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# The Influence of the Fin Set-up to the Cooling Output of the Floor Heating Convector

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***Abstract*****—**This article deals with the numerical simulation of the floor heating convector in 3D. Presented convector can operate in two modes – cooling mode and heating mode. This initial numerical simulation is focused on cooling mode of the convector. Models with different temperature of the fins are compared and three various shapes of the fins are examined as well. The objective of the work is to predict air flow and heat transfer inside convector for further optimalization of these devices. For the numerical simulation was used commercial software Ansys Fluent.

***Keywords*****—**Cooling output, floor heating convector, numerical simulation, optimalization.

## I. Introduction

THE heat convector systems have many construction variations [1]. One of them is installation of the convector to the floor. This is the type of examined convector.

Heating convector consists of the outer container, which is the shell placed to the floor. Inside the container is the water-air exchanger with axial radiator fan. The exchanger has system of pipes equipped with the lamellae. The pipes are separated to two independent sets, one for the cooling and the other for the heating mode. Above the heat exchanger is covering aluminium grid. The example of floor heating convector is on fig. 1.

Fig. 1 example of the floor heating convector

The temperature difference of the outer air and heating water in the heating mode is considerably higher, then the temperature difference in the cooling mode (surrounding air to coolant). That is why the set of the pipes for the cooling has more pipes then the set of the pipes for the heating (figure 2).

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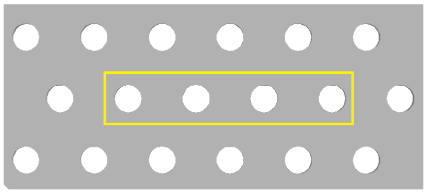

Fig. 2 lamella of the heat exchanger with marked pipes designated for heating

The cooperating company, which produces convectors of various types, had insufficient information about processes inside the convectors. Almost all their development was based on presumption of air flows and measurement of new convectors. The objective of the work is to use the numerical simulation methods for the optimalization of the floor heating and cooling convectors.

In the past were made initial 2D simulations [2], [3] focused on study of flow interaction with metal plates on the input and output of the exchanger, used for change of flow direction. Those simulations were aimed at flow around the pipes without involvement of the cooling effect of the fins.

3D simulation allows covering up the effect of the fins, so the first goal was to compare models with and without cooling of the fins.

The shape of the fins has significant effect to the heat transfer [1]. Three different fins – straight, sinusoidal and angular were examined for two different diameters of the pipes.

## II. Geometry and Computational Grid

The boundary of the model inside the heat convector is displayed on figure 3. Geometry of the model was created in Pro/Engineer from the model of the complete heat convector.

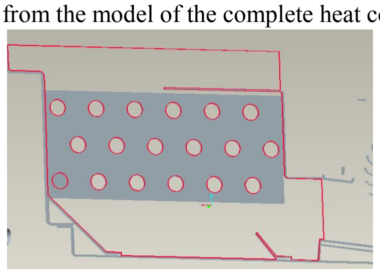

Fig. 3 part of the convector in Pro Engineer

The computational grid was created in Ansys Design Modeller and has about 70 000 hexahedral cells (fig. 4). Around the pipes was created mesh inflation. The thickness of the model is 2,5 mm.

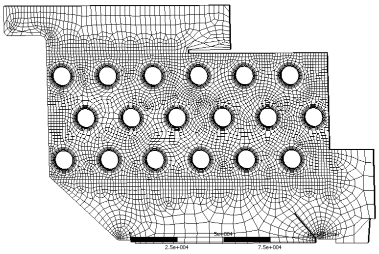

Fig. 4 computational grid

For examination of shape of the fins were created 3 models, representing the area between two fins (fig. 5), also in the Pro Engineer.

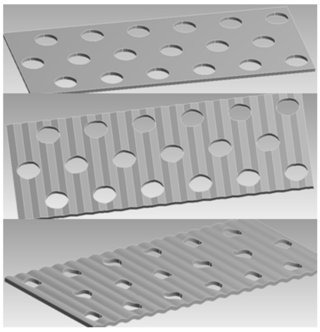

Fig. 5 separate models with different fin shape

The computational grid for sinusoidal fin contains 150 000 cells (figure 6).

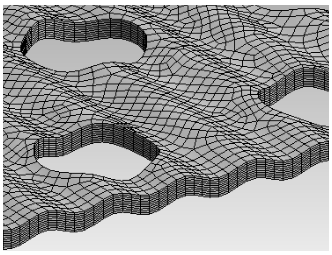

Fig. 6 computational grid for sinusoidal fin

## III. Numerical Simulation

For the numerical simulation was used commercial software Ansys Fluent. The flow is solved as an unsteady by the DNS model because of the low Reynolds number. All simulations were made in cooling mode. The temperature of the pipes and fins was set to 9 °C. Room temperature of ingoing air is 22 °C. Inlet velocity is 3,5 m/s. The calculations were made to the time of 1 second, then continued for 0,6 second for time average values, all with the time step 0,001 second.

The configuration of the boundary condition is on figure 7. Dotted line represents pressure output, long dash line velocity inlet. The grey parts above and under the fin are set to periodic boundary condition. Initial temperature in the convector is 22 °C.

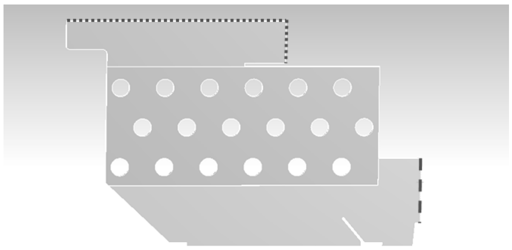

Fig. 7 configuration of the boundary condition

For the purpose of the comparison of results were measured values of the average temperature and average velocity on the plane beneath the heat exchanger fins (figure 8).

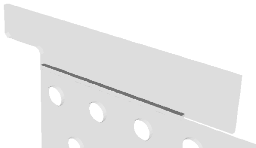

Fig. 8 measuring plane in the model

The theoretical cooling output of 1 gap between fins, $\dot{Q}(W)$ is given by the (1), where $\dot{m}$ (kg/s) is mass flow rate, $c_p$ (J/kg.K) specific heat capacity at constant pressure, $T_1$ is the temperature of the outer air and $T_2$ is the average temperature beneath the heat exchanger fin.

$$\dot{Q} = \dot{m}.c_p(T_1 - T_2) \tag{1}$$

The separate fin models have inlet velocity also 3,5 m/s. The entrance for the air is on the shorter side between the fins. Lateral planes are set to wall and on the rear side is set to the pressure output (fig. 9).

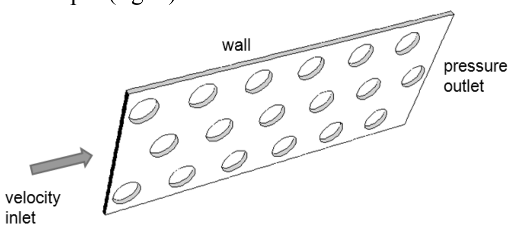


Fig. 9 boundary condition of the separate fin model

## IV. Results

At first were compared results of the complete model with and without cooling effect of the fins. The output velocity from the heat exchanger was in both cases identical, v = 2 m/s. On figure 10 are shown contours of time averaged velocity in section perpendicular to Z axis (z = 1,25 mm, Z is identical with the axis of the pipes).

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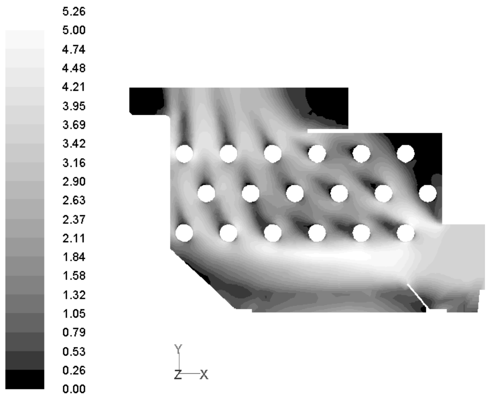


Fig. 10 contours of time averaged velocity

At the bottom of the convector is situated metal plate for directing of the flow to the pipes of the heat exchanger. Maximal velocity v = 5,26 m/s lies under the heat exchanger. On the right corner of the exchanger is the upper metal plate, used to isolate the outgoing air from the fan input, so the backward flow to the ventilator, situated on the right side, is lowered. Bottom metal plate also increases the static pressure on the input to the exchanger (figure 11).

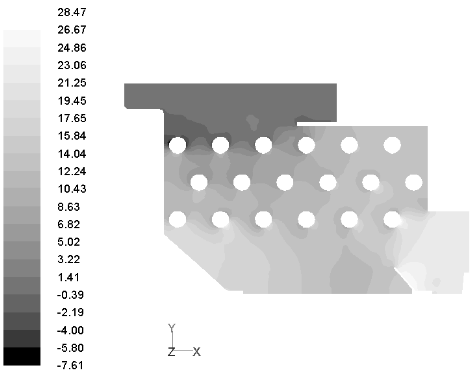


Fig. 11 contours of static pressure

On the figure 12 and 13 are presented contours of time averaged static temperature in t = 1,6 s.

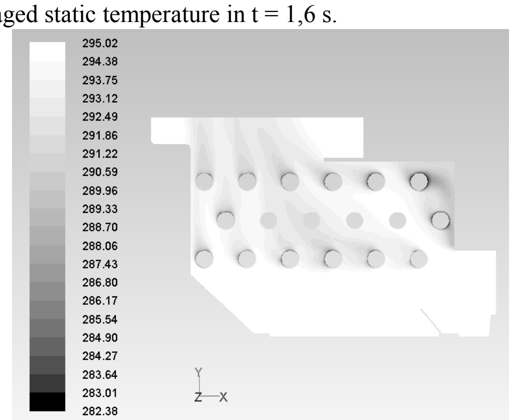


Fig. 12 static temperature without fin effect

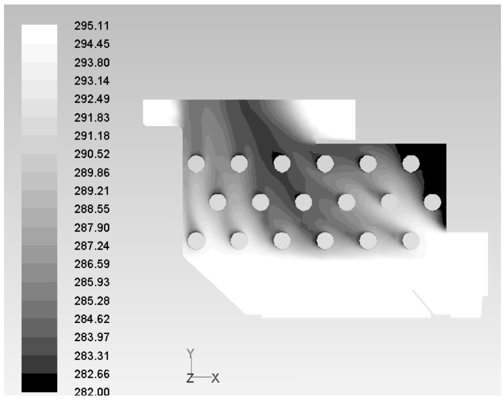


Fig. 13 static temperature with fin effect

We can observe that in case of model with cooling effect of the fin static temperature falls considerably higher, then in case of model with cooling effect of the pipes. The minimal temperature on the output is in case of model with pipes T = 291 K, in case of model with cooling fins T = 283 K. Theoretical cooling output of 1 gap between fins is presented in Table I.

The contours of time averaged velocity (fig. 10) shows the area in right upper corner of the heat exchanger, where insufficient air flow appears. This is documented in both figures of contours of static temperature by the lowest temperature fields in the model.

TABLE I

| Cooling mode | $T_2$ (K) | m (kg/s) | Q (W) |
|---|---|---|---|
| pipes | 292,7 | 3,984E-04 | 0,92 |
| pipes + lamellae | 284,1 | 3,984E-04 | 4,36 |

Figure 14 demonstrates the time averaged output velocity profiles from the heat exchanger (plane in fig. 8). Profiles were obtained in 3 different Z coordinates (0,5 mm; 1,25 mm and 2 mm). Maximal velocity is in the centre of the gap between fins, represented by (o), side profiles in 0,5mm and 2 mm, represented by (+) and (x), are almost identical.

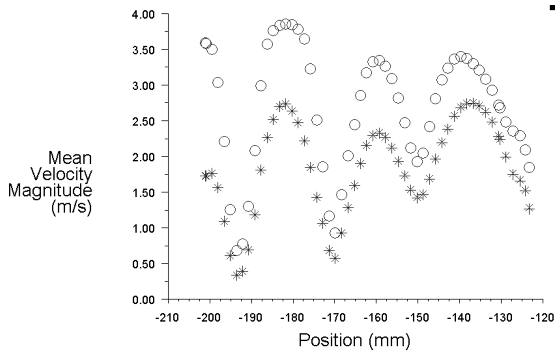


Fig. 14 output velocity profiles

For the study of fin shape were created 6 different models. For the diameters of pipes d = 12 mm and d = 9,25 we can see results of time averaged velocity fields for straight fins in

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figure 15. Average output velocity decreases from $v_{d12}$ = 4,58 m/s to $v_{d9}$ = 4,07 m/s. The angular and sinusoidal fins indicate approximately similar progress.

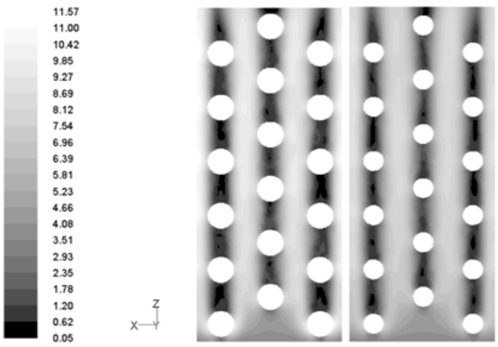


Fig. 15 temperature fields for different pipes diameter

If we look at the temperature fields of different fin shapes, there is significant difference (fig. 16). Angular and sinusoidal fins have better cooling effect, than the straight fin. Temperature field with minimum temperature T = 282 K appears in case of the angular fin at the upper 1/3 of length of fin, in case of the sinusoidal fin in the upper 1/2 of length of fin.

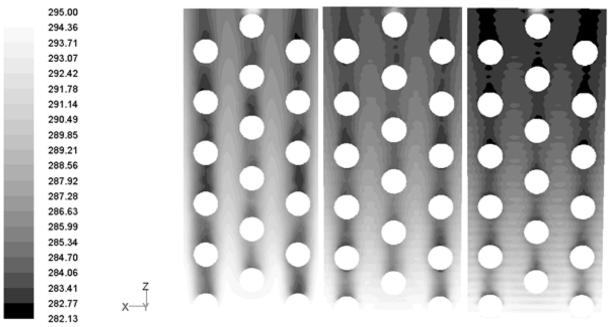


Fig. 16 temperature fields for different fin shapes

The increased cooling ability is compensated by the increased pressure drop through the fins (fig. 17)

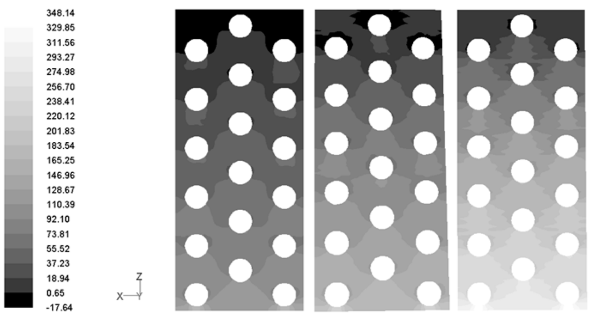


Fig. 17 contours of static pressure, from the left - straight, angular and sinusoidal

TABLE II
THE SINUSOIDAL SHAPE OF THE FIN AS THE MOST SUITABLE FROM THE PRESENTED VARIANTS

| Mode | $T_{out}$ (K) | m (kg/s) | Q (W) |
|---|---|---|---|
| straight - d 12 | 284,7 | 5,7E-04 | 5,9 |
| straight - d 9,25 | 284,6 | 5,7E-04 | 5,9 |
| angular - d 12 | 283,3 | 5,7E-04 | 6,6 |
| angular - d 9,25 | 283,1 | 5,7E-04 | 6,8 |
| sinusoidal - d 12 | 282,7 | 5,7E-04 | 7,0 |
| sinusoidal - d 9,25 | 282,7 | 5,7E-04 | 7,0 |

The set-up of the model for testing the fin shapes could be considered as ideal state. All air flows around the pipes and fins, so even the minimal value of theoretic output table II is higher, then the output of the model with fins in table I.

## V. Conclusion

The test of the fins shape showed the way of increasing the cooling output of the device.

Presented results will be used for comparison with the experimental data. The experimental laboratory is currently under construction and will be focused on the testing of the various types of the cooling and heating convectors.

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