FACTA UNIVERSITATIS
Series: **Working and Living Environmental Protection** Vol. 17, Nº 3, 2020, pp. 163 - 174
https://doi.org/10.22190/FUWLEP2003163I

CFD ANALYSIS OF TEMPERATURE FIELD IN PELLET STOVE AS A GENERATOR OF AN ABSORPTION HEAT PUMP

UDC 683.943:[621.313.12:621.575

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Abstract. *The paper presents an initial CFD study on adopting the biomass-pellet usage in a generator of an absorption heat pump by obtaining the temperature field inside the biomass furnace. Contemporary absorption technologies are mostly based on the use of gas and other waste heat as a driving force in the Generator, where the two-component working fluid splits into the refrigerant and the absorbent. There are few or no absorption heat pumps that work directly on biomass - pellets. In the Balkans, biomass - pellets are a frequent and renewable source of thermal energy. The aim of this paper is to initially research the possibility of an absorption generator to work directly on available pellets. Following this idea, a comprehensive overview of contemporary absorption technology is given with a physical and mathematical model of the small pellet stove in FLUENT, which will be modified to adapt the generator. In the beginning, temperature fields are obtained by simulation inside the furnace and on its surfaces. Work showed that the temperature field has enough potential for triggering the absorption process as temperatures in the upper part of the stove are above 400°C at the heating capacity of around 13 kW up to 20 kW. The implemented work and the obtained results could serve as a useful reference for further design and optimization of the generator of AHP for direct Biomass utilization for a middle size system.*

Key words: *absorption heat pump, generator, CFD, biomass, pellets*

1. INTRODUCTION

Energy and environmental issues are inevitable parts and cornerstones of modern society. The world population is increasing, and people are striving for higher standards of living. The amount of energy needed to maintain our society increases inevitably. At the same time, the availability of resources, especially non-renewables (liquid and solid fuels), is decreasing [1].

Received October 4, 2020 / Accepted December 6, 2020
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Therefore, there is a general agreement that, in order to avoid an energy crisis and ensure the existential security of civilization, mankind will have to do more, to reorient its consumption to renewable energy and exploit energy-efficient systems. absorption heat pumps are foreseen as a vital part of such systems in the future. They can be used for various purposes in heating, cooling, increasing the heat output over the input of the source, upgrading heat quality, to deliver higher temperature compared to the inlet stream, etc. [2]. Global trends in absorption heat pump - (hereinafter AHP) technologies are briefly discussed in the second chapter. In its essence, an AHP system is based on a cycle using the mixture of fluids (e.g. NH_3-H_20 ammonia-water, LiBr-H_2O lithium bromide - water, etc.) as the working medium, an external source of heat (generator - combustion chamber, usually gas-propane-butane), and a pump instead of a compressor, because it consumes 5-10 times less electricity than the standard compressor unit. The generator of AHP presents a part of the system that performs the splitting of the mixture components and converts fluid phases, i.e. the cooling fluid (e.g. ammonia-vapor) is separated from the absorbent (e.g. water) using an external heat source (usually gas). Since AHP mainly uses gaseous fuel, the aim of this paper is to simulate the use of biomass-pellet in a stove for the generator of AHP as a more accessible and frequent form of energy in Serbia. Currently, on the market, there are no solutions that possess a generator that works directly on pellet or another form of biomass instead of gas. Toward this goal, a small-scale pellet stove is used for analysis at the beginning. The results of the analysis should provide a direction for possible utilization of a pellet stove as the generator of AHP. The initial computational study was performed on a small-scale pellet stove (up to 30kW of heating power), to obtain a closer insight into the temperature field inside the pellet furnace. Up to now, there are few investigations about the heating of the absorption generator by generated heat from the biomass-pellet [1], and it was done indirectly by working fluid glycol heating the generator. In our work, we will discuss the possibility of placing a generator directly in the furnace of a pellet stove, to achieve better efficiency COP (coefficient of performance) of process. The obtained results will give starting information and a big picture for further attempts at modifying the furnace for the pellet-operated generator of AHP.

1.1. Overview of absorption heat pump technologies

Fig.1 and 2. show a basic scheme of AHP. It consists of indispensable parts enabling the work of an AHP system, like the generator for splitting fluid components, the absorber for mixing components and creating solutions, the condenser, the evaporator, and the pump. It can be seen that diverse external sources of heat may be used to drive the system, such as waste heat or hydrocarbon fuels, solar energy, geothermal, etc. The heat provided from the high-grade temperature source is combined with the heat extracted from the low-grade heat source such as ambient air, ground soil, and water, which eventually leads to an increased quantity of heat to the condenser and absorber from a relevant compression cycle. [3]. A company in Italy has developed a gas-fired AHP, Fig.2 working on NH_3-H_2O, for capacities of up to 30kW of heating, in Fig. 3 are presented components of Italian Robur absorption heat pump. It can produce water with a temperature of up to 60°C for low-grade heating of buildings, with the efficiency of Gas-Utilization G.U.E. up to 1.6. Fig. 4 present the heating capacity and G.U.E. depending on different external air temperatures [4].

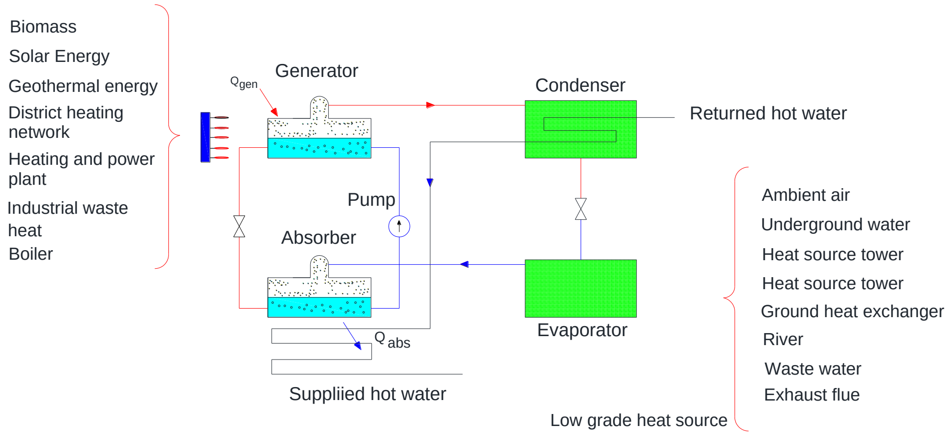


Fig. 1 Basic scheme of AHP

Garimela et al. [5] studied gas-fired generator-absorber heat Exchange (GAX) in the AHP cycle for cooling and heating modes for different outdoor air temperatures. The achieved heating COP at a proportionally low ambient temperature of 5°C was 1.4. For a further decrease in the ambient temperature, the acquired COP can be found in [5].

Fig. 2 Gas-fired AHP manufactured by the Italian company [4]

Kang et al. [6] investigated NH_3-H_2Ogas – fired AHP for wall and floor pipe heating of residential buildings. The needed temperature of the coolant was 65°C. Three operating modes, cooling, space heating and pipe floor-wall heating, were possible in only a single unit. The achieved heating COP was between 1.6-1.8.

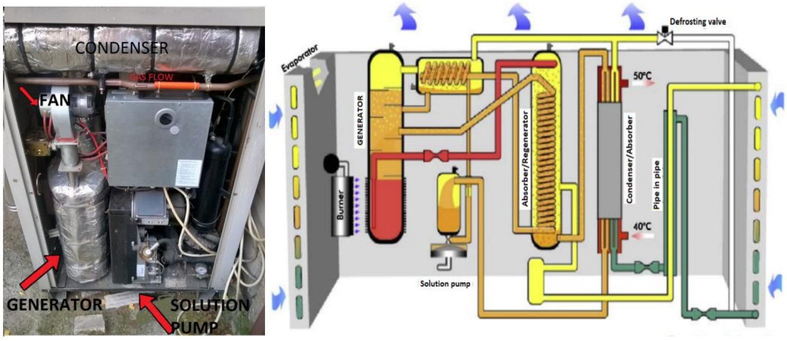
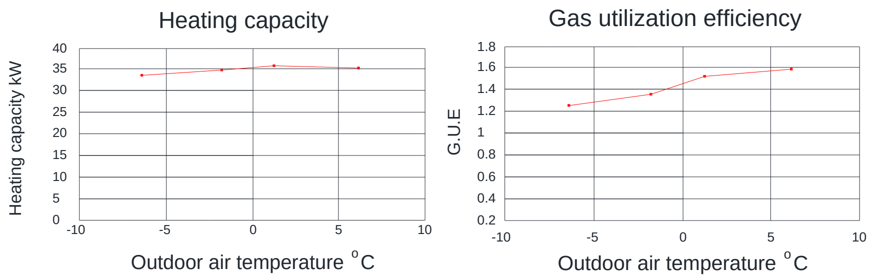


Fig. 3 Basic scheme of the gas-fired AHP manufactured by the Italian company, on site and description [4]

Philips [7] constructed a gas-fired absorption heat pump for middle-size energy systems. The achieved heating COP for 8°C of outdoor air temperature is around 1.8. Six different cycles and working fluids were under examination, and it was concluded that for such an AHP concept and boundary conditions, the most favorable working fluid is NH_3-H_2O with the best techno-economic ratio.



a b

Fig. 4 a. Heating capacity, b. G.U.E. for gas-fired ASAHP for water temperature of 50°C

Li et al. [8] gave the idea for using AHP systems in the district heating system, but in the manner, that hot water from the boiler or district heating is used as a driving source rather than to be delivered directly to consumers as in conventional heating systems. The manufactured and studied system was implemented in different cities and it obtained energy-saving rates of 18-42%. Wu et al. [9] investigated several working pairs of fluids for absorption cycles. The considered fluids were NH_3-H_2O, NH_3-$LiNO_3$; NH_3-NaSCN for both single-stage and double-stage gas-fired ASAHP. Results showed that NH_3-$LiNO_3$ needed a lower temperature in the generator and could work at a lower temperature in the evaporator with a higher temperature in the condenser [10]. The double-stage ASAHP had some advantages over the single-stage by using a low-temperature driving source, while operating in cold regions, and for delivering higher temperature of hot water. Garrabrant et al. [11] tested and improved two absorption cycles, single-effect, and generator-absorber heat exchanger cycles, for residential gas-fired heat pump applications. The experimental study of the single-stage cycle was done with micro-channel and conventional heat exchangers, and a conventional heat exchanger was tested for a generator-absorber heat exchanger cycle. The achieved efficiencies (COP) ranged from 1.8 to 1.38 for different temperatures of return water and the constant ambient temperature of 20°C. Heating capacity was between 2.93 kW for the recovery start and 2.2kW when water was heated to 57°C.

For industrial purposes, several authors investigated the use of AHP technology and its suitability for drying, evaporation, distillation, and heat recovery. Abrahamson et al. [12] tested a single-stage AHP (solution H_2O-NaOH, H_2O-LiBr) for drying. The principle was to extract latent heat from exhaust humid air of 58˚C and produce a hot stream of 80°C for paper drying. The achieved COP was between 1.66-1.8. Gidner et al. [13] examined the use of waste heat at 69 and 95°C for producing a hot stream of 125°Cfor for the evaporation process in the pulp and paper industry. The used solution was H_2O-NaOH and the achieved capacity of the system was 10.8 MW. Rivera et al. [14, 15] worked with an absorption heat Transformer on water purification with condensing heat recovery, for a capacity of 0.7kW and achieved COP in the range 0.23 - 0.33. The tested solution was H_2O-LiBr. Costa et al. [16] examined the use of waste heat in the Kraft pulping process to produce low-pressure steam. COP for absorption heat Transformer cycle was 0.35 and for absorption heat pump was 1.3. The tested solution was H_2O-LiBr.

Further improvement of middle scale AHP can be done by replacing diaphragm solution pump with a more efficient one as Wang Z.X. et al. [17] conducted in their research achieving improvements of net-work, thermal efficiency and exergy efficiency respectively to 4.87 %, 3.62 %, 10.6 %.

2. Physical and Mathematical Model of a pellet stove firebox for an Initial Insight into the Process of Adapting it as the Generator

From the previous overview of contemporary AHP technology, it can be seen that most driving heat sources for the AHP generator come from gas or waste heat, but very few or none from direct utilization of biomass – pellets in the generator. As the initial part of the investigation, the small-scale pellet stove and its firebox were numerically simulated on pellet combustion to acquire the temperature and pressure field inside the firebox, as the starting part in modifying it to a pellet-fired generator, Fig. 5. The stove was manufactured by the company “Megal a.d” Bujanovac, and the maximal tested heating capacity was up to 35 kW.

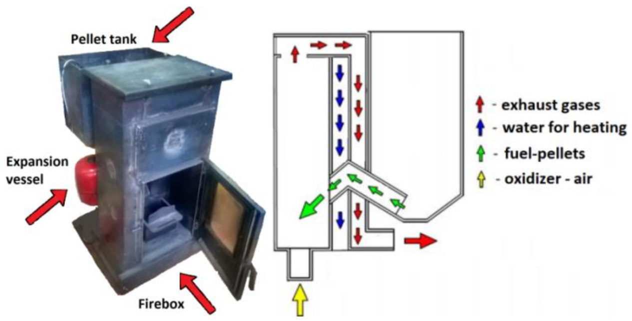


Fig. 5 Basic scheme of the pellet stove with the working cross-section

Firstly, the furnace will be tested and coupled with the Italian gas fired AHT as in [1], mentioned in the previous chapters, Fig. 6. Propylene-glycol will flow through the water side of the furnace, and further on after it was heated through the spiral heat exchanger inside the generator where the solution of NH_3-H_2O was found, for the absorption cycle. The shape and dimensions of the firebox are shown in Fig. 7. The evaporation temperature of the solution in the Italian GAHP generator is around 130°C at 30 bars for gas utilization efficiency G.U.E. of 1.3 for outlet water temperature of 50°C, and refrigerant-solution (NH_3-H_2O) mass ratio of 0.45.

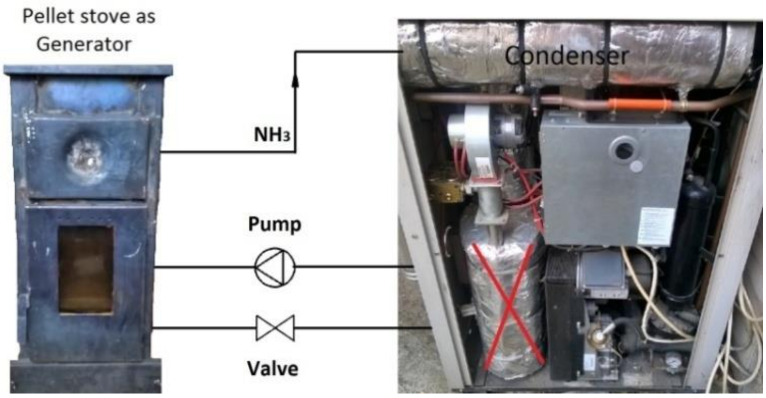


Fig. 6 Connection of pellet stove-generator to AHP

2.1. Physical and Mathematical model of the firebox for pellet combustion

Before the mathematical representation of the case, several physical assumptions had to be made:

- the model is three-dimensional and stationary, and fixed bed combustion is assumed inside the furnace firebox;
- a realistic case of the fuel feed into the combustion zone such as shown in Fig. 4, is replaced by introducing pieces of solid fuel (wood pellets from biomass) in the form of a dispersed phase within the domain of the mathematical model (using the "Discrete Phase" models within the FLUENT software) [1];
- regarding the adopted concept of mass flow in the software FLUENT, the mass flow of the dispersed phase is defined based on the fuel consumption for the tested CFD regime for 8, 12 up to 25 kW;
- It is assumed that a piece of fuel (dispersed phase) is spherical, with uniform density in all directions, wherein the size of the piece to be inserted is defined on the actual dimensions of the pellet (length and diameter) and the grindability of the model is taken into account via the Rosin-Rammler's distribution;
- resistances to heat and mass transfer of a piece of pellets from biomass, which are entered into firebox are ignored, because of the relatively small size pieces and small Biot's number lower then 1 (one), while the temperature of the entered items currently achieves fuel temperature value in the combustion zone [3];
- the interplay between the pieces of fuel (dispersed phase) is ignored, but they react only in the continuous phase;

- a two-stage combustion process is assumed that is described in the combined model of finite reaction rate / Eddy dissipation model in the software FLUENT [18];
- the volatiles for test fuel used in the model are represented in the form of fictitious formula based on the elemental composition and technical analysis.

Transport equations solved by FLUENT are shown below as the RANS model in a Cartesian coordinate system.

- The mass conservation equation:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{u}) = S_m \tag{1}$$

As it is assumed that $\frac{\partial \rho}{\partial t}$ is equal to zero we get the final form of equation 1:

$$\nabla(\rho \vec{u}) = S_m \tag{2}$$

ρ is the density of the fluid stream, $\vec{u}$ is the velocity vector, $\vec{u} = (u_x \vec{i} + u_y \vec{j} + u_z \vec{k})$, t is time and the S_m is the mass source term inside the firebox.

- Momentum conservation equation:

$$\frac{\partial(\rho \vec{u})}{\partial t} + \nabla(\rho \vec{u}\vec{u}) = -\nabla p + \nabla \cdot (\overline{\overline{\tau}}) + \rho \vec{g} + \vec{F} \tag{3}$$

where p is the static pressure, $\overline{\overline{\tau}}$ – the Reynolds turbulent stress tensor, and $\rho \vec{g}$, $\vec{F}$ are the gravitational body force and the external body forces and can be neglected, $\partial(\rho \vec{u}) / \partial t$ is equal zero. According to these assumptions the momentum equation takes the following form:

$$\nabla(\rho \vec{u}\vec{u}) = -\nabla p + \nabla \cdot (\overline{\overline{\tau}}) \tag{4}$$

while $\overline{\overline{\tau}} = \mu_{eff} (\nabla \vec{u} + \nabla \vec{u}^{T})$ and $\mu_{eff} = \mu_k + \mu_t$ [Pas] is the effective viscosity as the sum of μ_k molecular and μ_t turbulent viscosity. Molecular viscosity depends on temperature while turbulent on k-ε turbulent parameters.

- Energy conservation equation:

$$\frac{\partial(\rho H)}{\partial t} + \nabla(\rho \vec{u} H) = \nabla\left(\frac{\lambda_t}{c_p} \nabla H\right) + S_h \tag{5}$$

Under the assumption that Luis no. $Le = \lambda / \rho c_p Dm = 1$ where λ is conduction coefficienct, λ_t turbulent conduction coefficient, Dm - mass diffusion rate, cp – specific heat capacity under constant pressure, S_h – source term of heat production, and that the $\partial(\rho H) / \partial t$ is equal to zero and the total enthalpy is calculated by the equation:

$$H = \sum_j H_j Y_j \tag{6}$$

Where Y_j is the mass of the particle j, and its enthalpy is defined as:

$$H_j = \int_{T_{ref,j}}^{T} c_{p,j} dT + h_j^0 (T_{ref,j}) \tag{7}$$

here $h_j^0 (T_{ref,j})$ the enthalpy of formation of species *j* at referent absolute temperature $T_{ref,j}$.

- Conservation equation of chemical species:

$$\frac{\partial(\rho Y_j)}{\partial t} + \nabla \cdot (\rho \vec{u} Y_j) = -\nabla \bar{J}_i + R_i + S_i \tag{8}$$

Here R_i is the formation speed of species *i* during the chemical reaction and S_i is the speed of species formation because of the disperse phase. Mass transport for the turbulent model of combustion is represented as:

$$\vec{J}_i = -\left(\rho D_{i,m} + \frac{\mu_t}{Sc_t}\right)\nabla Y_i - D_{T,j}\frac{\nabla T}{T} \tag{9}$$

$Sc_t = \mu_t / \rho D_t$ is the Schmidt's number for turbulent flow, and D_t is the coefficient of turbulent diffusion. Other important equations for sub-models of the dispersed phase for solid parts of biomass during combustion and chemical kinetics can be found in [16].

2.2. Computational domain and Boundary conditions

The computational domain of this case is the firebox of the furnace, as combustion takes place inside the domain and the temperature field is important to obtain. For this geometry, three types of surfaces are identified as boundary conditions, one air inlet, pressure outlet and opaque wall shown below in Fig. 7. Fuel mass inlet is 3kg/h equivalent of around 12 kW simulated stove capacity up to 20 kW. On the top of the domain, the outlet surface is defined, as flue gases go through it. Computation of the case and graphical values were obtained on a Dell Precision T7810 Tower Workstation - Intel Xeon E5-2630 v3 2.40 GHz 462-9274, 32 GB RAM, 24 cores. The finite volume method is applied to the case and the discretization of the domain, as an integral part of the software package FLUENT. The number of tetrahedral finite volumes inside the domain is 304606, Fig. 8.

Discretization of partial differential equations is done by the 'Second Order-Upwind' scheme, and the SIMPLE algorithm was used for coupling pressure and velocity field. A convergence criterion of the iterative process was that the last two values do not differ more than 10^{-5}. Convergence was achieved after 10000 to 12000 iterations.

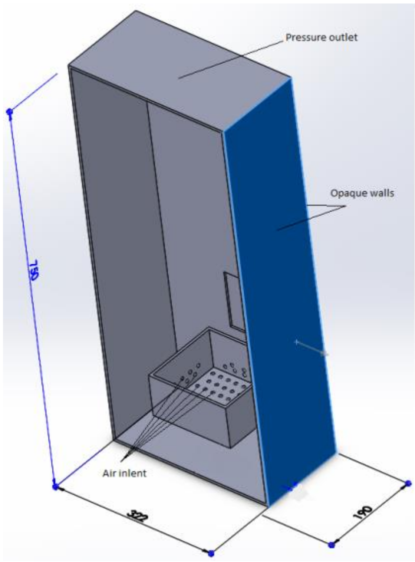


Fig. 7 Dimensions of computational domain in [mm]

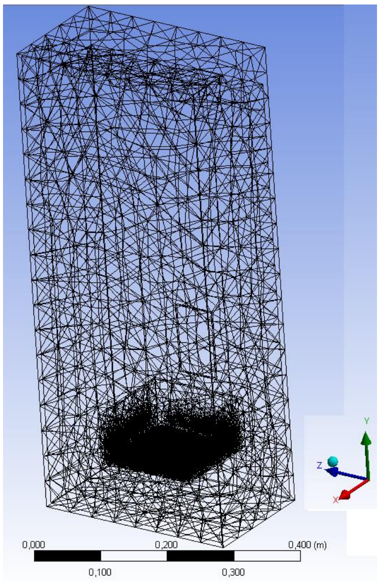


Fig 8 Tetrahedral mesh of domain

3. RESULTS

After the converged solution achieved results in form of a temperature field are shown below Fig. 9 and 10. It can be seen that in the upper part of the stove we have temperatures of around 900 – 1200 ºC, this position could be appropriate for placing the spiral generator of absorption heat pump for heat gains and triggering the absorption process, as the products of propane-butane combustion are in that range. The COP of the pellet stove was 89% acquired by simulation.

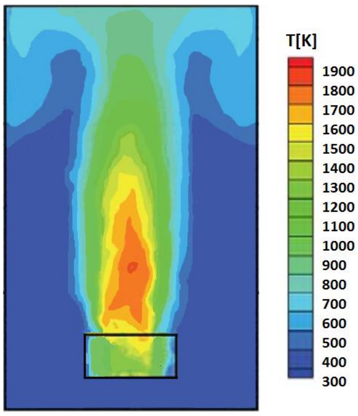


Fig. 9 Temperature distribution at front - midplane inside the pellet stove of pellet combustion at around 13 kW heating capacity

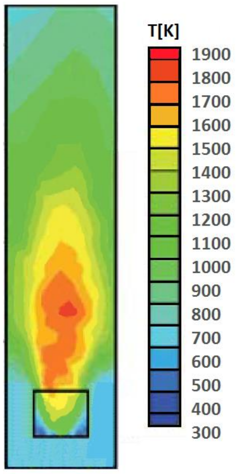


Fig. 10 Temperature distribution at the side - midplane inside the pellet stove of pellet combustion at around 13 kW heating capacity

Considering that for different conducted CFD test regimes - pellet mass input, and different heat loads of the furnace, the mean temperature of the furnace, averaged by volume $T_{st.ins.}$ [K] changes. For that reason, it was eager to define the functional approximate dependence between the heat load of the furnace, and the average volume temperature of the furnace. This dependence is shown in Fig. 11. Simulated conditions were at pellet mass inlet 1.6 kg/h, 2.9 kg/h and 4.3 kg/h, which lead to acquired heat loads and temperatures.

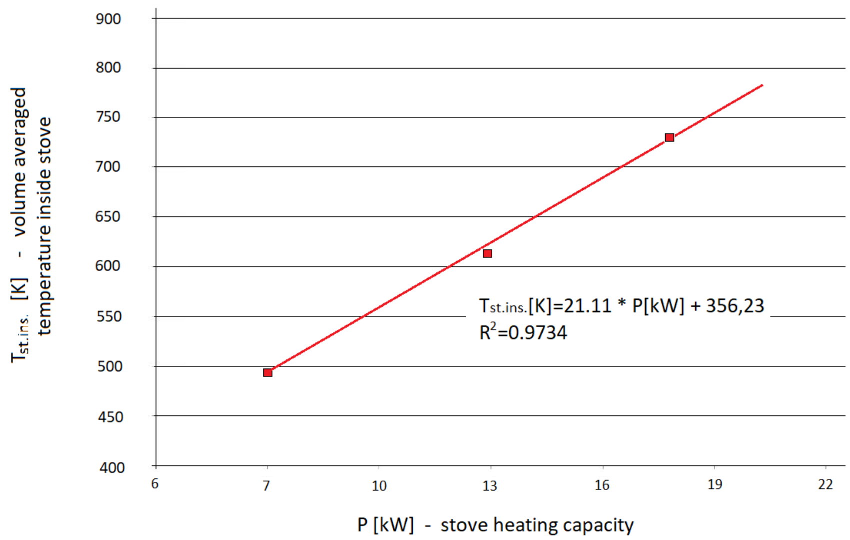


Fig. 11 The functional dependence of volume averaged stove temperature Tst.ins. [K] and the heat load of the stove P[kW].

4. Conclusion

As part of this paper, the simulation of pellet combustion was performed inside the stove, and it gave us the distribution of temperature inside the domain, as seen in Figures 9,10, and 11. This is important for predicting the most suitable locations for placing the generator of AHP at the appropriate spot inside the stove. We know that the adiabatic flame temperature of Gas – Propane Butane is around 1900 °C. As we can see in Fig. 9, 10, 11, this is the current range of temperatures, and it is very plausible to go further with the investigation. The working temperatures of ammonia/water solution in the generator are in the range from 110 to 160 °C for 30-35 bar pressure range, when ammonia evaporates from the solution; there are possibilities to create various shapes and sizes of the generator, which can be placed in upper positions in the stove, depending on temperature and heat flux. From Fig. 11 it is seeable that the average volume stove temperature for needed heat loads is above 400 °C which is enough for triggering the absorption process inside AHP. With these results, further research will proceed with manufacturing an areal generator and connecting it to the stove and the ammonia/water AHP heat pump for real performance testing.

Acknowledgement: *The paper is a part of the research done within the projects: III 42006-Research and development of energy and environmentally highly effective polygeneration systems based on renewable energy resources. Project is financed by the Ministry of Education, Science and Technological Development of the Republic of Serbia.*

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NUMERIČKA CFD ANALIZA TEMPERATURNOG POLJA U KOTLU NA PELET RADI PRIMENE KAO GENERATORA APSORPCIONE TOPLOTNE PUMPE

U radu je predstavljeno incijalno CFD ispitivanje malog kotla na biomasu-pelet kao mogućnost primene za generator apsorpcione toplotne pumpe, analizom dobijenog temperaturnog polja unutar kotla. Savremene apsorpcione tehnologije uglavnom se zasnivaju na upotrebi gasa i drugih otpadnih toplota kao pokretačke energije generatora, gde se dvokomponentna radna smeša deli na rashladni fluid i apsorbent. Postoji malo ili nimalo apsorpcionih toplotnih pumpi koje rade direktno na biomasu - pelete. Na Balkanu je pelet od biomase čest i može smatrati da je obnovljiv izvor toplotne energije. U skladu sa temom, u radu je dat sveobuhvatan pregled savremene literature iz apsorpcione tehnologije sa fizičkim i matematičkim modelom kotla u FLUENT-u, koji će biti modifikovan kako bi se prilagodio generatoru. U početku se temperaturna polja dobijaju simulacijom unutar peći i na njenim površinama. Radovi su pokazali da temperaturno polje ima dovoljno potencijala za pokretanje procesa apsorpcije, jer su temperature u gornjem delu peći iznad 400 ° C pri grejnom kapacitetu od oko 13 kW do 20 kW. Izvedeni rad i dobijeni rezultati mogli bi da posluže kao korisna referenca za dalje projektovanje i optimizaciju generatora ATP za direktno korišćenje biomase za termotehničke sisteme srednjih kapaciteta.

Ključne reči: *apsorpciona toplotna pumpa, generator, CFD, biomasa, peleti*