Computation of Nusselt numbers in plate heat exchangers using CFD with Reynolds-averaged Navier–Stokes equations

Mads Kyed
Fachbereich Maschinenbau
Flensburg University of Applied Sciences
Kanzleistrasse 91-93, 24943 Flensburg
Email: mads.kyed@hs-flensburg.de

Nico Šubić*
Fachbereich Maschinenbau
Flensburg University of Applied Sciences
Kanzleistrasse 91-93, 24943 Flensburg
Email: nico.subic@hs-flensburg.de

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A CFD simulation based on a Reynolds-averaged Navier–Stokes model is employed to compute Nusselt numbers in corrugated plate heat exchangers. The approach is verified via the Dittus-Boelter formula in a simplified tube geometry. A corrugated plate is analyzed and the effect of a chevron angle on the Nusselt number is investigated.

Keywords: Nusselt number, corrugated plate, heat exchanger, RANS models.

1 Introduction

A plate heat exchanger (PHE) is a device in which two fluids separated by solid plates exchange heat. Corrugation patterns in the plates induce turbulence in the fluids, thereby increasing the convective heat transfer, which enables PHEs to achieve higher levels of efficiency compared to regular shell and tube heat exchangers. An accurate prediction of the heat transfer is critical in many applications, for example in optimal sizing of PHEs. The classical LMTD method for PHE sizing relies on the Nusselt number to incorporate the effect of turbulence on the convective heat transfer. Computation of Nusselt numbers in PHEs to date typically employ heuristic correlations like the Dittus-Boelter formula and derivatives hereof, in which the coefficients are empirically adapted from measurements. In this paper we propose to use turbulence models and CFD to compute the Nusselt number in a PHE. We first verify our computation method on a simple tube geometry, on which the Dittus-Boelter formula is known to yield a sufficiently precise prediction, and then employ the method to demonstrate the effect of a chevron angle on the performance of a PHE.

Our results show that a CFD based numerical scheme can be established to robustly compute the Nusselt number of a PHE from the Reynolds number, Prandtl number and geometric parameters that describe the corrugation patterns in the plates. Algorithms based on such schemes may be used to generate large data sets from which improved analytic formulas for the Nussselt number of a PHE can be constructed. We consider this paper a step on the way towards such a framework.

The influence of geometrical parameters on PHE performance has been investigated in previous works. Dović and Švaić [2] have analyzed the influence the length-width relation of a single plate and the corrugation angle have on the heat transfer coefficient in a PHE. Skočilas and Palaziuk [7] have investigated the influence of the corrugation angle on the heat transfer coefficient and on the pressure drop for a single channel. Dović, Palm and Švaić [3] developed a generalized approach to computation of the Nusselt number from empirical measurements and CFD simulations. That CFD can be a viable source of data was also shown by Luan et al. [5] who created a correlation between Nusselt, Prandtl and Reynolds numbers in ANSYS for a special type of PHEs, not including, however, geometrical parameters.

We use the OpenFoam framework to carry out CFD simulations, and GMSH to create meshes.

2 Model

We use the $k-\omega-SST$ model, a two equation turbulence model within the RANS models, coupled with the standard energy equation to compute the heat transfer performance. The turbulent length scale is given in terms of the hydraulic diameter.

Initial experiments showed that the $k-\omega-SST$ model yields more accurate Nusselt number computations in our setting than both the $k-\varepsilon$ and the standard $k-\omega$ model. The $k-\varepsilon$ model in particular proved to be inadequate with very large deviations. From the inferior behavior of the $k-\varepsilon$ model we conclude that the more detailed treatment of wall effects in the $k-\omega$ models are critical to Nusselt number computation in heat exchangers.

3 Simulation

Initial simulations were carried out for a flow in a smooth pipe. In this simplified geometry, the Dittus Boelter formula

$$Nu = 0.023 Re^{0.8} Pr^{n} \tag{3.1}$$

($n = 0.3$ for cooling and $n = 0.4$ for heating) is known to be valid when the relation between the diameter D and the length L is at least 10, and the flow is fully developed with Reynolds number greater than 10000 and Prandtl number $0.6 \leq Pr \leq 160$.

The geometry for the pipe flow was generated with GMSH. We employed a pure hexaeder mesh based on a 2D hex mesh of

Property	value
Mol weight	18 mol
Density	997 $\frac{\text{kg}}{\text{m}^3}$
Specific heat capacity	4180 $\frac{\text{J}}{\text{kgK}}$
Prandtl number	6.09

Table 1: Fluid properties water

a circle. We extruded this surface twice to generate an entrance region and a fully developed region, and used the extrusion length as entrance length for the flow. We used the Reynolds based approach to ensure an entrance length in correspondence with the flow development:

$$l_{entrance} = 1.359 D Re^{0.25}. \tag{3.2}$$

We employed the material properties for the fluid given in table 1. As boundary conditions we assumed a constant heat flux of $1000 \frac{\text{W}}{\text{m}^2}$ on the outer cylindrical boundary, a fixed temperature of $298.15K$ on the inlet, and an adiabatic outlet (zero temperature gradient). The zero gradient condition was also employed at the outlet for the velocity field. The inlet velocity was set to be constant and directed in direction of the pipe. Simulations with Reynolds numbers in the range $20000 - 40000$ with a step size of 5000 were carried out. High Reynolds number were chosen since the Dittus Boelter equation is only valid for $Re \geq 10000$. The Nusselt number was calculated in post processing from the area average temperature of the heated wall in the developed region and the bulk temperature of the fluid. With the prescribed heat flux at the walls we computed the heat transfer

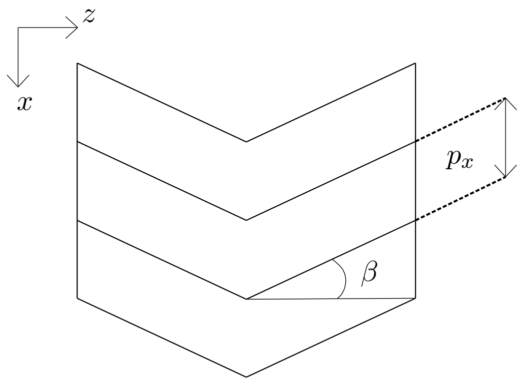

Figure 1: Geometric properties of the corrugation pattern

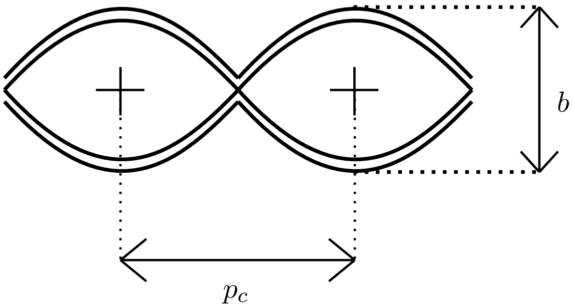

Figure 2: Geometric properties of the corrugation pattern

coefficient with

$$h = \frac{q}{t_{wall} - t_{fluid}}. \tag{3.3}$$

The heat transfer coefficient was used to obtain the Nusselt number

$$Nu = \frac{h d_{hyd}}{\kappa} \tag{3.4}$$

with

$$\kappa = \frac{\eta c_p}{Pr}. \tag{3.5}$$

For the plate simulations we employed a sinosodial profile extruded with corrugation angle β. The same scheme as for the pipe was used to compute the Nusselt number. Usually the hydraulic diameter of arbitrary geometries can be estimated with

$$d_{hyd} = \frac{4A_{channel}}{A_{contact}}, \quad (3.6)$$

where $A_{channel}$ represents the channel flow area and $A_{contact}$ the wetted surface. However [6] presents an approach based on the surface enlargement factor φ and the channel width b between two plates, which we employed with the additional relation for the corrugation pitch according to [4]

$$d_{hyd} = \frac{2b}{\varphi}, \quad (3.7)$$

where φ is given by

$$\varphi = \frac{1}{6}\left(1 + \left(1 + \left(\frac{\pi}{2cos(\beta)}\right)^2 \gamma^2\right)^{0.5} + 4\left(1 + \left(\frac{\pi}{2\sqrt{2}cos(\beta)}\right)^2 \gamma^2\right)^{0.5}\right) \quad (3.8)$$

and γ is the aspect ratio of the corrugation defined as

$$\gamma = \frac{2b}{p_x} \quad (3.9)$$

with p_x the corrugation pitch in main flow direction. To extract the Nusselt number from the simulation we employed again (3.3) and (3.4), with d_{hyd} given by (3.7). We simulated Reynolds numbers in the range $2500 - 4000$ with a step size of 500. Constant mass flows were used as boundary conditions on both the inlet and outlet. A constant heat flux of $500\frac{W}{m^2}$ was chosen as boundary condition on the fluid-plate interfaces, which corresponds to a total heat flux of $1000\frac{W}{m^2}$ similar to the pipe. The outer plate walls were assumed to be adiabatic. A no-slip boundary condition was used for the velocity field on the fluid-plate interfaces.

Re	Nu_{CFD}	Nu_{DB}	Deviation
20000	131.16	130.74	0.32 %
25000	157.19	156.29	0.58 %
30000	183.05	180.83	1.23 %
35000	208.01	204.56	1.68 %
40000	232.69	227.63	2.22 %

Table 2: Results of the simulation for the pipe

4 Results

4.1 Pipe

The results for the simulations are given in Table 2. We notice that the deviation from the Dittus-Boelter formula (3.1) is smaller for lower Reynolds numbers, which matches the results from [1]. Further we notice that the deviation is positive for every simulation. We deduce a slight overprediction of the turbulent effects, which is known to happen in the framework of $k - \omega$ models. In general the results show good agreement with the Dittus-Boelter formula (3.1), and we thus conclude that our scheme is viable to extract Nusselt numbers.

4.2 Corrugated plate

Concerning PHEs, we recall the following expected behavior:

- A steeper chevron angle reduces the pressure drop.
- A steeper chevron angle reduces the heat transfer coefficient and therefore the Nusselt number.

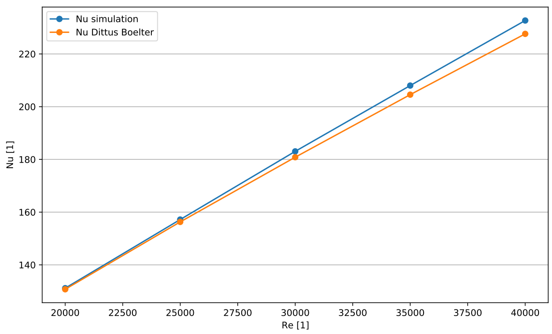

Figure 3: Nusselt numbers from simulation and from the Dittus Boelter formula

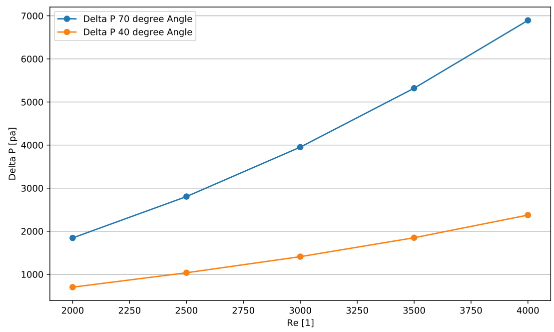

Figure 5: Pressure drop from simulations with a 70 and 40 degree chevron angle

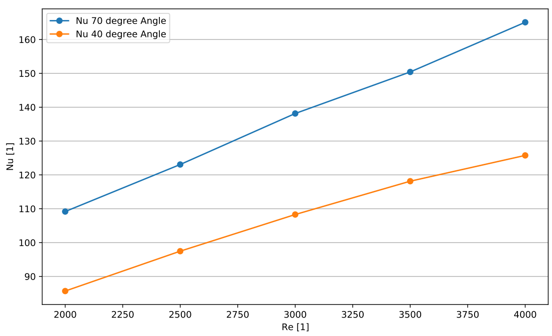

Figure 4: Nusselt numbers from simulations with a 70 and 40 degree chevron angle

- With increasing Reynolds number, the pressure drop and Nusselt number increase.
- With increasing Reynolds number, the increment in the Nusselt number for steeper angles is lower than that of broader angles.
- With increasing Reynolds number, the increment in pressure drop for steeper angles is lower than that of broader angles.

In our simulations, we find that the *Nusselt* number increases with increasing *Reynolds* number as expected. The same is observed for the pressure drop. The observed effect of changing chevron angles on the Nusselt number is also in accordance the expected behavior.

We note that a validation based on comparative measurements was not carried out. We intend to do so in future works.

5 Conclusion

We computed Nusselt numbers for PHEs with CFD simulations based on turbulence modelling. We first validated our scheme by computing the Nusselt number for a flow in a pipe and comparing the results with the Dittus Boelter formula. We then employed the algorithm on corrugated plates and investigated the effect of the corrugation angle on the Nusselt number, which we observed to be in accordance with expected physical behavior. We conclude that our scheme is valid.

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