# CFD model of regenerative heat exchanger

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**Abstract:** This paper is focused on the computational fluid dynamics (CFD) modeling of regenerative heat exchanger suitable for animal houses. Buildings used for housing of animals in farms with intensive breeding, like poultry or pig houses, are characterized by high generation of heat inside, partly produced by animals, and in the case of small young animals, supplemented also by heating. On the other side these buildings need intensive ventilation which causes big losses of energy by exhausted air. A good way how to reduce heat losses can be the use of technical systems of heat recovery.

There are two principal constructions of heat exchangers for heat recovery. There are either recuperative or regenerative heat exchangers. Industrially produced heat exchangers, commonly used in residential or industrial buildings, can be used in agricultural conditions only with difficulties, mainly because of the high dust concentration, which is extremely high in animal houses.

CFD modeling was used to calculate main parameters of a special heat exchanger, developed for application in animal houses. The construction of regenerative heat exchanger with fixed matrix is based on heat accumulation in material of matrix in the form of massive plates. The program Fluent was used for airflow and heat exchange simulations. Results of simulations were verified by measurement of prototype of real heat exchanger.

**Keywords:** animal houses, energy, parameters, ventilation, Czech Republic

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## 1 Introduction

Buildings used for housing of animals in farms with intensive breeding, like poultry or pig houses, are characterized by high generation of heat inside, mainly produced by animals, and in the case of young animals, supplemented also by heating. On the other side these buildings need intensive ventilation which causes large losses of energy by exhausted air. A good way to reduce heat losses can be the use of heat recovery technical systems (Kic and Gurdil, 1999; Kic and Pavlicek, 2006a; Kicet al., 2007). Whereas in recuperators, where heat is transferred directly and immediately through a partition wall of some kind, from a hot to a cold fluid, both of which flow simultaneously through the exchanger, the operation of the regenerative heat exchanger involves the temporary storage of the heat transferred in a packing which possesses the necessary thermal capacity.(Willmott 2011) One consequence of this is that in regenerative heat exchangers or thermal regenerators, the hot and cold fluids pass through the same channels in the packing, alternately, both fluids washing the same surface area. In recuperators, the hot and cold fluids pass simultaneously through different but adjacent channels.

In thermal regenerator operation the hot fluid passes through the channels of the packing for a length of time called the "hot period," at the end of which, the hot fluid is switched off. A reversal now takes place when the cold fluid is admitted into the channels of the packing, initially driving out any hot fluid still resident in these channels, thereby purging the regenerator. The cold fluid then flows through the regenerator for a length of time called the "cold period," at the end of which the cold fluid is switched off and another reversal occurs in which, this time, the hot fluid purges the channels of the packing of any remaining cold fluid. A fresh hot period then begins.

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During the hot period, heat is transferred from the hot fluid and is stored in the packing of the regenerator. In the subsequent cold period, this heat is regenerated and is transferred to the cold fluid passing through the exchanger.

A cycle of operation consists of a hot followed by a cold period of operation together with the necessary reversals as can be seen at Figure 1. After many cycles of identical operation, the temperature performance of the thermal regenerator in one cycle is identical to that in the next. When this condition is realized, the heat exchanger is said to have reached "cyclic equilibrium" or "periodic steady state." Should a step change be introduced in one or more of the operating parameters, in particular, the flow rate and entrance temperature of the fluid for either period of operation, or the duration of the hot and cold periods, the regenerator undergoes a number of transient cycles until the new cyclic equilibrium is reached.

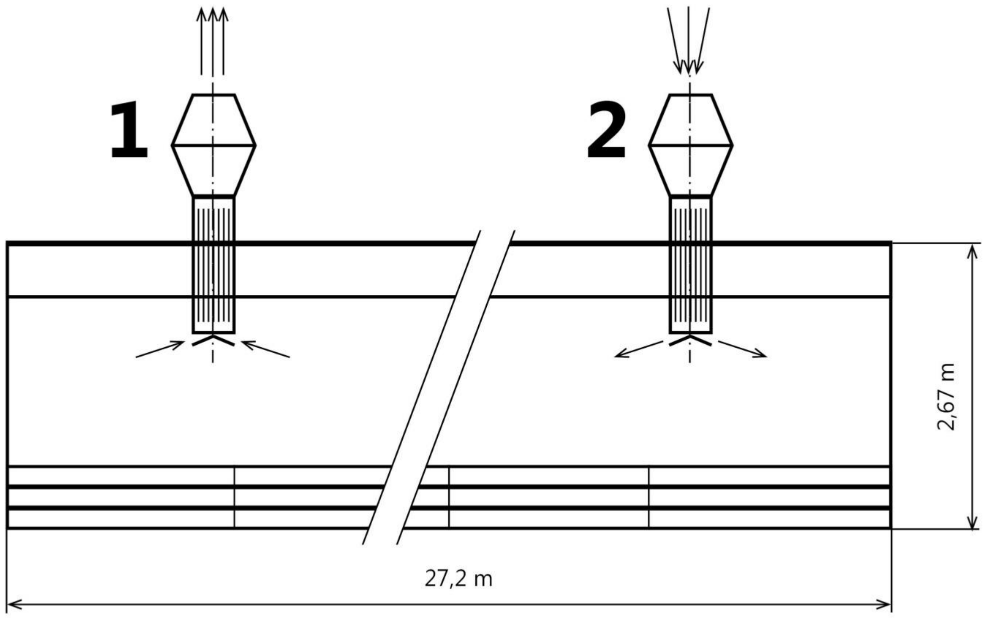


Figure 1 Use of the heat regenerator with cement-bonded particleboards for accumulating the heat produced by animals inside the stable. 1 - heating period, 2 – cooling period

Earlier work (Pavlicek, 2007), presented results of the regenerator’s measurement at the laboratory and also under real conditionsinpig and poultry farms. The unique design is presented there with its theoretical background (Kic and Pavlicek, 2006a; Kic and Pavlicek, 2006b; Kicet al., 2007). Regenerator is made from non-metallic materials (CETRIS and CEMVIN - cement-bonded particleboards), which are characterized by relatively high thermal capacity and low thermal conductivity.

Numerical simulation process presented here is the methodological attempt to obtain relevant information of regenerator’s thermal behavior. The aim is to obtain the possibility to solve non-stationary state of equipment for the changeable flow and thermal conditions. Obtaining such information will lead to acceleration of the design process of such kind of equipment for the next scheduled cases.

## 2 Materials and methods

### 2.1 Experimental apparatus

Experimental apparatus is a rectangular channel equipped with plate regenerator, axial fan with the flow direction reversion capability and regulation of rotor revolutions.

temperatures between center and the surface of the plate. The time dependent temperature profile arising inside the

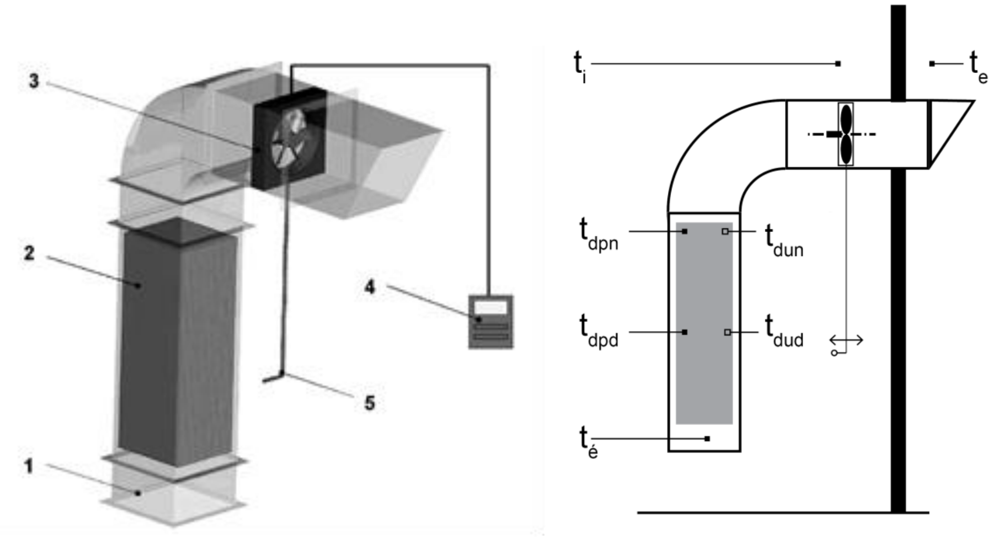


Figure 2Heat regenerator with cement-bonded particleboards and position of specific physical characteristics of system. 1-air inlet, 2- heat accumulating plates, 3 – fan, 4 – flowrate control, 5 – reverse of airflow direction$t_i$ – internal temperature, $t_e$ – external temperature, $t_é$ – inside exhaust/inlet temperature, $t_{dpn}t_{dpd}$ – upper and central temperature on the surface of the plate, $t_{dun}t_{dud}$ – upper and central temperature in the center of the plate (inside the material)

Measurements were taken using an Ahlborn Almemo 2290-8 data logger equipped with thermistor sensors, which are fixed inside the apparatus and also directly inside the heat accumulation plates. Such configuration is usable for obtaining time dependences of temperature for different flow conditions and plate sizes, thanks to the possibility of their replacement and position adjustment.

### 2.2 Theoretical background

Heat transfer inside a heat regenerator is a relatively complex problem characterized as a time dependent problem including convection and conduction principles along with geometrically dependent surface temperature of plates due to heating/cooling from the flowing air stream. Due to the relatively small heat transfer coefficient and thermal conductivity (for CEMVIN k=0.35 W/mK and $\alpha$ =20 W/m$^2$K) the Biot number for the plate is about 0.2, which means that the plate isn't heated uniformly. The temperature profile through the thickness of the plate has not negligible differences of material of plates is time dependent.It is therefore impossible to use simple analytical integral methods to predict an amount of heat accumulated/released from the plates. Non trivial analytical solution very close to this problem, but only for one plate can be found in Mladin (2011) and Cafagni(2013).

### 2.3 Computational model for ANSYS/Fluent

Computational model is based on the experimental apparatus dimensions. ANSYS/Design Modeler is used to generate a rectangular channel (Figure 3). The size of plates and the size of channel cross-section correspond exactly to the experimental apparatus. The only visible change is the higher total length of a channel. It is chosen due to better distribution of air between plates, to make it uniform as much as possible. Complete geometrical model is made as a parametrical and therefore the relatively long time of preparing this model is compensated with possibility to obtain different geometrical variants of this topology relatively quickly.

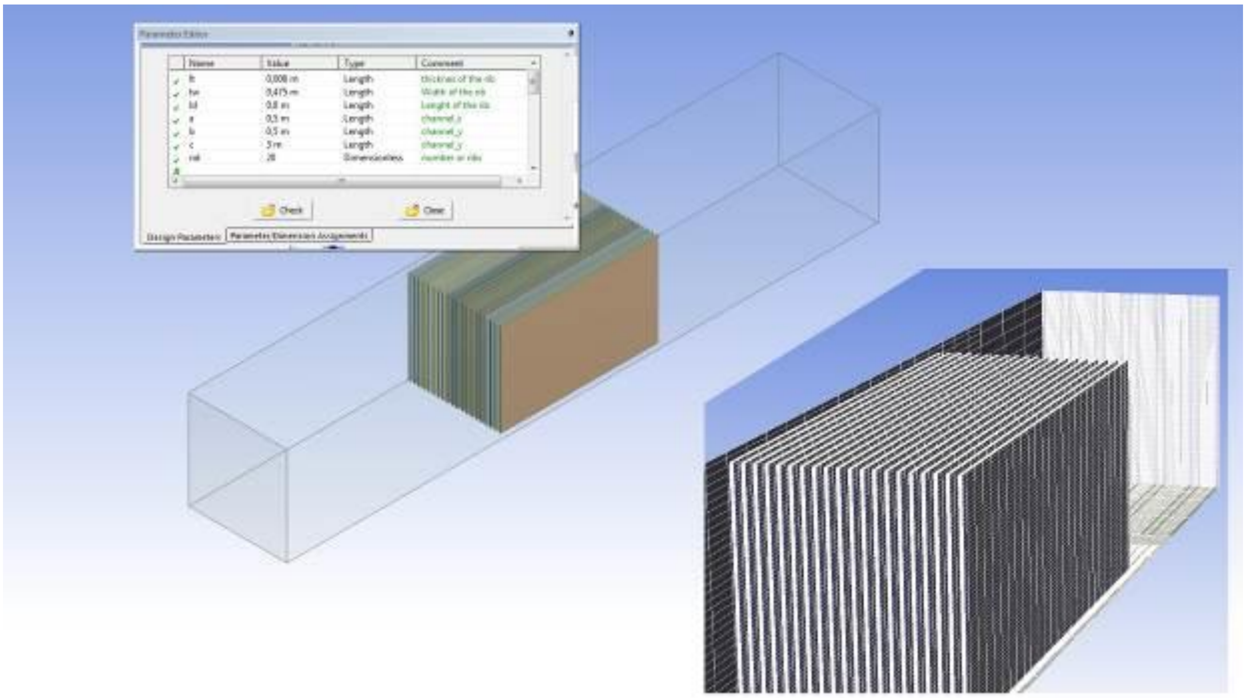

Figure 3 Computational domain,with geometrical parameters and resulting shape of computational grid

2.3.1 Geometry and Computational Grid Creation

The important problem with numerical analysis arises after a numerical experiment with the grid generated directly from ANSYS/Meshing software, because of the great ratio between length of the slit and its width. Automatic grid generator does not suffice to construct the grid with their quested details, uniformity and acceptable size. The great problem of this topology is that even a small difference among the computational grid shape in any slit leads to a significant non-uniformity in velocity profile (flowrate) through this slit. Therefore, ascaling grid method is proposed (Figure 4). The universal geometry (long channel of rectangular cross-section) was created with parametrically adjusted size and number of plates, which are placed equidistantly, exactly at the centre of channel. The grid sizing in one direction is performed and requested shape with regular mesh is obtained.

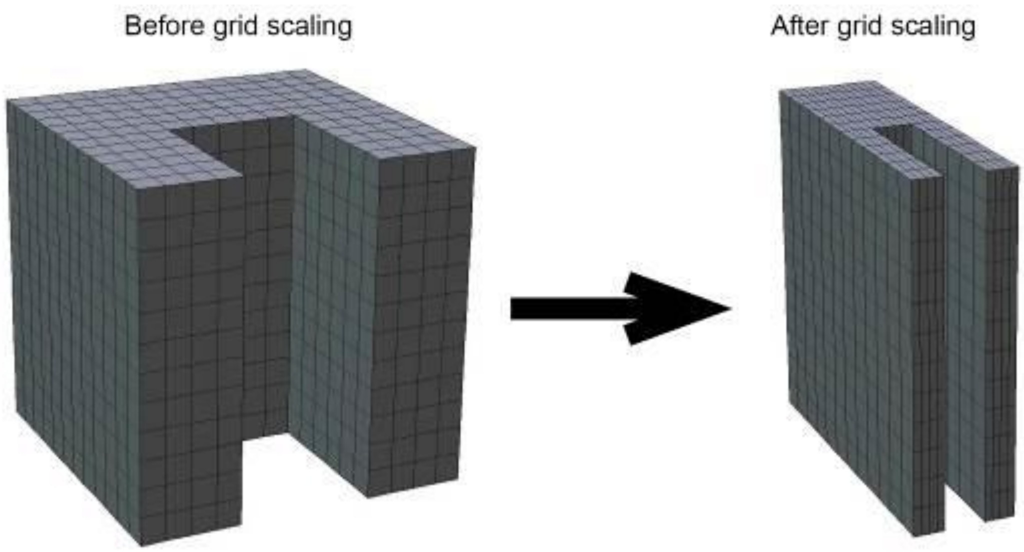


Figure 4 Principle of grid scaling leads to uniform grid distribution with defined number of elements in each direction used for making uniform mesh for all channels

The geometrical shape is three times symmetrical, assuming, that plates are inserted onto channel symmetrically. There are only two symmetries from physical point of view, because there is no symmetry in longitudinal direction for temperature profile. Finally the inapplicability of the symmetry boundary conditions was observed during first set of numerical simulations. Very poor convergence of solution was also observed when symmetry boundary conditions were used.

Also, the two dimensions model is not acceptable for this simulation, because it cannot calculate the total amount of energy accumulated inside plates and the spatial distribution of monitored quantities. Therefore, a complete domain is used for calculation. Table 1 shows the configuration parameters used for one given frequency of the flow direction reversion.

**Table 1 Numerically solved variants for one stream changing frequency**

| Flowrate | [m$^3$/s] | 0.3 | | | | | | | | | 0.6 | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Length | [m] | 0.6 | | | 1.2 | | | 1.8 | | | 0.6 | | | 1.2 | | | 1.8 | | |
| Thickness | [mm] | 3 | 5 | 8 | 3 | 5 | 8 | 3 | 5 | 8 | 3 | 5 | 8 | 3 | 5 | 8 | 3 | 5 | 8 |

2.3.2 Boundary Conditions

Simulation was run under the same conditions as laboratory measurements so as to be comparable for validation of the numerical model. The air flowrates (min. – flowrate 1 – 0.3 m$^3$/s and max – flowrate 2 – 0.6 m$^3$/s)obtained during the measurements are shown in Table 2 and correspond to three possible adjustable regimes of the used ventilation system.

**Table 2Air flow rates from measurement used as boundary conditions**

| Width of the plate | Air Flow Rate | | |
|---|---|---|---|
| | Low | average | high |
| mm | m$^3$/s | m$^3$/s | m$^3$/s |
| 8 | 0,28 | 0,35 | 0,60 |
| 5 | 0,29 | 0,36 | 0,61 |
| 3 | 0,30 | 0,39 | 0,62 |

Due to relatively high velocities (up 2.5 m/s) corresponding to these airflows at inlet between the plates, the flow is solved as a turbulent stream with k-$\varepsilon$ turbulence model. The walls of channel are assumed as an isothermal, because there was insulated during the measurements. Thermo physical properties for materials CEMVIN and CETRIS are shown at Table 3.

**Table 3 Important thermo physical properties for CEMVIN and CETRIS**

| | Density | Specificheat capacity | Thermal conductivity |
|---|---|---|---|
| | Kg/m$^3$ | J/kgK | W/mK |
| CEMVIN | 1520 | 1500 | 0,35 |
| CETRIS | 1350 | 1400 | 0,22 |

## 3 Results and discussions

Figure 5 depicts the typical shape of measured temperatures. The CFD calculation is performed as a time dependent cycle with respect to 1 minute or 5 minute heating/cooling period. Boundary conditions are switched at the end of each cycle. Discrete values of temperatures are exported from CFD model to be compared with measured data. Such comparison can be seen in Figure 6. It shows periodical oscillation of temperature inside the plate for 120 time period for cold side temperature 1℃ and hot side temperature 24℃.If the numerical model gives such good fit to measured data, it can be found, that physical properties, boundary conditions and also the computing mesh together represents the real behavior of observed system, e.g. model is tuned. The solution of variants can be based on such tuned model.

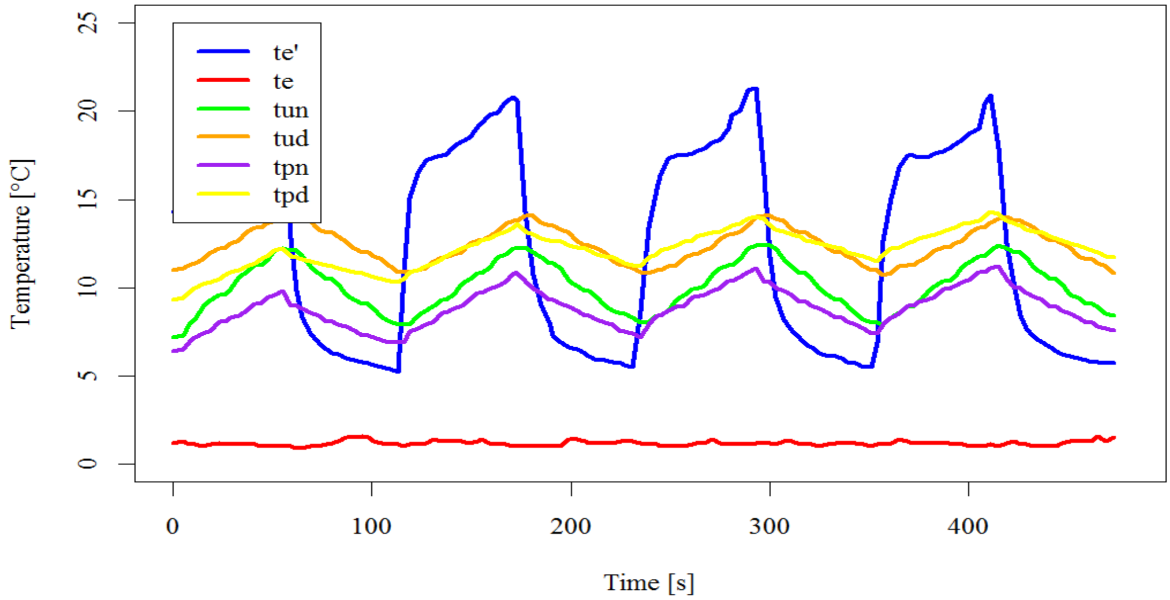


Figure 5Typical shape of temperature data from measurement (symbols correspond to Figure2)

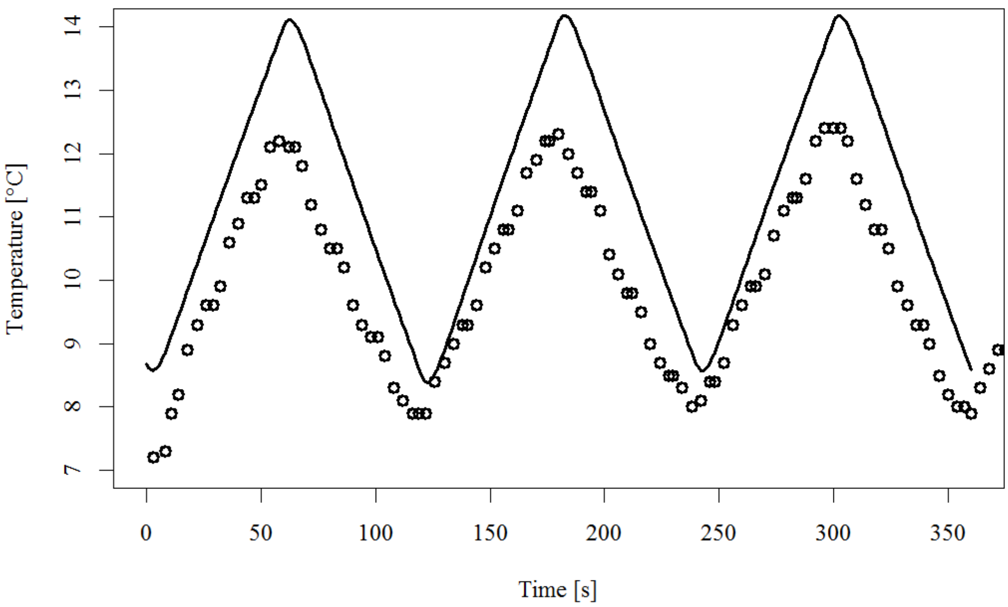


Figure 6 Comparison of temperature values inside the plates from the measurement (dots) and calculation (line) for 3 mm plates, flowrate 0.3 $m^3s^{-1}$ and steadyperiodic state.

Figure 7 shows the shape of temperature distribution inside the channel cross section perpendicular to plates. Similar results can be obtained for velocity, pressure and energy. Figure 8 shows discrete temperatures at points M, N, O, P and Q for a 3000 seconds temperature profile development corresponding to a8 mm thick plate, 1.8 m long with two different air flow direction periods, namely 120 and 600 s.

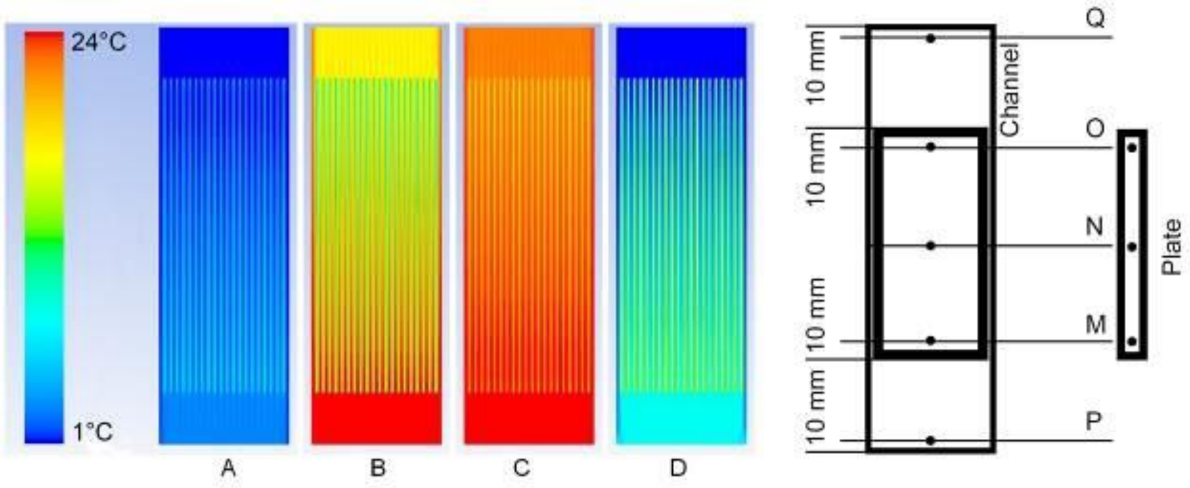


Figure 7One cycle of heating and cooling period for 3mm thick plates. A – end of cooling, B – heating, C – end of heating, D – cooling, with sketch of evaluation points positions of discrete values M, N, O, P, Q.

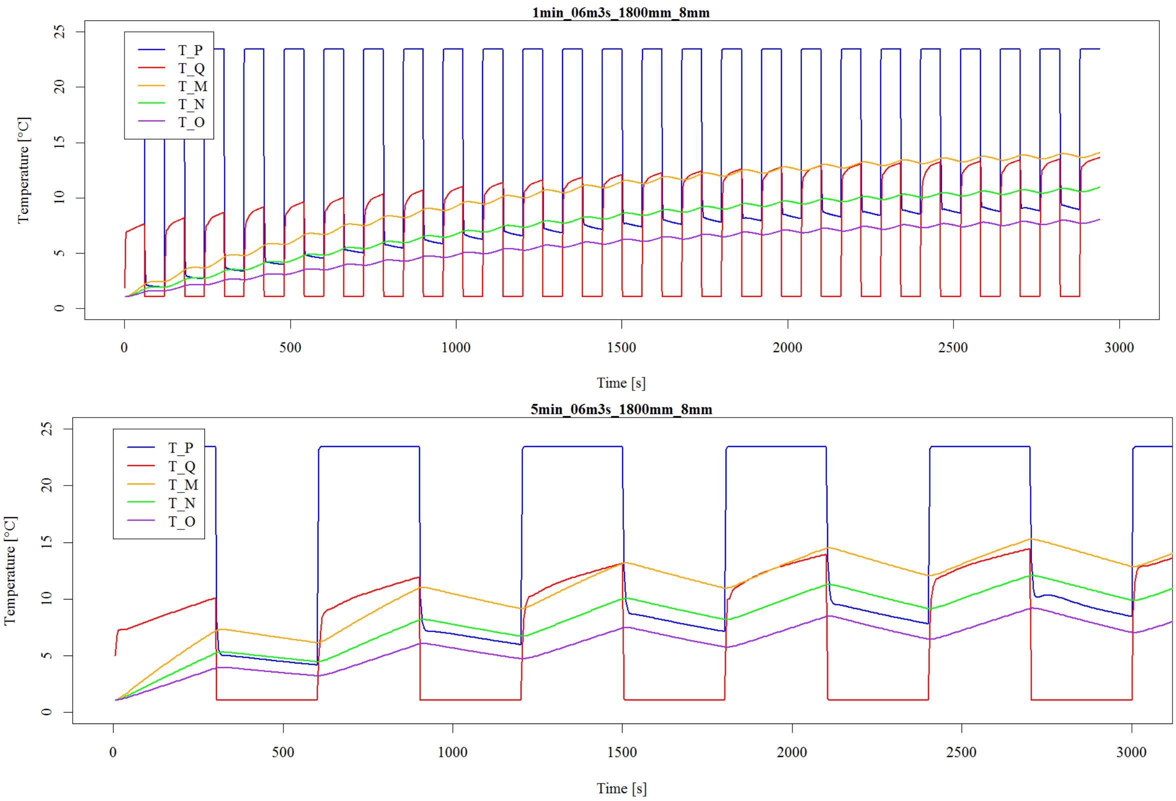


Figure 8 Temperature profile atpoints M, N, O, P and Q observed during the same configuration and time periodof 120 s and 600 s, starting from initial cold state

Figures 9, 10, 11 and 12 show the time dependency of temperature and specific internal plate energy for all plates together for 60 and 300 s reversing period, respectively. Nine cases were selected to compare influence of reversion period, air flowrate, length of the plate and plate thickness. Specific internal plate energy is related to the initial state, when the temperature of plates is equal to the cold side temperature. It is important to mention that the slope of this curve can be interpreted as an intensity of heat transfer from/into plates. It can be seen, that at approximately 1200 s there is no yet a fully developed stationary periodical state. Temperature and internal energy aren’t exactly the same in every time step.

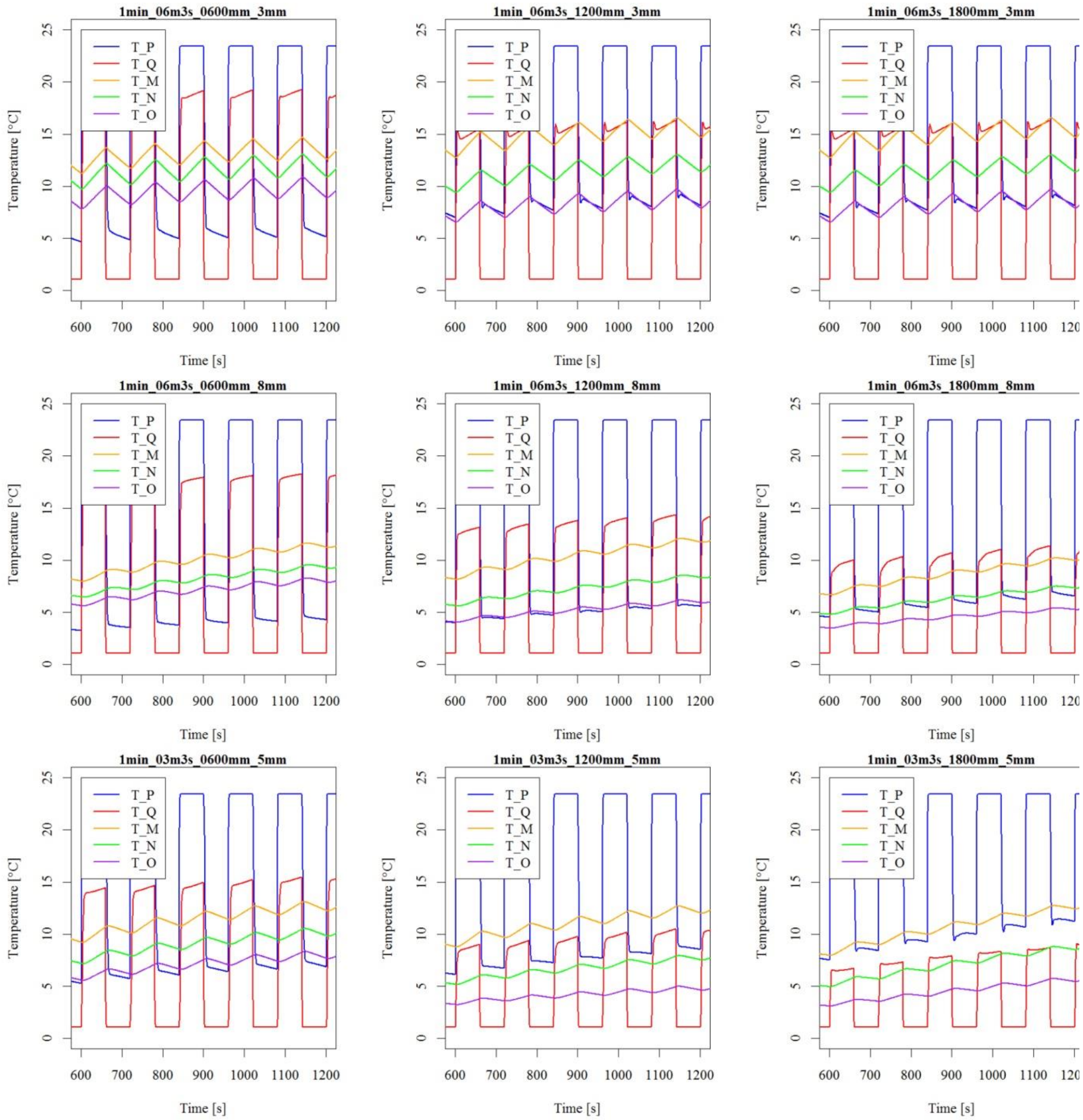


Figure 9 Temperatures at discrete points M, N, O, P, Q for 60s flow reversing period.

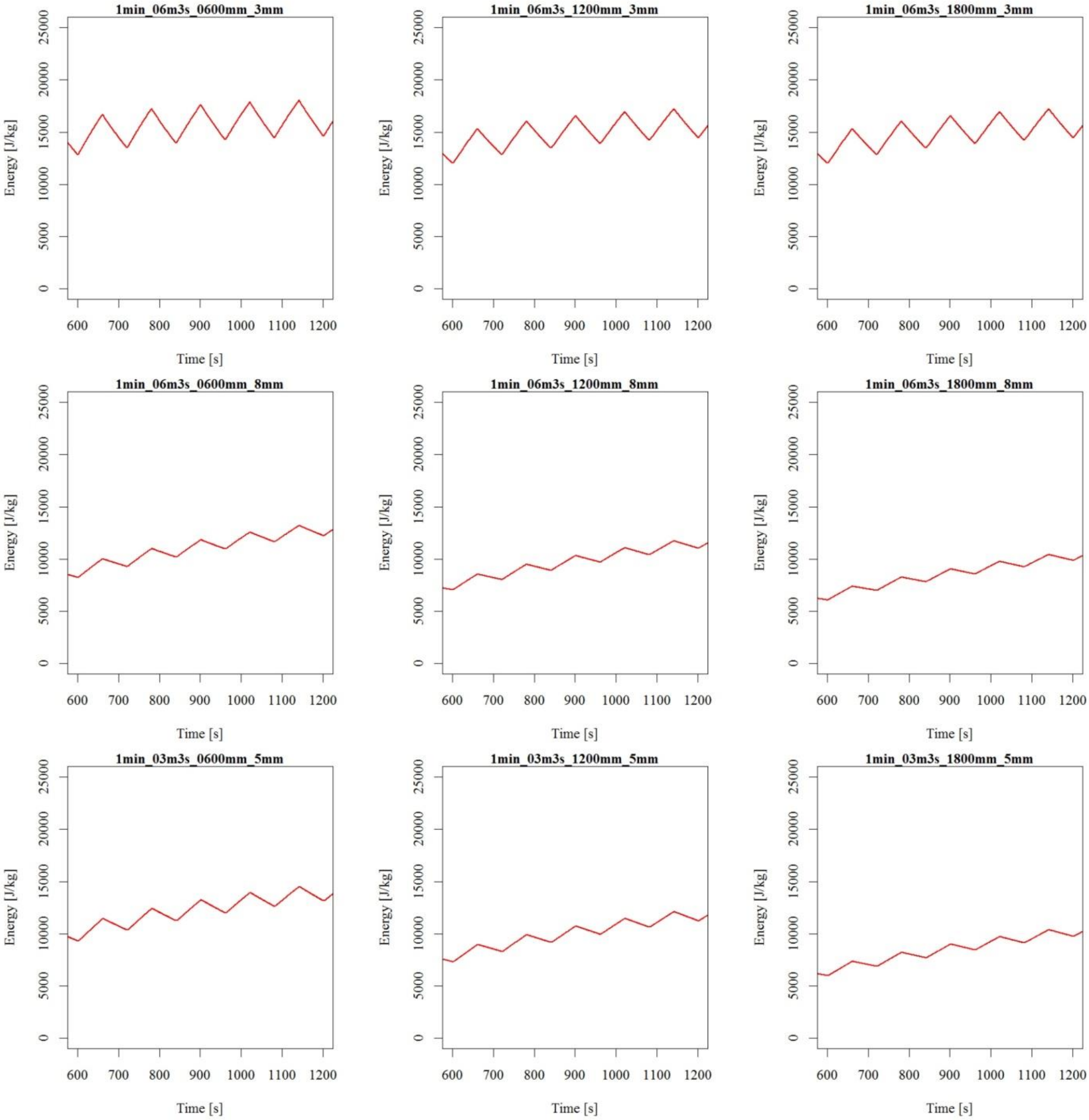


Figure10 Specific energy of platesin time dependence for 60s flow reversing period

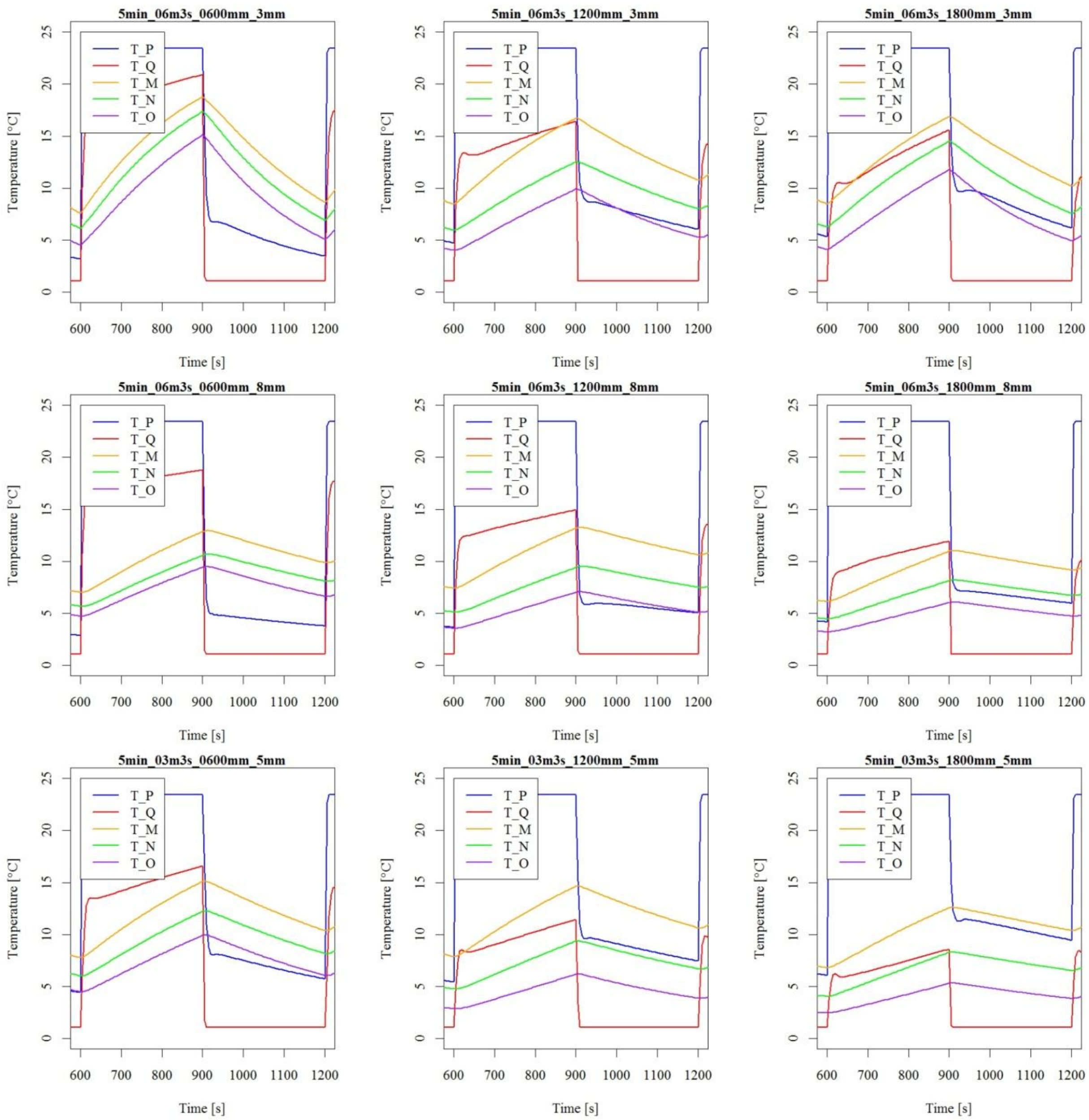


Figure 11Temperature at discrete points M, N, O, P and Q for 300s flow reversing period

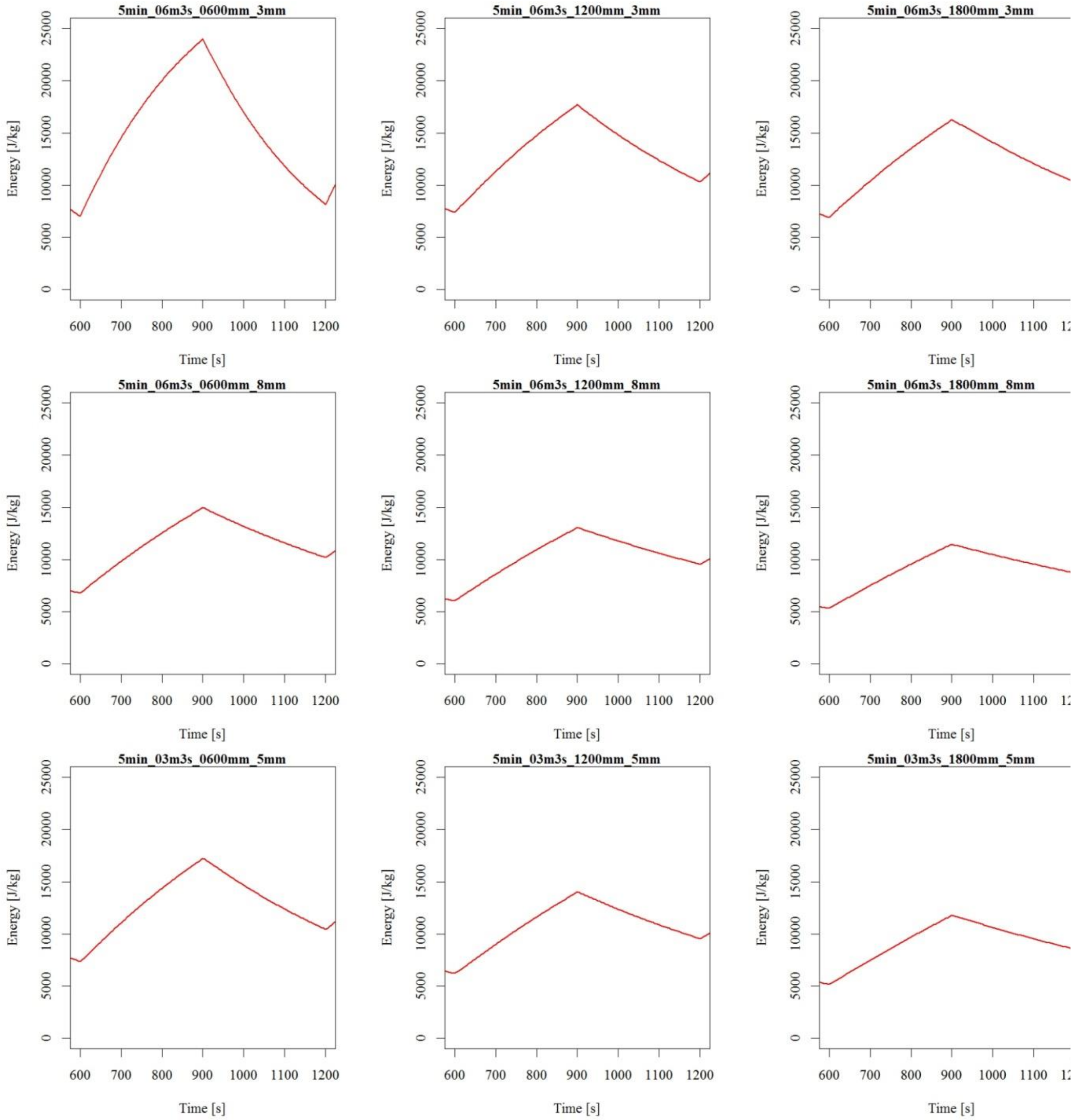


Figure 12 Specific energy of platesin time dependence for 60s flow reversing period

All simulations were made up to the time 3000 s, where the stationary periodicity was observed in all cases. For the long plates, there is an effect of more uniform temperature profiles at the hot side, because incoming air has a longer time to be pre-heated and after flow reversion. The shapes of plate's specific energy oscillations shows expected effect of more progressive heat transfer in every cycle for thin plates. This is because of the heat from the centre of thin plate is transferred more rapidly, as is also seen at temperature profiles. It can be seen, that the temperature difference between ends of long plates is relatively higher, that on short plates, but the difference isn't dramatically considerable. It leads to the recommendation, that for intensive heat regeneration the long thin plates are better to use, than the short and thick ones. It is clear, that in this case, the relatively higher frequency of flow reversion is needed.

The shape of the temperature profile within the plate is an important characteristic of the quality of the heating/cooling process for the designed structure. It is possible to obtain the time course of the temperature profile for different sizes of plates using the unsteady numerical simulation. Figure 13 shows a comparison of temperature profiles for the plates of 600 mm, 900 mm, 1200mm and 2100 mm. The plate is placed in the axis of the channel so that the distance from the edges of the channel on both (hot and cold) sides of heat exchanger is the same. The profile is evaluated in the longitudinal axis of the channel, passing through the center plate positioned closest to the centre of channel (in the case of twenty plates thecentral axis of the plate number 10 is analyzed).

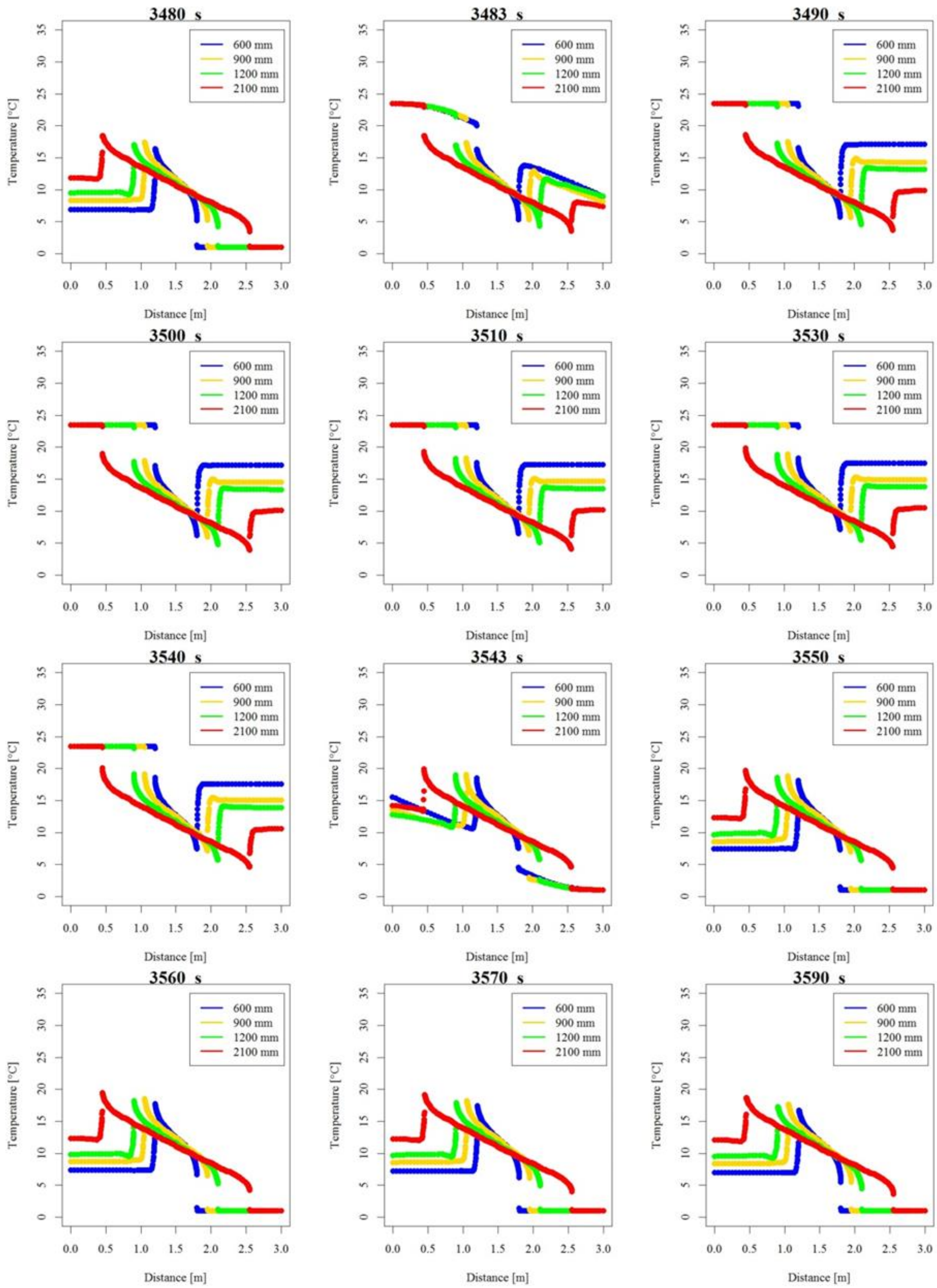


Figure 13 Longitudinal time dependent temperature profile for 120 s heating/cooling period

The graph shows the distance from the hot side on the x-axis and the y value of the air temperature outside panel and plate internal temperature inside the plate (in the central part of the graph). Relatively fast change of temperature of flowing air, and a slow change inside the plateis seen when the direction of flow is changed. This fact also corresponds to the temperature values at points M, N and O in Figure 8.

At time 3480 there is a switch to the charging mode. The warm air enters the heat exchanger and the internal energy of plates increases. The flow direction in Figure 13leads from left to right. At the time of 3540 s the flow is reversed and another 60s the heat exchanger is in a state of discharge. The flow direction leads from right to left. Interval from 3480 s to 3600 s corresponds to 120 s period of this case. The shape of the temperature profile inside the board has the stationary- periodic character. It has only a very subtle deviation seen as an almost invisible shift of profile curves upwardly and downwardly during the heating and cooling.

The analysis presented here with 5mm thick plate indicates that extension of boards can leads to increasing the temperature difference between the hot and cold side of the heat exchanger. It is also clear that in this reference case is acceptable for all plate lengths, because there is no effect of exhausting the heat capacity of plates – the temperatures inside the plate do not approach the temperature of the free air stream. This fact can be interpreted, that the plates have sufficient heat capacity to absorb the heat during the heating period.

## 4 Conclusions

The ANSYS/Fluent CFD software efficiently simulated the thermal and flow conditions inside a plate regenerative heat exchanger. Solved variants lead to the deduction, that using relatively thin plates with higher frequency of flow reversion is better for presented kind of application because of more intensive heat transfer from the whole plate mass. Also the relatively long plates can be recommended due higher inlet air temperatures, which are caused by longer air exposition to plates and also larger heat transfer area. The absorbing theheat fromthe airflow is the limiting factor forthe structure of theheat exchanger.Such ability of plates depends onsufficienttemperature gradientbetween the surface of plate and theflowing air.CFDcan determine theheating orcooling ofthe platewith respectthe structure of theheat exchangerof this kind, so thatthere will not arise excessiveheating of platesortheir insufficient warming. There can be also done optimization for other design parameters, such as thickness of plate, air stream temperature, heating/cooling period or material properties as requested.

## 5 Nomenclature

a, b, c [m] dimensions of channel for numerical analysis

Bi [-] Biot number

$c_p$ [J/kgK] specific heat capacity

k [W/mK] thermal conductivity

$l_{t,w,l}$ [m] dimensions of one plate, indexes t – thickness, w – width, l - length

$t_e$ [K] cold side temperature

$t_{e'}$ [K] hot side temperature

$t_i$ [K] internal temperature

V [$m^3/s$] volumetric flowrate

$t_{d,up,nd}$ [K] plate temperature, indexes u – inside, p – surface, n – cold side, d - hot side

$\alpha$ [$W/m^2K$] heat transfer coefficient

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