

PROPOSAL OF HEAT EXCHANGER IN MICRO COGENERATION UNIT, CONFIGURATION WITH BIOMASS COMBUSTION

Jozef HUŽVÁR, Patrik NEMEC

Authors: Jozef Hužvár, MSc. Eng. Patrik Nemeč, MSc. Eng.
Workplace: Faculty of Mechanical Engineering, Department of Power, University of Žilina
Address: Univerzitná 1, 010 26 Žilina, Slovak Republic
Phone: +421 41 513 2866
E-mail: jozefhuzvar@gmail.com, patrik.nemec@fstroj.uniza.sk

Abstract

Combined heat and power (CHP) production brings a lot of savings of primary energetic sources and it is suitable not only for big and medium operations, but for a small applications as well, for example for householders etc., where the power consumption is only a few kilowatts. This cogeneration area is marked like micro cogeneration. Micro cogeneration unit combusting biomass is alternative of decentralized energy production.

In area of cogeneration and micro cogeneration are used more technologies, like for example engines with internal combustion, Stirling engines, gas micro turbines, ORC (Organic Rankine cycles), fuel cells etc.

This article describe problem of micro cogeneration conversion way of chemically fixed energy of solid biomass on electric power and low – potential heat and optimal heat exchange between boiler and work space of two-cycle engine. Project subject is research of converse way and device development on its realization.

Key words

Micro Cogeneration Unit, Heat exchanger, CFD methods

Introduction

The EU is committed to reducing its overall emissions to at least 20% below 1990 levels by 2020, and is ready to scale up this reduction to as much as 30% under a new global climate change agreement when other developed countries make comparable efforts. It has also set itself the target of increasing the share of renewable in energy use to 20% by 2020.

One of the way how get this target is Cogeneration. Cogeneration is progressive technology of electricity and heat production, based on principle of conjugate production, what means, that electricity and heat are produced at the same time. Nowadays not only fossil fuels but also biomass began to be used.

Target and Proposal

The main target is to design minimally one suitable Micro-cogenerational process incl. the conversion from chemical energy to electrical one using biomass and the low potential heat using analytic-experimental methods and determines the functional and material requirements of optimal conversion process based on the technical and economical optimization. Since the working device is an experimental equipment, there are several ways for its construction. The device consists of the boiler for dendromass, the two-cycle engine and two heat exchangers (fig. 1).

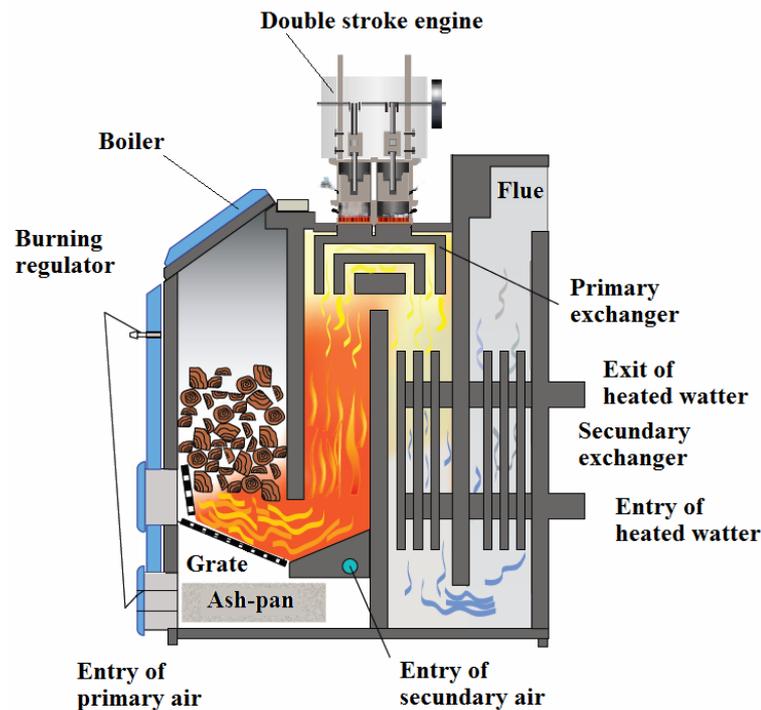


Fig. 1 Proposal of Micro-cogeneration unit

One of the proposal ways is based on the principle to exchange heat from the boiler to the working space by using of a copper exchanger. During the compress piston stroke a consistent application of sprayed liquid is done by the injection jet on the front area of the exchanger. The spraying of liquid into large surface area makes possible a quick heat transfer between exchanger and liquid (adding of heat into liquid form accumulated heat in exchanger). After the evaporation liquid is changed into the high temperature steam and extends its volume. Polytrophic or simplified isothermal-adiabatic expansion follows after that and engine starts to work. After finishing of expansion up-stroke steam leaves work space of engine.

Important aspect of optimal heat exchange is appropriate design of exchange area between boiler and work space of two-cycle engine (fig. 2) to reach sufficient capacity needed to warm up of liquid and its evaporation.

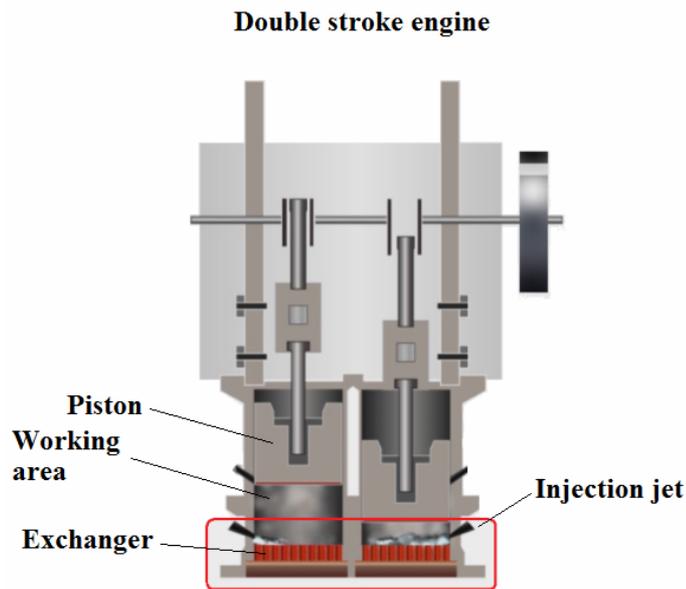


Fig. 2 Part of heat exchange

Working steps

1st Step: The following calculations were carried out: 1) the specific volume extension during the evaporation as a function of pressure (fig.3), 2) the required amount of injected water as a function of pressure and temperature (fig. 4) and 3) the required amount of heat for the heating and evaporation of water as a function of pressure (fig. 5), which all are based on the size of the double stroke engine.

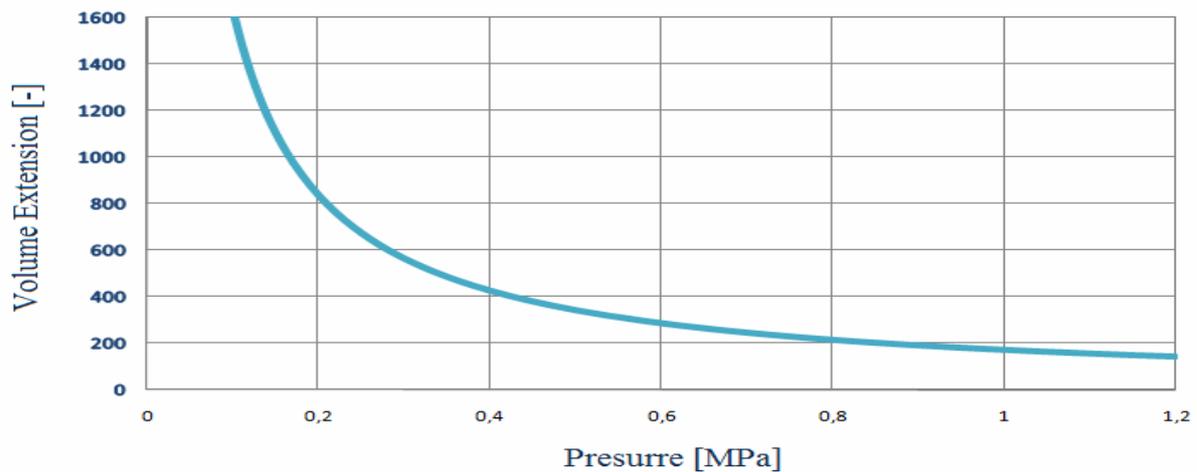


Fig. 3 Specific volume extension as a function of pressure during evaporation

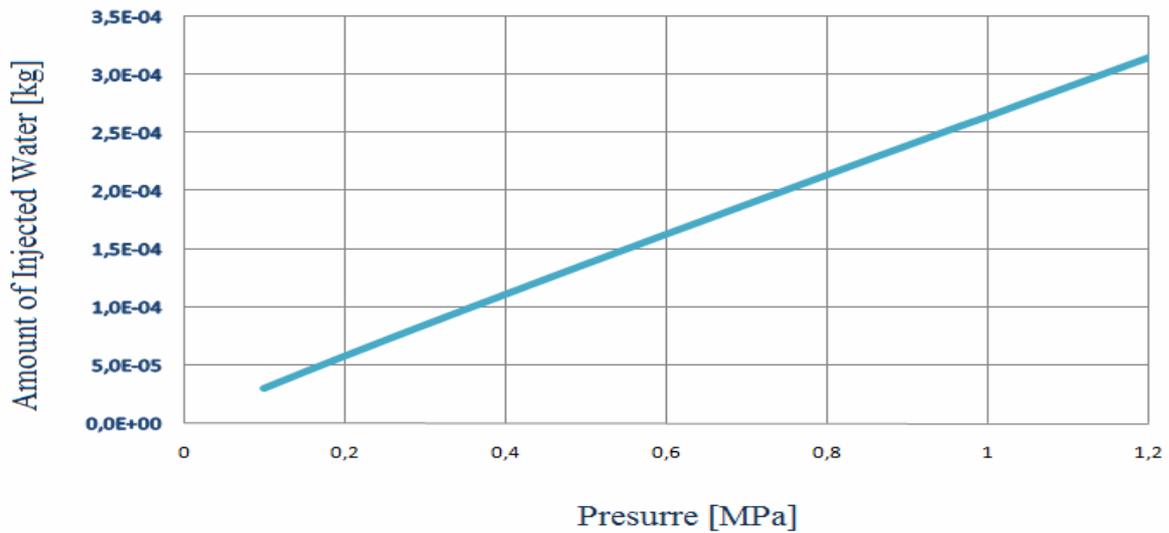


Fig. 4 Required amount of injected water as a function of pressure and temperature

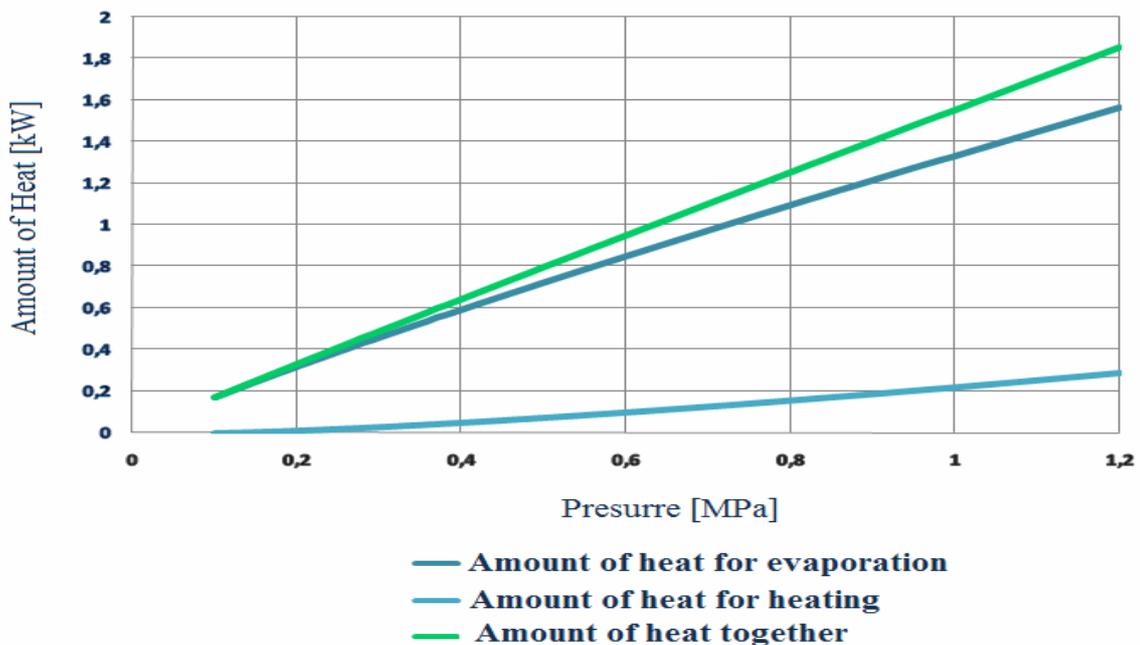


Fig. 5 Required amount of heat for heating and evaporation of water as a function of pressure

2nd Step: In the next step the volume of exhaust gas per hour, the quantity of fuel per hour, the velocity, the viscosity, the Reynolds and the Prandtl numbers as well as all other input variables were calculated where the stoichiometric equations, boiler characteristics and sizes of exchanger were used.

3rd Step: This stage was started with the drawing and meshing different kind of head pipe exchanger models (fig. 6). The pipes' diameter changed from 4 mm to 8 mm, and the ordering of the head pipes was both the in-line tube and the staggered tube arrangements.

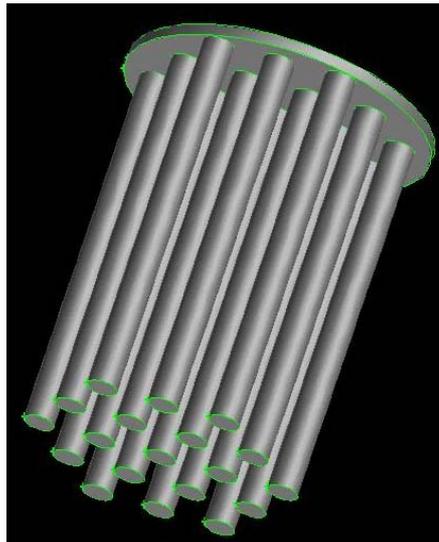


Fig. 6 Heat pipes exchanger

4th Step: All above calculations have been used for modeling the heat transport in copper heat exchanger by using the commercial software Fluent. In this work the standard $k-\varepsilon$ model and the $RNG\ k-\varepsilon$ model were used which are described in details below.

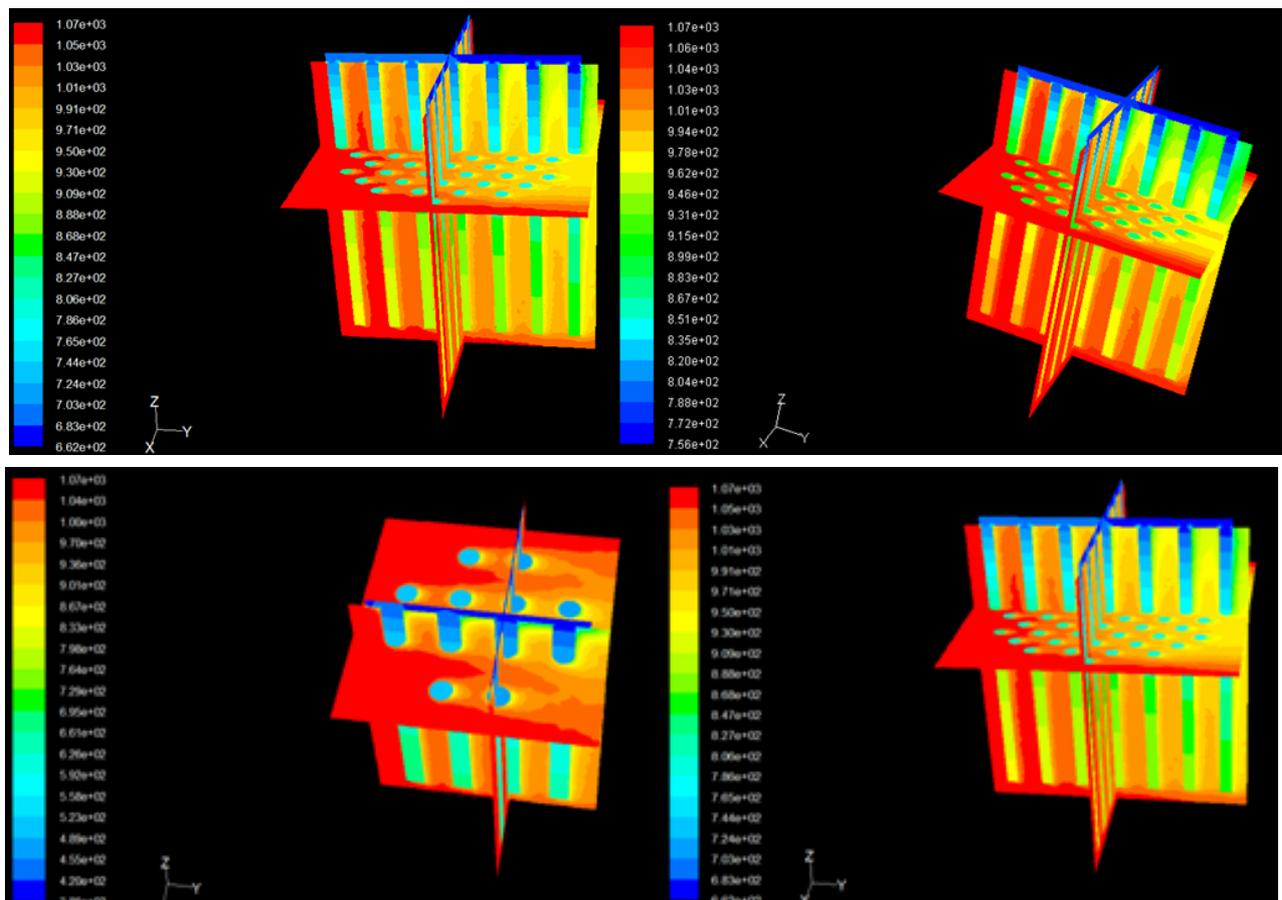


Fig. 7 Comparison of different kind of models

Governing equations

Computational Fluid Dynamics (CFD) is based on the solving of the conservation or transport equations for mass, momentum, energy and chemical species. The basic equations for a turbulent fluid flow are the Reynolds-Averaged Navier-Stokes (RANS) equations and their steady state forms are defined in this Chapter [1].

Continuity equation

The mass conservation equation of the gas phase is written as:

$$\nabla(\rho\bar{v}) = 0, \quad (1)$$

where ρ is the fluid density and \bar{v} is its ensemble-averaged velocity vector defined on a 3D domain.

Momentum equations

The Navier-Stokes (~momentum) conservation equations are defined as:

$$\frac{\partial}{\partial t}(\rho U_i) + \frac{\partial}{\partial x_j}(\rho U_i U_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho f_i, \quad (2)$$

where f_i are the sum of external forces and τ_{ij} is the viscous stress tensor described by the Newton law:

$$\tau_{ij} = \mu \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} \mu \delta_{ij} \frac{\partial U_l}{\partial x_l} \quad (3)$$

where δ_{ij} is the Kronecker symbol and μ is the molecular viscosity depending on the fluid properties [1].

Energy equation

When considering heat transfer within the fluid or solid regions of the domain, the software covers also the general form of the energy equation written below:

$$\frac{\partial}{\partial t}(\rho h) + \frac{\partial}{\partial x_i}(\rho h U_i) - \tau_{ij} \frac{\partial U_i}{\partial x_j} + \frac{\partial q_i}{\partial x_i} = \rho U_i f_i + S_h, \quad (4)$$

where h is the total specific enthalpy and which for a multicomponent medium takes the following form:

$$h = \sum Y_i h_i, \quad (5)$$

where Y_i is the mass fraction of the species i in the mixture and h_i is its total enthalpy written as:

$$h_i = h_{T_{ref},i}^0 + \int_{T_{ref}}^T C_{p,i}(T) dt \quad , \quad (6)$$

where h^0 , T_{ref} and $C_{p,i}(T)$ are the formation enthalpy, the reference temperature and the specific heat at constant pressure, respectively.

Species transport equations

Finally, the behaviour of the species i will be solved by the following equation:

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla(\rho \vec{v} Y_i) = -\nabla \vec{J}_i + R_i + S_i \quad (7)$$

where S_i is the rate of creation by the addition from the dispersed phase plus any user-defined sources, R_i is the net rate of production of species i by chemical reaction and \vec{J}_i is the mass transfer rate. The finite volume method to solve the governing flow equations described above is applied [1].

Modelling of Gas Phase

The modelling results of the turbulent reactive flow in this work are based on the *Standard* and the *RNG k-ε* turbulent model, the Discrete Ordinates (DO) radiation model and the Eddy Dissipation Combustion Model (EDM, turbulent gas phase combustion). The sensitivity of the combustion and turbulence models in CFD predictions is also evaluated. The calculations of the following turbulence and combustion models were done for the comparison:

Turbulence:

- the *Standard k-ε* model
- the *Realizable k-ε* model
- the *RNG k-ε* model

Combustion:

- the *Finite-Rate/Eddy-Dissipation* model
- the *Eddy-Dissipation Model*
- the *Eddy-Dissipation-Concept* model.

Turbulence model

The *k-ε* turbulence model has become widely used as it provides robustness, economy and reasonable accuracy for different kinds of turbulent flows. Three types of the *k-ε* model are introduced in this chapter. The form of transport equations for k and ε in standard, realizable and RNG models are similar. The main differences between the three models are as follows:

- The turbulent Prandtl Numbers controlling the turbulent diffusion of k and ε
- The production and destruction terms in the equation for ε
- The approach of calculating turbulent viscosity.

The **Standard k - ε** (SKE) model is the most widely used turbulence model in industrial applications. The equation for the turbulent kinetic energy k is derived directly from the Navier-Stokes equations and dissipation rate ε could be derived in the same way, usually a different approach is taken. The two-equations of turbulence models employ the Boussinesq hypothesis to relate the Reynolds stresses with the mean flow field gradients. In the SKE model, the turbulent kinetic and its dissipation rate are computed from the following equations:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k U_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \mu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \frac{\partial U_i}{\partial x_j} - \rho \varepsilon \quad (8)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon U_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} \mu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \frac{\partial U_i}{\partial x_j} - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad (9)$$

where μ_t is computed by combining k and ε as follows:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} . \quad (10)$$

The standard values for constants of the model are listed below:

$$C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92, \sigma_\mu = 1.0, \sigma_k = 1.0, \sigma_\varepsilon = 1.3$$

These default values have been determined from experiments on turbulent shear flows including homogeneous shear flows and decaying isotropic grid turbulence. They have been found to work fairly well for a wide range of wall-bounded and free shear flows [1].

The **RNG k - ε** (RNG) turbulence model is derived from the Navier-Stokes equations, using a mathematical technique called *renormalization group* (RNG) methods. The RNG model includes constants which are different to those used in the standard k - ε model and also adds new terms to the transport equations for the turbulent kinetic energy and its dissipation. This model has an additional term in its equation that significantly improves the accuracy for rapidly strained flows. The effect of swirl on turbulence is also accounted for in the RNG model, enhancing the accuracy for swirling flows. The RNG theory provides an analytical formula for turbulent Prandtl numbers, while the standard model relies on user-specific constant values. Finally, this model uses an analytically derived differential formula for the effective turbulent viscosity which accounts for low Reynolds number flows, while in the SKE a high Reynolds number model is including. Therefore, the RNG k - ε model is more accurate and more reliable [2].

The equation for k and ε are as follows:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k U_i) = \frac{\partial}{\partial x_j} \left[\alpha_k \mu_e \frac{\partial k}{\partial x_j} \right] + \rho (P_k - \varepsilon) \quad (11)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon U_i) = \frac{\partial}{\partial x_j} \left[\alpha_\varepsilon \mu_e \frac{\partial \varepsilon}{\partial x_j} \right] + \rho \frac{\varepsilon}{k} (C_{1\varepsilon}^* P_k - C_{2\varepsilon}^* \varepsilon) \quad (12)$$

where G_k denotes the generation of turbulent kinetic energy that arises due to mean velocity gradients, G_b represents the generation of turbulent kinetic energy due to buoyancy and Y_M is

the fluctuating dilatation in compressible turbulence that contributes to the overall dissipation rate. α_k and α_ε are the inverse effective Prandtl numbers for the turbulent kinetic and its dissipation.

The constants in the RNG model are defined as:

$$C_{1\varepsilon} = 1.42, C_{2\varepsilon} = 1.68, \alpha_k = \alpha_\varepsilon = 1.39$$

The RNG theory uses a scale elimination procedure that describes the turbulent viscosity in the following equation:

$$d\left(\frac{\rho^2 k}{\sqrt{\varepsilon\mu}}\right) = 1.72 \frac{\hat{\nu}}{\sqrt{\hat{\nu}^3 - 1 + C_\nu}} d\hat{\nu} \quad (13)$$

where

$$\hat{\nu} = \mu_{eff} / \mu$$

$$C_\nu \approx 100$$

Equation (13) incorporates the ability to accurately define how the effective turbulent transport varies with the effective Reynolds number to obtain better results for low-Re number and near wall flows. In the high-Re number, the effective viscosity is defined by the following equation:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (14)$$

the constant $C_\mu = 0.0845$ is derived using the RNG theory and it is interesting to note that this value is very close to the empirically value of 0.09 used in standard model.

The **Realizable k- ε** (REK) was used for simulations in this work. The term realizable means that the model satisfies certain mathematical constraints on the normal stresses, more consistent with the physics of turbulent flows [3].

The modelled transport equation for k is the same as for k - ε standard model and the equation for ε is:

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_i}(\rho\varepsilon U_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \rho S \varepsilon - C_2 \rho \frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} \quad (15)$$

where

$$C_1 = \max\left(0.43, \frac{\eta}{\eta + 5}\right) \quad (16)$$

The form of the ε equation is different from those in the standard and RNG models. Another difference between the realizable model and the standard and RNG models is that C_μ is no longer constant (the viscosity is computed in equation 10.). The coefficient of dynamic viscosity is calculated from:

$$C_\mu = \frac{1}{A_0 + A_S \frac{kU^*}{\varepsilon}} \quad (17)$$

where

$$U^* = \sqrt{S_{ij}S_{ij} + \overline{\Omega}_{ij}\overline{\Omega}_{ij}} \quad (18)$$

and

$$\overline{\Omega}_{ij} = \Omega_{ij} - 2\varepsilon_{ijk}\omega_k$$

$$\Omega_{ij} = \overline{\overline{\Omega}_{ij}} - \varepsilon_{ijk}\omega_k$$

In the above equation $\overline{\overline{\Omega}_{ij}}$ is the mean rate-of-rotation tensor viewed in a rotating reference frame with the angular velocity ω_k . The model constants A_0 and A_S are defined as:

$$A_0 = 4.04, \quad A_S = \sqrt{6} \cos \phi$$

where

$$\phi = \frac{1}{3} \cos^{-1} \left(\sqrt{6} \frac{S_{ij}S_{jk}S_{ki}}{\overline{S}^3} \right), \quad \overline{S} = \sqrt{S_{ij}S_{ij}}, \quad S_{ij} = \frac{1}{2} \left(\frac{\partial U_j}{\partial x_i} + \frac{\partial U_i}{\partial x_j} \right)$$

It has been seen that C_μ is a function of the mean strain and rotation rates, the angular velocity of the rotating system, and the turbulent fields. The standard value of $C_\mu = 0.09$ is calculated from equation 10 for inertial sub layer in the equilibrium boundary layer.

The model C_2 , σ_k and σ_ε constants have been determined to ensure that the model performs well for certain canonical flows and are defined as follows:

$$C_2 = 1.9, \quad \sigma_k = 1.0, \quad \sigma_\varepsilon = 1.2$$

Conclusion

This part of research was very important and useful for the next steps. Based on the CFD method the optimal sizes and ordering head pipes for heat exchange from boiler to double stroke engine were found when it is easier to design and measure the real head pipes exchanger.

Acknowledgement

Proposal of solution of microgenerational unit for biomass combustion is solved by APVV program in cooperation with GoldenSun Company.

References:

- [1] Mancini M., Analysis of mild combustion of Natural Gas with preheated air. PhD thesis, Clausthal University of Technology, Faculty of Energy and Management, 2007.
- [2] Yakhot, V., Orszag, S.A., Thangam, S., Gatski, T.B., Speziale, C.G., 1992. Development of turbulence models for shear flows by a double expansion technique. *Physics of Fluids A* 4, 1510_1520
- [3] T.H. Shih, W.W. Liou, A. Shabbir, Z. Yang and J. Zhu, A New $k-\varepsilon$ Eddy Viscosity Model For High Reynolds Number Turbulent Flows, *Comput Fluids* 24 (1995), pp. 227–238.