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# Numerical analysis of the tubular heat exchanger designed for co-generating units on the basis of microturbines

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ABSTRACT

The application of numerical simulations using the computational fluid dynamics (CFD) analysis when mapping processes in the course of which the heat transmission occurs has become an essential part of the heat transfer systems. The present contribution deals with the possibility to use the waste heat of the flue gas produced by small microturbines. The waste heat is mapped by means of both the numerical simulations applying the FLUENT software and the practical experiment. Utilizing a part of the waste heat for water heating and decreasing the outlet temperature of the flue gas into atmosphere when applying in co-generating units represents one of the partial benefits. The present paper brings information concerning the newly designed type of heat exchanger including the results of its numerical analysis.

The analysed heat exchanger designed in the system with microturbine (MT) C30 reached generally the efficiency of 75%. Both the results of simulations and the carried out practical experiment confirmed the temperature of the flue gas to be sufficient behind the exchanger to prevent the condensation of water from the flue gas. On the contrary, except for heating water the exchanger under consideration offers – thanks to its design – also other possibilities to use of the flue gas. The practical experiment confirmed the results of the CFD prediction with rather small differences as the temperature of water obtained from the exchanger was 359 K and the designed shape of the exchanger did not result in substantial pressure losses in flue gas approximately 50 Pa. The mean logarithmic temperature difference of the mapped and verified exchanger was ~203 K.



## 1. Introduction

One possibility how to increase the efficiency of the energetical processes based on gaseous fuels combustion is to design various types of heat exchangers which use the waste heat of the flue gas produced in various phases of the energetical processes. The heat exchangers specified for preheating the combustion air are solved regarding the combustion equipment. To reduce the heat loses these are constructed in vicinity to the combustion equipment [1–4]. In case of small microturbines (MT) the recuperator [5] is frequently used to achieve the efficiency exceeding 30%. Due to low initial costs the compactness of the solution is necessary with respect to the relatively small MT size. The combustion air preheating is carried out in the recuperator which forms the MT outer wall. For example the Capstone Company has been using the annular primary-surface recuperator for a relatively long time. Several thousand recuperators of this type have been fabricated for MT service (30 and 60 kW units) and they have accumulated over a million operating hours without a failure [6]. Another type of the exchanger of a different construction the installation of which strongly increases the equipment efficiency are the spirally wound recuperators with wavelike corrugation, the function of which is based on contra-flow of air and hot flue gas [7,8].

It was Wilson [9] who intensively studied the heat exchangers from the standpoint of their various constructions and pointed out especially the design of the exchanger concerning the pressure losses in the flowing air. However, the effort to increase the heat (or total) efficiency of the process concerns predominantly the problem of using the heat directly from the combustion equipment, e.g. from the MT wall or the combustion space. Such exchangers are based on the results of the numerical simulations which include the boundary conditions, such as the velocity, pressure and direction of the flowing combustion air and others.

Nevertheless, relatively little attention has been paid to various possibilities of getting the waste heat from the flue gas of small and middle MT. There exist studies dealing with heat exchangers from the standpoint of the heat transfer in dependence on the heat

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**Nomenclature**

| | |
|---|---|
| $a$ | velocity of sound (m s$^{-1}$) |
| $C_\mu$, $C_{1S}$, $C_2$, $C_{3S}$ | empiric values |
| $c_v$ | specific heat capacity at constant volume (J kg$^{-1}$ K$^{-1}$) |
| $c_p$ | specific heat capacity at constant pressure (J kg$^{-1}$ K$^{-1}$) |
| $E$ | total energy (J kg$^{-1}$) |
| $G_k$ | generation of turbulence kinetic energy due to the mean velocity gradients (kg m$^{-1}$ s$^{-3}$) |
| $G_b$ | generation of turbulence kinetic energy due to buoyancy (kg m$^{-1}$ s$^{-3}$) |
| $g$ | gravity (m s$^{-1}$) |
| $h$ | sensible enthalpy (J kg$^{-1}$) |
| $\bar{u}_i$ | $i$th component of medium velocity (m s$^{-1}$) |
| $\bar{x}_i$ | Cartesian coordinates in the system ($x_1$, $x_2$, $x_3$) or ($x$, $y$, $z$) (m) |
| $p$ | density (kg m$^{-3}$) |
| $p$ | pressure (Pa) |
| $\mu$ | dynamic viscosity (Pa s) |
| $\mu_t$ | turbulent viscosity (m$^2$ s$^{-1}$) |
| $k$ | turbulent kinetic energy (m$^2$ s$^{-2}$) |
| $\sigma_k$, $\sigma_s$ | empirical constants |
| $Y_m$ | dilatation dissipation term (kg m$^{-1}$ s$^{-3}$) |
| $\varepsilon$ | rate of dissipation (m$^2$ s$^{-3}$) |
| $M_t$ | turbulent Mach number |
| $\gamma$ | ratio of specific heat ($c_p/c_v$) |
| $R$ | specific gas constant (J kmol$^{-1}$ K$^{-1}$) |
| $T$ | absolute temperature (K) |
| $Pr$ | turbulent Prandtl number |
| $\upsilon$ | component of the flow velocity parallel to the gravitational vector (m s$^{-1}$) |
| $u$ | component of the flow velocity perpendicular to the gravitational vector (m s$^{-1}$) |
| $k_{eff}$ | effective thermal conductivity (W m$^{-1}$ K$^{-1}$) |
| $k$ | thermal conductivity (W m$^{-1}$ K$^{-1}$) |
| $(\tau_{ij})eff$ | deviatoric stress tensor (Pa) |
| $\mu_{eff}$ | effective viscosity (Pa s) |
| $\delta_{ij}$ | Kronecker delta |
| $Y_i$ | mass fraction of species $i'$ |
| $J_i$ | diffusion flux of species $i'$ (kg m$^{-2}$ s$^{-1}$) |
| $D_{i',m}$ | mass diffusion coefficient (m$^{-2}$ s$^{-1}$) |
| $S_{ct}$ | turbulent Schmidt number |
| $t$ | time (s) |
| $S$ | exchanger heat transfer surface (m$^2$) |
| $Q_m$ | mass flow rate (kg s$^{-1}$) |
| $\alpha$ | heat transfer coefficient (W m$^{-2}$ K$^{-1}$) |
| $k_c$ | total heat transfer coefficient (W m$^{-2}$ K$^{-1}$) |
| $\overline{\Delta T}$ | logarithmic mean temperature difference (K) |
| $P_{fg}$ | heat power in flue gas (W) |
| $P_w$ | heat power in water (W) |
| $P_l$ | waste heat (W) |
| $\eta$ | total exchanger efficiency (%) |

transfer surface walls [10], shape and size of fins [11]. Similarly also the design of exchangers as to the size or efficiency of the energetical units is drawn up [12,13]. As given above, a great part of the flue gas potential is passed to the inlet combustion air; however, there is still a relatively great part of heat which remains dormant. The temperature drop of the outgoing flue gas may see for an increased heat or total efficiency of the process, which is an indisputable energetical benefit. At the same time both the energetical exploitation of the waste heat of the flue gas and the temperature reduction of the outgoing flue gas into atmosphere are environmentally friendly.

The present paper offers a better exploitation of the waste heat of the flue gas on the condition that pressure losses of the flue gas are minimized. To achieve the given aim, a suitable heat exchanger had to be numerically verified. The designed and then constructed exchanger was tested in practice and the results of the numerical simulation were compared with the real measured values.

## 2. Experimental

One of the main objectives of this experimental was to evaluate the heat exchanger consisting of the MT C30. This co-generating unit worked with natural gas as fuel; however, it is able to burn also other kinds of fuel. The whole system is diagrammatically shown in Fig. 1(a). The experimental part of this contribution is divided into two parts. The first part represents the very design of the exchanger whereas the main attention was paid on its simple construction. In this stage the commercial software FLUENT [14] based on FVM (finite volume method) together with grid system software GAMBIT were applied. Using the results of this computational fluid dynamics (CFD) simulation the final shape was designed (Fig. 1(b)) and its practical manufacturing started. In the second part of this experimental the theoretically designed and a finally constructed heat exchanger was practically proved.

As evidently seen from the enclosed model, it is a sectional exchanger with 6 sections altogether, each section is created by tube of helical shape. Each section is interconnected with the following one. It means that all 6 sections are interconnected forming one unit

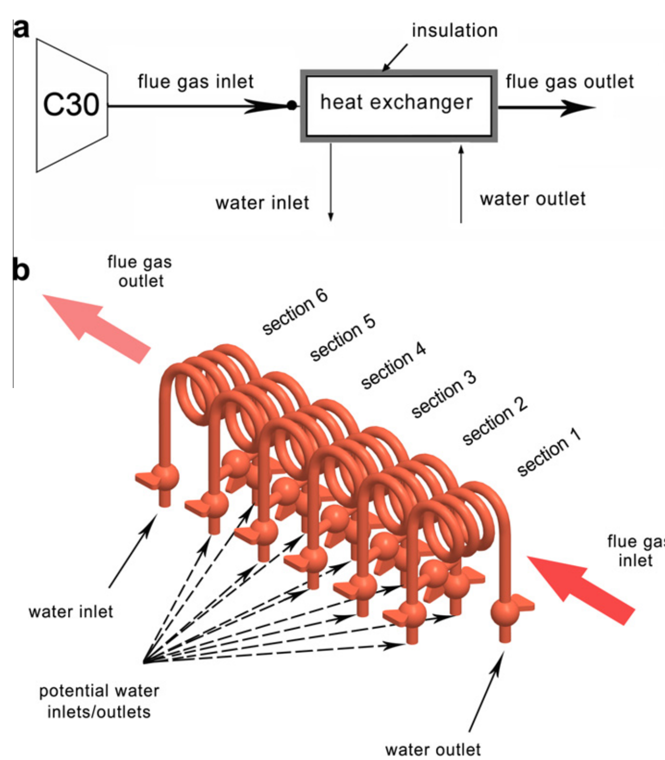


**Fig. 1.** The co-generating unit system layout (a) the analysed system layout (b) model of the actual heat exchanger under analysis.