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Heat Storage Performance of a Honeycomb Ceramic Monolith

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Abstract: Honeycomb ceramic is the key component of the regenerative system. The three-dimensional numerical model has been established for thermal process in honeycomb regenerator. The numerical simulation was performed using FLUENT, a commercial computational fluid dynamics (CFD) code, to compare simulation results to the test data. The regenerative process of a honeycomb ceramic regenerator was simulated under different conditions. The results under different flow rates, different flowing time, different materials and different wall thickness were investigated. The work in this paper provides a theory basis and guide to the exploitation and appliance of HTAC system and the results of the numerical calculation can be used as the foundation of engineering design. The results may be utilized for design of porous media reactors and process optimization.

Keywords: Ceramic honeycomb, heat transfer characteristics, numerical simulation, heat storage performance.

1. INTRODUCTION

High temperature air combustion (HiTAC) is characterized by reactants of high-temperature and low-oxygen concentration. Many researchers have recently realized significant energy savings, NOx emissions reduction and heat transfer uniformity in industrial heating furnaces resulting from such novel combustion 1. A regenerative burner is very effective for reducing fuel consumption in an industrial furnace [1-4]. A typical regenerative burner consists of at least two burners and two regenerators. While one of the burners is fired using cold air fed to the base of its regenerator, exhaust gas is drawn through the other burner and down into its associated regenerator to preheat the packing, then discharged to the atmosphere. When the regenerator being heated is sufficiently charged, the reversing system operates, cold air flows to the newly heated regenerator and is preheated, the previously cooled regenerator being reheated by the exhaust gas from the other burner firing. The regenerative burner shows a high heat recovery efficiency of 80-95% and the combustion air is preheated to over 1000°C.

The classical books [5-7] provide analytical methods and mathematical models for calculating temperature profiles, effectiveness and other thermal performance parameters of regenerators. H. Hausen and Rummel [5] presented the first approximate solution to determine the spatial temperature distribution in solid material at a certain time and at a certain cross-section in a heat regenerator. They assumed time independent fluid flow temperature, but the assumption does not hold well in practice. Rummel introduced some constants to the heat transfer coefficient to get away from this assumption. However, the constants should be determined by experiment. An analytical model limited to a small wall thickness was developed by Klein *et al*. [8]. End temperatures of this solution agree approximately with experiments for fast switching limiting case. But there were little comparison about temperatures along the regenerator.

Generally, the use of the dynamically operated regenerative burners was avoided and consequently measurements were made easier. Since a complete understanding of the heat transfer in industrial furnaces is always required in the design and optimization of dynamic heating processes. Our ambition in this work was to carry out studies on the start-up process in a furnace equipped with reverse regenerative burner systems.

In this paper a three-dimensional numerical simulation model for a ceramic honeycomb regenerator used for a HiTAC burner was developed. This kind of regenerator is either a fixed bed with randomly packed ceramic balls or having honeycomb structure with identical cells. In this work, it is based on finite volume methods and the commercial Computational Fluid Dynamics software Fluent is used to simulate the dynamic process. Moreover, the model considers the temperature dependent thermal properties. By comparison, one-dimensional numerical simulation is not capable of considering all the above effects. The aim of the model is to obtain the temperature distribution of solid matrix and gases along the regenerator, the pressure difference across the regenerator at different operating conditions. It was verified with experiments that were performed on the same regenerator.

2. SIMULATION MODEL

Fig. (**1**) shows the dimensions of one heat regenerator used in a HiTAC regenerative burner. Regenerator dimension is $150W \times 150H \times 300L$. All flue gases generated by combustion are sucked again by the burners and pass

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through the regenerators. The switching time can be set to a value between 15 and 60 s.

2.1. Geometry Parameters of Honeycomb Cell

A three-dimensional simulation model was developed to find out the temperature distribution of the solid storing material and flowing gases and other thermal and flow parameters for this regenerator and compare results with experiments. Because of geometric symmetry of the honeycomb structure, mathematical analysis was made on one honeycomb cell, or matrix, that formed a small part of the regenerator cross-section along the flow path (Fig. **1**).

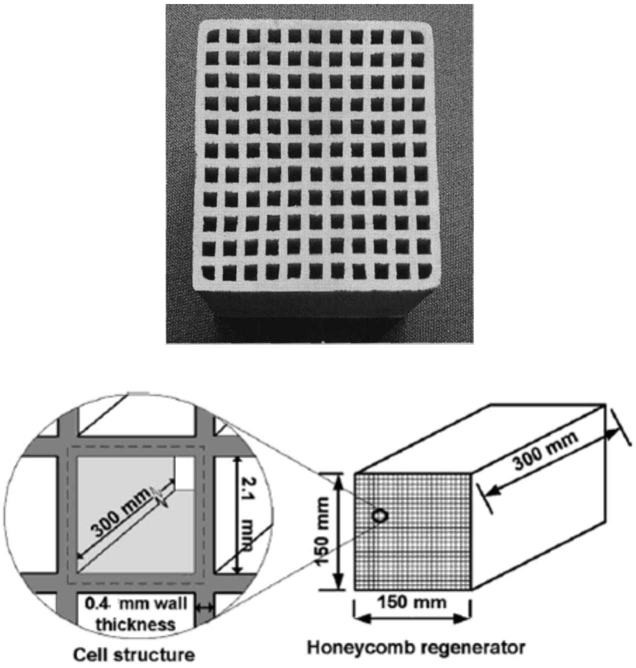

Fig. (1). Honeycomb regenerator and dimension of the ceramic material and flow path.

In our test lab, there are 10 types of honeycomb cells, the side of which ranges from 2.1 mm to 5.0 mm. For such simple geometry with very small side, the flow is always laminar at nominal operating conditions, and flue gases emissivity is low inside regenerator cells due to very short beam length, so the heat transfer by radiation is not taken into account in this work.

2.2. Computational Zone

Fig. (**2**) shows the three-dimensional zone in the simulation. It includes the solid material and flow path (the gas phase). The honeycomb cell is square--axisymmetric, the velocity, temperature and pressure distribution are all symmetric. So half the wall thickness, 1/4 flow path and the whole length of the cell is taken as the computational zone.

2.3. Physical Property Parameters of Gas and Honeycomb

The flue-gas and the preheated air flow through the honeycomb alternately. As the difference of the physical properties between the gas and the air has small influences on the heat transfer performance of the honeycomb regenerator, the physical properties of the air is used instead of the gas in the numerical simulation for convenience. In fact the specific heat of flue gas is a little larger than that of the air, so the heat transfer between the flue gas and the honeycomb is a little more than the heat transfer between the honeycomb and the air if both of the switching times are the same. The temperatures of the air and the gas flowing through the honeycomb vary greatly, so their physical properties vary greatly correspondingly. The physical properties are assumed to vary linearly with the temperature. And the solid material is assumed to be Mullite, the thermal conductivity of which changes linearly with temperature. The physical property parameters of air and Mullite are listed in Tables **1** and **2**, respectively.

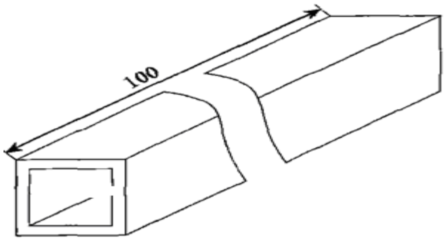

Fig. (2). Three-dimensional computational zone.

Table 1. Physical property parameters of air [19].

Temperature (T) 1473 K

Density (ρ) 0.239 kg / m^3

Isobaric specific heat (Cp) 1.210 $KJ / (kg \cdot K)$

Dynamic viscosity (μ) 53.5e- 6 $kg / (m \cdot s)$

Thermal conductivity (λ) 9.15e-2 $W (m \cdot K)$

Inlet velocity (V) 7 $m \cdot s$

Prandt number (Pr) 0.724

2.4. Governing Equations

In the honeycomb cell, gas attributes to the three-dimensional unsteady laminar flow. There are three different forms of heat transfer process, that is, the thermal conductivity of gas or air in itself; convection heat transfer between honeycomb and flue or air; thermal conductivity, thermal storage and thermal rejection internal honeycomb. In order to simplify the calculation, we made the following assumptions:

The differences between the flue gas and air is ignored and the physical parameters of air are used instead of that of flue gas; radiation heat transfer in the channel is ignored; Heat loss from the regenerator to the environment is ignored; The surface area and body mass distribution of the cell is uniform; The Cell surface is smooth.

A typical set of equations for regenerative problem was used: equation of mass conservation, Navier-Stokes, thermal conductivity equation for gas, thermal conductivity equation for porous media and Ideal gas state equation.

Table 2. Physical property parameters of Mullite.

Temperature K	**Density** kg/m^3	**Thermal Conductivity** $W(m \cdot K)$	**Isobaric Specific Heat** $J/(kg \cdot K)$
300	2500	2.14	800
1500	2500	4.37	800

Mass conservation equation:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \tag{1}$$

Navier-Stokes equations:

$$\frac{\partial(\rho u)}{\partial t}+div(\rho u\overline{U})=div(\eta\, gradu)-\frac{\partial p}{\partial x} \tag{2}$$

$$\frac{\partial(\rho v)}{\partial t}+div(\rho v\overline{U})=div(\eta\, gradv)-\frac{\partial p}{\partial y} \tag{3}$$

$$\frac{\partial(\rho w)}{\partial t}+div(\rho w\overline{U})=div(\eta\, gradw)-\frac{\partial p}{\partial z} \tag{4}$$

An energy balance was made in every domain in the channel, the flowing gases and solid domains.

The problem is transient and the general energy balance equation is:

$$\frac{\partial(\rho T)}{\partial t}+div(\rho u\overline{U})=div(\eta\, gradu)-\frac{\partial p}{\partial x} \tag{5}$$

where S_T, a sink or source heat, is zero in this case. The expression within brackets is the heat flux vector, which in this case includes the contributions from conduction and convection, respectively. The heat is transported by convection and conduction in the flow domain and only by conduction in solid domain. Therefore, the energy equation of the solid material domain is:

$$\frac{\partial(\rho T_s)}{\partial t}=div(\frac{\lambda}{c_p}gradT_s)$$

And the energy equation of the flowing gases domain is:

$$\frac{\partial(\rho T_g)}{\partial t}+div(\frac{\lambda}{c_p}gradT_g)=div(\frac{\lambda}{c_p}gradT_g)$$

Energy conservation Equation:

$$\frac{\partial(\rho T)}{\partial t}+div(\rho UT)=div(\frac{\lambda}{c_p}gradT)$$

Ideal gas state equation:

$$p=\rho R_g T \tag{6}$$

2.5. Boundary Conditions

Since the symmetry of the flow and heat transfer in the three-dimensional flow zone of the honeycomb, the four exterior sides along the direction of the gas flow are defined as symmetric boundary as shown in Fig. (**2**). The solid surface of both ends of honeycomb is defined as adiabatic boundary condition. Coupling boundary conditions are applied on the two interior surfaces where the honeycomb is in contact with the flowing gas. Because the heat transfer between the honeycomb and the gas is continuous and the temperature and the heat flux are both affected by the interaction with the wall, no matter if the temperature or the heat flux on the interface are parts of the results. Obviously the two contact surfaces cannot be used as the first, second or third type of pre-specified boundary conditions. Boundary conditions such as these dynamically determined by the thermal transformation process cannot be predetermined, is known as conjugate heat transfer. The coupled boundary conditions at the intersection of the two-phase interface can be listed as the following expressions:

$T_w|_{honeycomb}=T_w|_{gas}$ (Continuous Temperature)

$q_w|_{honeycomb}=q_w|_{gas}$ (Continuous heat flux)

where the subscript--honeycomb and gas mean the simulation regions respectively, w means the interface between the two regions.

The gas mass flow rates of the two heat-exchange stages in one working cycle should be the same. However, the inlet temperature of flue gas is 1473 K. So the inlet velocity of gas is V=7 *m/s*. During the heating period, the inlet end of hot flue gas is defined as the inlet boundary condition, where the speed and the pressure are set to a specific value and 0 Pa, respectively. While the other end of the cold-side is set to pressure outlet. During the cooling period, the inlet end of cold air is defined as velocity inlet and the pressure is 0 Pa, the other end is set to pressure outlet.

As the honeycomb hole is 5 mm, the gas inlet velocity is $7\,m/s$, the dynamic viscosity is 18.46 × 10-6 $Pa \cdot s$, the density is 0.235 kg/m^3, and the Reynolds number is about 60. For such simple geometry with very small diameter, the flow is always laminar since local Re ranges between 90 and 1000 at nominal operating conditions and therefore, the velocity profile and pressure drop across the regenerator can be determined from the solution of Navier-Stokes equation.

Since the local Remax is much smaller than 2000 (the critical Reynolds in a smooth tube), the flow can be considered as laminar. In practice, the actual critical Reynolds number may be reduced because of the wall roughness, and the flow near the wall region tends to be turbulent, which may strengthen the local heat transfer. However, this effect can be corrected by means of experiments, or suitable correction factor is used in the numerical heat transfer 19. The local heat transfer caused by the roughness is ignored and the surface of the cellular surface is assumed to be smooth.

2.6. Switching Condition

Heating or cooling period is half of one cycle time. The ending of heating period is the starting time of cooling period. Similarly, the ending of cooling period is the starting time of heating period. The switching operation is instantaneous and it did not take time. The physical properties and the temperature of honeycomb, the temperature, physical properties and flowing condition of gas do not change before and after the moment.

When the heating period is over, the cooling process makes end that is pressure outlet before the switching time changes to velocity inlet and the heating end that is velocity inlet before the switching time changes to pressure outlet. Correspondingly, when the cooling period is finished, the heating process makes end that is pressure outlet before the switching time changes to velocity inlet, and the cooling end that is velocity inlet before the switching time changes to pressure outlet. A commutation cycle consists of a heating period and a cooling period. The calculation repeats the cycle until the process becomes a stable state.

2.7. Initial Conditions

When the furnace starts working, the temperature of regenerator increases slowly from the ambient temperature, so the initial temperature values of honeycomb and air are set to ambient temperature, $T_0 = 300K$; the initial velocity of gas is set to 0, $u = v = w = 0m/s$; the initial pressure is 0, $P_0 = 0Pa$. The initial temperature of the cellular ceramic body is set to 300 K when the detailed starting characteristics is investigated. In other cases, the initial value of the temperatures are set to 900 K , which is the mean value of the inlet temperature of flue gas--1500 K and the inlet temperature of air. This way the time that the honeycomb regenerator takes to reach to the stable working conditions greatly reduces as well as it saves the computing time.

2.8. Computation Ending Condition

When the difference of the exit temperature of air and flue gas at the end of one cycle between those at the end of the previous cycle is less than 0.2°C, then we consider that the honeycomb regenerator has reached to stable working condition, and the calculation process is completed.

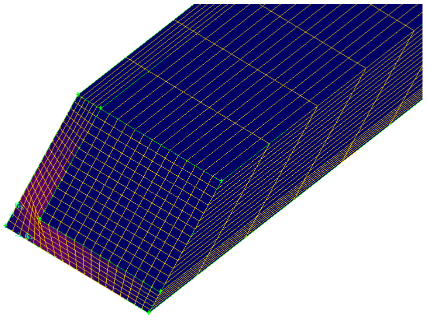

Fig. (3). Meshing of the computational zone.

2.9. Computational Process

The computations are initiated using cold start initial conditions and executed for one complete cycle of operation, i.e., the duration for both the regeneration and the combustion periods. The results are then saved in the computer and the final temperatures are set to be the initial conditions of the next run. The computations were executed repeatedly until the temperature profiles just before the changeover are the same as the previous data, periodic steady state.

The meshing of the computational zone is shown in Fig. (**3**). The governing integral equations for the conservation of mass and momentum, and for energy and other scalars such as turbulence and chemical species were solved through control-volume-based technique. Pressure and velocity field are computed by Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithms. The convergence criterion for energy equation was that the residual error was less than 10^{-6} , whereas for other governing equations it was 10^{-3} . In this study, all residual errors of the degree of freedom were set by 10^{-3} , which is satisfactory for simulation analysis.

Experiments were carried out on honeycomb regenerators that are contained in a methane oxidation reactor. For completeness, a simple description of the reactor is given here. Tests of temperatures were carried out in a reverse flow reactor. The reactor is 600 mm wide, 600 mm high and 300 mm long. The entire configuration of the test apparatus is shown in Fig. (**4**).

In a reverse flow reactor, the feed is periodically switched between the two reactor ends using switching valves. When switching valves 1 and 4 are open, the feed flows to the reactor from left to right (forward flow), indicated by the solid arrows. When switching valves 2 and 3 are open, the feed flows to the reactor from right to left (reverse flow), indicated by the dotted arrows. The total cycle consists of these two operations, and the term switch time denotes the time at which the flow is changed from forward to reverse flow or from reverse to forward flow. The sum of the times for forward and reverse flow is the cycle time.

In this experiment the flue gas is substituted by hot air. Thermal energy carried by hot air can be captured by the solid heat storage medium. Then, with switching the flow direction, the captured thermal energy within the heat storage medium can be used to preheat the air. In the experiments, the ceramic honeycomb monoliths were used as heat storage medium. The monoliths consist of a structure of parallel channels with porous walls. The monolith properties are as same as that used in the model.

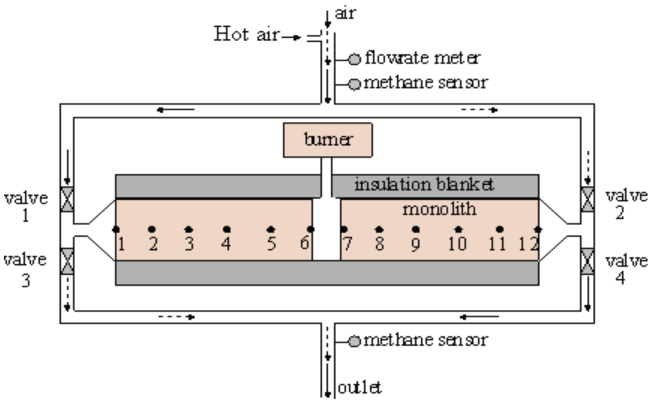

Fig. (4). Configuration of the test apparatus.

To prevent heat loss from the reactor to the surroundings, the reactor was surrounding with a 350 mm thick layer of insulation ceramic fiber blanket. Thermal conductivity of the ceramic fiber blanket is 0.144 $W(m \cdot K)$.

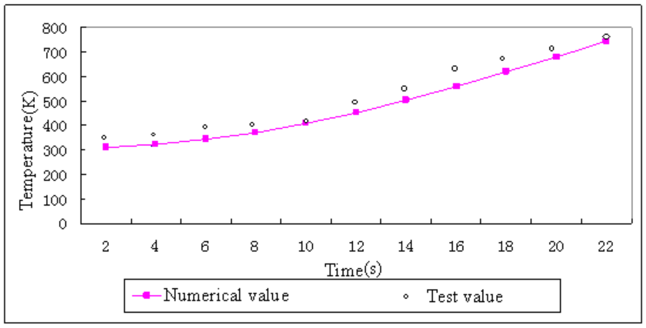

Fig. (5). Experimental and mathematical temperature of flue gas inlet.

Air and the hot air were supplied to the reactor using an air compressor. The flow rate of inlet was measured using a flow rate meter. Thermal profiles from the reactor were obtained using twelve thermocouples (denoted from 1 to 12 in Fig. **1**). All thermocouples were placed along the centerline of the reactors.

The data acquisition system recorded all sensor values. Data were saved as data files for analysis.

The calculated temperatures of flue gas inlet were compared with the ones measured as shown in Fig. (**5**). The temperature of those 12 points was measured by 12 thermocouples. During the measurements there should be no flame in the combustion space. Therefore, the measurements were performed in the 2 second gap between the intervals. It is shown in Fig. (**5**) that the measured temperatures of inlet flue gas are generally more than the simulated values. The reason may be that the radiation of the hot air is ignored in the model. However, the tendency of the temperature is the same as the experiment.

4. RESULTS

For a 3-D numerical simulation model, the structure and meshes were constructed by GAMBIT, and then numerical solution was obtained from the commercially available CFD code FLUENT to investigate the thermal storage behavior of the honeycomb regenerator. The thermal behavior of the channel has been simulated numerically for some cases. In this part the heat transfer performance of ceramic honeycomb will be discussed under different conditions by changing the different parameters. All conditions can be summarized in four cases: the gas inlet velocity, the specific heat of regenerator, the flowing time through the regenerative and the wall thickness of the ceramic body.

4.1. Influence of Different Flowing Time

Figs. (**6-9**) show the temperature and velocity distribution at different time. It is apparent that this is a transient process in which the temperature and the velocity change with time. It can be clearly seen from the color in the figures that the inlet temperature of the ceramic regenerator increases with the increasing time. This is due to the stronger heat transference between the ceramic and the hot gas, and the temperature around the inlet of the ceramic almost equals to the hot gas. In the early period of the heat exchange, the heat transfer between the regenerator and the hot gas occurs mainly in the inlet zone. The high temperature zone gradually goes to lower temperature zone with the carrying out of the heat transfer, and the ceramic regenerator temperature in the axial direction gradually increases. By comparing the temperature and the velocity map, it can be seen that both the distribution maps are very similar. This is because the relationship of the ideal gas $\rho v = nRT$. When the gas temperature decreases, its density decreases, and the gas velocity is bound to increase according to the mass conservative law. That is to say that the gas velocity varies with temperature. Therefore, the velocity distribution and temperature distribution are very similar. It can be seen from the temperature distribution of the outlet section that the outlet temperature is getting more and more higher. It means that the regenerative capacity of regenerator is getting worse over time, because the temperature difference between the gas and the wall is getting even more smaller, and the heat transfer coefficient decreases. Therefore, in order to obtain a higher recovery rate, the appropriate flowing time should be determined. If the flowing time is too

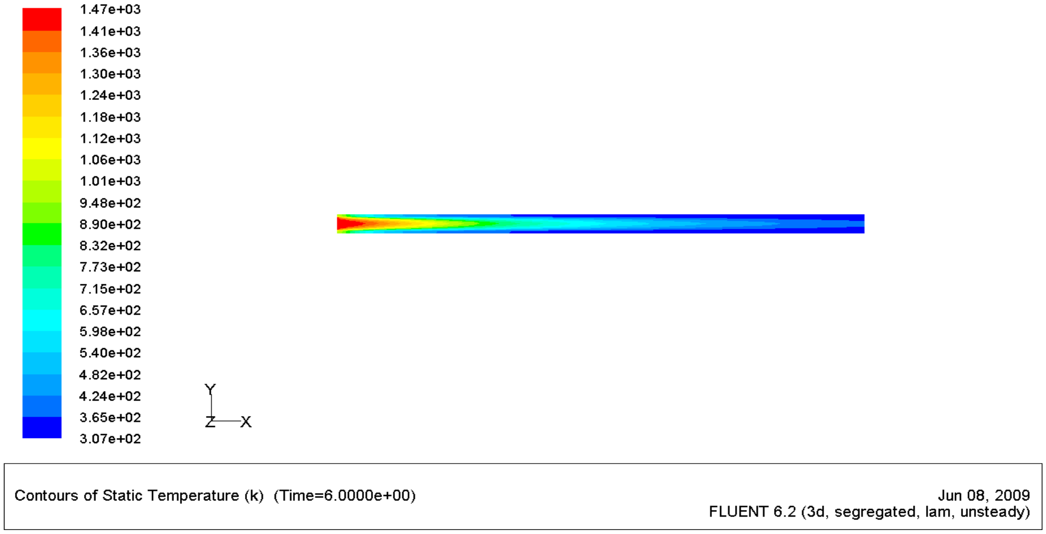

Fig. (6). Temperature distribution in the central section (7 *m/s*, 6 *s*).

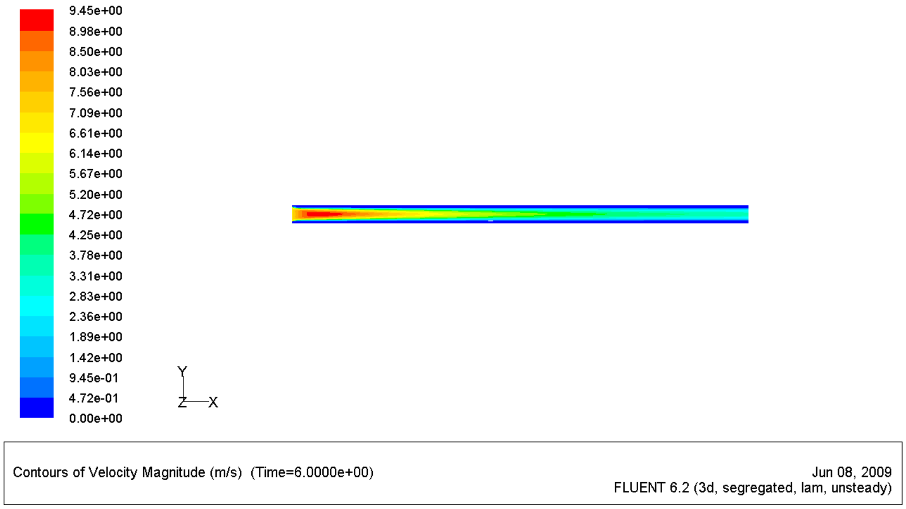

Fig. (7). Velocity distribution in the central section (7 *m/s*, 6 *s*).

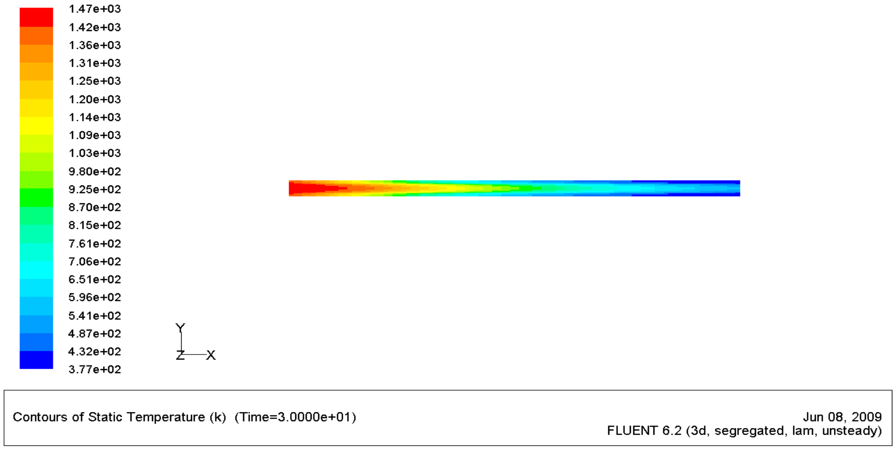

Fig. (8). Temperature distribution in the central section (7 *m/s*, 30 *s*).

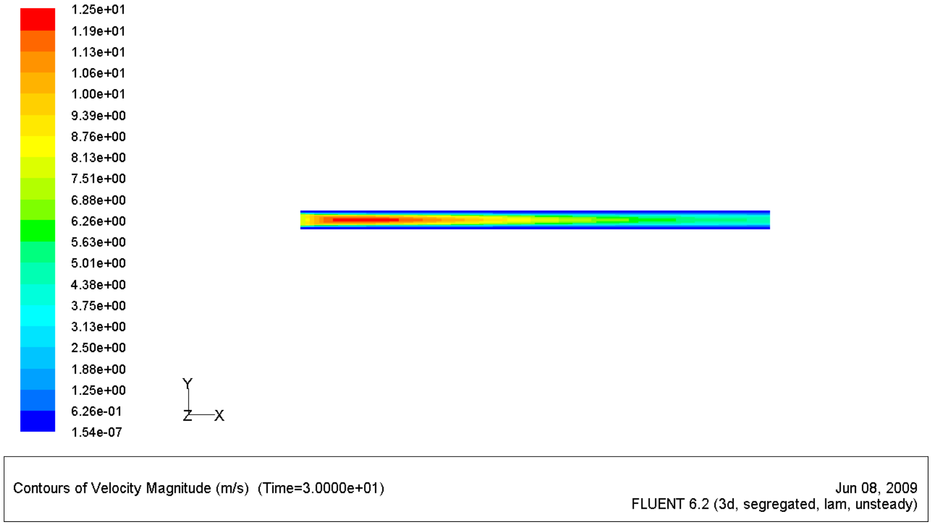

Fig. (9). Velocity distribution in the central section (7 *m/s*, 30 *s*).

long, the heat in the gas cannot be fully absorbed by the wall, the outlet temperature is high; if the flowing time is too short and the regenerator cannot be fully heated, it is difficult to obtain a good warm-up effect, the temperature of regenerator reduces.

4.2. Influence of Inlet Velocity

Figs. (**10**, **11**) show temperature distributions as t=50 s under v=7 *m/s* and v=15*m/s*, respectively. The temperature distribution differs greatly under different velocities. As the inlet velocity of gas increases, the high temperature zone moves gradually to the outlet direction and the high temperature zone becomes bigger. The outlet temperature of the gas is greater when the inlet velocity of gas is greater. As the inlet velocity is greater, the mass flow rate unit time is greater and the heat carrying by the gas is greater, since the time of heat transfer between the gas and the wall is not sufficient. Therefore, in order to prevent the gas outlet temperature from being higher, the influence of the gas flow rate on heat transfer must be fully considered, and the combustion system must be reasonable in practical application.

5. SUMMARY AND CONCLUSION

Honeycomb regenerator is an important part of high temperature air combustion system. This paper focuses on the flow and heat transfer of the flue gas and ceramic. In addition, the heat transfer of honeycomb is considered as well. In this paper, numerical simulation method is used to study the factors that influence the heat transfer characteristics of the gas and the honeycomb in the start-up period. Moreover, the model was validated with the experimental data. The following conclusions were obtained:

(1) Numerical model of honeycomb regenerator was presented and the whole start-up process was simulated by means of CFD commercial software—Fluent.

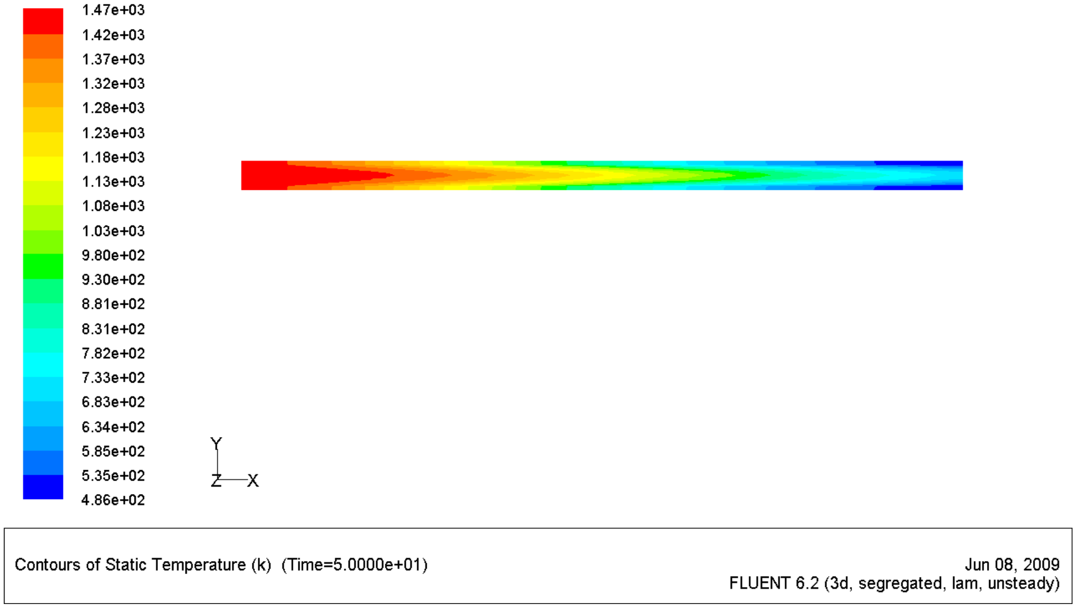

Fig. (10). Temperature distribution in the central section (t=50 *s*, v=7 *m/s*).

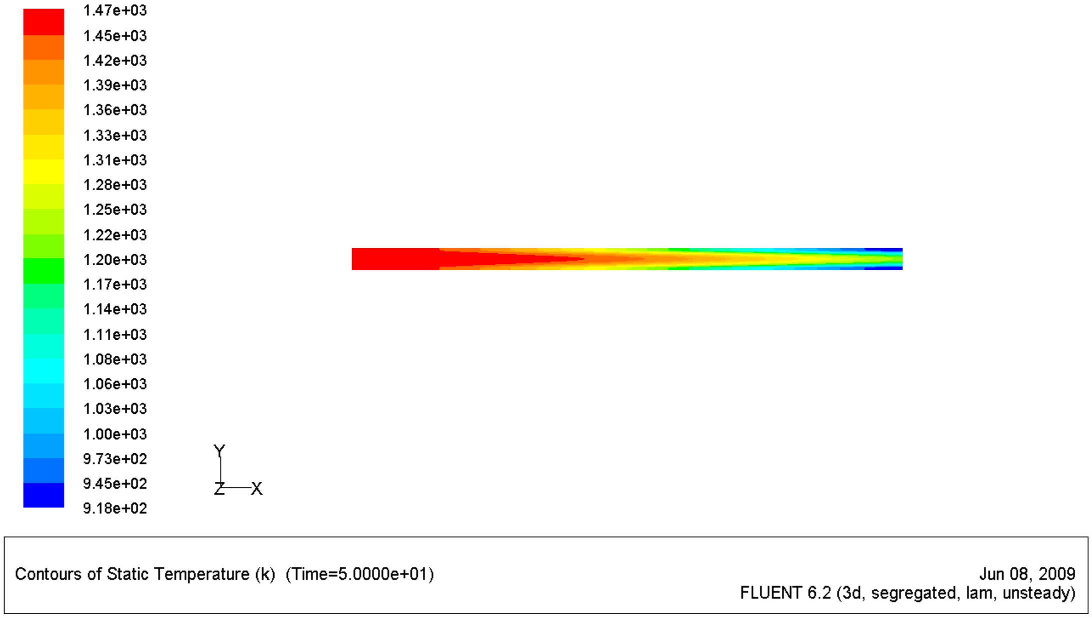

Fig. (11). Temperature distribution in the central section (t=50 *s*, *v*=15 *m/s*).

(2) The flue gas and air inlet temperature curves change with the cycle and the changing discipline of the gas inlet temperatures were obtained. And then the heat transfer between the gas and the honeycomb was analyzed. According to the analysis, the duration time of start-up period can be roughly calculated, which provides reference for further study.

(3) The velocity distributions on the cross-section or the section along the length were obtained, which verified the laminar flow of the channel. This conclusion provides reference to optimizing the flow.

(4) The gas and the honeycomb temperature distributions on the cross-section or the section along the length at the end of the heating period were obtained. The characteristics and the reason of the temperature distribution were analyzed, which provides a reference for further research.

(5) The pressure distribution along the honeycomb were obtained, the reasons for the loss of production and the method to reduce the pressure loss were analyzed.

(6) The transient temperature curves of flue gas inlet and air inlet were obtained, and the temperature fluctuations by the intersection of cold and hot airstream forward were analyzed.

(7) The influence of switching time on the heat transfer efficiency and temperature efficiency was analyzed. And the principle of best switching time was presented, which can be used as the foundation of engineering design.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflict of interest.

ACKNOWLEDGEMENTS

Declared none.

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