Dimple Shape Design to Enhance Heat Transfer in Plate Heat Exchangers

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ABSTRACT

This article studies four dimple shapes: spherical, smoothed-spherical, normal distribution, and error distribution and how they enhance heat transfer on a plate within a plate heat exchanger using computational fluid dynamics. The dimple that showed the greatest efficiency of heat transfer was the normal distribution dimple, giving a temperature increase of 7.5 times of the smoothed-spherical and 15% more than the error distribution dimple shape. This was primarily due to the large increase in the turbulent kinetic energy caused by the eddies created upon the flow over the normal distribution shape. With the normal distribution shape being found to be the most effective in enhancing heat transfer, a layout of multiple normal distribution dimples based on the stage of flow development was also studied. It was found that a fully developed flow resulted in 9.5% more efficiency than half developed flow and 31% more efficient than placing dimples directly next to each other.

Keywords: Ansys Fluent, CFD, Dimple, Heat transfer enhancement, Plate Heat Exchangers

INTRODUCTION

Heat exchangers play an important role in both everyday life and large-scale production. Heat exchangers can be found in many places, from some of the world's largest companies to the radiators in your very home. There are many kinds of heat exchangers, such as the ones in a refrigerator which use a coolant fluid to absorb heat [1], or a plate heat exchanger (PHE) of which was studied in this paper. The main function of a heat exchanger is to transfer heat from one source to another and they are generally used to heat or cool a required element via conduction [1, 2]. In an industrial setting, heat exchangers are often found between multiple reactors in order to increase or decrease the temperature of a fluid after one reaction, before entering the following reactor [3]. This is often done to reduce costs as less fluid material will be required and/or to reduce waste by recycling the stream. For these reasons, the efficiency of a heat exchanger is very important.

Plates within PHEs, are produced as a large sheet of metal with many indentations. These indentations are known as "dimples" and serve to increase the surface area of the plate and encourage turbulent flow [4]. When the surface area available for heat transfer as well as the Reynolds number increases, the temperature change should also increase. This makes the PHE more efficient than it would be otherwise.

In this paper, the plates within a plate heat exchanger were assessed and computationally experimented upon in order to provide an extensive coverage of how to improve the efficiency of a PHE by solely changing the shape of the dimples upon the plates within it. This was done via a computational fluid dynamics (CFD) program known as Ansys Fluent.

Computational fluid dynamics (CFD) is defined as "the method of mathematically modelling a physical phenomenon involving fluid flow and solving it numerically using analytical prowess" [5]. The primary benefit to using CFD in an industrial setting is that it is faster than and comparable to a physical experiment. This allows companies to run simulations in order to gain a realistic idea of what the final process of their production will be.

This paper aimed to simulate realistic fluid flow and dynamics over four different dimple shapes: spherical, smoothed-spherical, normal distribution and error distribution. This was done to obtain values for the Reynolds number, pressure and temperature change across a set

time and distance. A fair comparison of dimple shapes was to be achieved to effectively display which shapes and configurations were more efficient for the transfer of heat in a PHE.

METHODOLOGY

This study was performed with 2D models and simulations due to the symmetrical structure of the dimple shape and flow channels in radial direction. Nonetheless, 2D models were deemed appropriate to realistically simulate what would occur in an industrial setting, as the results can still be used to compare the heat exchange between different dimple shapes. 2D models also accurately replicate a side-on view of a 3D model without the necessity of a third dimension, i.e. the temperature and velocity profiles would be similar between a 2D and a 3D simulation if viewed from the perspective where the z axis is equal to zero.

For the simulations whereby dimple shape was compared, a pipe was set up with a length of 1 m and a diameter of 0.1 m with the pipe walls being aluminium. The inlet pressure was set to 101325 Pa (1 atm) for all simulations.

For the method, two key areas were explored: simulations over different dimple shapes; and simulating over multiple dimples. The first of which was to find the most efficient dimple shape, and the latter to find the best layout for these dimples over a single plate in a PHE.

SIMULATING OVER DIFFERENT DIMPLE SHAPES

Set-up

A 0.5 m/s flowrate was selected, giving a Reynolds number of 48,500, and thus a very turbulent flow. The inlet streams were all set to 300 K, with the walls at a constant 350 K. Dimples were added separately to their respective simulations. Figure 1 demonstrates a sketch of a single spherical dimple.
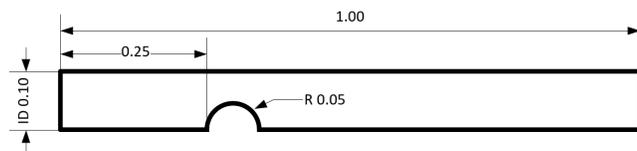

Figure 1. A sketch of a 2D pipe with a single spherical dimple, with dimensions in metres.

The cross-sectional area (CSA) of this dimple was then calculated to be 3.927×10^{-3} m^2. This CSA was to be used across all dimple sizes in order to maintain a fair test, as well as all dimples having a peak height of 0.05 m, while being 0.3 m from the pipe inlet. Figure 2 shows the sketch for the smoothed-spherical dimple, while

Figure 3 shows a diagram of the finalised normal and error distribution shapes respectively.
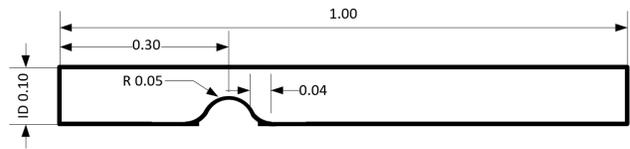

Figure 2. A sketch of a 2D pipe with a single spherical dimple, with dimensions in metres.

Figure 3. A diagram of the final pipe containing the normal and error distribution curves, respectively.

Results

Of the different dimple shapes tested: spherical, smoothed-spherical (SS), normal distribution (ND), and error distribution (ED), the ND dimple demonstrated the greatest heat exchange rate, with the average water temperature increasing by 0.53 K in 1.5 s. This is displayed clearly in Figure 4.
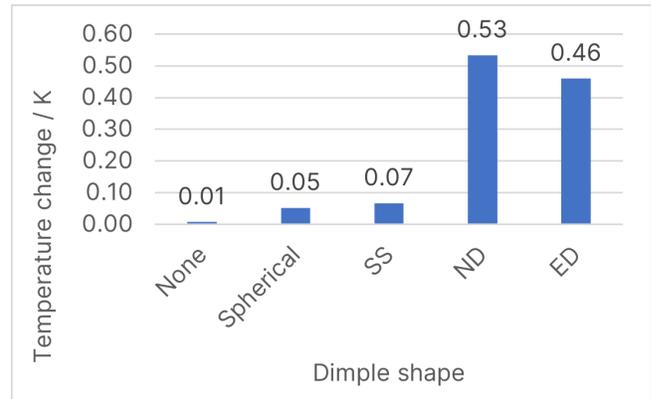

Figure 4. Average temperature change of water across different dimple shapes.

It was found that the primary driver for the temperature increase over the ND and ED dimples compared to the rest was the intensity of the eddies created. This was measured by comparing the average values for post-dimple turbulence kinetic energy (TKE), which represent "the mean kinetic energy per unit mass associated with eddies in turbulent flow" [6]. The values for post-dimple TKE are shown in Figure 5.
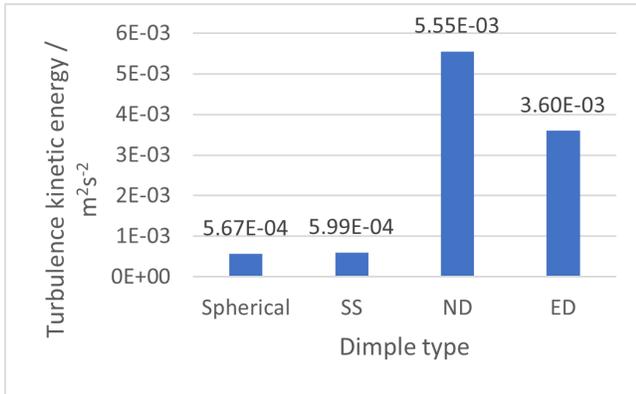
Figure 5. Post-dimple TKE for different dimple shapes.

Figure 5 demonstrates the effect eddies have on heat transfer as the values are proportional to the temperature values in Figure 4. Therefore, as the TKE and thus eddy intensity increases across the fluid flow, the heat transfer also increases.

Regarding the other variables that were measured – Reynolds number, pressure, and velocity – the normal distribution (ND) dimple showed surprisingly little change in most of these areas in comparison to the other dimple shapes. This is shown in Table 1.

Table 1. A comparison after 1.5 s between the simulation of a pipe with different dimple shapes while the fluid is heated by the pipe walls.

Dimple type	Re change	P change / Pa	T change / K	u change / m s^{-1}
None	0	-0.3	0.01	0.00
Spherical	13536	-82.7	0.05	0.14
SS	12597	-78.9	0.07	0.13
ND	8480	-142.6	0.53	0.09
ED	13354	-141.8	0.46	0.14

Compared to the spherical and smoothed-spherical dimples, the ND dimple provides, at the least, 7.5 times more efficiency in terms of heat exchange with a pressure change that is less than doubled. Notably, the ND had the smallest velocity change, of the dimpled pipes, meaning the fluid had more time to heat up. The Reynolds number also had the lowest change of the dimpled pipes. However, since the fluid flow was very turbulent to begin with, and remained very turbulent throughout the pipe in all cases, a difference less than 4000 in the Re between the ND dimple and the spherical is negligible. Finally, while the ND dimple does prove to have the greatest pressure drop, the great increase in efficiency outweighs this minor drawback. Taking these factors into account, the ND dimple was clearly the most efficient overall and is the recommended shape, of these four, for a dimpled

plate in a PHE despite the pressure drop.

SIMULATING OVER MULTIPLE DIMPLES

Set-up

Once the most efficient dimple shape for heat transfer was found to be the ND dimple, it was implemented multiple times into a 5 m long pipe. The distances between the dimples were varied in relation to the hydrodynamic entrance length for turbulent flow [7]:

$$\frac{L_e}{D} = 1.36 Re^{\frac{1}{4}} \qquad (1)$$

where L_e is the entrance distance, D is the inner diameter of the pipe and Re is the Reynolds number. The Re at the peak of the normal distribution dimple was found to be approximately 96000, giving a L_e of 2.39 m. However, Çengel and Cimbala [8] write "in many pipe flows of practical engineering interest, the entrance effects become insignificant beyond a pipe length of 10 diameters" before showing that

$$L_e \approx 10D \qquad (2)$$

In this case, the result obtained for the hydrodynamic entrance length was 1.0 m, thus the pipe was to be extended to 5 m for the resultant simulations. When comparing different dimple shapes, this did mean that fully developed flow had not recovered by the end of the 1 m pipe in Figure 1 and Figure 3. This made little difference to the previous single dimple shape comparison results however, as the difference in heat transfer values from one dimple shape to another will have been equally impacted.

Both L_e distances were compared and gave average velocities approximately 4% from the average outlet velocity 5 m along the pipe, by which time the fluid had certainly fully developed. As both regions gave the same accuracy, the shorter 1.0 m distance was used. In industry, the shorter hydrodynamic entrance length will be favoured as more dimples can be used over a set area. This is because the fluid flow will be able to fully develop more often, and thus be disrupted more often, resulting in an increased rate of heat transfer.

Once the hydrodynamic entrance length of 1.0 m was established, simulations were run over a 5 m long pipe consisting of three ND dimples. For this setup, the following three simulations were run:

- Three ND dimples directly next to each other to simulate flow which was not at all developed. This meant that their peaks were 0.2 m from one another as the radius of the dimples was 0.1 m.

- Three ND dimples with their peaks 0.5 m apart from one another to simulate half developed (or developing) flow.

- Three ND dimples with their peaks 1.0 m apart from one another to simulate fully developed flow.

For all three of these simulations, the initial dimple was setup as before, with its peak 0.3 m into the pipe. An example of the half developed flow setup is shown below.
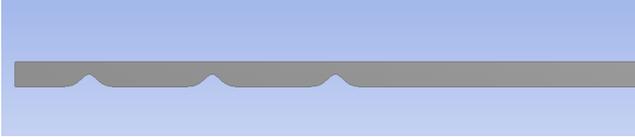

Figure 6. The first half of the final model for half developed flow over three ND dimples. The remainder of the pipe contains no dimples.

Results

For multiple ND dimples, the temperature change across the entire pipe was 3.79 K, 4.54 K, and 4.97 K for a none developed flow, a half-developed flow, and a fully developed flow respectively. From this it is clear that the more developed the fluid flow is before being disrupted, the more heat is exchanged from the pipe walls to the fluid.

This is useful because it can be applied to design plates for PHEs in industry. Knowing that fully developed flow provides the best heat transfer between the plates and fluids within a PHE, companies that require specialist PHEs designed for their process should purchase plates that are distanced such to ensure fully developed flow between dimples. In this case for example, each dimple should be imprinted upon a plate 1 m apart in order to maximise heat transfer. Additionally, as the dimples are more spread out when compared to a less developed flow, firms could reduce their cost per plate as they will require a lower surface area, meaning less material is being used. Depending on the specific material required, the number of plates, and the size of the plates, the cost savings could differ significantly.

REPEATING DIMPLES

In order to more accurately translate these results into industrial heat exchangers, ND dimples were also repeated as much as possible over a 5 m pipe, as opposed to using only three dimples. For these simulations, the dimples were repeated at distances of 0.2 m (directly next to one another), 0.4 m, 0.6 m, and 1.0 m (fully developed velocity profile).

In order from the least to most temperature change, these repeated dimples for the layouts of no dimples, 1 m between dimple peaks, 0.6 m between dimple peaks, 0.4 m between dimple peaks, and 0.2 m between dimple peaks (without the final dimple), showed average outlet temperatures of 306.4 K, 308.3 K, 309.5 K, 310.8 K, and 311.5 K respectively.

Contrary to the previous results, which showed fully developed flow to have the greatest temperature increase, these results argue in favour of the dimples being directly next to one-another, as is currently common practice in industry. However, each of these pipes, unlike in the previous simulations, have drastically different lengths – which is comparable to the surface area of a 3D model. The length of the pipe containing the dimples with their peaks 0.2 m apart had now increased by 13.9% from the original 5 m length of pipe, and a 6.2% increase in length from the pipe with dimples with 0.4 m between their peaks. As these simulations contain 0.3 m of pipe before a dimple peak is present, in an industrial setting these percentages would be even greater.

What this means is that while maintaining none-developed flow between dimples still shows promise in an industrial setting, this temperature change is offset somewhat by the increase in material required. Instead, a good alternative would be to remove every other dimple – which in this case is represented by the dimple peaks being 0.4 m apart – as it is still one of the most temperature-efficient models. This layout only has a decrease of 0.2% temperature change compared to completely none-developed flow, while saving half of the material used in the imprinting of dimples upon a plate.

A MAJOR DRAWBACK OF 2D MODELLING

3D models are clearly the next step to further develop and analyse the results of this work.

3D models have been attempted over the course of this project. The models were created such that the different dimples to compare were the same as in the 2D models. This was done by revolving the dimple shapes around the y-axis. For these experiments, only the ND and spherical dimples were compared.

Simulations were performed over a square based pipe, and a circular based pipe. The circular based pipe however suffered from having the dimples clip through the wall, thus causing inequal losses in both volume and surface area between the different dimple shapes. This is demonstrated in Figure 7.
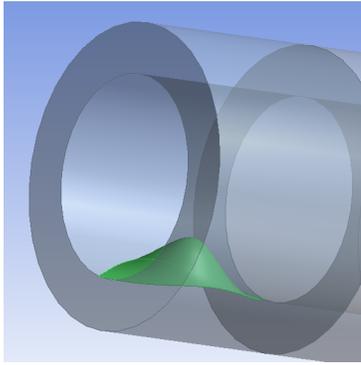
Figure 7. The ND dimple being cut due from the wall of a spherical pipe.

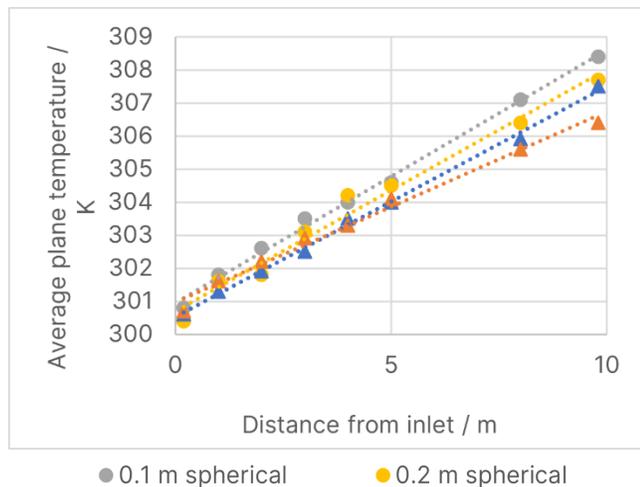Due to this, for the following results, only the square based pipe results are presented.

● 0.1 m spherical   ● 0.2 m spherical

Figure 8. Average temperature of the fluid at different distances, both for the two dimple shapes and differing distances between dimples.

The heat transfer results shown in Figure 8 do not line up with the results achieved from the 2D simulations. This is clear as the spherical dimples outperform the ND dimples regardless of distance between dimple peaks – whether it be 0.1 m or 0.2 m for the spherical dimples, or 0.2 m and 0.4 m in the case of the ND dimples. The reason for this was that the control variable carried forward from the 2D simulations to 3D regarding dimple shape was the cross-sectional area (CSA) of the dimple. This worked well for a 2D simulation, but not when a third dimension was added.

When scaling the CSA of a 2D model up into a 3D model, the resultant measurements correlate to the volume of the dimple. These 3D simulations therefore unfairly compared the two dimple types, as they had different surface areas. This is important because the surface area should be a control variable, in order to maintain the same amount of surface in which the water could be heated over.

For future work, 3D simulations should be run with the surface areas of the dimples equal, as opposed to the cross-sectional area, which was carried over directly from its 2D counterpart.

CONCLUSIONS & RECOMMENDATIONS FOR FUTURE WORK

The normal distribution (ND) dimple shape provided the greatest heat transfer showing an increase in temper-ature 7.5× the smoothed-spherical, with a 15% uplift from the error distribution (ED) shape. Despite having the greatest pressure drop of 142.6 Pa, the large increase in temperature far outweighed this downside.

Eddy intensity was the primary cause for this efficiency of heat exchange. The ND dimple demonstrated a 54% increase in turbulent kinetic energy over the next greatest – the ED dimple. This meant that the intensity and number of eddies produced proved more meaningful for heat transfer than the surface area of the plates.

Fully developed flow was more susceptible to being disrupted than other flow states, being 9.5% more heat transfer efficient than disrupting half developed flow, and 31% more efficient than none developed flow. However, when repeated over a larger scale, it is not wholly viable to have repeated dimples up to 1 m apart. Instead, a compromise should be made where dimples are separated by one-dimple length. In practice, this would mean removing every other dimple from the current plate-printing process. This would still ensure a high temperature change (a 0.2% decrease from none developed flow), while saving half the material used to imprint dimples.

The primary recommendation for future work is to develop and simulate 3D models for a more realistic approach and more easily applicable results to industry. This would be more demonstrative of how dimple shapes and flow development can affect heat exchange, due to its real-world nature. In contrast with the initial 3D models presented, future simulations would have dimple surface area as a control variable, in order to achieve fair an reliable results.

ACKNOWLEDGEMENT

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