Influence of mesh quality and density on numerical calculation of heat exchanger with undulation in herringbone pattern

Václav Dvořák, Jan Novosád

***Abstract*—** Research of devices for heat recovery is currently focused on increasing the temperature and heat efficiency of plate heat exchangers. The goal of optimization is not only to increase the heat transfer or even moisture but also reduce the pressure loss and possibly material costs. During the optimization of plate heat exchangers using CFD, we are struggling with the problem of how to create a quality computational mesh inside complex and irregular channels. These channels are formed by combining individual plates or blades that are shaped by molding, vacuum forming, or similar technology. Creating computational mesh from the bottom up manually is time consuming and does not help later optimization. The paper presents a comparison of results obtained from numerical simulations using meshes created by two different ways. The first way is creating the mesh manually. This method is quite slow and difficult. It is necessary to create new mesh for each variant of heat exchange surface. The second way is creating meshes based on dynamic mesh method provided by software Fluent. Creating of mesh by pulling is similar to the own production process, i.e. it is perpendicular to the plates. The paper discusses the differences in results of numerical simulations using meshes creating by different methods with various element size. It was found that generally finer meshes bring lower obtained efficiency and lower pressure loss.

***Keywords*—** Dynamic mesh, heat exchanger, CFD.

I. Introduction

The development of recuperative heat exchangers in recent years focused on increasing efficiency. Another challenge is the development of so-called enthalpy exchangers for simultaneous heat and moisture transport, i.e. transport of both sensible and latent heat, as presented by Vít et al. in work [1].

To simulate a heat exchanger, we have to create a model and a computational mesh and use computational fluid dynamic (CFD) software. By assembling the heat exchanger, complicated and irregular narrow channels are created. These channels are split into small volumes (elements). Final mesh should be structured or unstructured with different element size.

A lot of others researchers dealt with design and investigation of performance and pressure drop of plate heat exchangers based on numerical simulations. Most of them used the unstructured mesh for calculations.

Gherasim et al. in work [2] presented the comparison of various grids for plate heat exchanger modelled by tetrahedral mesh. In order to assess the influence of the grid resolution on the solution, five grids were created and tested by meshing the volumes with different interval sizes. The laminar and turbulent regimes were simulated. The evolutions of the average pressure and average temperature of the hot fluid over transversal sections along the length of the plate was investigated. In general, the differences between the series for the turbulent case are larger than those for the laminar case. It was founded that the two grids with smallest elements give very close results. In terms of temperature the obtained results were closed for grids with smaller elements. For the pressure, there were founded a quite large difference between the grid with smallest elements and the grid with the largest ones.

There are some next researchers who dealt with numerical simulations of plate heat exchangers with the chevron (undulated) profile. E.g. Tsai [3], Liu [4] dealt with these heat exchangers with different geometries. The conclusions about temperature and pressure drop were similar to Gherasim [2].

Novosád in work [5] investigated the influence of oblique waves on the heat transfer surface. The biggest problem in this work was the creation of custom geometry. Each option had to be modeled separately and a meshed. Each model had to be loaded into the solver, set the boundary conditions and subsequently evaluated by calculation.

Disadvantages of repeated generation of computational meshes are: It is slow, meshes made in different models are not similar and parameterization of the model is problematic. Further, even a small change of geometry requires to go through the whole process of model creation and mesh generation again. As a result, there is high probability of creation errors of model and low quality of mesh cells. It is necessary to setup the solver, boundary conditions and all models for all computed variants. Furthermore meshes are not similar, i.e. the size, shape, height of wall adjacent cells are not the same for different topologies.

Therefore Dvořák in works [6] and [7] developed a new

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method for generation of computational variants. This method was based on dynamic mesh which is provided by software Fluent. Meshes were created by pulling, which is similar to the own production process, i.e. it is perpendicular to the plates. The main advantages is that such generation of variant is automatic and controlled by in-house software. All computational variants thus have similar mesh.

The aim of this work is to compare numerical results obtained by using two different ways of generation of computational meshes and to find correct mesh parameters to gain accurate numerical results.

II. Methods

A. Methods of grid generation

In this paper, we discuss the case of a counter flow heat exchanger, which does not have a symmetrical heat transfer area. It is caused by undulations in herringbone pattern. Processes in such heat exchanger can be investigated by modeling the flow around at least two plates using periodical boundary conditions. The first plate has undulations inclined by angle α, while the second one by angle $-\alpha$. How such a model appears can be seen from Fig. 1 and Fig. 2.

In this work we investigated 13 cases of undulations, three different angles of 30°, 45° and 60° and five different pitches of (4, 5, 6, 7 and 8) mm. Definitions of the pitch and angle of undulation are obvious from Fig. 1.

There were 30 waves on each plate. The width of the model B was adjusted for each variant of pitch p and angle α to maintain four crossings of wave peaks of both plates across the model.

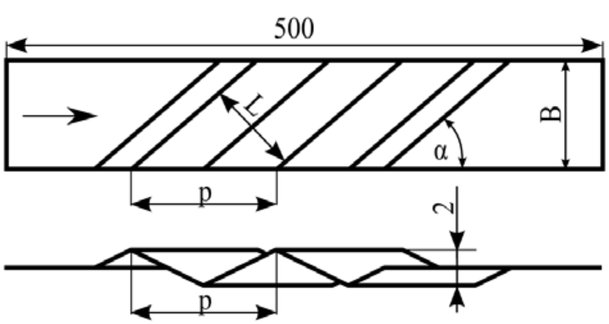

Fig. 1 Plate with oblique ridges - sketch, count of ridges n = 2.

The numerical model is in Fig. 2. The heat transfer surface is divided into two parts. Input and output portions (reported as wall) is fixed, and serve to develop the velocity profiles before the central portion (main wall) which will be provided by undulation. Input boundary conditions are specified by velocities (velocity inlets), the output boundary conditions are specified as pressure outlets with static pressure 0 Pa.

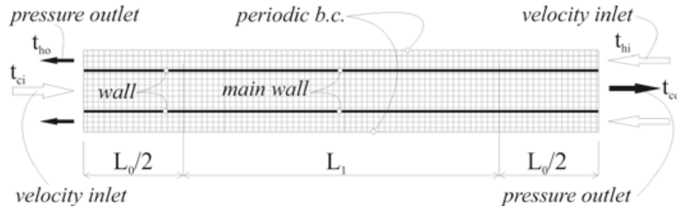

Fig. 2 Model of heat exchange surface of counter flow recuperative heat exchanger.

In this study, we used turbulence model SST κ-ω, medium was air considered as incompressible gas. As a results, we obtained pressure, velocity, turbulence and temperature fields inside the computational domain for average inlet velocity of air 2.5 m/s.

Creating computational meshes is based on two different methods. First method is creating the mesh manually. Second method is based on dynamic mesh method provided by software Fluent. The algorithm for mesh creating was described by Dvořák in work [6], [7].

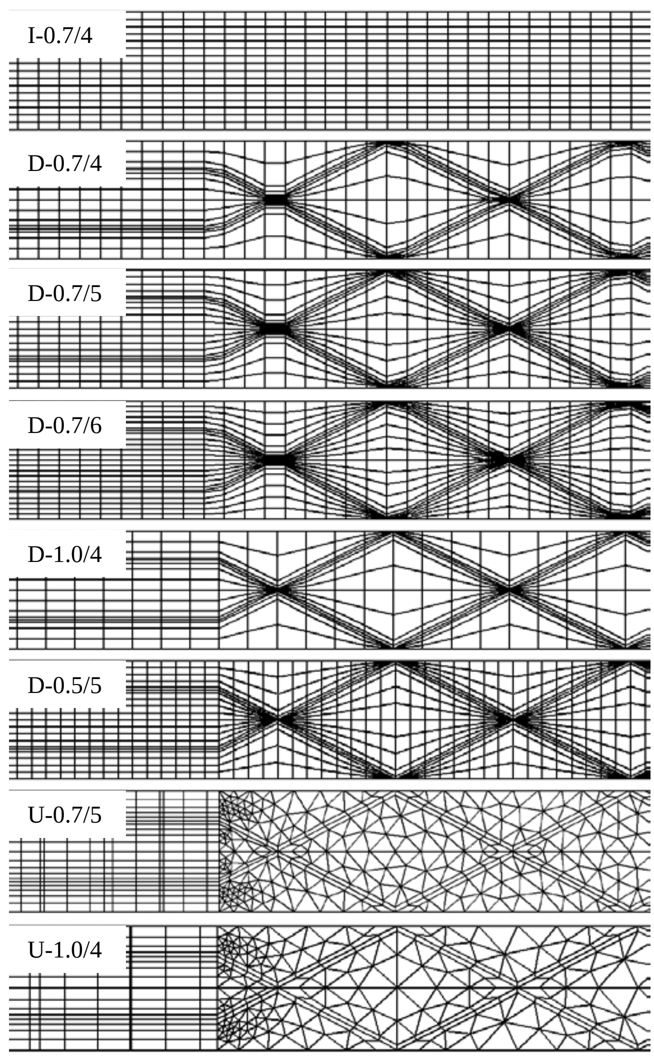

Fig. 3 Computational meshes, from up to bottom – initial mesh before deforming (I-0.7/4), meshes after deforming (D-0.7/4, D-0.7/5, D-0.7/6, D-1.0/4, D-0.5/5, unstructured meshes (U-0.7/5 and U-1.0/4).

Meshes of various density and element size were created. We used very coarse mesh with element size of 1.0 mm with 4 layers of computational cells across the half of the channel, middle fine meshes with element size of 0.7 mm with 4, 5 and 6 layers and very fine mesh with element size of 0.5 mm and 5 layers of computational cells.

We also used two meshes generated manually, a coarse one with element size of 1.0 mm with 4 layer of computational cells and a middle fine mesh with element size of 0.7 mm with 5 layers of computational cells.

To compare obtained numerical results, all meshes had the same width of wall adjacent cells which was 0.1 (mm) and value of y+ was around 1.5. Comparison of used meshes are in Fig. 3, where cuts of computational meshes are for undulation with angle of 30° and pitch of 4 (mm).

Fig. 4 shows the comparison of the mesh after deforming and unstructured mesh in isometric view, where the meshes are for undulation with angle of 45°, element size of 0.7 (mm), and pitch of 5 (mm).

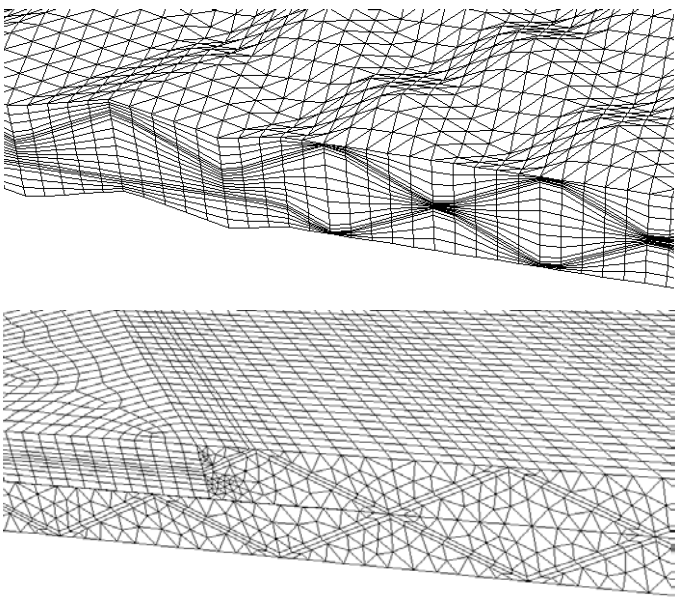

Fig. 4 Computational meshes, from up to bottom –mesh after deforming (D-0.7/5), unstructured meshes (U-0.7/5).

We should mention limitation of both methods. For deformed meshes, it is not possible to ensure that all peaks of undulations are modelled properly, see variants D-0.7/5, D-0.7/6, D-1.0/4 in Fig. 3. Meshes generated manually are generated by pulling meshes, see variants U-0.7/4 and U-1.0/4 in Fig. 2, in direction given by angle α, which means that the quality of the mesh decreases for low angles of undulation.

B. *Theory of counter flow heat exchangers*

Most of the recuperative heat exchangers in air conditioning works in the isobaric mode, where mass flow rates of warm and cold air are equal, i.e. $\dot{m}_c = \dot{m}_h$. Assuming equality of specific heat capacities, $c_{pc} = c_{ph}$, we can write the coefficient of efficiency as

$$\eta = \frac{t_{hi} - t_{ho}}{t_{hi} - t_{ci}}, \tag{1}$$

where t_{hi} (°C) is the inlet temperature of hot air. Furthermore index c denotes a cold stream, index i inlet into the heat exchanger and index o the outlet of the heat exchanger.

For the pressure drop assessment, it is used the ratio of total pressure loss between the inlet and outlet

$$\Delta p = \overline{p}_i - \overline{p}_o, \tag{4}$$

where $\overline{p}_i$ (Pa) is mass averaged total pressure in the inlet and $\overline{p}_o$ (Pa) is mass averaged total pressure at the pressure outlet.

The dependence between the heat balance and efficiency η is expressed as

$$k\ A = \dot{m}\ c_p \frac{\eta}{1-\eta} \tag{5}$$

where $\dot{m}$ (kg s^{-1}) is the mass flow rate, c_p (J kg^{-1} K^{-1})) is isobaric specific heat capacity, k (W m^{-2} K^{-1})) heat transfer coefficient and A (m^2) is the area of heat transfer surface.

III. Results

Dependency of efficiency on the pitch of undulation for various angles of undulation is shown in Fig. 5, Fig. 6 and Fig. 7. These figures describes the effect of mesh type and elements quality on efficiency value. Mesh types created by deforming (dynamic mesh) are labelled by letter "D". Unstructured meshes are labelled by letter "U". Element size and the count of layers of computational cells across the channel are labeled by numbers, e.g. "0.5/5" means that the element is 0.5 mm and there are 5 layers across the channel.

We can see that the efficiency increases for higher pitch of undulations for all angles, because the air flows around the undulations in better way for high pitches.

Generally we can see that difference between all meshes are more significant for lower pitch of undulation.

As we can see there is significant difference between the case "D-1.0/4", which is the coarsest mesh, and other cases. The efficiency value for this case is higher than for the other ones, so we can assume that this mesh variant is not suitable to use for computations. That could be caused by poor mesh quality, while comparing other results the wall friction is overvalued.

We can observe, by comparing cases for element size 0.7 mm, the effect of count of cell layers. Lower efficiency and so lower heat transfer coefficient is obtained for higher count of computational cells across the channel. Remind that all cases

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have the same width of 0.1 mm of adjacent cells and the deformed cases have also the same width of the second cell of 0.3 mm.

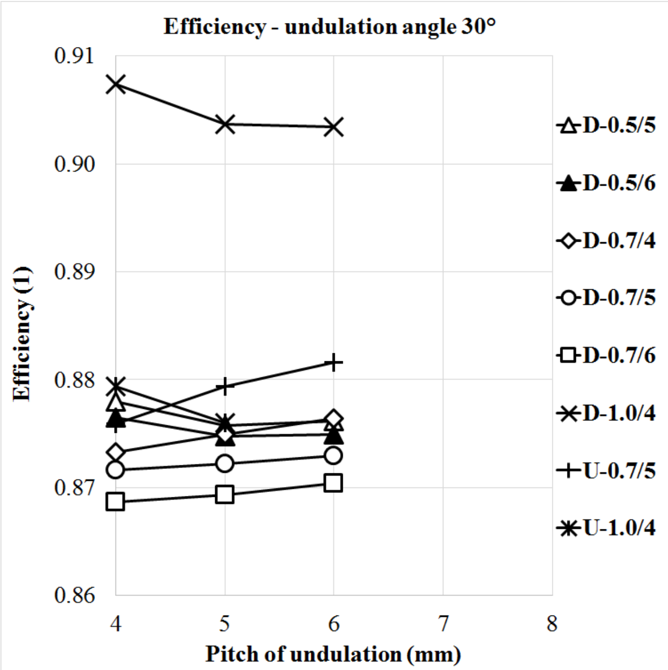

Fig. 5 Dependency of the efficiency of the heat exchanger on pitch of undulations for undulation angle 30°.

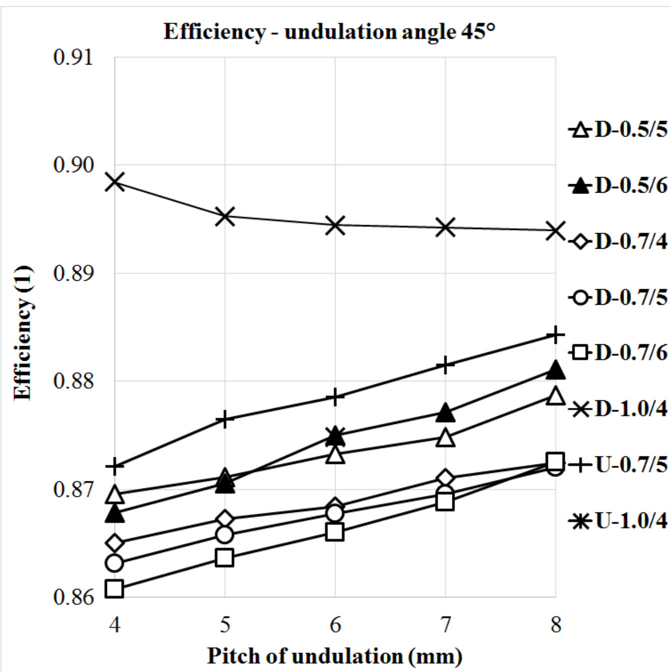

Fig. 6 Dependency of the efficiency of the heat exchanger on pitch of undulations for undulation angle 45°.

An important observation is that the best match of the efficiency values between the mesh creation ways for undulation angle 30° is for case “D-0.5/5” and case “U-0.7/5”.

As we can see from Fig. 6 and Fig. 7, the higher undulation angle causes higher difference between case “D-0.5/5” and case “U-0.7/5”. That could be caused by deformation of elements of unstructured mesh in the direction along the width of heat transfer area.

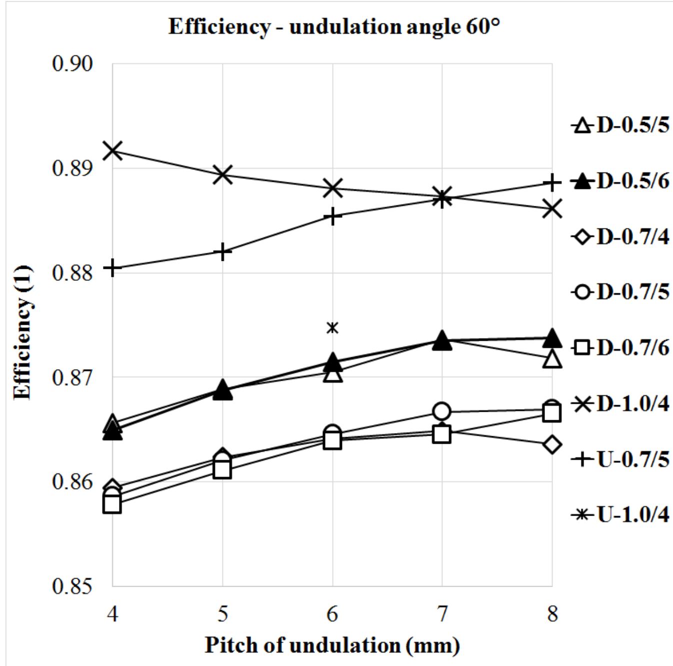

Fig. 7 Dependency of the efficiency of the heat exchanger on pitch of undulations for undulation angle 60°.

Dependency of pressure drop on the pitch of undulation for various angles of undulation is shown in Fig. 8, Fig. 9 and Fig. 10. These figures describes the effect of mesh type and elements size on pressure drop. Labels of each variant of mesh are done in the same way as for dependency of efficiency.

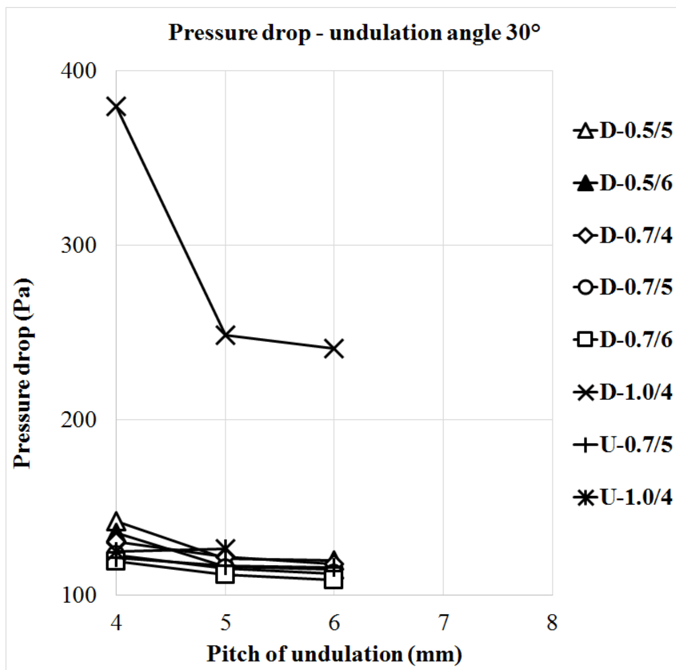

Fig. 8 Dependency of the pressure drop of the heat exchanger on pitch of undulations for undulation angle 30°.

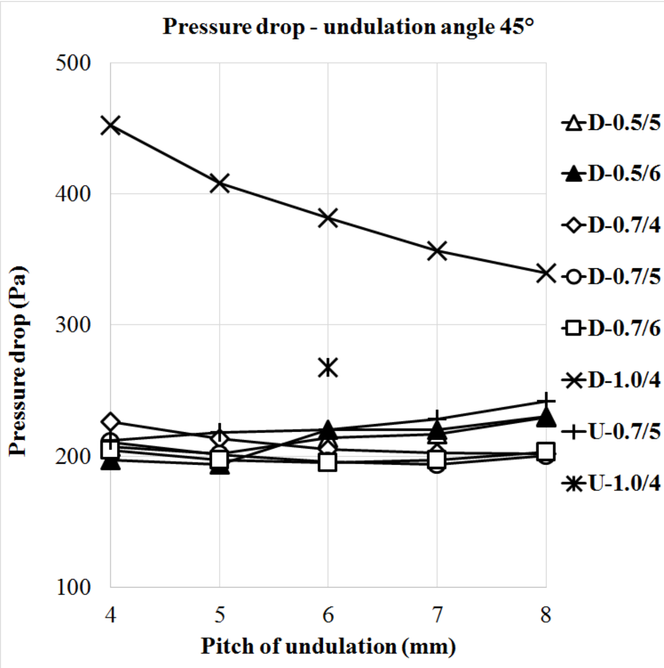

Fig. 9 Dependency of the pressure drop of the heat exchanger on pitch of undulations for undulation angle 45°.

As we can see there is significant difference between the case “D-1.0/4” and other cases for undulation angle 30° and 45°. The pressure drop for this case is higher than for the other ones.

As we can see from Fig. 8 and 9, pressure drop for all cases (except “D-1.0/4”) are in very narrow range. Generally the obtained pressure loss is lower for higher count of layers of computational cells and form finer meshes.

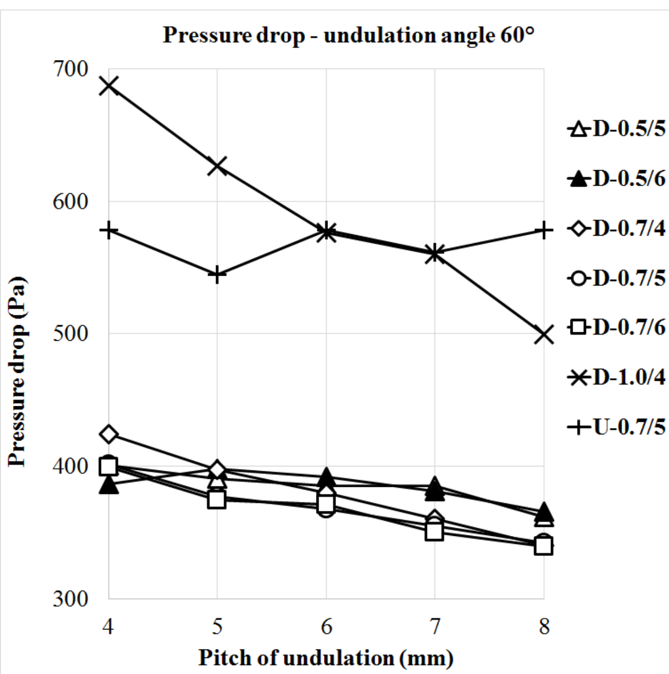

Fig. 10 Dependency of the pressure drop of the heat exchanger on pitch of undulations for undulation angle 60°.

Another situation is shown in Fig. 10. We can see there are two groups of results. A good match between the two coarsest cases “D-1.0/4” and “U-0.7/5”. It is possible that there are some effects which are coupled with the poor mesh quality. These effects caused higher pressure drop, e.g. we evaluated high wall friction compare to other cases.

The second group of results are for finer deformed meshes. As for efficiency, we can see that obtained pressure loss decreases with count of layers of computational cells, but it is higher for the finest mesh D-0.5/5. It seems that differences between variants D-0.7/5 and D-0.7/6 are negligible.

IV. Conclusions

We investigated flow in counter flow plate heat exchanger. We used CFD and two different method of generation of computational meshes and examined effect of computational cells size and count of layers of computational cells across the half of the channel. For accurate comparison, all meshes had the same width of wall adjacent cells. We used manually generated unstructured meshes and automatically generated dynamic meshes. We investigated flow around plates with various undulations with different angle and pitch of undulation and evaluated pressure loss and efficiency of heat transfer of the exchanger.

Generally the obtained efficiency and obtained pressure loss decreased with higher mesh density, i.e. with lower size of computational cells.

We found that very coarse meshes can yield both too high and unrealistic efficiency and pressure loss compare too other results. It seems that for deformed meshes, which are structured, the satisfactory size of elements is 0.7 mm, while this size can be unsatisfactory for unstructured meshes in some cases.

Comparing meshes with the same density, we can conclude that higher count of layers of computational cells brings lower efficiency and lower pressure loss. While there is still obvious difference in efficiency for counts of layers 5 and 6, it seems that 5 layers are enough to obtain the lowest pressure loss.

The finest structured mesh with element size of 0.5 mm and 5 layers brings slightly higher efficiency and pressure loss than structured meshes with element size 0.7. We can hardly state whether it is because of finer mesh or because of better reproduction of peaks of undulation. Therefore, it could be beneficial to have data also for finest mesh with higher count of layers.

Finally, from trends visible in our results, we can say that meshes with lower efficiency and lower pressure loss seem to be more correct than meshes yielding higher efficiencies and higher pressure losses. This is also confirmed by our general experience and may be explain by more significant effect of numerical dissipation while using coarser meshes. Of course, to confirm this statement, the numerical data should be compared with experiments.

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