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Fin Spacing Effect of the Tube Fin Heat Exchanger at the Floor Heating Convector

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***Abstract*—**This article deals with numerical simulation of the floor heating convector in 3D. Numerical simulation is focused on cooling mode of the floor heating convector. Geometrical model represents section of the heat exchanger – two fins with the gap between, pipes are not involved. Two types of fin are examined – sinusoidal and angular shape with different fin spacing. Results of fin spacing in case of constant Reynolds number are presented. For the numerical simulation was used commercial software Ansys Fluent.

***Keywords*—**fin spacing, cooling output, floor heating convector, numerical simulation.

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 heating (fig. 2).

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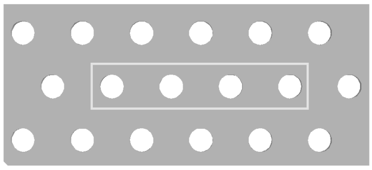

Fig. 2 lamella of the heat exchanger with marked pipes designated for heating

The cooperating company, which produces convectors of various types, had insufficient information about processes inside the convectors. That is why the numerical simulation is used to show the effects inside the convector. The objective of the work is to find possibility of optimization for the floor heating and cooling convectors.

One of the parameters which affect cooling output of the heat exchanger is fin spacing. Create the heat exchanger with modified fin spacing is possible, so this parameter was examined in a range from 2 mm to 4 mm.

Simulations were made for sinusoidal and angular shape of the fin. Results are separated to two parts – first with constant Reynolds number and second with constant inlet velocity.

Because major part of the cooling output is carried by the fins [1], for comparison of the spacing effect were created models without pipes.

II. Geometry and Computational Grid

Model of the angular fin is shown on the fig. 3. On the left side we can see the straight entering area with length of 50 mm, then starts the fin with length corresponding to the convectors heat exchanger. Models were created with different spacing for both sinusoidal and angular fin for 2; 2,5; 3; 3,5 and 4 mm.

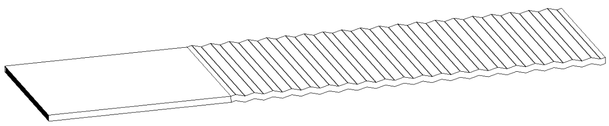

Fig. 3 model of the sinusoidal fin

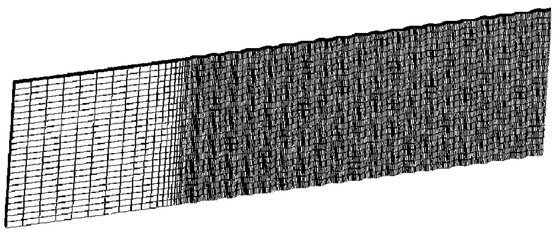

Fig. 4 computational grid

The computational grids were created in the Ansys Design Modeller and have from 70 000 to 90 000 hexahedral cells. On

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the fig. 4 is presented the computational grid for sinusoidal fin with 2,5 mm spacing. Volume between fins has higher sizing factor then the straight entering area on the left side of model.

III. Numerical Simulation

For the numerical simulation was used commercial software Ansys Fluent. The flow is solved as an unsteady by the LES model. All simulations were made in cooling mode.

Because the pipes for cooling in a real heat-exchanger covers almost whole area of the fin (see fig. 2), temperature of the fins was set to 9 °C as a constant value. Room temperature of ingoing air is 22 °C. Inlet velocity is 3,65 m/s in case of Re = 500 with initial fin spacing h = 2 mm. The calculations were made to the time of 1 second, and then continued for 0,6 second for time average values, all with the time step 0,001 second.

The configuration of boundary conditions is on fig. 5. Straight entering area has heat flux = 0 and has no effect to the temperature of ingoing air.

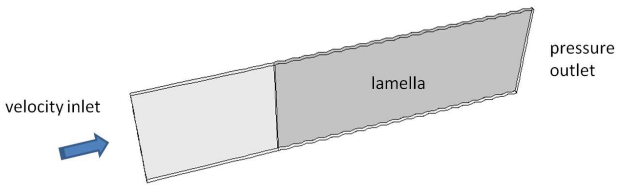

Fig. 5 configuration of boundary conditions

For the purpose of the results comparison were measured values of the average temperature and static pressure on several planes along the fins (fig. 6).

In case of the sinusoidal fin is first plane placed after 1st period, second after 5th period and third before last period of the sinusoid. Shape of angular fin respects the sinusoidal one.

The velocity and temperature field is compared on the plane placed simultaneously to the direction of the flow in the centre of the model.

If we want to maintain constant Reynolds number in case of different fin spacing, we have to change the inlet velocity. Velocity decreases from 3,65 m/s in case of 2 mm spacing, to 1,83 m/s in case of 4 mm spacing.

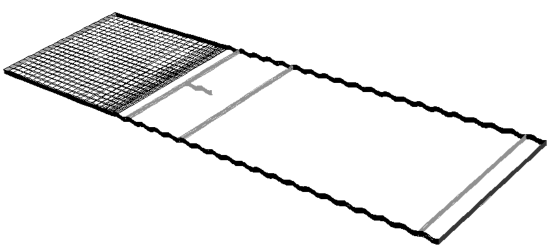

Fig. 6 3 monitored planes inside model

IV. Results

First part of results presents contours of velocity field in case of sinusoidal fins. Inlet velocity v = 3,62 m/s. On the fig. 7 we can see 5 different fins with spacing from 2 mm to 4 mm.

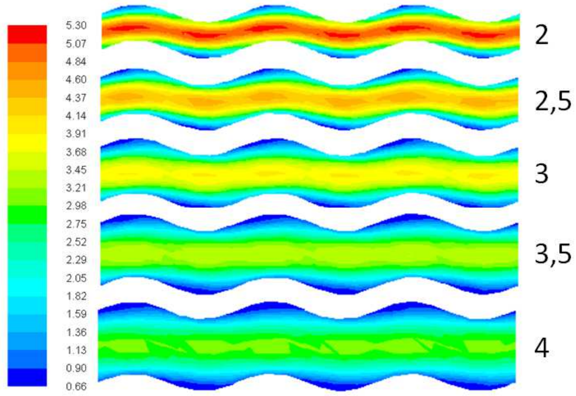

Fig. 7 velocity field for sinusoidal fins

Velocity profiles are time averaged. With expansion of the gap between fins we can observe straightening of the main flow as the influence of the shape of the fin to it decreases.

Second part of results presents contours of velocity field in case of angular fin. On the fig. 8 we can see 5 different spacing of fins.

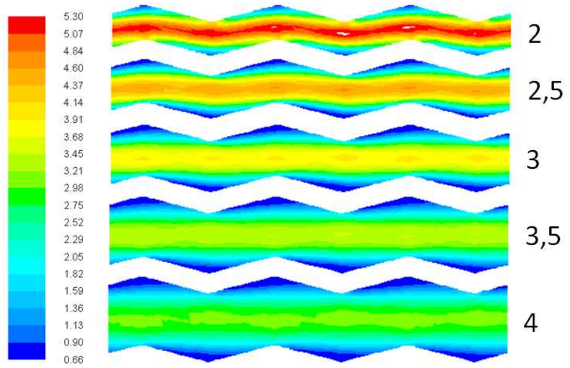

Fig. 8 velocity field for angular fins

Shape of the fin affects the shape of main flow mostly at lower fin spacing. With comparison to the sinusoidal fin is effect of straightening of main flow at higher fin spacing slightly intensive at angular fin. In case of 4 mm spacing we can observe minimal influence of the shape to the flow.

On fig. 9 we can see contours of temperature for set of sinusoidal fins. Observed area starts after 1st period of the fin because at this part of the fins we can observe high temperature differences. As velocity declines with higher fin spacing, cooling ability in observed area of fin decreases. This effect is represented by the higher temperature fields along the fins on fig. 9. Maximal temperature is limited to the 288 °C which is displayed by red colour. Higher temperature is presented by a white area.

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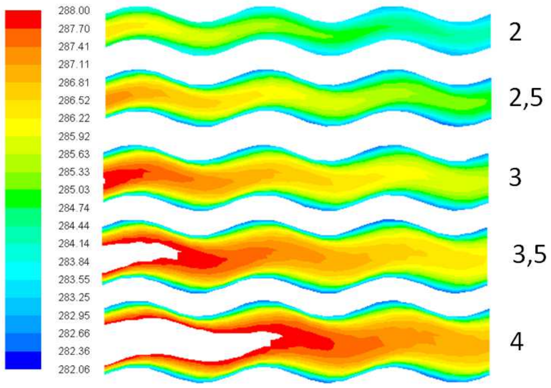

Fig. 9 temperature field for sinusoidal fins

On fig. 10 are presented results for angular shape of the fin. The best cooling ability at this part of fin is achieved by 2 mm spacing.

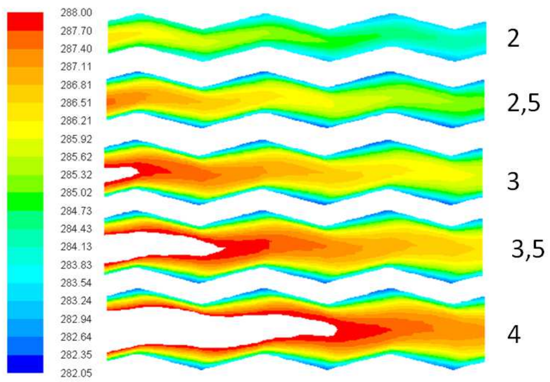

Fig. 10 temperature field for angular fins

If we compare results from angular and sinusoidal fin for spacing 4 mm, we can observe that in case of sinusoidal fin is cooling ability slightly better than at the angular fin, approximately by ½ T (period). The same development we can observe at 3 and 3,5 mm spacing.

On the fig. 11 we can see graphs of time averaged temperature development for sinusoidal fin. Values are measured on plane placed in the center of the fin and *x* coordinate goes from 0 to 154 mm (length of fin). Inlet temperature is 295 K and decreases to nearly 282 K at the end of the fin.

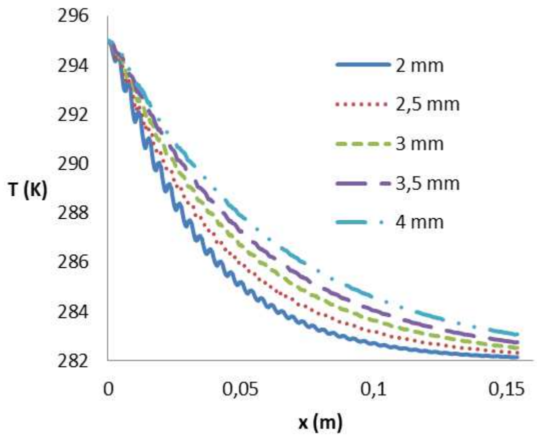

Fig. 11 temperature development along the sinusoidal fins

The most inthensive temperature drop is at 2 mm spacing. With increasing of distance between fins temperature rises in the whole part of the fin and result to the higher temperature value at the fin output. The highest temperature difference appears approximately at one half of the fin length.

On fig. 12 is presented temperature development for angular shape of the fin. Fluctuation of temperature in case of 2 mm spacing was caused by fin shape effect to the air flow between fins at lower spacing. As distance between fins increases, this effect disappears.

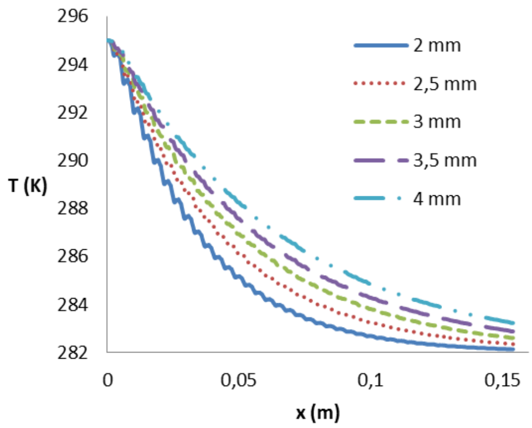

Fig. 12 development of temperature along the angular fins

Temperature development of angular and sinusoidal fin looks similar, but in case of higher spacing shows sinusoidal fin slightly lower temperature in direction of the flow.

On the fig. 13 and 14 are presented pressure drops along the fins in case of sinusoidal and angular fin for 5 different spacing values.

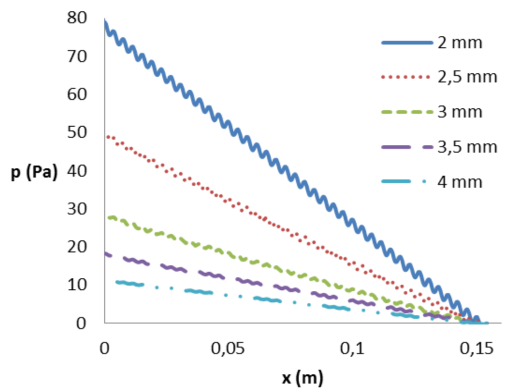

Fig. 13 pressure drop along the sinusoidal fins

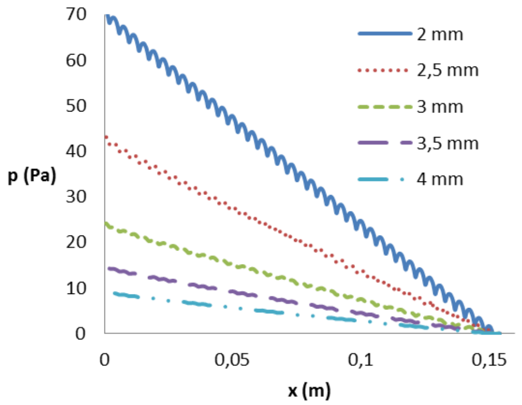

Fig. 14 pressure drop along the angular fins

We can observe that initial pressure depends on the distance between fins. With higher distance initial pressure decreases. Pressure fluctuation at 2 mm spacing is also caused by fin shape effect to the air flow. Initial value of pressure drop is higher in case of sinusoidal fin.

TABLE I
THEORETICAL COOLING OUTPUTS FOR SET OF SINUSOIDAL AND ANGULAR FINS

	t_{out} (K)	m (kg/s)	Q (W)
sin 2	282,15	5,9E-04	7,62
sin 2,5	282,29	5,9E-04	7,53
sin 3	282,48	5,9E-04	7,42
sin 3,5	282,70	5,9E-04	7,29
sin 4	282,98	5,9E-04	7,13
ang 2	282,14	5,9E-04	7,63
ang 2,5	282,32	5,9E-04	7,52
ang 3	282,55	5,9E-04	7,38
ang 3,5	282,79	5,9E-04	7,24
ang 4	283,08	5,9E-04	7,06

Cooling output decreases with higher fin spacing. In case of higher spacing is cooling output of sinusoidal fin slightly better than output of angular fin.

V. Conclusion

The spacing between fins has significant effect to the temperature field inside heat-exchanger and it also affects theoretical cooling output of the fin pair.

With higher spacing temperature rises inside heat-exchanger and theoretical cooling output is decreasing. On the other side initial pressure decreases with higher spacing to and resistance to the fluid flow lowers to.

Determination of the optimal fin spacing for this type of floor heating convectors will be the objective of future simulations and measurements.

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