IMPROVEMENT OF EFFICIENCY IN PLATE HEAT EXCHANGER BY CREATING DIMPLES IN CORRUGATED CHANNELS

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Abstract - The development of new heat exchanger patterns is ever changing. Companies are constantly pushing innovation to satisfy customer needs in terms of thermal efficiency, reliability, and serviceability. In this thesis, the implementation of dimples in the corrugated pattern of a plate heat exchanger is investigated to increase thermal efficiency. The thesis comprises three studies, moving from research of a dimpled surface to implementation of dimples on a real-life heat exchanger model used by Alfa Laval. Computational fluid dynamics was applied to model heat transfer, pressure drop and thermal efficiency through three Reynolds Averaged Navier Stokes (RANS) models and one Large Eddy Simulation (LES). The RANS-models were shown to provide insufficient resolution which heavily affected mixing, vortex formation and flow distribution in the heat exchanger channel. The RANS-model suggested that the thermal efficiency decreases when dimples are added, but failed to accurately capture key phenomena such as vortex formation and mixing. In the more advanced LES model, dimples were shown to increase the thermal efficiency by 18 % compared to a flat channel. This study not only provides insight into the flow behavior in dimpled heat exchanger channels, but also shows the importance of using the correct turbulence model for a given problem. The flow dynamics in a dimpled plate heat exchanger is complex, and a RANS-model is not sufficient for an accurate description of the velocity field. In this regard, LES performs better and the result of the LES simulation is therefore more trustworthy than results from the RANS. As no experimental data exists, it is difficult to compare models to reality, laboratory trials are necessary.

***Key Words*:** Flat Plate Heat exchanger, Dimples, Herringbone pattern, large eddy simulation, Thermal Efficiency, Nusselt number, Reynold number, CFD etc.

1.INTRODUCTION

The development of new heat exchanger patterns is ever changing. New products are constantly developed to satisfy customer needs in terms of thermal efficiency, reliability, and serviceability. To increase the thermal efficiency of a heat exchanger, the corrugated pattern is often an area of great focus, as this is where most of the heat transfer occurs (Alfa Laval Corporate AB. 2020). The usage of dimples in corrugated plate heat exchanger patterns is an area that has not yet been explored sufficiently, but recent scientific articles has sparked the interest of an investigation.

The design of a heat exchanger is an exercise in thermodynamics, which is the science that deals with heat energy flow, temperature, and the relationships to other forms of energy. To understand heat exchanger thermodynamics, a good starting point is to learn about the three ways in which heat can be transferred -conduction, convection, and radiation. A heat exchanger is a device, which transfers thermal energy between two fluids at different temperatures. In most of the thermal engineering applications, both fluids are in motion and the main mode of heat transfer is convection. Examples are automobile radiators, condenser coil in the refrigerator, air conditioner, solar water heater, chemical industries, domestic boilers, oil coolers in a heat engine, and milk chillers in pasteurizing plan. Heat exchanger, any of several devices that transfer heat from a hot to a cold fluid. In many engineering applications it is desirable to increase the temperature of one fluid while cooling another. This double action is economically accomplished by a heat exchanger. Among its sassier the cooling of one petroleum fraction while warming another, the cooling of air or other gases with water between stages of compression, and the preheating of combustion air supplied to a boiler furnace using hot flue gas as the heating medium .Other uses include the transfer of heat from metals to water in atomic power plants and the reclaiming of heat energy from the exhaust of agas turbine by transferring heat to the compresseda1r on its way to the combustion chambers .Heat exchangers are used extensively fossil-fuel and nuclear powerplants, gas turbines, heating and air- conditioning, refrigeration, and the chemical industry. The devices are given different names when they serve a special purpose. Thus boilers, evaporators, superheaters, condensers, and coolers may all be considered heat exchangers. Heat exchangers are manufactured with various flow arrangements and in different designs.

Transport in Plate Heat exchangers

Plate heat exchangers (PHE) find their main applications in liquid-liquid duties where high heat transfer is of importance. They are most used in the food industry such as dairy and beverage processing but also find use in paper mills, petrochemical plants, closed-circuit cooling systems and process heaters. The ability to remove, replace and clean individual plates and the separation of fluids by a physical wall makes the PHE ideal for sterile processing such as pasteurization. One of the major advantages of the PHE compared to the shell-and-tube heat exchanger is its flexibility. Individual plates can be taken out and changed to fit a different task through plate size, corrugation pattern and arrangement.

The heat transfer coefficient is much larger for a PHE compared to a shell-and-tube heat exchanger, which makes the required surface area about 1/3 for a given heat duty. This means that the PHE takes up much less space compared to a shell-and- tube heat exchanger, which can be an important factor as customers value space inside their factories (Shah, Subbarao, and Mashelkar 1986).

Operation

The plate heat exchanger consists of several rectangular plates held together with a frame. Each plate is pressed with a corrugated surface pattern on a piece of sheet metal. The corrugated plates are mounted successively as to form passages between the plates to guide the flow of media from one end to the other (Shah, Subbarao, and Mashelkar 1986). A

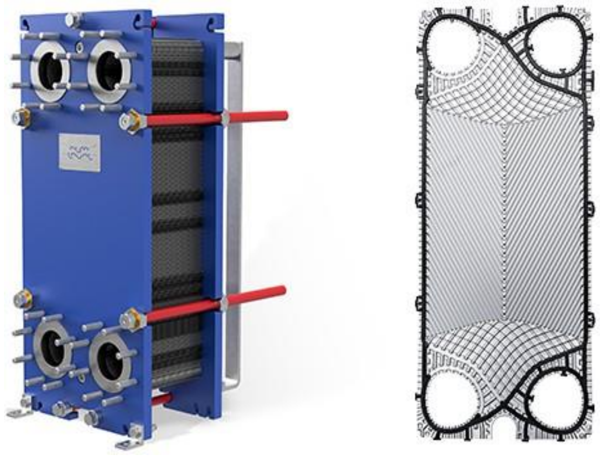

plate heat exchanger together with a corrugated pattern is shown in Figure 1 below.

Figure 1. An example of a PHE from Alfa Laval (left) and a corrugated plate (right).

The corrugated pattern provides mechanical support through multiple contact points as well as enhances the heat transfer by increasing the level of turbulence. There are many different plate patterns available today, one of which is the Herringbone design. The Herringbone design consists of corrugated plates inclined at an angle β to the flow direction. The assembly is done with every other plate pointing in the opposite direction, creating narrow flow passages. A figure of a herring bone pattern with an illustration of a flow passage is shown in Figure 2 below.

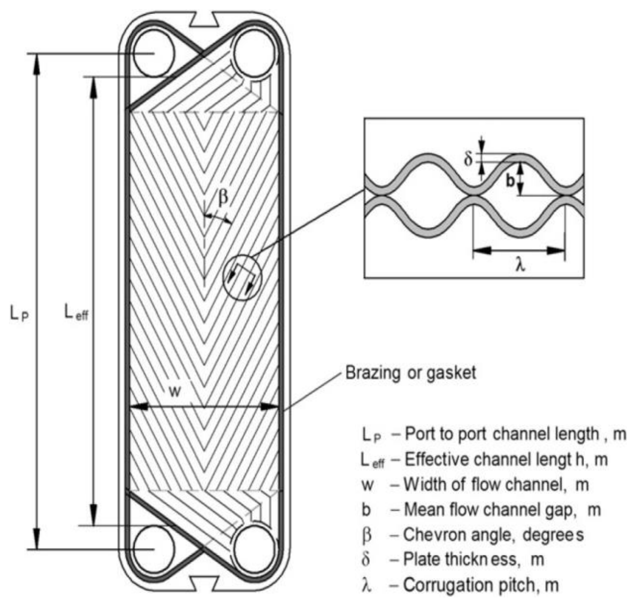

Figure 1.2 Herringbone pattern with corrugation angle β and illustration of flow channel geometry.

Several known mechanisms contribute to the high heat transfer coefficient inside PHEs. Any design containing protrusions and intrusions will cause boundary layer separation and reattachment which leads to earlier transition to turbulent flow. This will lead to enhanced heat transfer but with the cost of increased pressure drop. In herringbone design, the crisscrossing nature of the corrugations produces swirl flows which enhance mixing. This is one of the reasons that the herringbone pattern is generally superior to other designs in terms of heat transfer over friction and is the most used design today (Shah, Subbarao, and Mashelkar 1986).

2. Literature Review

Ferhat et al. 2019 [1] introduces a concept of" artificial roughness" to improve the heat transfer within corrugated channels. The goal of the paper was to investigate artificial roughness and how it affects the performance of plate heat exchangers. The results show a 50 % increase in thermal performance by using artificial roughness and added undulations compared to a flat channel. Several other studies have been conducted, also showing promising results (Chudnovsky and Kozlov 2006) (Elyyan 2008).

Alfa Laval is now interested in preforming their own study, confirming the findings in the article by Ferhat, and investigating the usage of micropatterns in the form of adding dimples to a corrugated heat exchanger channel.

Shah, Subbarao, and Mashelkar 1986 [2]. The heat transfer coefficient is much larger for a PHE compared to a shell-and-tube heat exchanger, which makes the required surface area about 1/3 for a given heat duty. This means that the PHE takes up much less space compared to a shell-and- tube heat exchanger, which can be an important factor as customers value space inside their factories

Xuan and Roetzel et.al. [3] Xuan and Roetzel et al. has noticed an increase in energy transfer rate in their investigation on random motion of nanoparticles in NF. An experimental study on the convectional and flow characteristics of water-Cu NF through a straight pipe with constant thermal flow under laminar and turbulent regimes has been reported. Nanoparticles of Cu with less than 100 nm diameter were employed. The results show that Nano-suspended particles substantially improved the performance of conventional base fluid HT. The volume fraction of base fluid in NF fits well with that of water. Pak and Cho (1998) found in their experiment the turbulent forced convection heat transfer of Al2O3 water is higher than TiO2 /water nanofluids inside a circular tube. Li and Xuan (2002) concluded that in laminar and turbulent flow regime in forced convection, the heat transfer coefficient of Cu/water nanofluids flowing inside a uniformly heated tube remarkably increased compared to that of pure water.

. Lotti et. al. [4] Lotfi et al. Have compared the single-phase with the Mixture and Eulerian two-phase models for the forced convection flow of Al2O3/Water nanofluid with temperature independent properties. Also, they have compared the Nusselt number predictions for a 1% value concentration of nanoparticles with several correlations and one set of experimental values. They have also studied the effect of volume concentration on the wall temperature. Their results showed that the Mixture model is more precise than the other two models.

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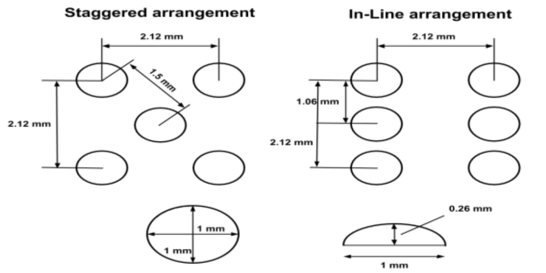
Reza Aghayari et. al.[5] Effect of Nanoparticles on Thermal Efficiency of Double Tube Heat Exchanger Reza Aghayari et al. did the experiments to find an Overall Heat Transfer Coefficient of Nano Fluids (OHTCNF)in heat exchangers and other relevant effective parameters. An improvement in Heat Transfer (HT)and OHTCNF containing Nano-aluminum oxide with ca. 20 nm particle size and particular volume fraction in the range of 0.001-0.002 was reported. The effects of temperature and concentration of nanoparticles on HT variation as well as Overall Heat Transfer Coefficient (OHTC) in a counter current double tube heat exchanger with turbulent flow have been studied. The experimental results fig:1shows a remarkable 8%-10% rise in the mean HT and the OHTC. In general, there are three mechanisms to improve heat transfer by introducing nanoparticles into the base fluid. Nano-particles benefit higher heat transfer rate; therefore, as nanoparticle concentration in the base fluid increases the heat transfer rate increases accordingly. The collisions occur between nanoparticles and the base fluid molecules on the one hand and the impacts of the particles to the heat exchanger wall on the other hand result in an energy increase. The friction between the wall and fluid increases if NFs are dealt with and, therefore, heat transfer improves.

Hassani et. al. [6] The effect of nanoparticles on the heat transfer properties of drilling fluids Hassani et al. has been found that the velocity and temperature have an important effect on the thermal property of mud. The thermal performance factor for all the cases is greater than base mud (5-22% for 0.01-2 wt.% nano-material) and convection results showed that the maximum thermal performance was found for the hybrid of CNT-silica nano-particle in higher Reynolds number. The heat transfer enhancement in 4200 Reynolds number, is 31%. The effects of temperature and concentration of nanoparticles on HT variation as well as Overall Heat Transfer Coefficient (OHTC) in a countercurrent double tube heat exchanger with turbulent flow have been studied. The experimental results fig:1 shows a remarkable 8%-10% rise in the mean HT and the OHTC. In general, there are three mechanisms to improve heat transfer by introducing nanoparticles into the base fluid. Nano-particles benefit higher heat transfer rate; therefore, as nanoparticle concentration in the base fluid increases the heat transfer rate increases accordingly. The collisions occur between nanoparticles and the base fluid molecules on the one hand and the impacts of the particles to the heat exchanger wall on the other hand result in an energy increase. The friction between the wall and fluid increases if NFs are dealt with and, therefore, heat transfer improves. The pioneering work on the problem of natural convection in vertical parallel plate is traced back by W. Elenbass [7] who analysed the laminar natural convection heat transfer in a smooth parallel plate vertical channels without internal bodies & a detailed study of the thermal characteristics of cooling by natural convection was reported. Followed by many experimental, theoretical & numerical investigations for both laminar & turbulent flow regimes. Only the configuration of natural convection in vertical channel with internal objects will be reviewed here.

I.H. Toruka [8] performed experimental study free convection from a cylinder array arranged in a vertical line between parallel walls. Empirical formulas were proposed to predict the average heat transfer coefficient. An enhancement of average Nusselt Number for an entire array of cylinders between parallel walls by 10% to 15% in comparison with the case of free space.

Y. Shen, P. Tong [9] explained light scattering experiment of turbulent convection in water is carried out in a convection cell with rough upper and lower surfaces. The vertical heat flux is found to be increased by ,20% when the Rayleigh number becomes larger than a transition value. The experiment reveals that the main effect of the surface



roughness is to increase the emission of large thermal plumes, which travel vertically through the central region. These extra thermal plumes enhance the heat transport, and they are responsible for the anisotropic behaviour of velocity fluctuations at the cell centre.

Y.B. Du [10], performed novel convection experiment is in a cell with rough upper and lower surfaces. The heat transport across the rough cell is found to be increased by more than 76%. Flow visualization and near wall temperature measurements reveal new dynamics for the emission of thermal plumes. The discovery of the enhanced heat transport has important applications in engineering and atmospheric convection.

N. Onur and M.K. Akta [11] performed study on natural convection between inclined plates. The plate inclinations were chosen to be 0^0,30^0,45^0 and measured with respect to vertical position. Hot plate is facing downwards and heated isothermally. The lower plate is insulated and unheated. Experiments were performed for various temperature differences in air to determine the effect of plate spacing ranging from 2mm to 33 mm and inclination on natural convection heat transfer. It was observed that heat transfer results do not depend on plate inclination strongly.

Ademola A. Dare and Moses O. Petinrin [12] explained diffusion velocity method (DVM), a version of vortex element method (VEM), was used to model the steady state, laminar natural convection flows along isothermal vertical plates and in isothermal vertical channels. This study shows that the diffusion velocity method is a viable numerical tool at modelling not only fluid flow problems but also the heat transfer problems. However, a large deviation recorded for correlation of Nusselt number and Rayleigh number for both the plate and the channel with existing correlations may stem from convergence difficulties encountered at the plate surface. From the results obtained, it is established that as the wall temperature increases while keeping the mainstream fluid temperature constant, the thermal boundary layer thickness increases. The study has also established that the diffusion velocity method is a viable numerical tool capable of modelling fluid and heat transfer problems.

3. Methodology

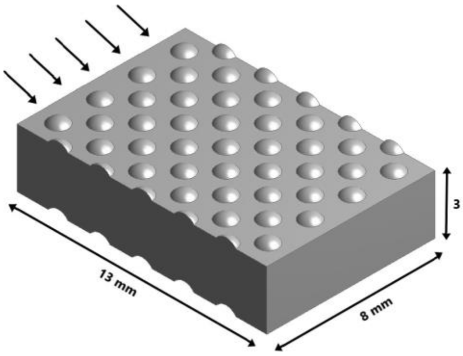
Setup-Guide

This part will contain a setup-guide that describes a general methodology which is used in case 1, 2 and 3. The goal is to collect information that is valid throughout the cases under a single heading, so that repetition is minimized. The geometric domain is different for each case, but with some similarities. In general, the dimensions of the domain are small compared to the size of a heat exchanger plate. This is done intentionally to allow increased meshing resolution while keep- ing computational effort at a minimum. A high meshing resolution is important to resolve flow structures around the dimples at a sufficient level. Numeric quantities obtained from the simulation can be extrapolated to fit the size of a real heat exchanger product. The spherical dimple geometry considered in this report were developed with guidance from previous studies from (Turnow, Zhdanov, and Hassel 2012) and (Banekar, Bhegade, and Sandbhor 2015), with a height-diameter ratio of 0.26. The diameter and height of each dimple was kept consistent in all cases. Two arrangements of dimples were considered, one with dimples placed in a staggered way and one with dimples placed in a line (Giram and Patil 2013). The dimple arrangements with spacing and dimensions are presented in Figure 3.

Figure 3 Arrangements, spacing, height and diameter of dimples.

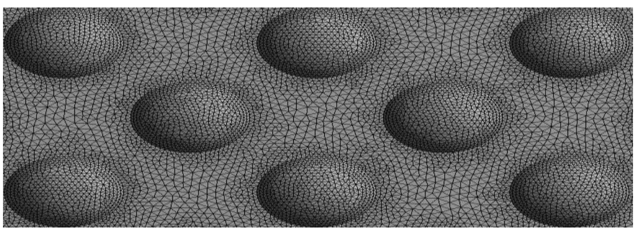
Flow structure on Dimpled Surfaces

As a pre study to the investigation of dimples in corrugated channels, basic research was conducted to study how a surface is affected by adding dimples. This is the first of three cases which will result in a recommendation for Alfa Lavals heat exchanger. This case will contain two sections: An arrangement study with flow structure analysis and an investigation of the dependency of Nusselts number, Fanning friction factor and thermal efficiency on Reynolds number. The computational domain throughout this case is rectangular with dimples penetrating the bottom surface, and protrusions added to the top surface. The domain is 13 mm long, 8 mm



wide and 3 mm high and the fluid is flowing inside the rectangular domain from one short end to the other. The geometric domain as well as dimensions is presented in Figure 4.

Figure 4. Geometric domain for flow structure on Dimpled Surfaces with dimensions presented in the table.

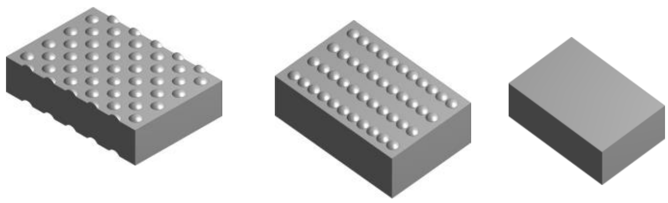


As consistency between the cases was considered an important parameter, the number of dimples were kept the same between configurations. Therefore, the volume and therefore also the time of stay is equivalent between the cases, as a protrusion adds the same amount of volume as an unobtrusive dimple removes. Including the inflation layer, the computational mesh for this case had 2.7 million elements, with 1.5 million being tetrahedral and 1.2 million being rectangular. The elements inside a single dimple were around 10000 cells. A relative quantity that can describe how well resolved a mesh is, is the cell density which has the unit [cells/volume]. The volume of the domain is 320 mm^3 and 2.7 million cells results in a total cell density of 8440 $cells/mm^3$. The computational mesh for the entire domain, including a close of a dimple, can be seen in Figure 5.

Figure 5 Computational mesh for case study 1

3.2 Arrangement

The dimples can be arranged in different configurations which will influence the secondary flow structures around each dimple. There are several arrangements to be considered but the scope was limited to two arrangements. The two dimpled configurations were staggered and in-line, which were compared to a baseline of a flat surface. The arrangements can be seen in Figure 6. The configurations will be evaluated in terms of Nusselt number for heat transfer, fanning friction factor for pressure drop and thermal effectivity coefficient for thermal effectivity.



(a) Staggered (b) In Line (c) Flat

Figure 6 Geometric domain for arrangement study in case 1.

3.3 Singular Heat exchanger Channel

As the second out of three cases, a singular corrugated heat exchanger channel is investigated, before implementing dimples on a herringbone pattern. A herringbone pattern would contain two plates inclined at an angle, β, with the fluid flowing in narrow passages between the plates, were as this study contains one corrugated channel without an inclination. This is a natural step towards the implementation of dimples on a herringbone pattern, as researching a singular channel will contribute to increased understanding how the flow behaves inside the corrugated channel. This is necessary as the flow inside a herringbone pattern is irregular and chaotic, and can be difficult to interpreted without investigating a singular channel first. The geometric domain was done by extracting parts of a singular channel from a herringbone pattern and adding dimples to channel. Protruding dimples were added to the top the channel, penetrating the channel and obtrusions were added to the bottom. Equivalent to section 4.2, the same number of dimples were added on top and bottom to ensure that the volume is kept constant. The fluid is flowing through the channel, from back to front. The rectangular extension at the front of the geometry was added to ensure that backflow was minimized by allowing the flow profile to develop fully before extracting data such as temperature, pressure, and velocity. Note that the extension is modeled as no-slip and adiabatic. The geometric domain for a dimpled channel is presented in Figure 7.

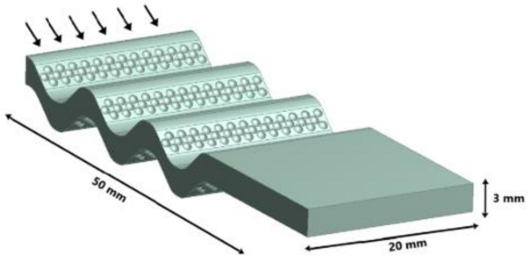


Figure 7 Geometric domain for the singular heat exchanger channel study.

Boundary conditions, selection of turbulence model and meshing methodology was done according to previous section. The mesh had 8.9 million elements including the inflation layer, with 4.8 million being tetrahedral and 4.1 million being rectangular. The volume of the domain was 3360 mm^3 which results in a cell density of 2650 cells/mm^3 . The cell density is smaller compared to the study in previous section, however the resolution around the dimples is unchanged. The decrease in cell density is a result of the singular channel consisting mostly of bulk flow in the center of the channel, where cell density is less important. The cell resolution around the boundary layers close to the wall is equivalent to the previous study involving dimples surfaces. The geometry is slightly more complex in this study compared to previous section which contained a flat surface with dimples. This study features a corrugated channel which requires a longer domain to allow the flow profile to develop fully, this is the reason for the increase in domain size, as compared to the study in previous section. Figure 8 shows the mesh for the singular channel.

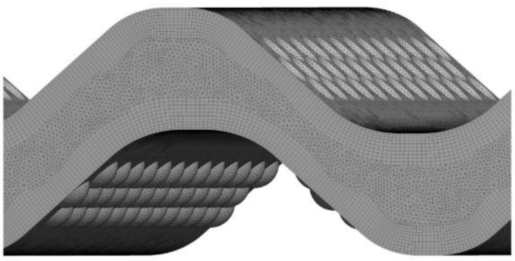


Figure 8 Dimpled single heat exchanger channel.

The inflation layer can be seen in Figure 8 and allow for high resolution around the dimples. The mesh is structured and of good quality without a single element having a skewness above 0.9. The solver was set to steady state with the residual target of 0.00005 residuals. Root square mean (RSM) was applied with a maximum 500 iterations if the convergence criteria was not met.

4. Experimentation

The dimples can be placed at different points throughout the channel. Without considering the production difficulty of pressing the heat exchanger plate with dimples, the position of dimples can affect the pressure drop and thermal performance of the plate. This study is therefore dedicated to investigating where the dimples should be placed for optimal performance. Several possible positions are available, on the flanks of the insulations and on the apex at the top of the channel curvature. Additionally, one could consider that the dimples should be placed with alternating direction regarding obtruding and protruding dimples. After considering possible arrangements, the scope was narrowed down to 4 geometries to be investigated, which are named: Apex-position all down, Apex-position alternating, Flank-position and Flat. Illustrations of all four geometries can be found on the next page, in Figure 9. As can be seen in Figure 9 there are four geometries that have dimples places at different positions throughout the channel. The first geometry has dimples, all facing the same direction, located at the apex of the channel. The second geometry has alternating protruding and obtrusive groups of dimples located at the apex of the channel. The third geometry has dimples located on the flanks of the channel and the fourth geometry is flat, without dimples. The fourth geometry will be used as a base case to compare against.

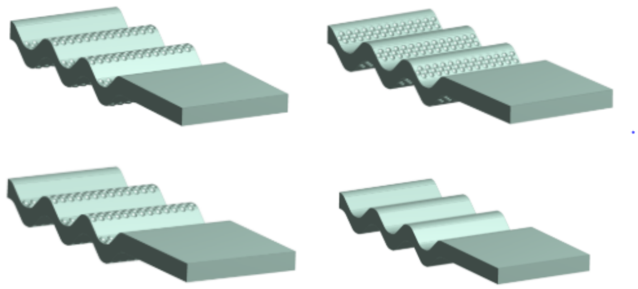


Figure 9 Four channel geometries with dimples placed at different positions.

4.1 Herringbone Pattern

This is the third and final case in the study of dimples in heat exchanger channels. This case will cover the implementation of dimples in a model of a real-life herring- bone pattern used in one of Alfa Lavals heat exchanger products. A herringbone between the plates, giving rise to high heat transfer and high pressure drop. Compared to the first and second case, the flow structure inside the herringbone pattern is complex and chaotic, making a detailed analysis challenging.

The geometric domain for this study was a herringbone pattern with dimples added to the flanks of the channel. The geometric domain is presented in Figure 10.

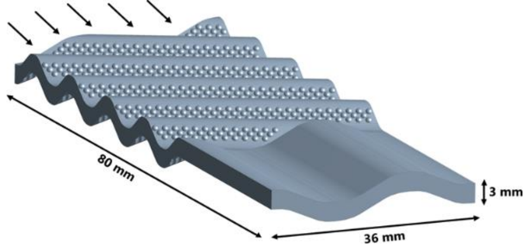


Figure 10 Geometric domain for Herringbone pattern.

Figure 4.10 shows the herringbone pattern with dimensions and flow direction from back to front. Like previous studies, the extension to the right of the geometry was added to allow the flow profile to develop fully before extracting temperature, pressure, and velocity. The extension was modeled as no-slip and adiabatic ensure that it does not interfere with the result of the simulation. Boundary conditions, selection of turbulence model and meshing methodology was done according to section 3.2. Note that the herringbone has undulations facing different directions between top and bottom. The undulations are inclined at a 60° angle, making the corrugation angle 60°. The corrugation angle is the main difference between the singular heat exchanger channel in case 2 and the herring bone pattern in case 3.

5. Results and Discussion

The post-processing software in Ansys CFD allows the user to visually analyse scalar quantities through contour plots located throughout the domain. This can provide increased understanding to the flow structure around the dimples and its contribution to heat transfer enhancement. A thermal analysis can be performed based on Figure 5.1, which includes temperature contours of the domain for the staggered arrangement including a sideways view and a close of a protruding and obtruding dimple.

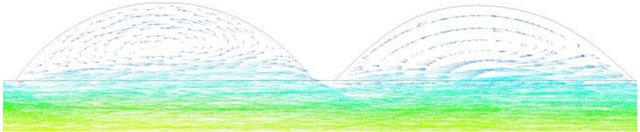
(a) Obtruding dimple (b) Protruding dimple

(c) Obtruding dimple, upwards

(d)

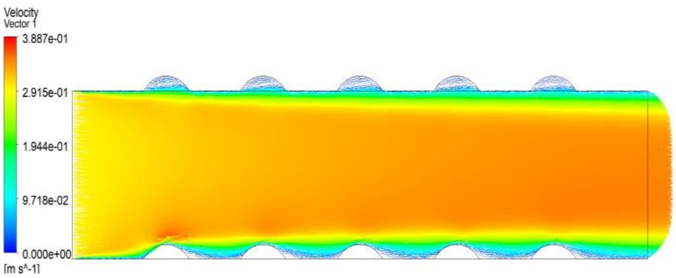
Graph. 1 Temperature contour plot with flow direction from left to right

Graph 1 shows the contribution that dimples have towards heat transfer enhancement. As can be seen in Graph 1 (d) the temperature is significantly higher in the wakes on the protruding side of each dimple and in the cavities of each dimple on the obtruding side. Graph 1 (b) and (c) show a close-up view within the dimple cavity. The temperature reaches a maximum at the front half of the obtrusion and shows a minimum at the downstream rim. Graph 1 (a) shows



a close of a protruding dimple, which reveals a temperature maximum between adjacent protruding dimples. Bear in mind that a low temperature represents a region of high heat transfer, as this is where energy has been transported from the wall into the fluid. The regions of high and low heat transfer can be explained by studying the velocity field. The velocity field is presented in Graph 2.

(a) Protruding dimple



(b) Sideways view

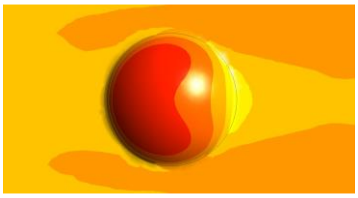
Graph 2 Velocity vector plot with flow direction from left to right.

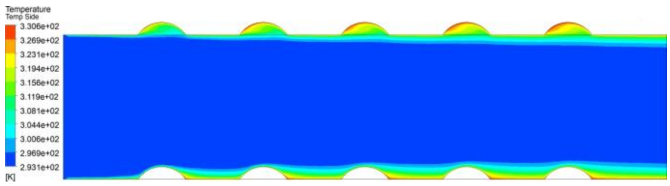
Graph 2 (b) shows the velocity field of the domain. The velocity reaches its maximum in the middle of the domain and tends towards zero as it approaches the wall, as can be expected when no slip boundary conditions are applied. Close to the wall, as the flow moves parallel with a dimple, it hits the downstream edge of the dimple causing flow separation. The separated flow moves upwards in the dimple cavity approaching the apex of the dimple, and extends to the upstream edge to form a recirculation zone. The recirculation zone contains hot, slow-moving media which is eventually remixed with the bulk of the flow at the upstream edge of the dimple. The recirculation zone in the dimple cavity can clearly be seen in Graph 2 (a). The regions of low heat transfer in Graph 1 corresponds to the regions of low velocity in Graph 2. Specifically in the protruding dimple cavity, the heat transfer at the downstream rim is maximum due to the flow hitting this edge as parts of the flow is separated upwards into the dimple. The heat transfer is minimum in the front half of the dimple due to the recirculation zone trapping hot fluid, which prevents the driving force of heat transfer.

Reynolds Sweep

According to previous studies, see section 3.3, dimples act as turbulent generators that push the transition from laminar to turbulent flow. Naturally, dimples have different contributions to heat transfer and pressure drop at different Reynolds numbers and it is interesting to investigate this dependency. Therefore, Nusselt number and the Fanning friction factor was obtained at different velocities to investigate where dimples have the most effect.

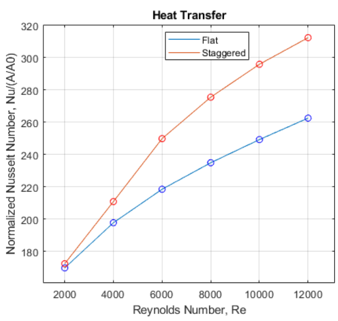
The Nusselt number and fanning friction factor can be

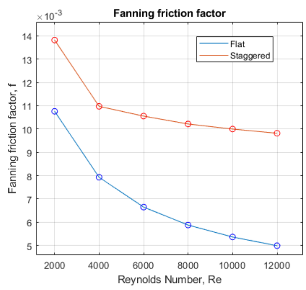




presented in Graph to further examine the trends, which is done in Graph 3. Graph 3 (a) shows that the heat transfer is enhanced by adding dimples. The dimpled model (red line) has a higher heat transfer than the undimpled model (blue line) at every design point in Graph 1 (a). As the Reynolds number decreases and laminar flow is approached, the effect of the dimples also decreases and the heat transfer of the dimpled channel approaches the heat transfer of the undimpled channel. As the Reynolds number increases, the heat transfer of the dimpled channel increases compared to the undimpled channel. The heat transfer at Reynolds number 12000 is 26 % larger for the dimpled channel compared to the undimpled channel.

(a) (b)





Graph 3 Heat transfer(left) and fanning friction factor(right) with dimples in a staggered arrangement (red line) and without dimples (blue line).

Graph 3 (b) shows the fanning friction factor, with the red line being the dimpled model and the blue line being the undimpled model. Just like heat transfer, the friction factor is consistently larger for the dimpled channel with the difference increasing as the Reynolds number increases. The largest difference can be seen at high Reynolds number 12000 with the friction factor being almost double that of the undimpled channel.

Conclusion

In this project, the objective was to investigate if dimpled geometries can increase the performance of plate heat exchangers. The investigation has shown mixed results. Computational Fluid Dynamic Simulations using a RANS model has shown that the thermal efficiency tends to decrease when dimpled are added to a flat channel. The dimples were shown to have limited effect when added to a herringbone pattern due to the high heat transfer and large pressure drop already present in the original pattern. More advanced LES simulations on a single heat exchanger channel showed that the RANS model is insufficient in its description of turbulent structures. LES proved that the thermal efficiency does in fact increase when dimples are added, due to enhanced mixing, formation of vortexes and reduction of recirculation zones inside the channel. As LES simulations have not been performed on a herringbone pattern, the effect of dimples on such a pattern is still uncertain. Without experimental data, it is difficult to draw explicit conclusions from the study connected to reality, as the model data need to be validated against laboratory trials. The project has also shown that performing Computational Fluid Dynamic simulations of heat exchanger channels is challenging, due to complex and dynamic flow structures located inside the channel. For most R&D applications, RANS-models does not provide the necessary resolution to accurately describe the velocity profile. LES captures many of the desired structures, but is computationally more expensive than RANS resulting in a need for purpose-built computers with high computational capacity. Recommendation for Alfa Laval: Based on the results of this paper, the most advanced simulation has shown that dimples increase the thermal performance of plate heat exchanger.

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