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COMPUTER FLOW SIMULATION THROUGH A SINGLE COOLING UNIT IN THE CONTEXT OF ITS CONSTRUCTION

Summary

The article includes the results obtained by the fluid flow simulation through a single candy cooling unit. The aim of this investigation was to verify the tabletop construction. It was found that there is a necessity for redesigning the original construction. The introduced modifications allowed reduction of the stagnation areas nearby the turn canals and the heat exchanger corners.
Key words*: candy mass cooling, cooling module, cooling fluid, simulation research*

KOMPUTEROWE BADANIA SYMULACYJNE PRZEPŁYWU CIECZY CHŁODZĄCEJ W ASPEKCIE BUDOWY KONSTRUKCJI MODUŁU CHŁODZĄCEGO

Streszczenie

W artykule zamieszczono wyniki przeprowadzonych badań symulacyjnych przepływu cieczy chłodzącej na przekroju pojedynczego modułu chłodzącego stołu do schładzania mas cukierniczych. Badania przeprowadzono na potrzebę weryfikacji konstrukcji stołu. Wykazano, że zaistniała konieczność wprowadzenia zmian w konstrukcji, które pozwoli na zmniejszenie stref stagnacji przepływu w okolicach kanałów nawrotnych oraz narożników płaszcza wodnego.
Słowa kluczowe*: schładzanie mas cukierniczych, moduł chłodzący, ciecz chłodząca, badania symulacyjne*

1. Introduction

Soft candies, such as „irysy”, „krówka” or „toffi” have been manufactured from ages. The production process is based mainly on the two steps: preparation of a sugar mass and cooling the mass down to adequate temperature in order to obtain desired outward appearance. Up to now, a mass has been cooled down on a cooling table in an open heat transfer system with a single use of coolant. The main drawback of this solution is high costs generation by high coolant consumption. Taking into consideration the environmental aspect, there is a necessity for a new table construction which will allow applying a closed heat exchange system. To evenly balance and increase the heat exchange rate between the sugar mass and the cold fluid, the table is equipped with a single embossed pillow-plate heat exchanger. This type of heat exchanger can be manufactured in the following steps: a thick and a thin plate are welded together, then, by use of water or compressed air, under 2-100 bar [2] a thin plate is being inflated until the required shape is obtained. A cross section of a plate after inflation is shown in fig. 1.

2. Scope of the research

In the study, the fluid flow simulation through a single cooling unit was conducted. Obtained data provided crucial information about the tabletop geometry influence on the fluid flow. As a result of the research the stagnation zones were indicated which may have a great influence on heat exchange intensity.

3. Virtual model

The cooling tabletop was analyzed using the Finite Element Method (FEM) in COMSOL Multiphysics software. The primal tabletop geometry, with circular and linear welds is shown in fig. 2.

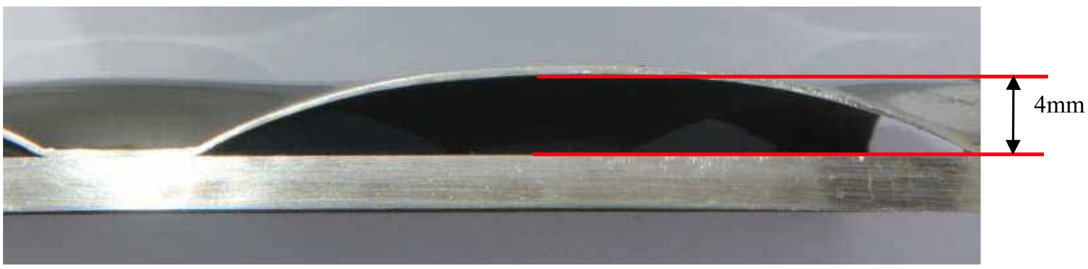


Fig. 1. Cross section of the single embossed pillow plate panel [6]
Rys. 1. Przekrój płaszcza „pillow-plate”[6]

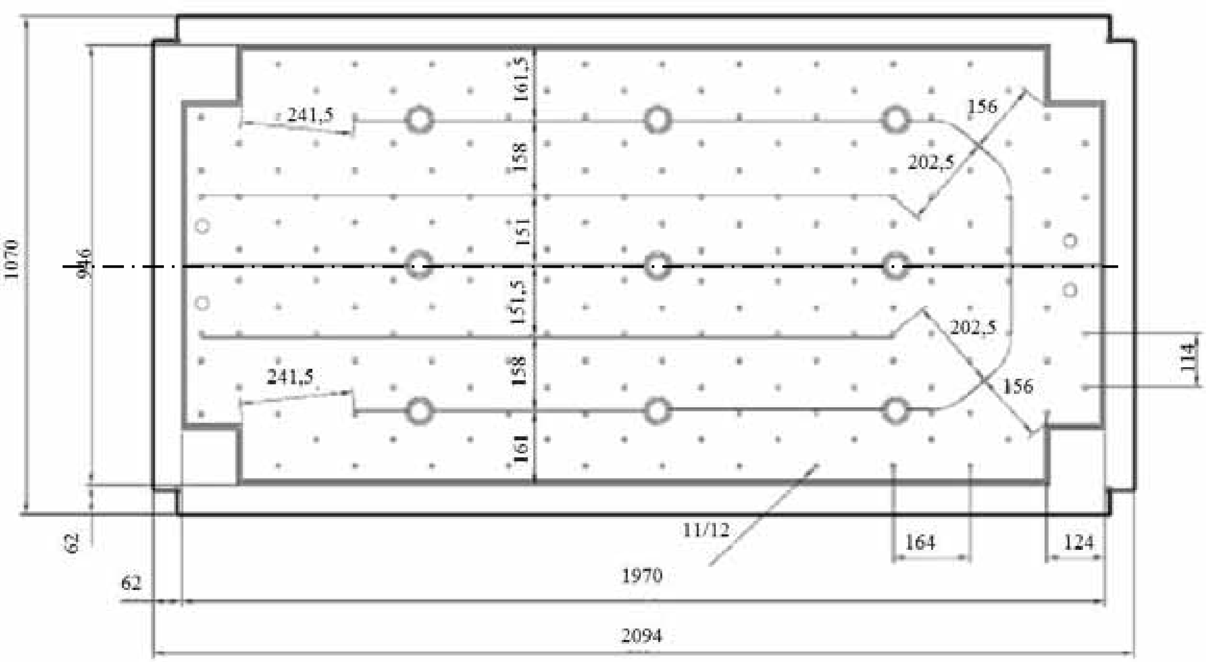
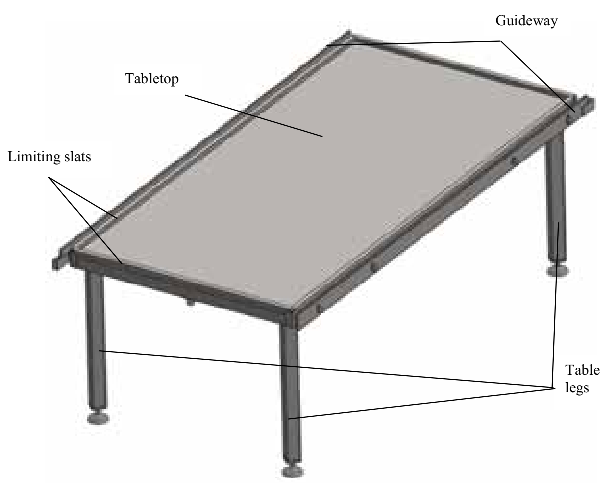


Fig. 2. The geometry of the confectionery cooling tabletop [1]
Rys. 2. Geometria blatu stołu do schładzania mas cukierniczych [1]

The virtual model of the candy cooling table with a pillow-plate type heat exchanger is shown in fig. 3.

a)



b)

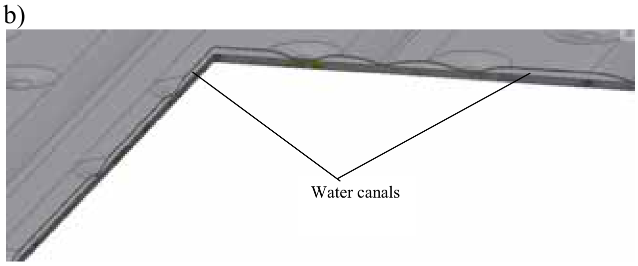


Fig. 3. The virtual model of the confectionery cooling table; a) general view of the virtual mode, b) cross section of the cooling plate [1]
Rys. 3. Wirtualny model stołu do schładzania mas cukierniczych; a) widok ogólny stołu, b) częściowy przekrój przez płytę chłodzącą stołu [1]

4. Calculation model and study methodology

As a result of the cooling table unit symmetry, along its longer side (fig. 2), only one part was subjected to the numerical simulations. Assigned boundary conditions are listed below:

- Constant input velocity,
- Pressure at the outlet p = 0 Pa,
- Flow fully developed.

For the purpose of numerical simulations, following assumptions and simplifications were established:

- Water with constant density and viscosity is used as a coolant,
- Fluid is incompressible,
- Newtonian fluid,
- Isothermal flow conditions,
- Steady-state flow,
- Turbulent flow with standard turbulence models for boundary layer,
- 1.4301 stainless steel roughness s = 0,046 mm.

Mentioned assumptions provide the following governing equations [3, 5]:

- Navier – Stokes equation (1):

$$\begin{aligned}
\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)&=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)\\
\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)&=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)\qquad(1)\\
\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)&=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)
\end{aligned}$$

- Continuity equation (2):

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)+\frac{\partial}{\partial z}(\rho w)=0 \qquad (2)$$

The cooling tabletop virtual model with pillow-plate heat exchanger, used in computational fluid simulation is shown in fig. 4. Arrows indicates the flow direction in the cooling unit.

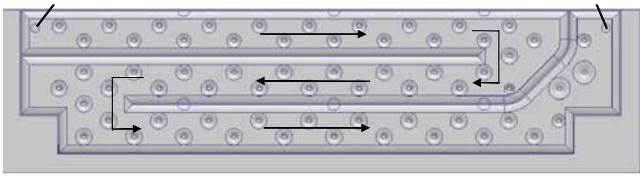


Fig. 4. Virtual model of the tabletop used in the flow simulation [1]
Rys. 4. Wirtualny model blatu wykorzystany do symulacji przepływu [1]

Figure 5 shows upper view of the discretized volume of fluid which fills the space limited by pillow-plate. The FEM grid is concentrated near the circular welds and turn canals. It is made of 3-D tetrahedral elements.

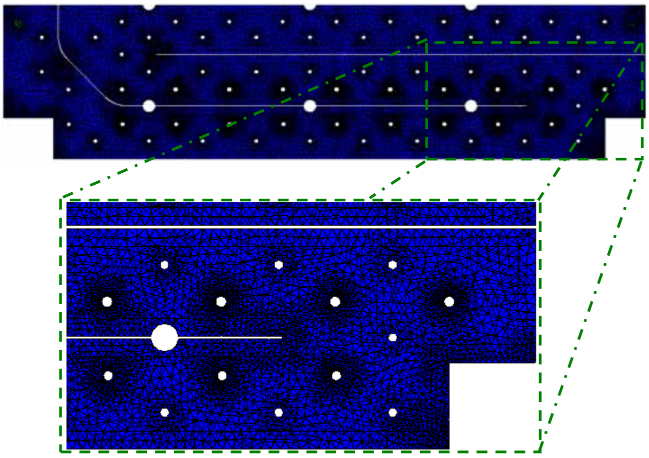


Fig. 5. The calculation model of a cooling fluid (top view and an enlarged view near circles welding and a turn zone) [1]
Rys. 5. Model obliczeniowy cieczy chłodzącej (widok od góry oraz powiększony widok w pobliżu spoin kołowych oraz strefy nawrotu) [1]

5. Results

The results of numerical simulations for inlet velocity $v = 1\ m \cdot s^{-1}$, for symmetrical part of the tabletop are shown in fig. 6. Obtained results indicate the stagnation areas near the turn canals and the bottom corners of the tabletop. The main cause of the stagnation areas occurrence are too wide turn canals and sharp ended corners of the pillow-plate directed to the flow. As a result fluid flows through too wide canals, in which, according to Bernoulli equation, fluid velocity decreases. Additionally, the sharp pillow-plate corners impede the flow near them.

The existing stagnation areas may affect the heat transfer intensity during a cooling process. Heat transfer, in the following table is due to conduction and convection (ignoring radiation). In the case of conduction, the heat transfer is heavily dependent on (among others) the temperature gradient and the tabletop and pillow-plate material properties. On the other hand, in convection, beside heat exchanger construction, fluid density, viscosity etc. the fluid velocity has a great influence on the heat transfer. Along with velocity rise, in a specific range, the convection coefficient rise as well, simultaneously improving the heat exchange [4].

The conducted analysis showed that the table construction had to be changed. One solution, among others, can be the linear welds lengthening and pillow-plate corners changing. According to this suggestion, new calculation model was prepared, in which the turn canals width is about 20% smaller then the longitudinal canals width. Additionally, the pillow-plate corners geometry were changed. The new table geometry was then subjected to analysis. Obtained results are shown in fig. 7. It can be seen, that stagnation areas were significantly reduced. In the cross-section X = 0,18 (fig. 6) the resultant velocity is equal to $0{,}5\ m \cdot s^{-1}$, whereas after introduced changes the velocity increases, reaching $1{,}2\ m \cdot s^{-1}$ (fig. 7). Analogous tendency was observed in the second cross-section. Here value of the velocity increased from 0,39 to $1{,}63\ m \cdot s^{-1}$.

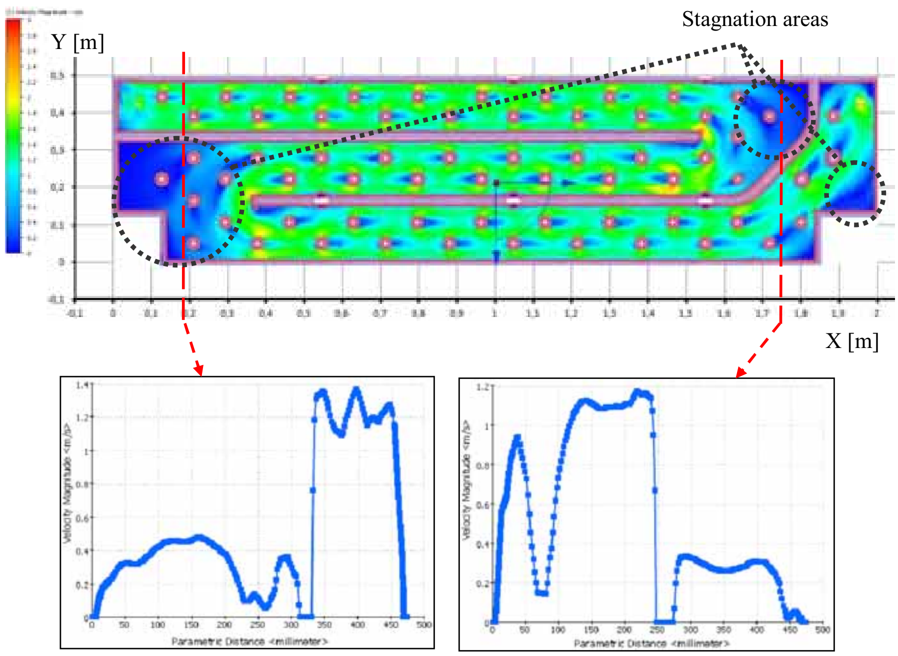


Fig. 6. The fluid velocity distribution in the cooling system and velocity profiles in sections of concern [1]
Rys. 6. Rozkład prędkości cieczy w układzie chłodzenia oraz profile prędkości wypadkowej w rozpatrywanych przekrojach [1]

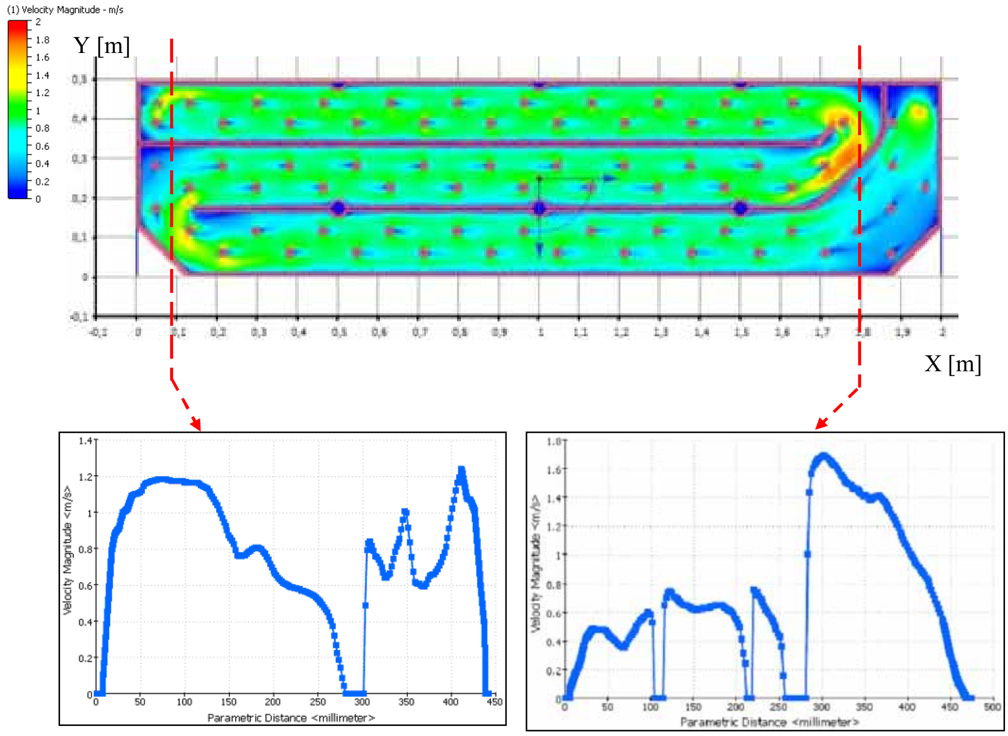


Fig. 7. The fluid velocity distribution in the cooling system and velocity profiles in sections of concern [1]
Rys. 7. Rozkład prędkości cieczy w układzie chłodzenia oraz profile prędkości wypadkowej w rozpatrywanych przekrojach [1]

6. Conclusion

The primal cooling tabletop had some significant construction drawbacks due to the presence of the stagnation areas which could heavily influence the heat transfer intensity. After introducing mentioned changes, concerning the turn canals and the pillow-plate corners, the stagnation areas were significantly reduced.

7. References

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