the values of coefficient $k_T$.

The values of the coefficient $k_p$ are set for dry air by temperature constant temperature. In Table 2, some values of coefficient $k_p$ can be seen.

There are many methods for calculation of flue gas density. Two of them were used in this work. At first it is possible to read the right values in the tables, which are calculated or measured. In Table 3 there are some values of air properties. The second column is density. The values that are not in the table can be calculated.

Density can be calculated based on known parameters. In the following equation density is expressed as a function of dynamic and kinematic viscosity:

$$\rho = \frac{\mu}{\nu}.$$ \hspace{1cm} (4)

![Figure 1: Scheme of Ericsson-Brayton hot-air engine with open cycle.](image-url)
So in this way we can define the properties of flowing medium. Specifying the geometrical properties or features of the chosen type of heat exchanger is also very important. There are many basic concepts of heat exchangers. Based on the geometrical features or heat transfer, methods can be classified into many classes. For the purpose of this work, pipe exchanger was selected. The heat exchanger in this category differs in the arrangement of tubes. The tubes can be organized straight or staggered or partly staggered. It is characterized with the dimensionless constants “a” and “b.”

If the tube bundle has horizontal spacing “s1,” and vertical spacing “s2,” as in Figure 3, the bundle can be characterized with these constants:

\[
a = \frac{s_1}{d_0},
\]

\[
b = \frac{s_2}{d_0},
\]

\[
\psi = 1 - \frac{\pi}{4} \cdot a.
\]

The streamed length “l” can be expressed as length of flow path across over a single tube [7]:

\[
l = \frac{\pi}{2} \cdot d_0.
\]

Another difference is in the nondimensional criteria. Reynolds number is characterizing the flowing medium and the type of flow. It depends on flow velocity and also on the geometry. For heat transfer through tubes in bundle the following Reynolds number criteria were used:

\[
Re = \frac{\omega \cdot l}{\nu}.
\]

Nusselt number is characterizing the heat transfer. If the turbulence in the inflowing medium is low, deviations in the Nusselt number may occur. The average Nusselt number in a cross-flow over a bundle of smooth tubes can be calculated from that in a cross-flow over a single tube. For the purpose of this work, the criteria equation was used according to [7, 8]. The heat transfer is described by the 2 parts of flow, the turbulent part and the laminar part of the flow near the walls, as follows:

\[
\text{Nu}_{\text{lam}} = 0.664 \cdot \sqrt{Re_{\psi J} \cdot \sqrt{Pr}},
\]

\[
\text{Nu}_{\text{turb}} = \frac{0.037 \cdot Re_{\psi J}^{0.8} \cdot Pr}{1 + 2.443 \cdot Re_{\psi J}^{-0.1} \cdot (Pr^{2/3} - 1)}.
\]

Turbulent flow in pipe sets in at Re > 10⁴. In the transition region of Reynolds number from 2300 to 10⁴ the type of flow
is also influenced by the nature of inlet stream and the form of pipe inlet. Tube bundles with in-line tubes behave more like parallel channels, which are formed by the tube rows. An expected increase in heat transfer coefficient due to the turbulence enhancement caused by the tube rows does not occur [7].

The application for hot-air Ericsson-Brayton engine will use as primary heat exchanger tube with staggered tubes.

The average Nusselt number for this type of heat transfer through tube bundle is defined according to the following equation [7]:

$$N_u_{bundle} = \frac{1 + (n - 1) \cdot f_a}{n} \cdot N_u_{l,0},$$

(9)

where

$$f_a,stag = 1 + \frac{2}{3b},$$

(10)

$$N_u_{l,0} = 0.3 + \sqrt{N_u_{l,\text{lam}}^2 + N_u_{l,\text{turb}}^2}.$$

Then followed the estimation of overall coefficient of heat transfer, which is depending on the Nusselt number, is

$$\alpha = \frac{N_u_{\text{bundle}} \cdot \lambda_{TM}}{l},$$

(11)

When both sides of equation are known, the overall heat transfer coefficient and the required heat transfer surface can be estimated. Subsequently, 3D model of heat exchanger was created. The model was in the first step created with wall thickness of tubes and inlet tube. But this solution sets major requirements for computing hardware, so a simplified model with tubes as full material was created, the proposal is in Figure 4.

### 3. Heat Exchanger Verification

#### Using Ansys Fluent

The model for Ansys Fluent was created using 3D modeling software. By the creation of the model the substitution of all the construction elements by simple geometrical features was very important [8].

This means that the whole exchanger was created as one volume with tubes as full material. The tubes have multiple collectors at inlet and outlet. No construction tolerances are reflected.

The exterior of the heat exchanger was created by cutting out the material from volume. In the first step, the tetrahedron mesh was used to fill the whole volume. The generated mesh is displayed in Figure 5. Quality of the generated mesh is determined by skewness of elements and by minimal orthogonal quality. Skewness of the model was 7.1833×10⁻⁷, where lower value is representing worse quality [8]. Based on this, the tetrahedron mesh was converted to polyhedral mesh.
The details of the generated mesh are in Figure 6. The model was solved with polyhedral mesh and \( K-\varepsilon \) model. The flow was predicted as turbulent. In Figure 7 the velocity contour is shown and in Figure 8 the temperature field can be observed. The current model has confirmed the mathematical model and also accuracy of chosen geometry.

4. Conclusion

Hot-air Ericsson-Brayton engine used in cogeneration unit is a nonconventional engine, which produces electric energy by using different types of fuel, for example, biomass, wood pellets, and so forth. Heat exchanger design for hot-air Ericsson-Brayton engine sets a wide range of specifications. At the first step, the working conditions of whole unit and the required power and temperatures for each part of this device were defined. The basic dimensions of heat exchanger were set using criterion formula. With this calculation, the inlet and outlet temperatures of the heat exchanger were verified. Then, the calculation using Ansys Fluent has followed. As the next step was to complete the construction documentation and finish all design fundamentals, the construction and real measurements can follow.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.
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