

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

MSc THESIS

Cenk BÜYÜKAŞIK

**COMPUTATIONAL HEAT AND FLUID FLOW ANALYSIS OF
AN INNOVATIVE PLATE FOR A PLATE HEAT
EXCHANGER**

DEPARTMENT OF AUTOMOTIVE ENGINEERING

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ABSTRACT

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In this study, heat transfer and fluid flow are numerically investigated for interrupted sinusoidal channels with ANSYS Fluent software under the assumptions of 2D viscous, incompressible, fully developed laminar flow. For this aim, an innovative sinusoidal geometry was created and used. Since the flow was periodically repeated, the periodic boundary condition was used to save computational time. Before the main computations, validations were performed to ensure the accuracy of the solutions. Effects of the sinusoidal wave amplitude, the thickness of the wall, space between interrupted channels, and vertical distances on Nusselt number, friction factor, and thermal enhancement factor were investigated for different Reynolds numbers. Velocity vectors, velocity contours, and temperature contours were also shown. Results from the study shown that this type of channel has increased the Nusselt number and friction factor due to the vortex generation.

Key Words: Sinusoidal channels, heat transfer, flow, the thermal enhancement factor

ÖZ

YÜKSEK LİSANS TEZİ

İNOVATİF GEOMETRİYE SAHİP BİR PLAKA TİPİ ISI EŞANJÖRÜNÜN NUMERİK ISI VE AKIŞ ANALİZİ

Cenk BÜYÜKAŞIK

ÇUKUROVA ÜNİVERSİTESİ
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Bu çalışmada ANSYS Fluent programı kullanılarak kesikli sinüzoidal kanallarda iki boyutlu, viskoz, sıkıştırılmaz, tam gelişmiş laminar akış davranışı numerik olarak incelenmiştir. Çalışma için inovatif sinüzoidal bir geometri oluşturulmuş ve kullanılmıştır. Akış periyodik olarak tekrarladığından hesaplama sürecini kısaltmak adına periyodik sınır şartı kullanılmıştır. Ana hesaplamalar öncesi sonuçların güvenilirliği için validasyonlar yapılmıştır. Sinüzoidal dalga şiddeti, duvar kalınlığı, kesikli çizgiler arası yatay ve dikey boşlukların Nusselt sayısı, sürtünme faktörü ve ısıl iyileştirme sayısına etkisi incelenmiştir. Seçilen geometrinin farklı Reynolds sayılarındaki davranışı gözlemlenmiştir. Hız vektörleri ve hız konturları ile sıcaklık konturları da araştırılmıştır. Çalışmadan elde edilen sonuçlar bu kanal türünün Nusselt sayısını ve sürtünme faktörünü arttırdığını göstermiştir.

Anahtar Kelimeler: Sinüzoidal kanallar, ısı transferi, akış, ısıl iyileştirme sayısı

EXTENDED SUMMARY

Heat exchanger is a device that transfers thermal energy between two or more fluids via a surface, which might be a plate, a tube's internal surface, or some kind of geometry within the flow direction as fins, etc. They are widely used in various industrial fields spanning from the automotive sector to heating, refrigeration and air-conditioning sectors with different types and capacities. Beginning from the early 2000s, the heat exchanger industry is affected by the petroleum extinction and global warming threats in such a way that new, more effective designs are compelled by the industries.

A plate heat exchanger comprises of thin plates that involve small channels. It has two sides of flow that possess different temperatures; by this mentality, the system generates heat transfer. Regenerating the boundary layer to enhance the heat transfer and increasing the surface area in which the heat transfer happens, creating mixing regions to generate higher heat transfer coefficients are plausible for higher values.

Laminar flow inside tubes can be investigated in two groups. As the flow enters downstream from the mouth, the flow develops a boundary layer region that coats the inner surface of the tube wall. For this reason, the longitudinal position, where the velocity boundary layer thickness has grown to the same size as the tube radius.

This study is an approach for an innovative plate geometry. After the calculations, it is claimed that the newly created shape is more efficient than some geometries in literature. A parallel plate PHE is examined to show this suggestion. Velocity contours (figure 4.7 and figure 4.8) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.9) presents the heat increment around the sinusoidal walls. Comparison between the radius and sharp-edged geometries. It is observed that both geometries have the same behavior in terms of Nusselt number; it increases with wave amplitude; however, about 20% - 25% change is observed as a gain for Nu number due to radius edge usage. The higher value of sharp edges Nusselt number is achieved on the 0.7 wave amplitude, which has a value of 10.92, and for radius shaped edge, the higher Nusselt value again achieved in 0.7 wave amplitude, and it is 11.1. It can be concluded from figure 4.25 that friction factor is in direct proportion to wave amplitude and internal edge space, which means friction factor increases with internal edge space and wave amplitude. Therefore, the highest value obtained for 0.7 wave amplitude and 7mm edge space. From figures 4.24, and 4.25, one can

conclude that the best geometry among them is the 0.3 wave and 0.7 internal edge space. Velocity contours (figure 4.27, figure 4.28, and figure 4.30) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.29) displays the heat increment around the sinusoidal walls. Velocity contours (figure 4.40 and figure 4.41) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.42) displays the heat increment around the sinusoidal walls. As seen in the figure 4.43, the velocity is higher in vertically interrupted wave geometries than the normal pattern. This process highly affects the boundary layer regeneration and provides better heat transfer, which also numerically validated.

Ultimate geometry among the researched interval is found as 0.3 Wave amplitude, 5 mm vertical distance, and 7 mm internal edge space. Figure 4.49 shows the increment of the Nusselt number with a raise in the Reynolds number. Table 4.23 is a consolidation for Nusselt number, friction factor and thermal enhancement factor versus Reynolds number increment.

In this study, an innovative plate geometry was investigated numerically. ANSYS Fluent software was used for these analyses. Seventy different calculations were performed, and all results consolidated. In the first section, a validation process performed to assure the accuracy of the further results. In the second section, the thickness effect for sharp-edged wavy walls had been investigated. In the third section, the radius intercalated edges effect investigated concerning sharp edges for different wave amplitudes under the circumstances of thickness $t=1\text{mm}$ and Reynolds number of 100. In the fourth section, internal space edge variety effect on performance, investigated for three different values, which are 3, 5 and 7. In the fifth section, interrupted waves have been investigated. A vertical distance is implemented. performed best for this investigation among the others. In the last section, the best geometry had been chosen among 70 analyses. At first, the last part of the induction is executed. Nu , f , and η values compared for 0.3 wave amplitude 5mm vertical distance for internal edge spaces of 3, 5 and 7 mm. The 3D version of this study might be a further research idea. Also, the results can be validated empirically and an experimental study can be performed. In addition to that, the turbulent version of the analyses can be conducted to reach more precise results. These are the future ideas which this work will lead.

GENİŞLETİLMİŞ ÖZET

Isı eşanjörleri ikiden fazla yüzeyde termal enerji transferi yapan cihazlardır. Bu bir plaka, tüp veya akış yönündeki bir geometri olabilir. Otomotiv sektöründen ısıtma, soğutma ,iklimlendirme sitemlerine kadar birçok farklı tipte ve endüstride kullanılmaktadır.

2000li yılların başından itibaren ısı eşanjörü endüstrisi petrol azalması ve küresel ısınmadan dolayı tehdit altındadır. Bu sebeple daha verimli tasarımlar yapma ihtiyacı doğmuştur.

Plaka tipi eşanjör ince plakalardan ve küçük kanallardan oluşur. İki farklı sıcaklığa sahip iki akışın bulunduğu sistem bu şekilde ısı transferi yapar. Sınır tabakanın yenilenmesi ısı transferini artırır ve karışım bölgeleri oluşturarak ısı transfer katsayısını yüksek değerlere çıkarır.

Laminar akış iki farklı grupta incelenir. Akış boruya girdiğinde bir sınır tabaka oluşturur.bu sebeple boru yarıçapı sınır tabaka ile ilişkilidir. bu çalışma inovatif bir plaka geometrisi yaklaşımıdır.hesaplamalardan sonra yeni oluşturulan geometrenin literatürdeki bazı geometrilere göre daha verimli olduğu ortaya çıkmıştır,bunun için bir paralel plaka ile karşılaştırma yapılmıştır. Hız konturları akış karışımları ve sınır tabakanın yeniden oluşumunu göstermektedir. Sıcaklık konturu ise sinüzoidal duvarların çevresindeki ısı artışını göstermektedir. Radiuslu ve keskin kenarlı geometriler karşılaştırılmıştır.Her iki geometride nusselt sayısının dalga şiddeti ile %20-%25 oranında arttığı gözlemlenmiştir.en yüksek keskin kenar nusslt değeri 0.7 dalga şiddetinde gerçekleşmiştir ve 10.92 dir. Yine en yüksek dalga boyu olan 0.7 de en yüksek radiuslu kenar nusselt sayısı değeri de elde edilmiştir ve 11.1 dir. Ayrıca sürtünme faktörünün de dalga şiddeti ile ve iç boşluk ile doğru orantılı olduğu gözlenmiştir. Böylelikle 0.7 dalga şiddeti ve 7 mm iç boşluk en iyi değerleri almıştır. Tüm sonuçlar düşünüldüğünde ise en geometrinin 0.3 dalga şiddeti vbe 0.7 iç boşluğa sahip geometrinin olduğu ortaya çıkmıştır. Hız konturları akış ayrılmasını göstermektedir ve bu şekilde sınır tabakanın

yenilenmesi ve ısı transferinin arttırılması gözlemlenmiştir. Sıcaklık konturu sinizoidal kanallar çevresindeki ısı artışını göstermektedir. Yine tezde geçen figürlerde dikey olarak yerleştirilen dalga geometrilerin normal yerleştirilenlere göre hız anlamında daha etkin olduğu gösterilmiştir. Bu işlem sınır tabaka yenilenmesi ve ısı transferi arttırmaktadır. Bu işlem ayrıca numerik olarak da doğrulanmıştır. İncelemeler sonucunda nihai geometrinin 0.3 dalga şiddeti 5mm dikey boşluk 7 mm iç boşluklu yapıda olduğu gösterilmiştir. Nusselt numarasının Reynolds sayısı ile arttığı gösterilmiştir. Ayrıca Nusselt sayısı sürtünme faktörü ve ısıl iyileştirme sayısının Reynolds numarası ile ilişkisi belirtilmiştir. Bu çalışmada bir inovatif plaka geometrisi numerik olarak incelenmiştir. Analizlerin yapılması için Ansys Fluent yazılımı kullanılmıştır. 70 farklı hesaplama yapılmış ve sonuçlar konsolide edilmiştir. Birinci bölümde sonuçların güvewnilirliği için doğrulama işlemi yapılmıştır. İkinci bölümde kalınlık etkisi incelenmiştir. Üçüncü bölümde radius eklenen kenarlar incelenmiştir yine farklı dalga şiddetleri kullanılmıştır. Kalınlık 1 ve Reynolds sayısı 100 kabul edilmiştir. Ayrıca keskin kenarlı sonuçlar ile karşılaştırma yapılmıştır. Dördüncü bölümde üç farklı iç boşluk ölçüsünün performansa etkisi incelenmiştir. Beşinci bölümde dikey boşluklu yapılar incelenmiştir. Son bölümde 70 farklı analizden en iyi geometri seçilerek farklı Reynolds sayıları için inceleme yapılmıştır. Bu çalışmanın 3 boyutlu hali yeni bir çalışma fikri olabilir. Yine aynı geometrinin deneysel olarak incelenmesi de ayrı bir çalışma konusu olabilir. Son olarak türbülanslı akış karakteristikleri ile daha hassas sonuçlar elde edilebilir. Bu fikirler bu çalışmanın gelecekte liderlik edebileceği fikirlerdir.

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NOMENCLATURE

u_m	Mean fluid velocity	m/s
Re	Reynolds number	-
d_e	Equivalent diameter	m
f	Friction factor	-
L	Length of the computation area	m
H	Hydrodynamic entry length	m
T_s	Surface fluid temperature	K
T_m	Mean fluid temperature	K
A	Area	m ²
h	Heat transfer coefficient	W/m ² K
$\dot{q}$	Heat flux	W/m ²
T_w	Wall Temperature	K
k	Thermal conductivity of the air	W/mK
$\dot{M}$	Mass flow rate	kg/s
ΔP	Pressure drop	Pa
C_p	Specific heat	J/kg K
Nu	Nusselt number	-
F_p	Wavy walls circumference	m
x	Axial coordinate	m
ν	Kinematic viscosity	m ² /s
μ	Dynamic viscosity	kg/ms
ρ	Density	kg/m ³
V	The volume of calculation area	mm ³
ΔT_m	Logaritmic mean temperature	K
t	Thickness	mm



1. INTRODUCTION

Heat exchanger is a device that transfers thermal energy between two or more fluids via a surface, which might be a plate, a tube's internal surface, or some kind of geometry within the flow direction as fins, etc.

Heat exchangers are one of the most utilized devices in the industry. They are widely used in various industrial fields spanning from the automotive sector to heating, refrigeration and air-conditioning sectors with different types and capacities. Beginning from the early 2000s, the heat exchanger industry is affected by the petroleum extinction and global warming threats in such a way that new, more effective designs are compelled by the industries.

The first simple consideration regarding heat exchanger is positioning Stones around a fire inside our early shelters. Stones consume and keep heat, which afterwards moved another place to warm up without any fear of fire. Our ancestries used this method unaware, yet that was an innovation back in those days.

In today's industry, there are various heat exchanger types from the size of a hand to 25 meters long and 450 tons of weight.

The Evolution of heat exchangers started thousands of years ago from homes, and nowadays, it leads humankind to reach interstellar activities. In this study, a new innovative shape of a plate heat exchanger will be examined and compared to the conventional types.

1.1. Heat exchanger Classifications

Heat exchangers are categorized in many different ways, due to transfer processes, the number of liquids, surface compactness, flow arrangements, heat transfer mechanisms, type of fluids (gas-gas, gas-liquid, liquid-liquid, gas-two-phase, liquid-two-phase, etc.) and industry. These devices could be categorized due to the construction type and process function, as well.

Most commonly used heat exchangers are: shell-and-tube for fluids, vaporization and condensation executions, and gases at high pressures and temperatures; plate-and-frame and spiral plate for fluids at in relation to low pressures and temperatures; and regenerators (rotary and mixed-matrix type) and ventilator (prime surface, plate-fin, tube-fin type) for flue gas and other contaminated gas implementations that do not have heavy fouling. When the hereinbefore traditional exchangers are used for corruption detainment applications, they do not require any extra design features in general. The classification of heat exchangers is shown in figures 1.1 and 1.2. (Anonymous 1)

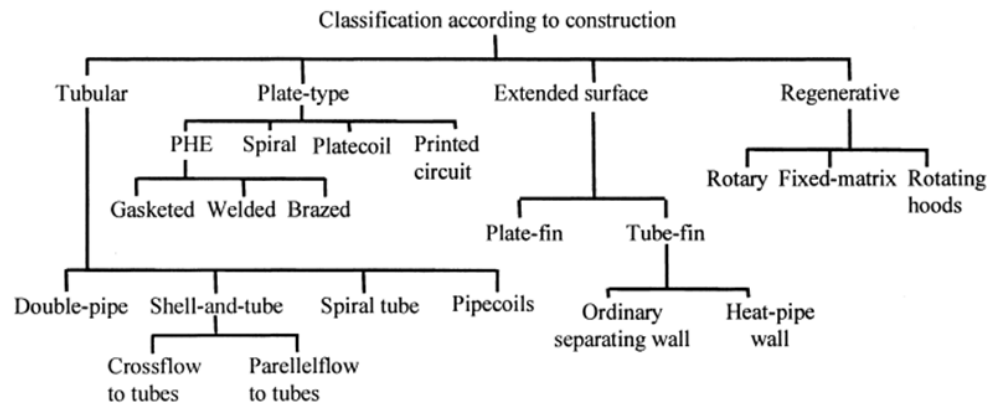


Figure 1.1. Classification of heat exchangers due to construction (Anonymous 1)

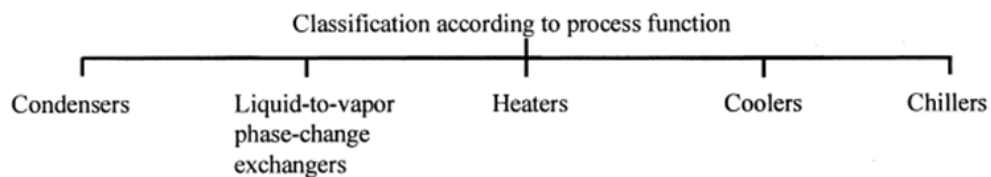


Figure 1.2. Classification of heat exchangers due to process function (Anonymous2)

1.2. Plate Heat Exchanger

In this study, a new plate heat exchanger geometry is examined. A plate heat exchanger system consists of some numbers of plates aligned together with two diagonal flows within. These flows are separated by plates, as shown in Figure 1.3. The surfaces in flow directions are the places that perform heat transfer. In need of a more compact system such as high-pressure liquids welded, semi-welded, and brazed heat exchangers are used. Instead of the pipe passing through a chamber, there are two exchange rooms, generally thin in-depth, divided at their most prominent facet by a corrugated metal plate. The plates used in a plate and frame heat exchanger are acquired by one-piece pressing of metal plates. Stainless steel is a standard metal for the plaques since it is capable of withstanding high temperatures; also, its strength and corrosion resistance is remarkable.

The plates are often tiered by rubber sealing gaskets, which creates a section around the edge of the plates. The plates are pressed to shape gutters at right angles to the stream of the fluid which flows through the channels in the heat exchanger. The plates are pressed together in a compact frame to create an arrangement of parallel flow channels with interchanging hot and cold fluids. The plates propagate an extensive surface area, which permits the quickest possible transfer. Thin-walled chambers increase the volume of the fluid contacts the plate and therefore increases the heat exchange. The gutters also create turbulence over the flow, which maximizes heat transfer in the exchanger. High degrees of turbulence could be obtained at low flow rates, and a high heat transfer coefficient can then be reached.

Comparing with shell and tube heat exchangers, the temperature in plate heat exchangers may be as low as 1 °C, whereas shell and tube heat exchangers need an approximation of 5 °C or more.

For the same amount of heat exchanged, the size of the plate heat exchanger is smaller, because of the large heat transfer area afforded by the plates (the vast area through which heat can travel). An increase and reduction of the heat transfer area are simple in a plate heat exchanger, through the addition or removal of plates from the stack. (Anonymous 2) A plate and frame heat exchanger system is shown in Figure 1.3.

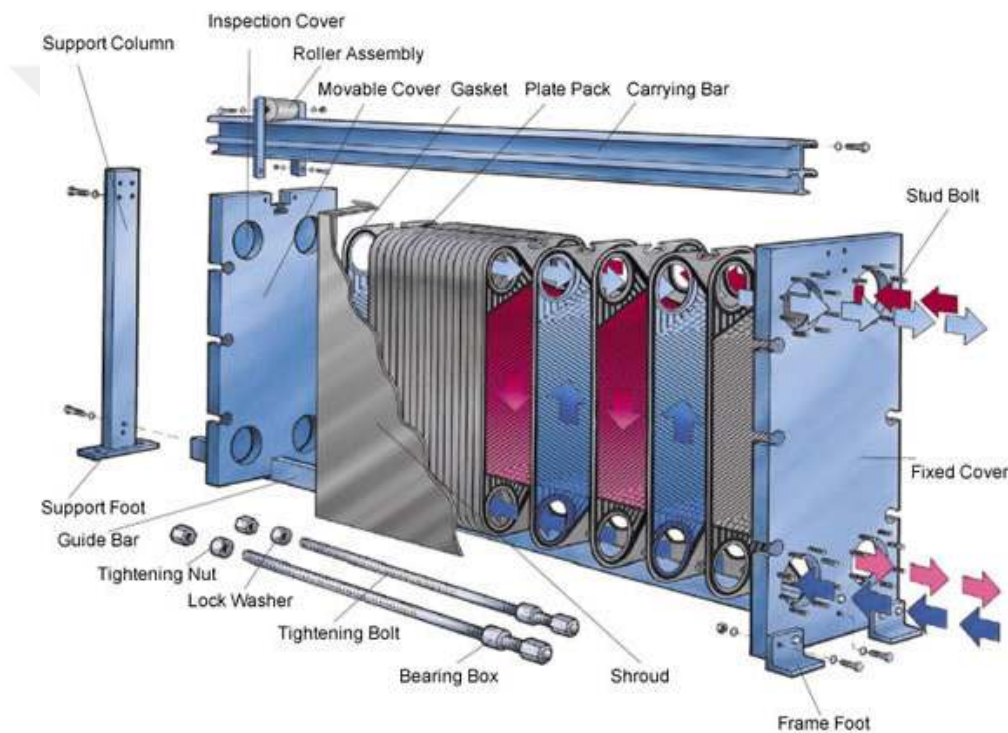


Figure 1.3. Plate and Frame Heat Exchanger System (Anonymous 3)

1.3. The idea of the Study

A plate heat exchanger comprises of thin plates that involve small channels. It has two sides of flow that possess different temperatures; by this mentality, the system generates heat transfer. When designing and operating such a system, one must consider the following restrictions. Flow turbulence control,

pumping power, dimensions, surface geometry, materials to use both as fluid and the system itself. This study had been dealt with the surface geometry aspect. Some experiences inspired this work here is some necessary information regarding this study.

Regenerating the boundary layer to enhance the heat transfer and increasing the surface area in which the heat transfer happens, creating mixing regions to generate higher heat transfer coefficients are plausible for higher values. To reach this idea, the approach of converging and diverging channels is new innovative thinking.

These channels lead to higher surface areas in unit volume and create larger mixing spaces, which considered to be led to higher heat transfer. Pressure drop and mass flow rate need to be contemplated since a drop in these concepts is somehow expected. The real aim of this work must be figured out for the system, and the optimal specifications should have been implemented to achieve the best results.

Heat and flow characteristics of converging and diverging channels are bound up with many parameters as plate size, plate thickness, the distance between plates, plate attack angle, plate temperature, and Reynolds number.

The CFD technique was used to solve the suggested situation. CFD is a branch that uses computer programs, which include a mathematical background and various algorithms to work out and/or analyze a flow problem. The program divides the examining element or system into small meshes and iteratively solve the flow problem for each grid. This approach would lead to very precise solutions if the boundary conditions identified, the meshes generated appropriately, and the solutions interpreted. In this study, ANSYS Fluent is used to solve the problem. At first, the “Space Claim” interface operated to draw the geometry than proceeded the “Meshing,” and after proper progress, the file opened with the Fluent to initiate the analyses.

A study conducted by Erdinc and Yılmaz in 2014 revealed that the validity of the results could be controlled with five main parameters which are: Solutions convergence, independent from the iteration, satisfy the momentum, continuity and energy equations, independent from the cell structure and suitable with the previous studies.

At first, some validations have performed to show the reliability of numerical calculations. Results for parallel plate channel are compared with the numerical solutions to be obtained in the literature. Interrupted plate calculations are validated with a chapter. After these backstage works, the original interrupted sinusoidal plate calculations have been performed.

2. PREVIOUS STUDIES

There are numerous studies and have been still conducting related heat exchanger concepts. Here are some scientific field works inspired this thesis:

Tsuzuki et al. (2009) had examined a microchannel heat exchanger with S-Shaped fins. They used the CFD program on 3D to perform this application. S-Shaped fins were located parallel to the stream direction; by this, they intended to receive the low-pressure drop. The study was about the effects of five different parameters which are: Fin angle, guide wing effect, fin width, fin length, and fin roundness to heat and flow characteristics. Results were stated as; the fin angle is the most effective parameter; the pressure drop increases mostly with the fin angle. The guide wing obtained with radial position and arc perimeter affects the characteristics in a way that if the circumference of the fin arc is 30% - 50% short, then the pressure drop increases. Fin narrowness provides more heat transfer yet reduces the fin efficiency; therefore, the optimal degree shall be found in case of any further application. Longer fins reduce pressure drop; however, it was the less effective parameter observed during this work. Fin roundness has a slight effect on heat transfer; however, it causes a significant pressure drop performance. Lesser fin roundness must be selected. (Tsuzuki, 2009)

Zhang et al. (2015), were looked into the heat and flow characteristics of S-shaped fin channels as well. An s-CO₂ test contrivance built up for this experimental work. Nine different S-shaped fin ducts internal geometries were used to form an optimal geometry. The illustration was designed with the help of NSGA-II, which is a multi-objective evolutionary algorithm. The program developed the optimal design based on two parameters; fin angle and fin length. The study showed that small fin angle ducts with big fin lengths are successful at decreasing pressure drop as big fin angle ducts with small fin lengths are better increasing the thermal effectiveness. (Zhang, 2015)

Ngo et al. (2006) investigated a new PCHE with an S-shaped fin configuration in a hot water supplier operated with CO₂. It is concluded that the new S-shaped finned PCHE provides 3.3 times less volume, 37% less pressure drop in the CO₂ side, and ten times lower in the H₂O side. The results verified in the test facility, as well.

(Ngo, 2006)

Alnak et al. (2018) investigated converging and diverging channels. 3D, unsteady, incompressible, laminar, fully developed flow was used in a channel equipped with semi-spheres in such a way, creating converging and diverging ducts. The ANSYS FLUENT program used to obtain results. This work revealed that the higher Reynolds numbers generate higher heat transfer, with better liquid streaming, and the renovating boundary layer provides higher thermal and hydraulic characteristics. The number and layout of the spheres affect the mixing areas and heat transfer. (Alnak, 2018)

Erdinc and Yılmaz (2014) released a study regarding converging and diverging channels as well. Thermal and hydraulic behavior of 2D, viscous, incompressible, fully developed laminar flow was investigated. A calculation area was determined on the converging and diverging duct, then certain boundary conditions as flow direction inlet outlet and wall had been identified. With the help of ANSYS FLUENT software, momentum continuity and energy equations solved. The investigation was composed of 4 parts. 1st step was parallel plate solution, comparing and verifying the results with literature to ensure further outcomes would be accurate. 2nd step was determining the parameters which affect heat and fluid flow characteristics. 3rd step was about looking into the mixing proportion pressure drop and heat transfer by changing the geometry angle under constant Reynolds number. Furthermore, the 4th step was doing this under different Reynolds numbers. It is concluded that a higher Reynolds number generates higher heat transfer. Creating mixing areas with simple fins in flow direction and changing the angle to generate higher heat transfers was the idea, and it is reached,

yet after 20 degrees, it collapses so there is an optimum value. Thicker fins provided higher Nusselt numbers, and pressure drop increased as well. (Erdinc and Yılmaz, 2014)

Durmuş et al. (2008) investigated pressure drop and heat transfer in plate heat exchangers with different surface profiles. Three different heat exchangers with different surface geometries as flat, corrugated, and asterisk were researched. The reference was a single channel heat exchanger, and flow was laminar, parallel, and counter. In this study, Nusselt number, efficiency, pressure drop, and exergy loss were examined. Upon the experimental results, the corrugated plate heat exchanger had the most heat transfer value. However, pressure drop increases in this type of heat exchanger. The efficiency is many times more valuable than the pressure drop because more efficiency means smaller products to generate the same energy. (Durmuş, 2008)

Zhang et al. (2016) examined the thermal and hydraulic characteristics of new genuine double delta winglets and double delta winglets with holes. The angle of configuration and distance were the effecting parameters. It is concluded that bigger angles and higher distances could enhance heat transfer. In this study, double delta winglets, double delta winglets with holes rectangular wings, and delta wings were compared. Delta and rectangular wings achieved better heat transfer intensification and higher resistance, yet double delta winglets and double delta winglets with holes reached better overall heat transfer performance. (Zhang, 2016)

Min et al. (2010) investigated a rectangular vortex generator and another vortex generator, which obtained cutting the same generators four corners. These two plates were experimentally compared. The results revealed that the modified wings have finer hydraulic and thermal characteristics. (Min, 2010)

Wang and Wanka (1995) studied an accurate numerical scheme that contains a periodic array of wavy passages which looks like sinusoidal located. The heat transfer and flow rates of this system were investigated. The flow was steady until the Reynolds 180 after this value it started to get turbulent. These turbulences provided subrogating near-wall fluid with the fluid in the core area. Thus heat and mass transfer increased. (Wang and Wanka, 1995)

Pehlivan (2013) empirically studied the thermal characteristics of the sinusoidal corrugated duct. The Reynolds numbers were in the range of 1500 to 8000; heat flux was constant at four different angles, and two various channel heights considered upon this study. It is concluded that the Nusselt number depends on the inclination angle. Higher angle, higher Nusselt. Channels extend the flow path and provide better mixing, which leads to higher heat transfer. The corrugated channel is an excellent alternative for high heat flux applications. Converging diverging channels heat transfer performance is relatively higher than the phase arrangement. (Pehlivan, 2013)

Aliabadi et al. (2014) investigated new correlations between heat transfer and pressure drop with three different fluids; water, air, ethylene glycol. The CFD used for this study and values like; fin pitch, fin height, wavelength, fin thickness, wave amplitude, etc. have been taken from previous applications, and the Taguchi method had been performed. Conclusions had been compared with the experimental data. The solutions: Mean deviations regarding Colburn factor “ j ” and Fanning factor “ f ” 3.74 and 9.07 percent respectively. In air correlations, 95 percent of the experimental data were correlated within ± 12 percent. The j factor varied for different fluids, while f is not changed considerably. (Aliabadi, 2014)

Elshafei et al. (2010) experimentally investigated the heat and pressure drop in corrugated channels. Channel spacing and phase shift variations effects on heat transfer and pressure drop discussed. Reynolds number was taken between 3220 to 9420, and wall temperature was uniform.

The study revealed that a significant difference is available comparing the corrugated channel to a conventional parallel plate. Nusselt number increased. However, pressure drop and friction losses are increased as well. Pressure drop decreased while channel spacing increase. For all corrugated channels, the friction factor is much higher than the parallel plate. The effect of phase shift variation on corrugated channels is related to channel spacing. (Elshafei, 2010)

Sriromreun et al. (2012) investigated the effect of Z shaped baffles on heat transfer in a rectangular channel. Z Baffles located as zig-zags to create co-rotating vortex flows, thus increasing turbulence and enhancing the heat transfer. Z baffle heights and pitch spacing lengths were audited to gain the optimal value between the Re 4400 to 20400. Baffles had 45 degrees direct to flow and had three different baffles to channel height ratios; 0,1 – 0,2 – 0,3. Baffle pitch ratios were 1,5 – 2 – 3. In this experimental and numerical study, results showed that the presence of Z baffles dramatically effected the heat transfer and friction loss. The Nusselt number, friction factor, and thermal performance enhancement factor for the in-phase 45° Z baffles are higher than the out phase 45° Z baffles. Larger channel height ratios provided higher heat transfer and friction loss. Shorter pitch lengths meant higher Nu, f, and TEF. (Sriromreun, 2012)

Kim et al. (2011) investigated the heat and flow features of a heat exchanger with offset-strip fins with various fin geometries and different fluids. They concluded the f value is not varying with changing of the working fluid; however, j value changes with the fluid. New j complements are presented as the function of the Prandtl number. The study achieved a 24% higher JF value than the reference offset-strip geometry. (Kim, 2011)

Promvonge (2010) examined the heat and flow features of a channel containing 60° V baffles. The system has 3 various baffle height ratios: 0,1 – 0,2 – 0,3. And three baffle pitch ratios 1– 2 – 3. Reynolds number is taken in the range of 5000 – 25000. Due to experimental results, V baffle increases Nusselt number, friction factor, and thermal enhancement factor dramatically over the smooth wall

channel. Considering the V baffle channels, heat transfer performance using the baffle pitch ratio one and baffle height ratio 0,1 provides the best thermal enhancement factor at lower Re. (Promvonge, 2010)

Peng et al. (2014) examined the performance of innovative offset strip fin array in compact heat exchangers. It has done experimentally and numerically. Five different fin schemes are investigated in different airflow speeds and constant inlet steam pressure. Reynolds ranged from 500 to 5000. Thermal performance highly depends on fin pitch, fin length, and fin bending distance. The j , f , and point of transition calculated from the empirical data. Then heat flow performances had been calculated numerically. Maximum Nusselt number observed at the channel inlet. Vertical whirlwinds near the fin area increase the velocity gradient and decrease the thickness of the boundary layer to enhance heat transfer. (Peng, 2014)

Sinha et al. (2013) investigated the heat transfer augmentation of a plate-fin heat exchanger with the help of winglet type vortex generators. Five different configurations took part in this study. These were common-flow up in series, common-flow down in series, combined, inline rows of winglet and staggered rows of the winglet. Reynolds number considered in a range of 250 to 1580. The conclusion is the best result obtained in common-flow up in series in terms of heat transfer and quality factor. (Sinha, 2013)

Gut and Pinto (2004) investigated an optimal configuration design for a plate heat exchanger. A Formula created which minimizing the heat transfer area concerning constraints on the number of channels, pressure drops, flow speed, and thermal effectiveness. (Gut and Pinto, 2004)

Saha and Sarkar (2014) investigated a plate-fin exchanger performance from the point of view regarding the flow regime effects on heat transfer. Delta and rectangular vortex generators used, and the study performed both streamwise and spanwise. Shape and flow arrangements affect the heat transfer observed. An innovative new approach called field synergy principle used for explaining the heat transfer growth. The performance assessed with the help of; Nusselt number,

pressure loss, performance evaluation factor, and domain averaged synergy angle. The research revealed that rectangular vortex generators create more heat transfer enhancement; higher Nusselt number is associated with smaller synergy angle or better coordination between the velocity vector and temperature gradient. (Saha and Sarkar, 2014)

Aktürk et al. (2011) empirically examined a chevron type plate heat exchanger in different Reynolds numbers from 500 to 5500. Hydraulic and thermal values have taken on entry and exit ports, and pressure drop values are also calculated. Reynolds number effect on Overall heat transfer coefficient and pressure loss is investigated and compared with former approaches. (Aktürk, 2011)

Oviedo-Tolentino (2009) experimentally researched the sinusoidal shaped converging and diverging configuration effect on the flow regime. The investigation concluded as follows; the diverge area increased chaotic mixing while the converge area decreased even in high Reynolds numbers. Escalation of the mixing may promote the heat transfer; however, it also increases the pressure drop. (Oviedo-Tolentino, 2009)

Yin et al. (2012) numerically observed heat and flow behaviors of corrugated sinusoidal wavy channels have different phase shifts for the upper and lower element. Wall temperature was stable, and Reynolds was varying from 2000 to 10000. The study revealed that corrugated channels had a great heat transfer value with a large pressure loss and phase shift activity is more effective on high Reynolds number. The optimum scenario achieved at 0° at lower Re. (Yin, 2012)

Naphon (2009) investigated the heat and fluid flow distributions for different shaped wavy plates. Plate forms, plate layout, and airflow rate effects on thermal and hydraulic behavior are observed. An unstable thermal boundary is achieved. Sharp edges promoted the heat transfer as V-shape. (Naphon, 2009)

Bahaidarah (2013) examined the sinusoidal wavy passages and entropy generation relations on 2D steady laminar flow. The features that lead to this investigation were height ratio, module length, and Re. The study concluded that

Reynolds number has an eloquent effect on entropy generation. The height ratio took part in entropy generation distribution; higher values increase the uniformity of the allotment. Module length ratio plays a role in distribution as well, however not as much as the height ratio. (Bahaidarah, 2013)

Ismail and Velraj (2009) investigated j and f factors on offset and wavy fins, compact plate-fin heat exchangers. ANSYS FLUENT is used for this approach. Both laminar and turbulent flow scenario is examined. The research showed that the f factor must be multiplied with 1.6 to generate the real value. The j factor is more accurate comparing the f . (Ismail and Velraj, 2009)

Mitra and Blomerius (2000) investigated flow and heat characteristics numerically in corrugated ducts with the help of Navier-Stokes equations for Re 600 to 2000. 2D and 3D analyses have been performed separately, and the results have been noted. The angles between corrugations were the other criteria. The 2D approach revealed the beginning of flow oscillation and heat transfer increase relation. The self-sustained oscillation occurs on higher Re in 3D than 2D. 3D plates with a 45° inclination angle had the best results because it needs the lowest heat transfer area. (Mitra and Blomerius, 2000)

Wang et al. (2019) investigated the effect of inlet temperature and structure parameters as wave amplitude, wavelength on flow boiling heat transfer of half sinusoidal corrugated copper microchannels experimentally at a constant mass flux value. The study concluded that; geometric structures have joint actions on the heat transfer performance. Higher input power causes instabilities. Higher inlet temperature decreases the oscillation period and the fluctuation amplitude. (Wang, 2019)

Udaiyakumar et al. (2018) simulated a compact heat exchanger and analyzed the effect of fin profile to heat transfer without increasing the volume of CHE. Then study supported with experimental results. CFD simulation used for this work to achieve better geometries. The main aim of the study was to enhance heat transfer without increasing pressure drop extensively. The best results

obtained in the CFD program were then experimentally implemented. The improvement found 28% in terms of heat transfer, and 4.5% is found for pressure drop. (Udaiyakumar, 2018)

Ionescu and Neagu (2017) numerically investigated heat transfer and fluid flow characteristics of various sinusoidal wavy channel models with different spacing ratios. Channel dimensions had four different values, and wave amplitude also had different values. The results showed that the thermal boundary layer gets thinner in concave positions, and the local Nusselt number increases as the spacing ratio increases. Friction coefficient from concave to the convex position of channels, increased with waviness parameter was raised from 0.084 to 1 and decreased from 2.5 to 1. (Ionescu and Neagu, 2017)

Harikrishnan and Tiwari (2019) investigated flow and heat transfer characteristics of the 3D sinusoidal wavy channel with secondary corrugations. CFD technique is used for this work. Different wave amplitudes and the number of waves in streamwise as well as spanwise directions for fixed width and height of the channel are computed by the CFD program. Flow and heat transfer characteristics of secondary corrugated channels were compared with channels having spanwise and streamwise corrugations alone. Volume-weighted average of secondary flow shown to point out the effect of secondary flow in different corrugated channels. (Harikrishnan and Tiwari, 2019)

Zontul et al. (2017) numerically investigated heat transfer enhancement of laminar pulsating flow in a wavy channel. The wavy channel had constant wall temperature, and its geometric parameters are constant, as well. The SIMPLE algorithm was used for the study. Reynolds numbers were used as follows $200 \leq Re \leq 800$. The differences between flow geometries and pulsating and steady flow were compared. The study concluded that pulsating flow considerably enhances heat transfer. (Zontul, 2017)

Arasteh et al. (2019) numerically investigated heat and fluid flow characteristics of a counter-flow sinusoidal parallel plate heat exchanger using

metal foam in ducts different zones. Two different wave amplitudes, two wavelengths, three various particle diameters, and three different thickness values had been used. A dimensionless number called performance evaluation criteria (PEC) had been defined to rate the heat transfer increment vs pumping power. The investigation concluded that the heat transfer rate, effectiveness and overall heat transfer coefficient of the heat exchanger were raised to 19.2%, and the PEC is enhanced to 1.171 in a particular optimum case which wave amplitude, wavelength, metal foam particle diameter and thickness are; 0.6 cm, 6 cm, 0.01 mm and $2A/3$, respectively. Metal foam effect on pumping power found as indirect. (Arasteh, 2019)

Kurtulmuş (2019) investigated thermal performance characteristics and hydrodynamic of the fluid, which improves the rate of heat transfer and pressure drop through streamlines topology, Velocity behaviors, vorticity concentrations, and turbulent Reynolds stress for a sinusoidal channel. Comprehensive experiments had been done with various Reynolds numbers between 4×10^3 to 1×10^4 to state heat transfer rate and friction factor, with varying channel height ratios, which defined as $M = H_{\min}/H_{\max}$. The experiments concluded that a tremendous heat transfer rate obtained; however, pressure drop also raised significantly. The highest value obtained for the thermal performance factor was 1.46 for $M=0.5$. Then experiments performed for $M=0.5$ pulsating flow and Reynolds number $4 \times 10^3 \leq Re \leq 7 \times 10^4$ and non-dimensional pulsation frequency (Strouhal number) range having $0.12 \leq St \leq 2.06$. The study concluded that, 50% heat transfer enhancement achieved with a significant pressure drop, steady-state flow to pulsating flow. Particle image velocimetry(PIV) used for flow physics investigation. (Kurtulmuş, 2019)

3. MATERIAL AND METHOD

In this study, CFD is used to analyze an innovative plate's behavior under different geometry parameters. The flow is under the assumption of 2D, laminar, incompressible, viscous, and fully developed.

3.1. Flow Inside Tubes

The flow in a tube can be laminar or turbulent, depending on the flow conditions. The Reynolds number provides a convenient criterion for determining the flow regime in a tube, although the roughness of the tube surface and the fluctuations in the flow have considerable influence. The critical Reynolds number for flow in a tube is generally accepted to be 2300. Therefore,

$Re < 2300$	laminar flow
$2300 \leq Re \leq 4000$	transition to turbulence
$Re > 4000$	turbulent flow

3.1.1. Laminar Flow Through A Duct

Laminar flow inside tubes can be investigated in two groups. These are the *Entrance region* (developing flow) and *Fully developed flow*. As the flow enters downstream from the mouth, the flow develops a boundary layer region that coats the inner surface of the tube wall. For this reason, the longitudinal position, where the velocity boundary layer thickness has grown to the same size as the tube radius. In other words, the longitudinal position where the inflowing stream has had enough time to feel the effect of wall friction (transversal viscous diffusion) to its centerline. (Akgun, 2003)

3.1.2. Friction Factor and Pressure Drop for Laminar Flow

The longitudinal velocity u is not a function of longitudinal position x ; for this reason, the wall shear effect τ_w is also independent of x . So τ_w can be calculated from the below equation. (Akgun, 2003)

τ_w value can be express in the dimensionless form:

$$f = \frac{\tau_w}{\frac{1}{2}\rho V^2} \quad (3.1)$$

3.2. Convective Heat Transfer Inside Tubes

Energy is transferred from a surface to liquid streaming over it because of the temperature difference between the surface and the fluid, called convection. The convective heat transfer rate depends on the specifications of the flow field.

If the temperature at some point on the surface is T_w and if the rate is being transferred locally at this point from the surface to the liquid per unit surface area is q , then it is usual to define a quantity, h , such that:

$$q = h(T_w - T_f) \quad (3.2)$$

It is appropriate to determine the *mean temperature difference* between the wall and the flow as:

$$\Delta T = (T_w - T_m) \quad (3.3)$$

This study numerically analyzes a specific geometry using the CFD method, as it has mentioned at the beginning of this chapter. First, the geometry identified afterwards the method applied to define the behavior under some variable parameters.

To sum up, the instrument is ANSYS Workbench plus some modules of this software, and the idea is Computational Fluid Dynamics. In the thinking phase of the work, previous experiences have taken into consideration; geometry had been chosen via these experiences.

3.3. CFD Phenomenon

CFD is an Engineering tool applied to complex fluid problems and intends to solve or simulate the correct behavior of the system. It uses finite difference, finite element, and finite volume methods (FDM – FEM - FVM) to achieve the results. In addition to that, momentum, continuity, and energy equations are using to achieve the results. CFD mentality gives motivation and simplification to figure out problems; however, the initial values, boundaries, calculation method, problem definition should be known and written accurately.

3.3.1. ANSYS Workbench

ANSYS is the most common and relatively accurate program used in the industry. It helps to solve the fluid programs correctly and assess the results to conclude. The software shows some diagrams, vivid simulations, and flow characteristics; therefore, one can easily animate the reviewed situation. ANSYS 19 has been used for this work with the modules of Space Claim, Fluent, Meshing, and CFD Post.

3.3.2. Solving The Problem

Figure 3.1 shows the stages one by one during the solution process. All steps performed previously mentioned ANSYS Workbench.

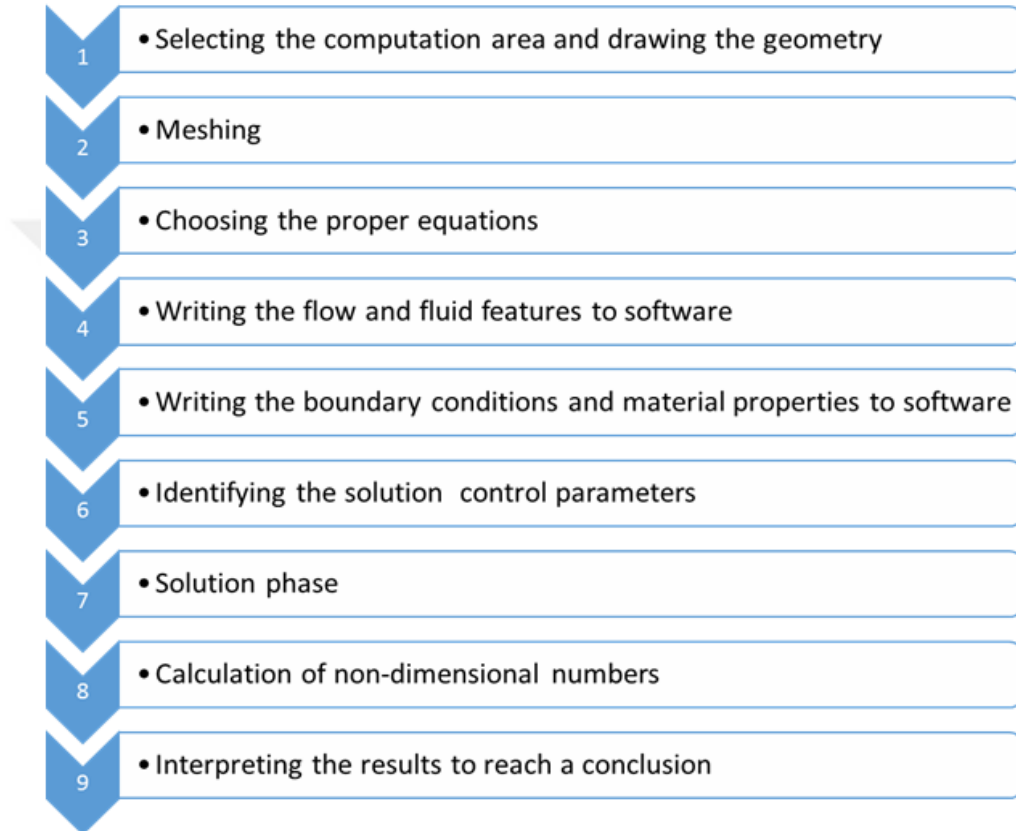


Figure 3.1. Solution phases for the identified problem

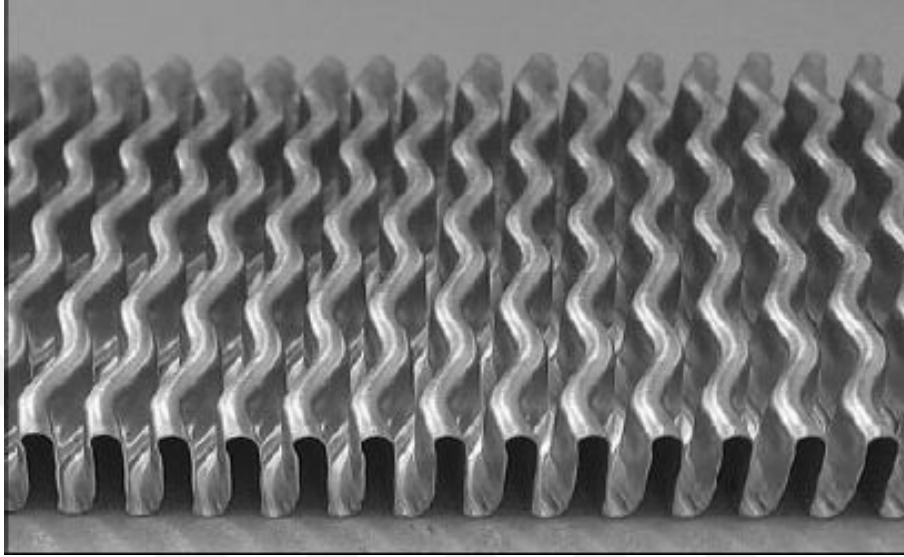
3.3.3. Computation Area and Parameters Affect the Problem

Figure 3.2. Representative picture of the wavy passages (Anonymous 4)

This study deals with the converging and diverging channels from a different point of view, wavy or sinusoidal channels. Figure 3.2 is a representative picture of imagination. For this problem, heat transfer performed with forced convection through these channels will be examined. A 2D representation for this subject shown in figure 3.3.

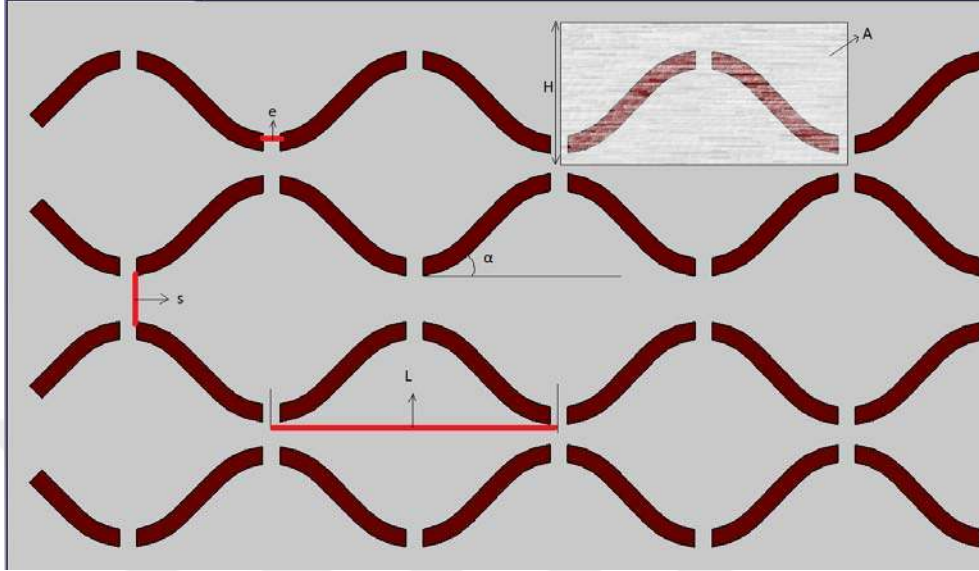


Figure 3.3. Variables affecting the study

Figure 3.3 shows the variables affecting this study. L , s , α , and e are identified. L and s are constant values. Other values are parametric. In addition to that, thickness and edge curves are implemented. Figure 3.4 shows the problem-solving procedure on FLUENT.

3.3.3.1 Wave Amplitude Definition and Wave Drawing

Wave amplitude frequently mentioned and a big part of this study. This object based on a cosine wave, which is shown in figure 3.4. It had been drawn in space claim with the help of the equation interface. The Wave amplitude represents a cosine wave amplitude. It usually is 1, and a period (length) is 2π . However, in this work, the scale had been chosen ten, which means the wavelength is ten times a cosine wave 20π , and the amplitude has chosen four different values, which are 0.3 – 0.4 – 0.5 and 0.7.

A brief explanation of wave drawing is available on attachments.

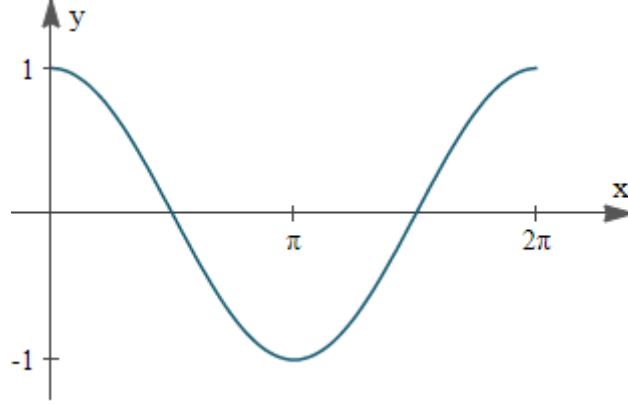


Figure 3.4. Cosine wave representation

3.3.4. Meshing Process

Meshing is the critical process for an accurate solution to a CFD problem. The quality of mesh should be convenient to reach a proper solution. Considering this, a mesh must be performed to the problem precisely. Skewness should be low; it takes values between 0 to 1. Closer to 0 means low skewness and quality mesh, and closer to 1 means high skewness and low quality. In this study, meshes are varying between 0.2-0.6. Figure 3.5 and 3.6 represents the meshing process for 0.4 wave amplitude, sharp edge, 5 mm internal space. The figure is attached to claim the quality of the mesh.

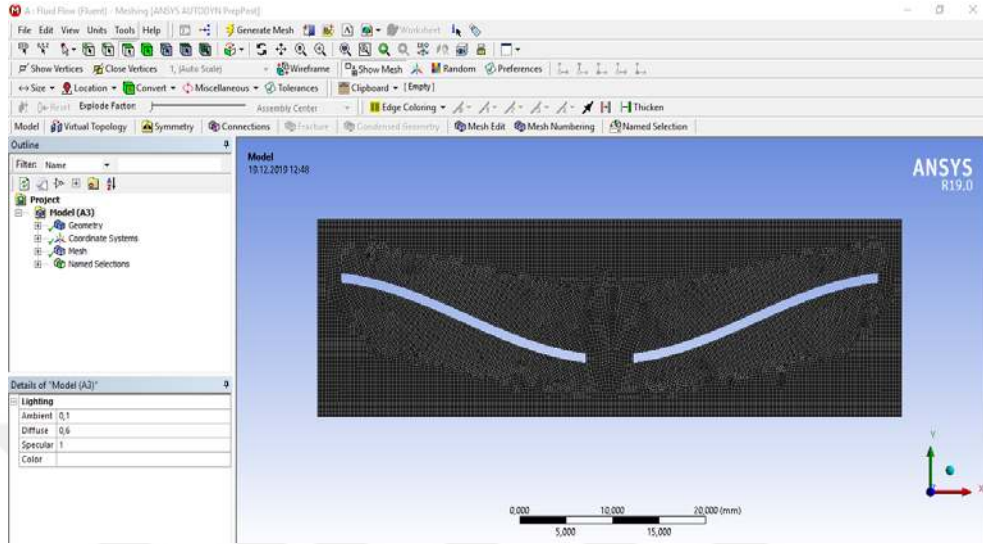


Figure 3.5. Meshing Process for 0.4 wave amplitude sharp edge, 5mm internal space

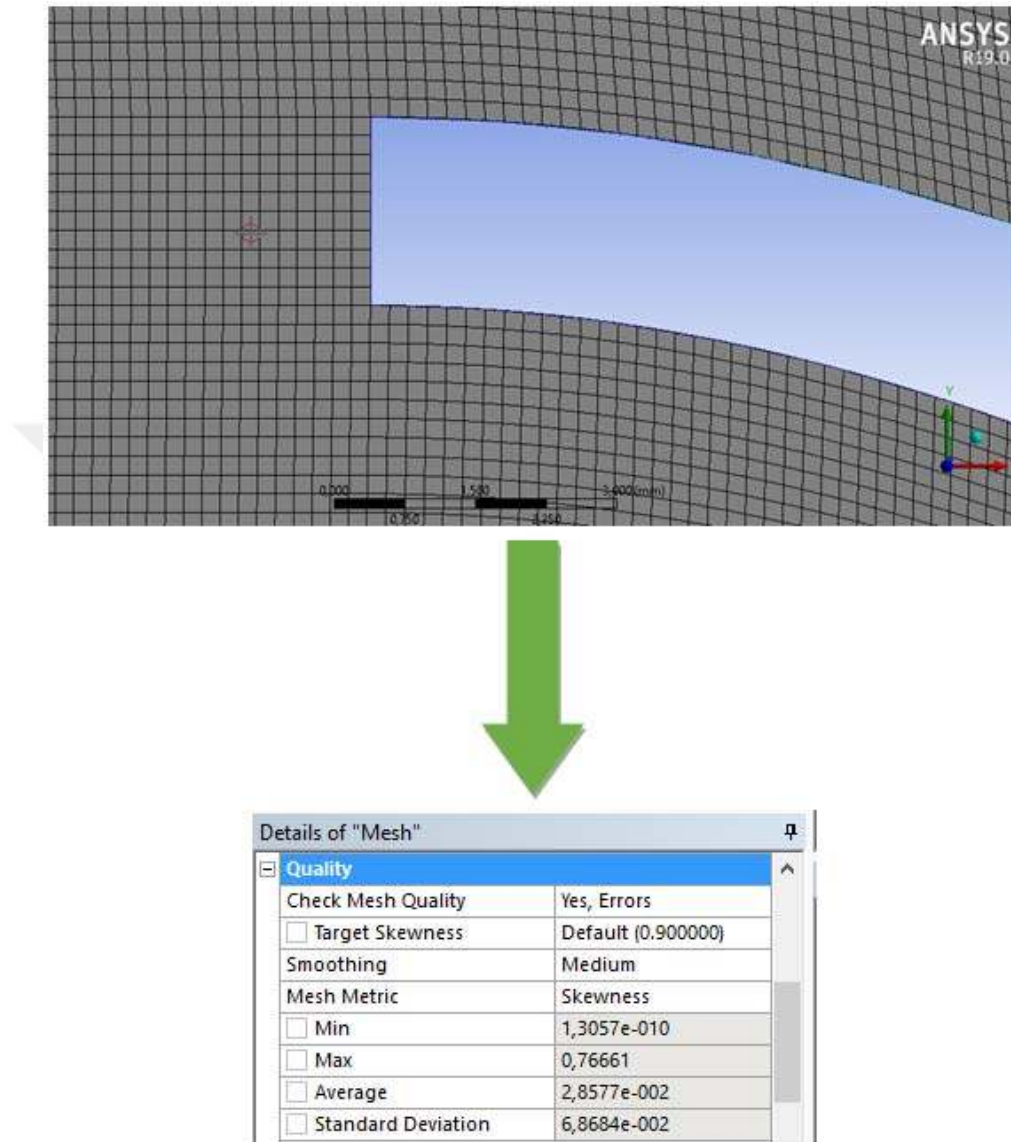


Figure 3.6. Mesh quality for 0.4 wave amplitude sharp edge, 5mm internal space

3.4. Mathematical Modeling

The fluid features used for this study are; 2D, Laminar, incompressible. Thus, the equations regarding the mentality are (Cengel, 2005):

Continuity Equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (3.4)$$

Where x is the dimension through the stream, y is the dimension longitudinal to stream, u is the velocity along the x-direction, and v is the velocity along the y-direction.

Other equations concerning the flow are the momentum equations of x and y components. (Cengel, 2005)

Momentum Equation on x-direction:

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (3.5)$$

Momentum Equation on y-direction (Cengel, 2005):

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \quad (3.6)$$

Where P is pressure ρ is the fluid density, and ν is kinematic viscosity.

Lastly, the energy equation for 2D, steady, incompressible, laminar flow is:

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (3.7)$$

Where α is the convection factor.

$$\alpha = \frac{k}{\rho} \cdot c_p \quad (3.8)$$

k = Thermal conductivity coefficient

c_p = Specific heat capacity

Some assumptions had been made to reach a comparable result. These criteria are 2D, incompressible, fully developed, viscous, and laminar, shall be selected on the FLUENT program in the beginning. (These conditions are shown in attachments)

3.5. Boundary Conditions

In figure 3.7, the representative computation area and computation mentality are shown.

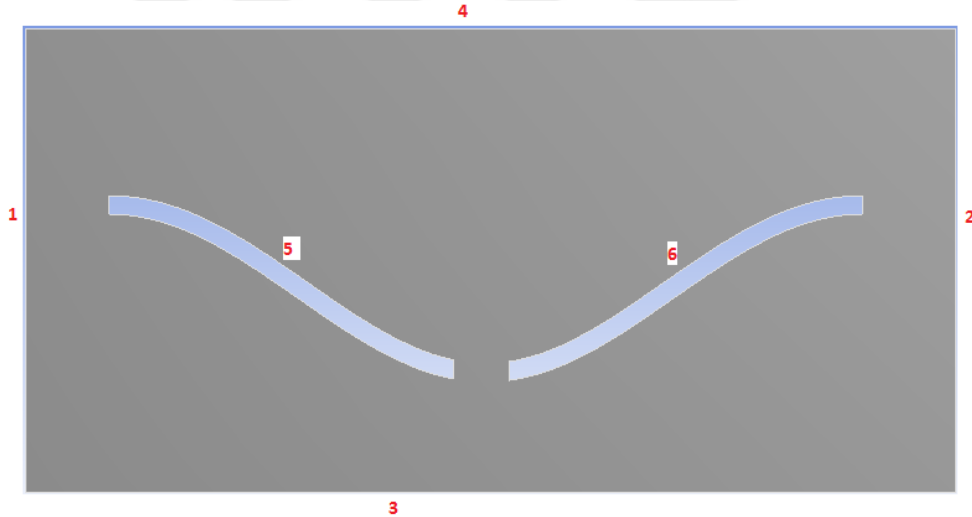


Figure 3.7. The solution area representation

These edges were defined during the meshing process. Region 1 is the inlet, and region 2 is outlet, regions 3 and 4 are symmetry boundaries, 5 and 6 are the internal walls which perform the heat transfer. Since the 3 and 4 numbered regions are the walls the velocity over that regions are 0, it can be shown as (Cengel, 2005):

$$v = 0 \quad (3.9)$$

$$\frac{\partial u}{\partial y} = 0 \quad (3.10)$$

$$\frac{\partial T}{\partial y} = 0 \quad (3.11)$$

In boundary conditions interface; inlet-outlet, walls, and symmetry edges should be defined. For this study, walls were taken as the no-slip condition, and wall temperature was taken as 330 K. (The figures and detailed information could be found in attachments)

3.5.1. Constant Temperature condition on the Sinusoidal Walls

Regions 1 and 2 represent the inlet and outlet of the fluid flow. In this study, the flow is considered/assumed hydrodynamic and thermally fully developed, which shown in figure 3.8.

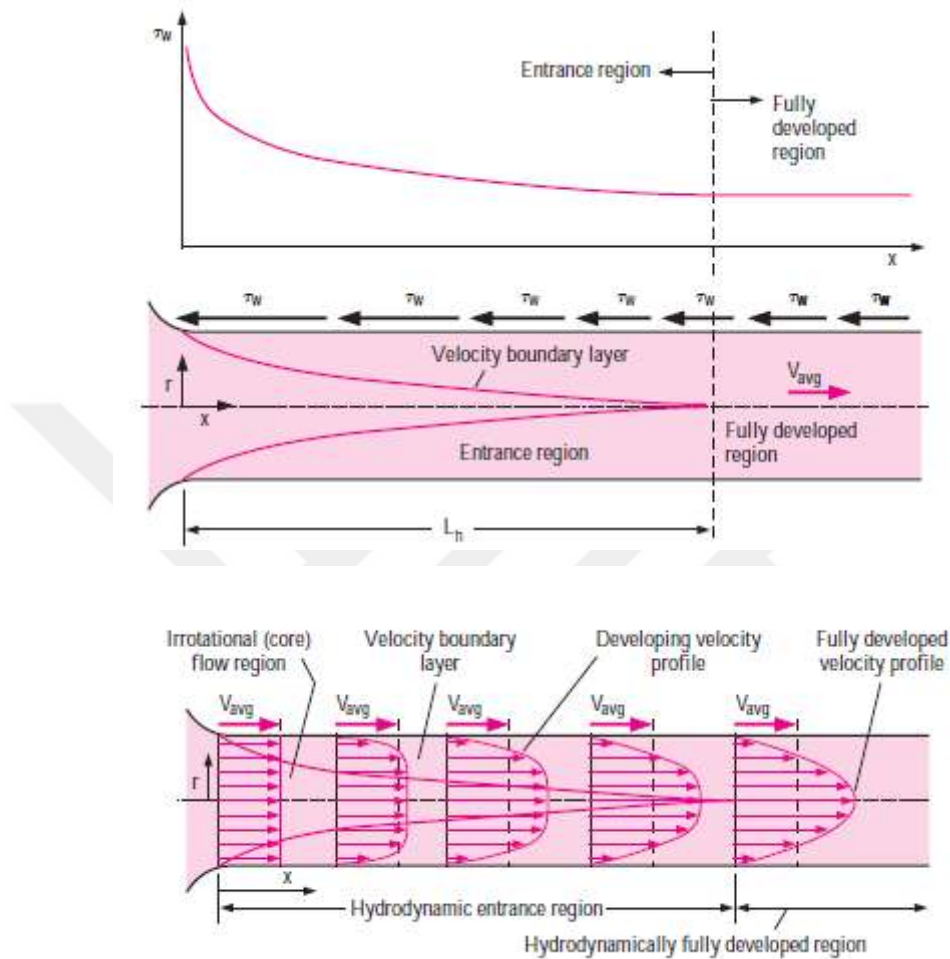


Figure 3.8. Hydrodynamically and thermally fully developed flow in channels (Cengel, 2005)

Hydrodynamic fully developed flow (Cengel, 2005):

$$u(y, x) = u(y, x + L) \quad (3.12)$$

$$v(y, x) = v(y, x + L) \quad (3.13)$$

H is height L is the length of the calculation area.

Thermally fully developed flow is represented as (Cengel, 2005):

$$\frac{T(y, x)}{T(x)} = \frac{T(y, x + L)}{T(x + L)} \quad (3.14)$$

Hydrodynamically and thermally fully developed flow restriction is given by periodic boundary condition on the FLUENT program. One needs to write the following elements through the command line:

**Mesh → modify-zones → make-period → no → yes → yes

After this implementation, the periodic condition is ready. Mass flow rate is given as inlet condition which found by:

$$\dot{M} = \rho u_m A \quad (3.15)$$

Reynolds number and equivalent diameter are represented as:

$$Re = \frac{\rho u_m d_e}{\mu} \quad (3.16)$$

$$d_e = \frac{4.V}{F_p} \quad (3.17)$$

Figure 3.9 shows the representations of the symbols given above.

$\dot{M}$	=	Mass flow rate
ρ	=	Density
A	=	Cross-sectional area
Re	=	Reynolds Number
u_m	=	Mean fluid velocity
d_e	=	Equivalent diameter
V	=	Volume of calculation area
μ	=	Dynamic viscosity
F_p	=	Wavy walls total circumference

Figure 3.9. Representation of the symbols

These equations are the background applicators. The inlet shall be chosen, and under the periodic conditions transaction, the mass flow rate shall be written, which calculated from the excel document prepared for the study, to implement the periodic boundary condition. (Further read-Attachments)

3.6. Solution Algorithm

To obtain correct and precise results, at first, some information must have been given concerning the CFD. The CFD application used in this thesis is ANSYS Fluent. The necessary mentality for ANSYS FLUENT solution is currently mentioned in figure 3.4. A detailed and profound approach to the topic is the solution algorithm. Fluent has five different solution types that might have been chosen for the relevant case. These are:

- SIMPLE
- SIMPLEC
- PISO
- FSM
- Coupled

SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) technique was chosen among these five versions for the innovative plate fluid and heat characteristics solution. It is compatible with periodic flows. The software uses a numerical approach to solve the Navier-Stokes equations for the defined case. It uses iterations to converge and derive a conclusion.

SIMPLE:

The algorithm is iterative. The necessary steps in the solution update are as follows:

1. To define boundary conditions.
2. Calculating the gradients of pressure and velocity
3. Solving the momentum equation to calculate the velocity zone.
4. Calculating the unreformed mass fluxes at surfaces.
5. Solving the pressure correction equation to produce cell values of the pressure correction.
6. Updating the pressure zone.
7. Updating the boundary pressure corrections
8. Correcting the surface mass fluxes:
9. Correcting the cell velocities
10. Updating density according to pressure variation. (Anonymous 3)

SIMPLE simply:

- An estimation of the velocity field is acquired by computing the momentum equation. The pressure gradient is obtained using the pressure distribution from an earlier iteration or an initial guess.
- The pressure equation is formulated and calculated to get the new pressure distribution.
- Velocities are retrieved, and a new set of conservative fluxes is gained.

The discretized momentum equation and the pressure correction equation are solved implicitly, where the velocity correction is solved explicitly.

(Anonymous 4)

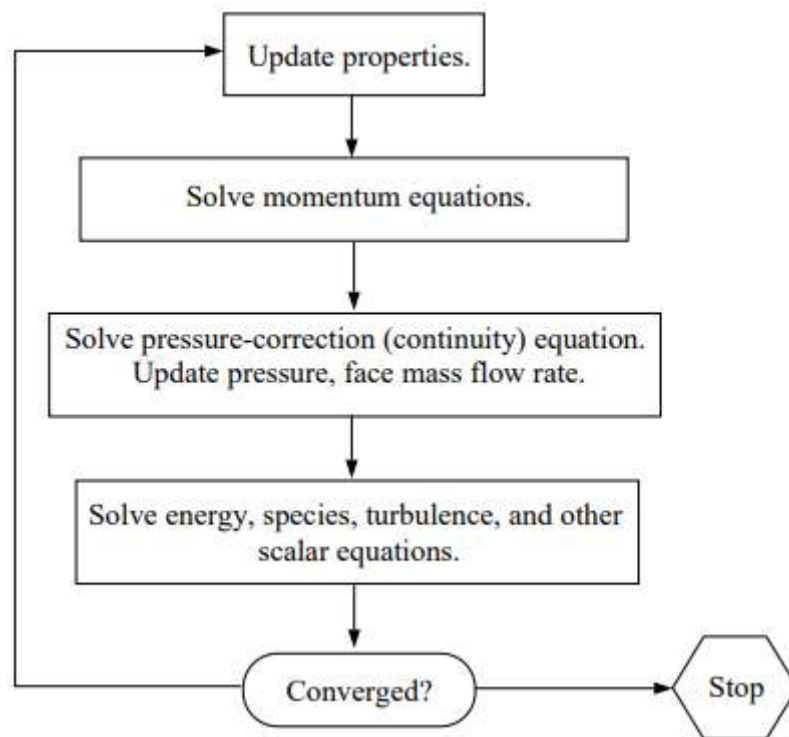


Figure 3.10. SIMPLE technique solution algorithm (Anonymous5)

3.7. Convergence

Convergence is one of the main criteria in Computational Fluid Dynamics. The iterative algorithm always includes an error. To minimize errors, a method named “under relaxation” is used. This factor is 0.3 for pressure equations and 0.7 for momentum equations. It has been selected 1 for energy equation and structural force and density. Convergence degree is $10e-6$ for continuity and momentum equations; however, its $10e-9$ for energy equation. The analysis shall never be ended before it converges and solution completed icon displayed. Figure 3.10 shows this topic.

3.8. Calculating the Dimensionless Numbers

CFD Post Program was used to determine the Nusselt number and friction factor.

3.8.1 Average Nusselt Number Calculation

Convective heat transfer Formula:

$$h = \frac{Q}{F_p \Delta T_m} \quad (3.18)$$

$Q =$ Heat passing through the Wall

$\Delta T_m =$ Logarithmic mean temperature

$F_p =$ Plate Surface Area

The heat on the channel edge

$$Q = \dot{M} \cdot c_p (T_o - T_i) \quad (3.19)$$

Logarithmic heat difference:

$$\Delta T_m = \frac{(T_w - T_i) - (T_w - T_o)}{\ln \frac{(T_w - T_i)}{(T_w - T_o)}} \quad (3.20)$$

c_p = Specific heat capacity

T_o = Temperature Outlet

T_i = Temperature Inlet

T_w = Wall temperature

So the Nu is:

$$Nu = \frac{h d_s}{k} \quad (3.21)$$

3.8.2. Friction Factor Calculation

Friction Factor Formula:

$$f = \frac{\Delta P}{L} \frac{d_s}{\rho \frac{u_m^2}{2}} \quad (3.22)$$

ΔP = Pressure loss in the computational area

$\frac{\Delta P}{L}$ = Pressure gradient.

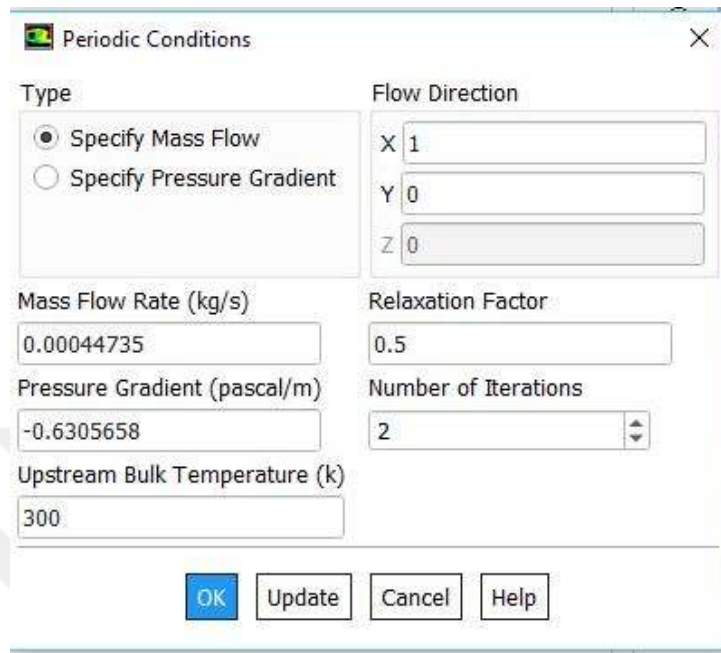


Figure 3.11. The pressure gradient is calculated from Fluent

It had been found in fluent, as shown in figure 3.11. This value was obtained after the analysis. The pressure gradient helps to get the f value so that thermohydraulic performance could have been calculated.

4. RESULTS AND DISCUSSION

This study is an approach for an innovative plate geometry. After the calculations, it is claimed that the newly created shape is more efficient than some geometries in literature. A parallel plate PHE is examined to show this suggestion.

4.1. Validation

4.1.1. Parallel Plate Validation

In this chapter, a validation had been executed to show the reliability of solutions. At first, a 25x100mm parallel plate was examined, and the solutions were compared with the literature. Figure 4.1 presents a parallel plate geometry. It had been drawn in “Spaceclaim.”



Figure 4.1. Parallel plate geometry projection

Boundary conditions table:

Table 4.2 Boundary conditions for parallel plate

Boundary Conditions	Value
T_{wall}	330 K
T_{inlet}	300 K

$$f = \frac{Re}{96} \quad (4.1)$$

$$Nu = 7.541 \quad (4.2)$$

Table 4.2 and 4.3 represents the friction factor and Nusselt number comparisons for different Reynolds Numbers between the analytical literature results and computational results.

Table 4.3 Comparison between computed and literature results for friction factor

RE	f(eq.4.1)	f(computed)
100	0.96	0.96089
200	0.48	0.48103
400	0.24	0.24024
800	0.12	0.12008

Table 4.4 Comparison between computed and literature results for Nu

RE	Nu(eq.4.2)	Nu(computed)
100	7.541	7.5403
200	7.541	7.5301
400	7.541	7.529
800	7.541	7.5251

Error level is acceptable, which has a value of lower than 0.25 both for Nu and f. Figure 4.2 and Figure 4.3 present the velocity and temperature distributions.

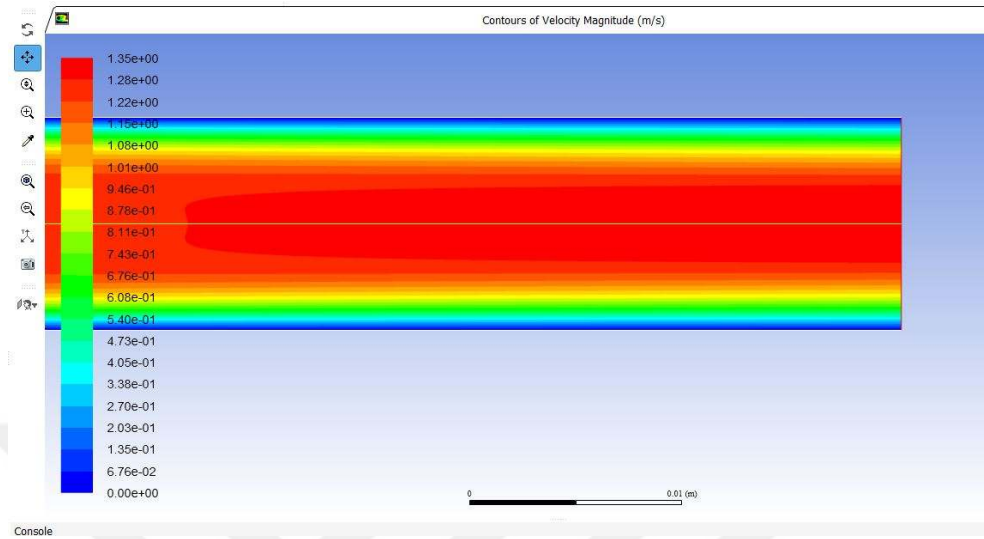


Figure 4.2. Parallel plate velocity contour

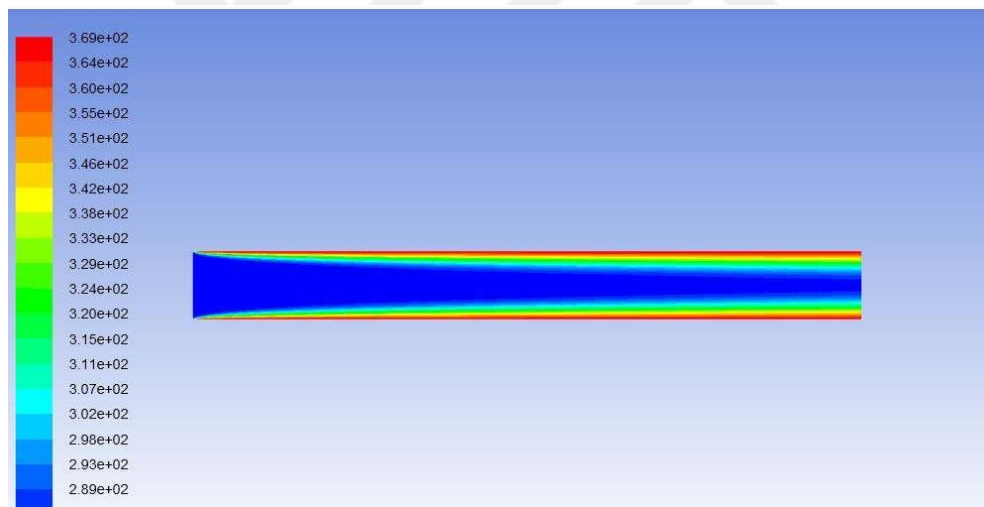


Figure 4.3. Parallel plate temperature contour

4.1.2. The Second Phase of Validation, Interrupted Plates

For this section of the study, Literature examinations investigated and verified. In the mentioned work, there are tables for Nu and f values for interrupted plates. Some of these results are validated as follows.

The geometry is shown in figure 4.4. Thicknesses and spaces between plates were parametric in the mentioned work. Validations are done due to two different spaces and just one thickness value for this study.

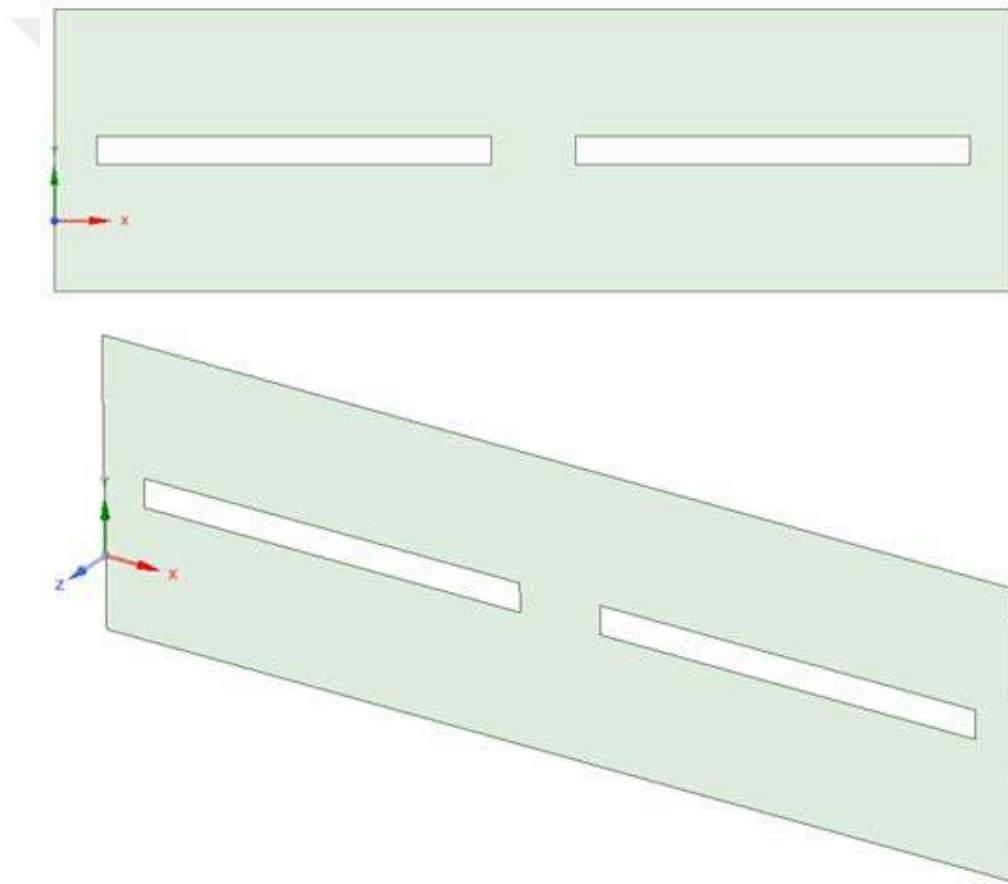


Figure 4.4. Interrupted Plate projection

Nusselt Values For $t=2$, $e=10$ are given in table 4.4

Table 4.5 Comparison between computed and literature results for Nu

RE	Nu(literature)	Nu(computed)
100	8.412	8.422
200	8.536	8.549
400	8.659	8.670
800	8.792	8.801

Nusselt values for $t=2$, $e=2$ are given in table 4.5

Table 4.6 Comparison between computed and literature results for Nu

RE	Nu(literature)	Nu(computed)
100	8.220	8.230
200	8.300	8.312
400	8.340	8.359
800	8.680	8.695

f values for $t=2$, $e=10$ are given in table 4.6

Table 4.7 Comparison between computed and literature results for f

RE	f(literature)	f(computed)
100	1.262	1.260
200	0.625	0.629
400	0.310	0.314
800	0.156	0.157

f values for $t=2$ $e=2$ are given in table 4.7

Table 4.8 Comparison between computed and literature results for f

RE	f(literature)	f(computed)
100	1.217	1.2178
200	0.608	0.6089
400	0.304	0.3044
800	0.152	0.1522

Validations show that the following data, including genuine work, are reliable. These confirmations also contain the thinking of mesh independence. In the following chapter, this topic will be discussed.

4.2. Mesh Independence

When the percentage difference of two successful meshes is negligible, one can continue with the “worse one” of both meshes. A mesh independent solution can be obtained with this projection, keeping the computational economy intact. One can have mesh independence by an excellent uniform mesh. Nevertheless, that would take a considerable amount of time. That is why a mesh independence approach with a non-uniform coarser mesh shall be made the size of the element is varied, and its effect on the final solution after convergence is checked.

In this study, this had been effectively implemented for all generated meshes. At first, a simple non-conformed mesh is used with a small number between 10000-15000 (Changes due to geometry) than averagely 35000-40000 element mesh is used, and afterwards, 70000-80000 element mesh performed. After the solutions checking, the average 35000-40000 seemed to be accurate enough to proceed. All other calculations have performed regarding this data.

The work relies on a series of parameters as:

- Thickness effect
- Wave amplitude
- Radius effect on walls
- Space between walls
- Vertically interrupted wall geometries
- Reynolds Number

The first topic is the thickness effect.

4.3. Calculations

4.3.1. Wall Thickness Investigation

Two different thickness values had been examined for this study. Thickness values are 1 and 3 millimeters. Sharp shaped, four different wave amplitude geometry with the same internal space of 5 mm investigated. Figures 4.5 and 4.6 show the thickness difference.

Figure 4.5 shows the distribution of Nu with respect to wave amplitude for different thicknesses. In this figure, it is seen that with an increase in wave amplitude, the Nusselt number increases. Furthermore, the thinner geometry has better Nusselt values than the thicker one. Nearly 0.4 Nusselt number difference is seen between 1mm and 3mm geometry in every wave amplitude.

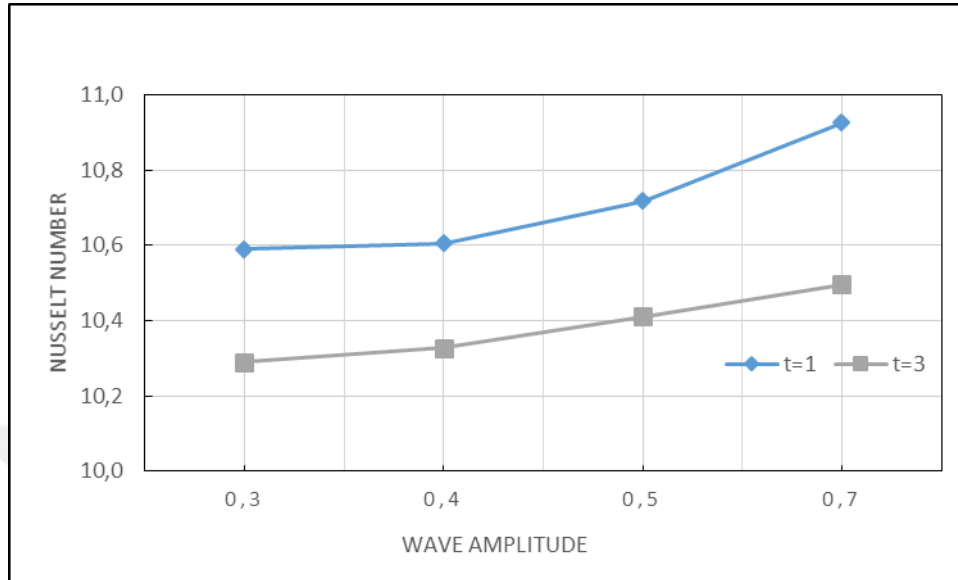


Figure 4.5. Nusselt numbers versus thickness and wave amplitude

Figure 4.6 represents the friction factor changes between 1mm and 3mm geometry due to different wave amplitudes. The friction factor increases with wave amplitude and thickness. There is a direct proportion between friction factor and wave amplitude, which the geometry possesses. Moreover, the thickness value of 3mm has higher friction factor results than 1mm.

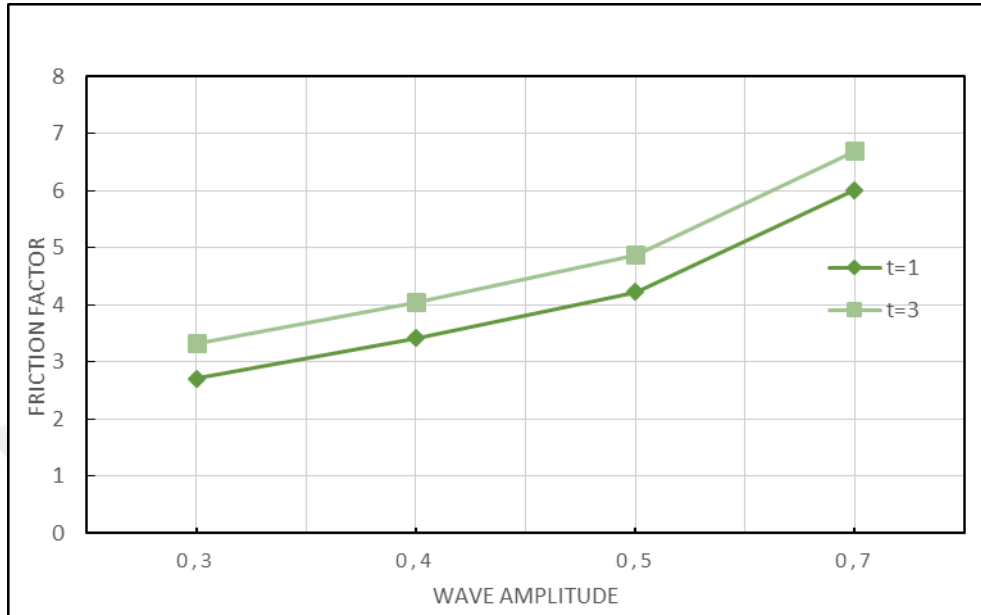


Figure 4.6. Friction factor versus thickness and wave amplitude

4.3.1.1 Contours and Vectors for $t=3$ mm

Here the 3mm thickness equipped duct's behavior is represented in terms of velocity and temperature. Figures 4.7, figure 4.8 and figure 4.9 show this topic.

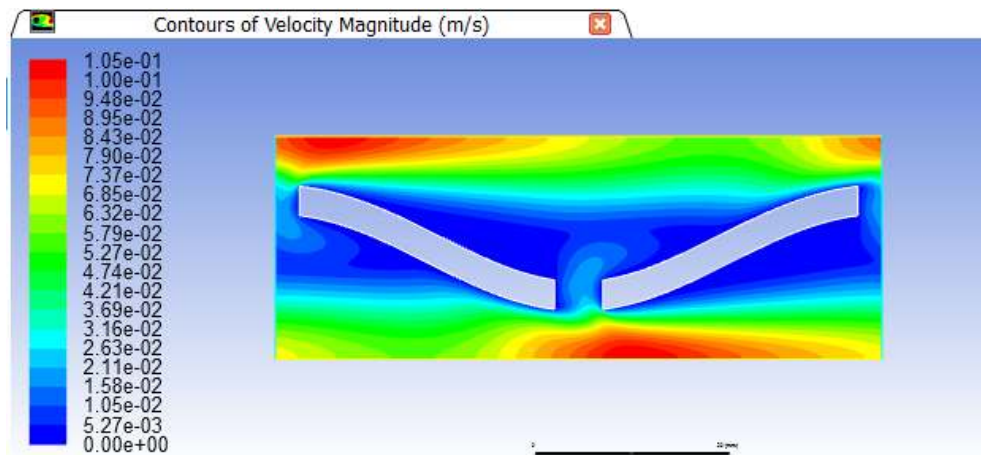


Figure 4.7. Velocity contour for $t=3$, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

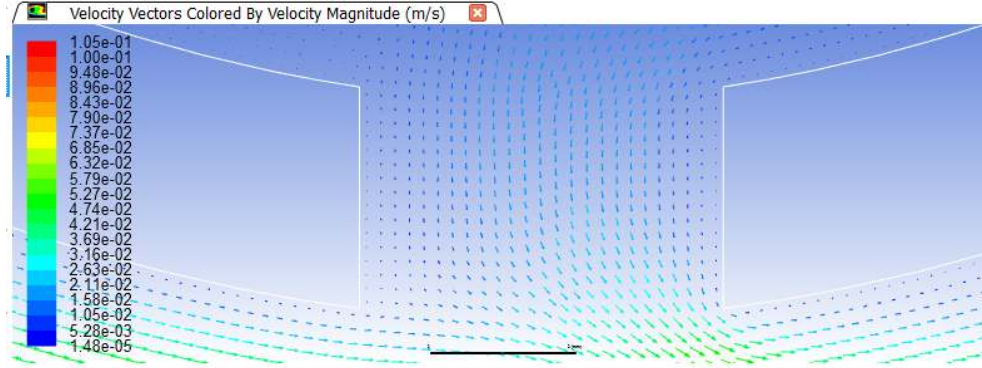


Figure 4.8. Velocity vectors for $t=3$, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

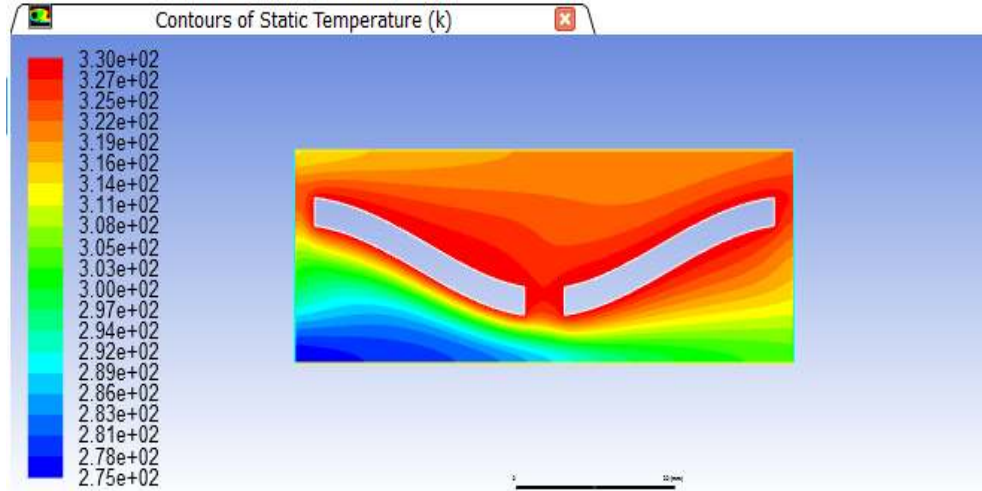


Figure 4.9. Temperature contour for $t=3$, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

Velocity contours (figure 4.7 and figure 4.8) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.9) presents the heat increment around the sinusoidal walls.

4.3.2. Radius Intercalated Edges Investigation

After the sharp edge results, an idea of augmenting the aerodynamic structure is considered. Due to this idea, the corners of the wall had been reconstructed as a curve. The geometry, as mentioned earlier, is represented in figure 4.10.

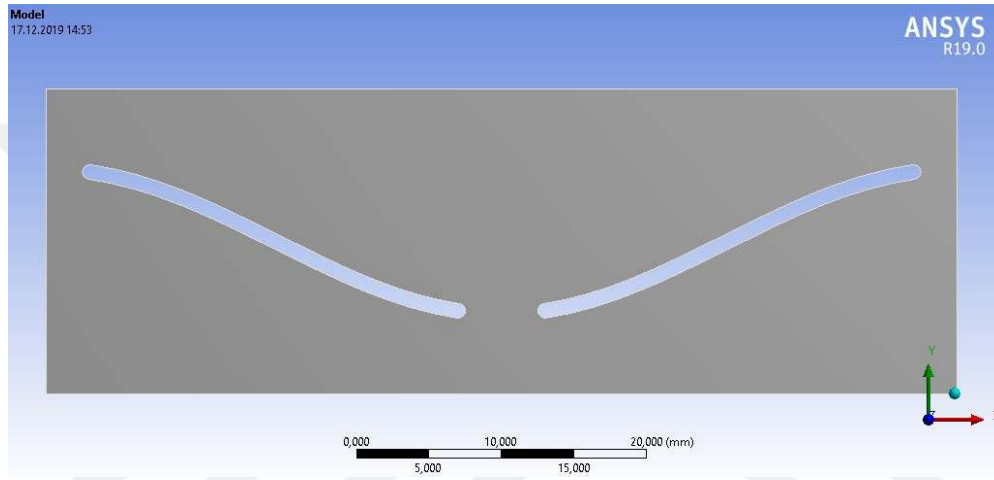


Figure 4.10. Radius shaped walls projection

Radius edges investigation concludes the " η " values, which represents the thermal enhancement factor is increased with the curvy passages. " η " value is represented as:

$$\eta = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} > 1 \quad (4.3)$$

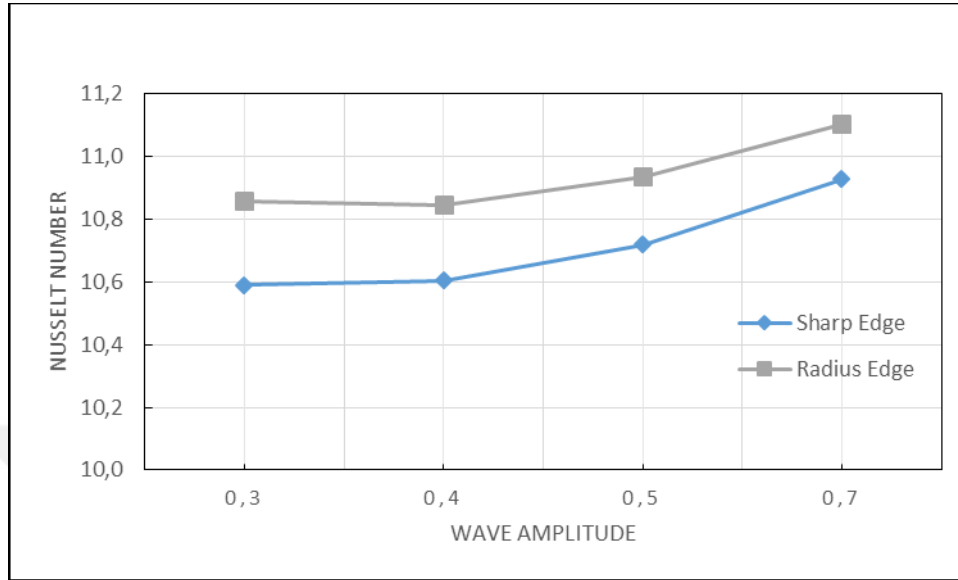


Figure 4.11. Nusselt number comparisons of sharp and curvy edges

Figure 4.11 represents the comparison between the radius and sharp-edged geometries. It is observed that both geometries have the same behavior in terms of Nusselt number; it increases with wave amplitude; however, about 20% - 25% change is observed as a gain for Nu number due to radius edge usage. The higher value of sharp edges Nusselt number is achieved on the 0.7 wave amplitude, which has a value of 10.92, and for radius shaped edge, the higher Nusselt value again achieved in 0.7 wave amplitude, and it is 11.1.

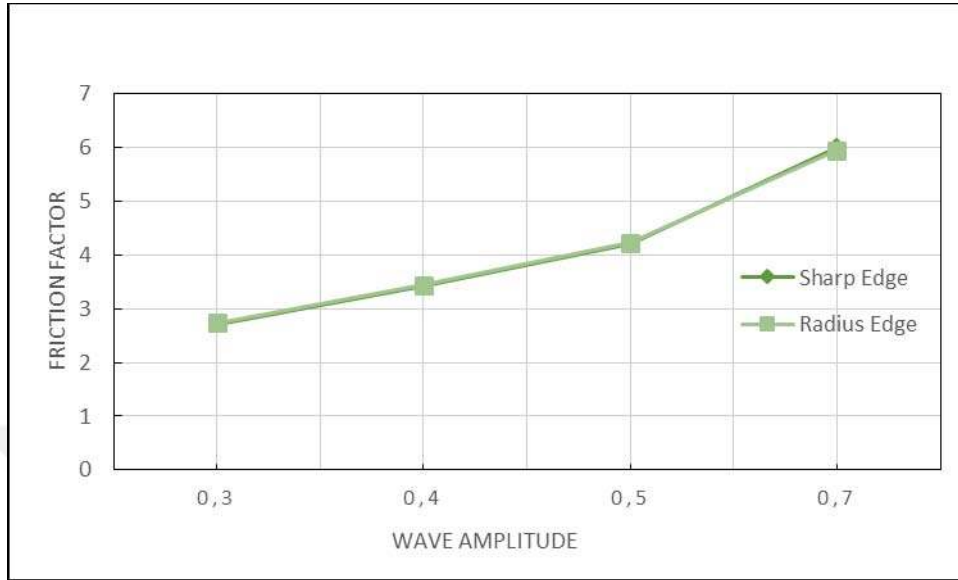


Figure 4.12. Friction factor comparison of sharp and curvy edges

Figure 4.12 shows friction factor values for sharp end radius edges. As seen from the figure, the friction factor does not change considerably for sharp and radius edges; however, it is evident that f increases with wave amplitude. For the lowest wave amplitude, f has a value of 2.7, and for the highest wave amplitude, f value advances to 6. All computations are done for $Re=100$.

The thermal enhancement factor examined to reach an adequate comparison. “ η ” value shows the innovative plate’s real performance. Here the generated data is compared with a parallel plate’s performance. Equation 4.3 had been used for these results. Nusselt and f values had been written to the equation, and results are compared in table 4.8 and 4.9 the figures 4.13 and 4.14 present this comparison visually.

Table 4.8 Sharp edge thermal enhancement values for different wave amplitudes

η	Sharp Edge
0,3	0,99380
0,4	0,92131
0,5	0,86804
0,7	0,78653

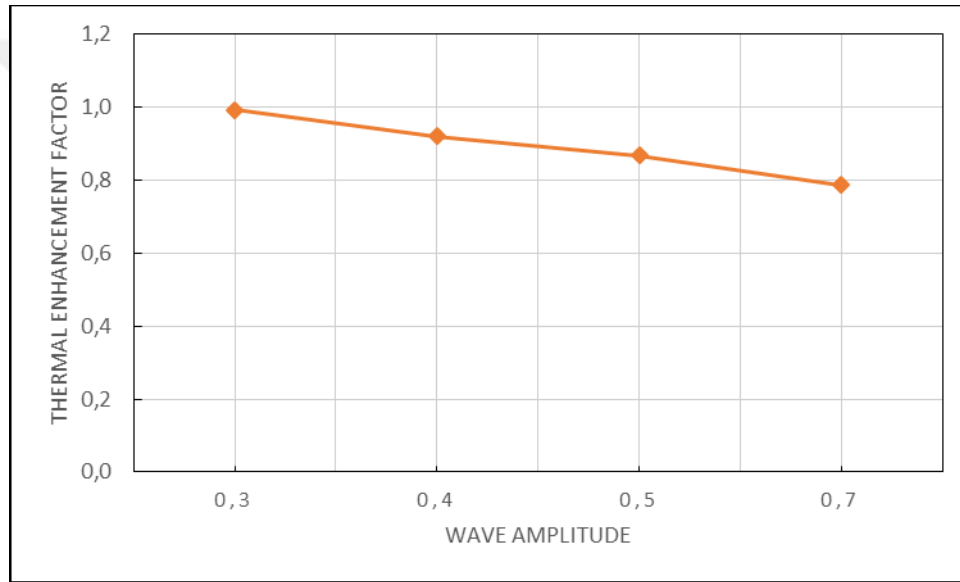


Figure 4.13. Sharp edge thermal enhancement factor distribution versus wave amplitude

Table 4.9 Radius edge efficiency values for different wave amplitudes

η	Radius Edge
0,3	1,01540
0,4	0,93986
0,5	0,88506
0,7	0,80157

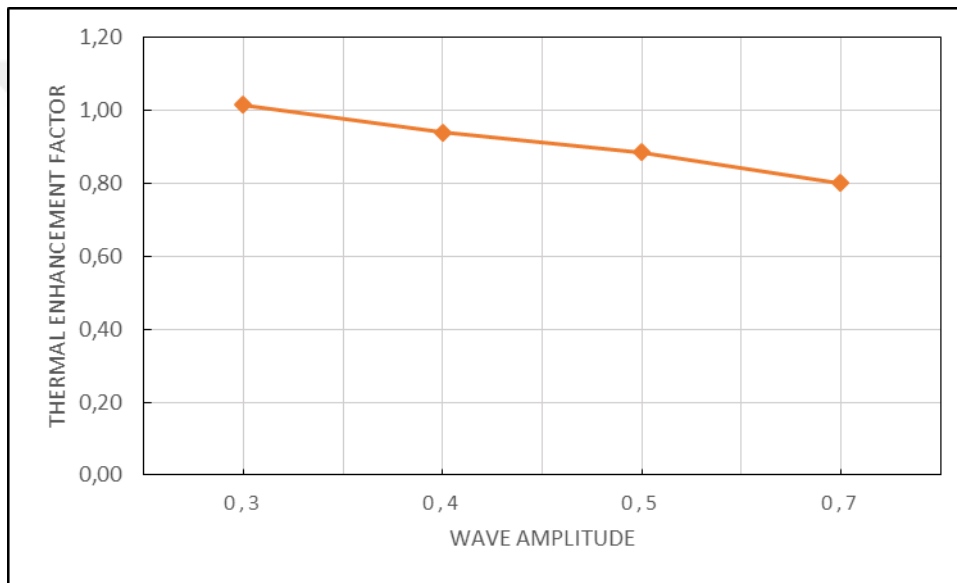


Figure 4.14. Radius edge thermal enhancement factor distribution versus wave amplitude

Thermal enhancement factors are slightly decreased as the wave amplitude increases. Combining and associating the thermal enhancement factor distributions for radius and sharp edges, the increment about 20% is evident for each value of wave amplitude. The best outcome is observed for the wave amplitude of 0.3. These calculations had been done for an absolute value of internal space “5,” as mentioned before. In the next step, wave amplitudes with different internal spaces are compared.

4.3.3. Internal Space Investigation

One of the main aims of this study is to look through the sinusoidal abrupt walled innovative plate behavior under some defined parameters. From this section upwards, this title will be investigated. Since the radius edge shape resulted better in the previous section, the calculations and analyses are done with radius edge geometry.

Figure 4.15, includes three different internal edge spaces 3-5-7 mm and to retain the consistency one specific wave amplitude of 0.5 selected to represent the main idea.

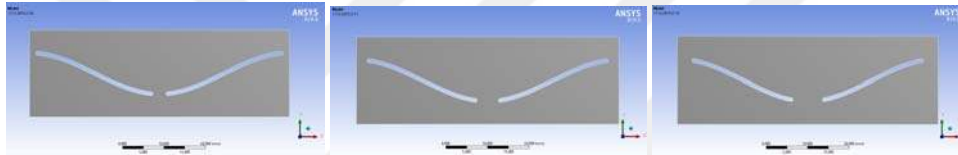


Figure 4.15. 0.5 Wave, 3mm, 5mm, and 7mm internal edge, 1 mm thickness and radius edge geometry

The following chapter compares distributions of Nu and f values for each wave amplitude. There are four different wave amplitudes and three different internal edge spaces. After that comparison, the best geometry will be examined in different ways.

4.3.3.1. Wave Amplitude of 0.3

The values for 0.3 Wave as follows. The calculations are done for $Re=100$.

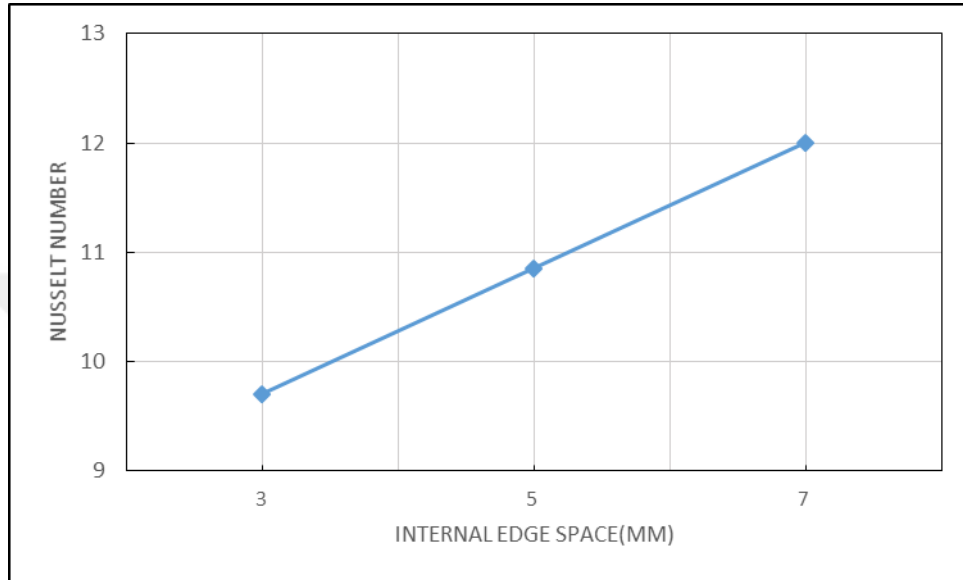


Figure 4.16. Nusselt numbers for different internal edges on 0.3 Wave amplitude

As mentioned before, three different edge space values are examined, and the first value of the four different wave amplitudes is 0.3. Figure 4.16 shows that the Nusselt number increases with internal edge space. The Nusselt number for 3mm edge space is 9.7, and for 7 mm, it raises to 12.

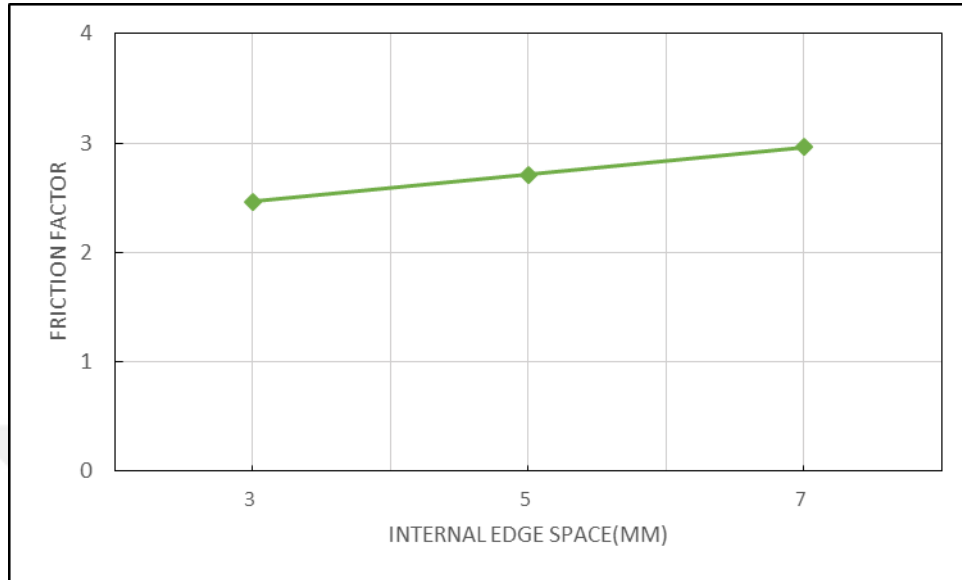


Figure 4.17. Friction factor for different internal edges on 0.3 wave amplitude

For friction factor values, the distribution is again directly proportional. As seen in figure 4.18, “f” increases with internal edge space. For 3mm internal edge space friction factor is 2.46, and for 7 mm internal edge space, the value raises nearly 3. Assessment of figures 4.16 and 4.17 concludes that a thermal enhancement factor calculation is required, which will be done in further chapters.

4.3.3.2. Wave Amplitude of 0.4

The values for 0.4 Wave as follows. The calculations are done for $Re=100$.

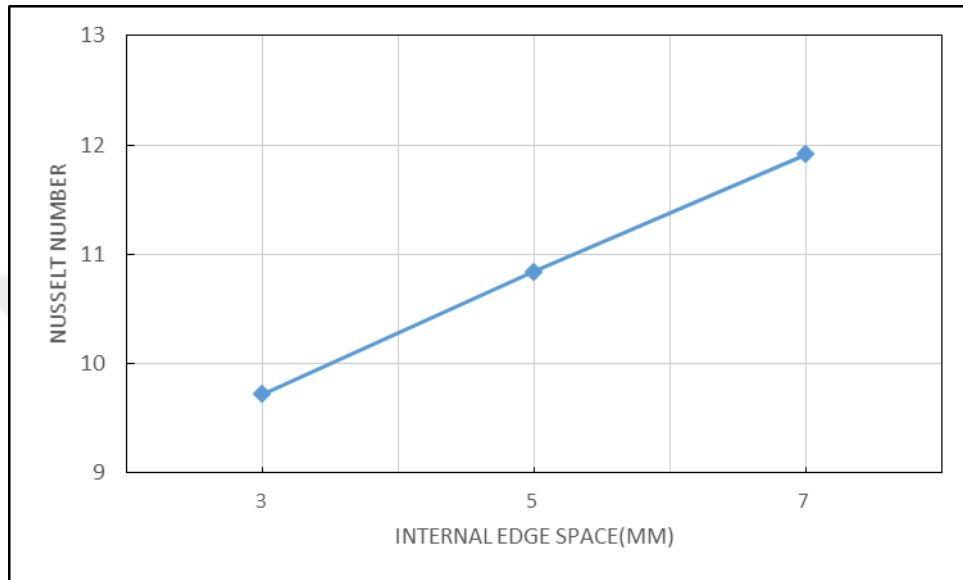


Figure 4.18. Nusselt numbers for different internal edges on 0.4 Wave amplitude

Again, three different values are compared, and Figure 4.18 shows that the Nusselt number increases with internal edge space. The Nusselt number for 3mm edge space is 9.72, and for 7 mm, it raises 11.91.

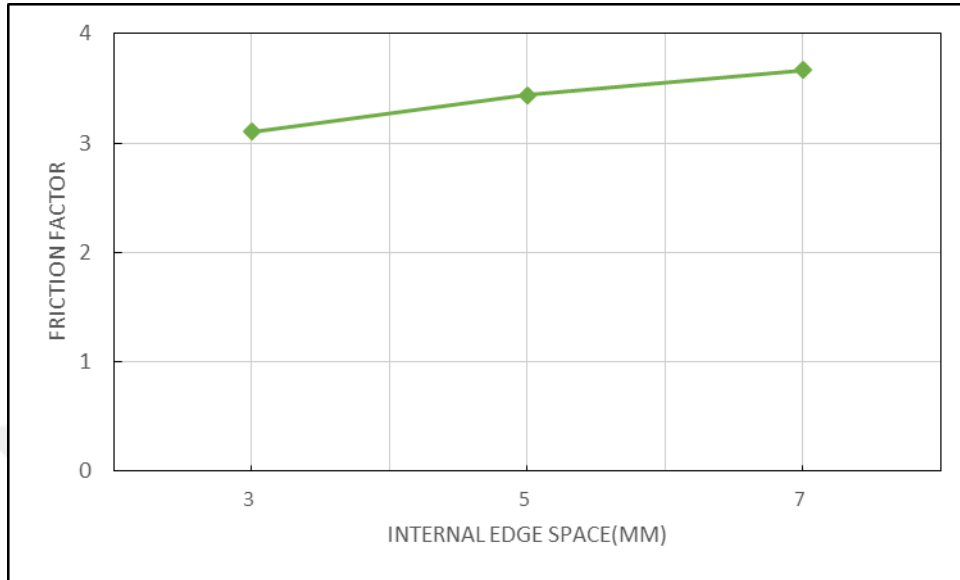


Figure 4.19. Friction factor for different internal edges on 0.4 wave amplitude

As seen in figure 4.19, “ f ” increases with internal edge space. For 3mm internal edge space friction factor is 3.1, and for 7 mm internal edge space, the value raises nearly 3.67. Assessment of figures 4.18 and 4.19 concludes that a thermal enhancement factor calculation is required, again. The data behavior both for Nu and f is compatible with the 0.3 Wave amplitude assessment.

4.3.3.3. Wave Amplitude of 0.5

The values for 0.5 Wave as follows. The calculations are done for $Re=100$.

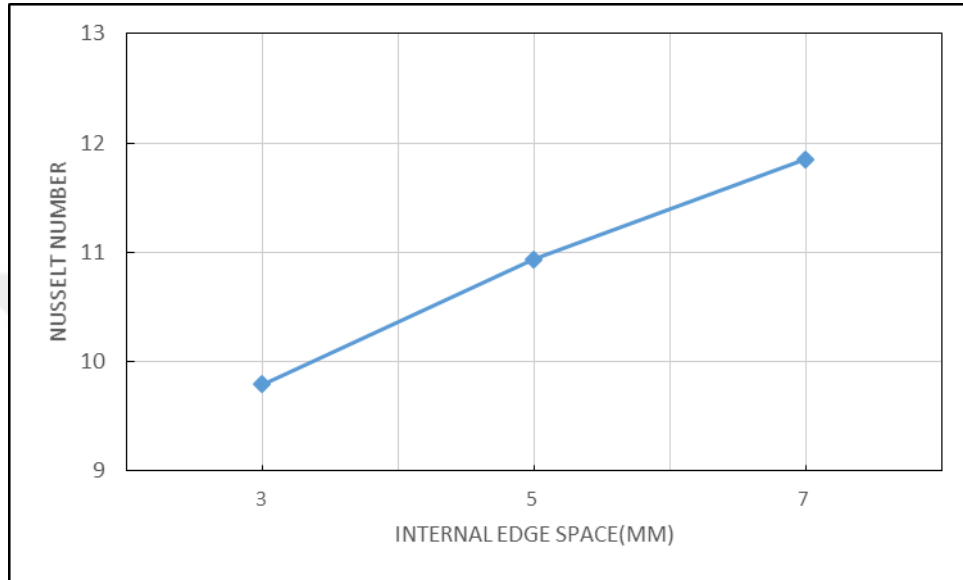


Figure 4.20. Nusselt numbers for different internal edges on 0.5 Wave amplitude

Figure 4.20 shows that the Nusselt number increases with internal edge space. The Nusselt number for 3mm edge space is 9.79, and for 7 mm, it raises 11.85. The results are compatible with respect to 0.3 and 0.4 wave amplitudes.

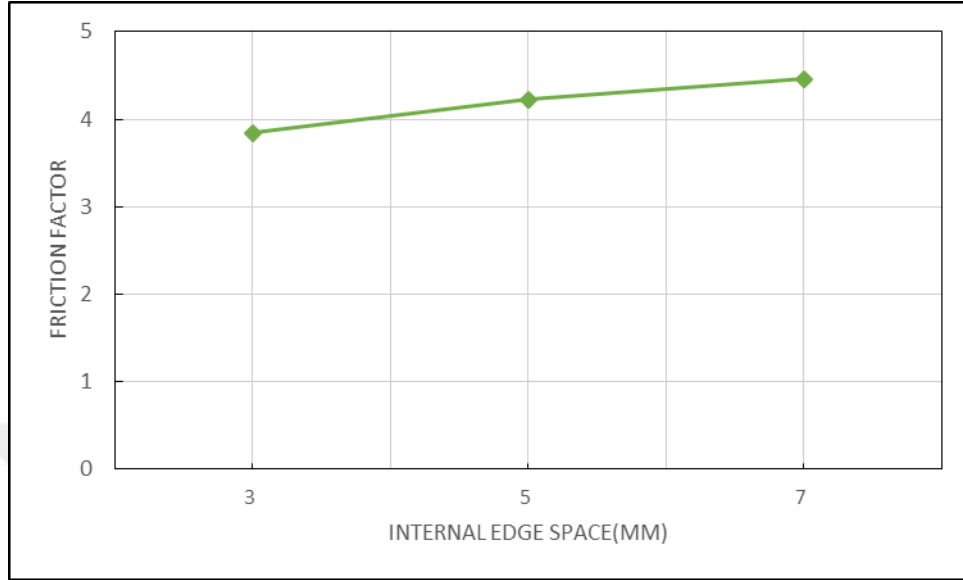


Figure 4.21. Friction factor for different internal edges on 0.5 wave amplitude

As seen in figure 4.21, “ f ” increases with internal edge space. For 3mm internal edge space friction factor is 3.84, and for 7 mm internal edge space, the value raises 4.5. Assessment of figures 4.20 and 4.21 concludes that a thermal enhancement factor calculation is required, again. The data behavior both for Nu and f is compatible with the 0.3 and 0.4 Wave amplitude assessment.

4.3.3.4. Wave Amplitude of 0.7

The values for 0.7 Wave as follows. The calculations are done for $Re=100$.

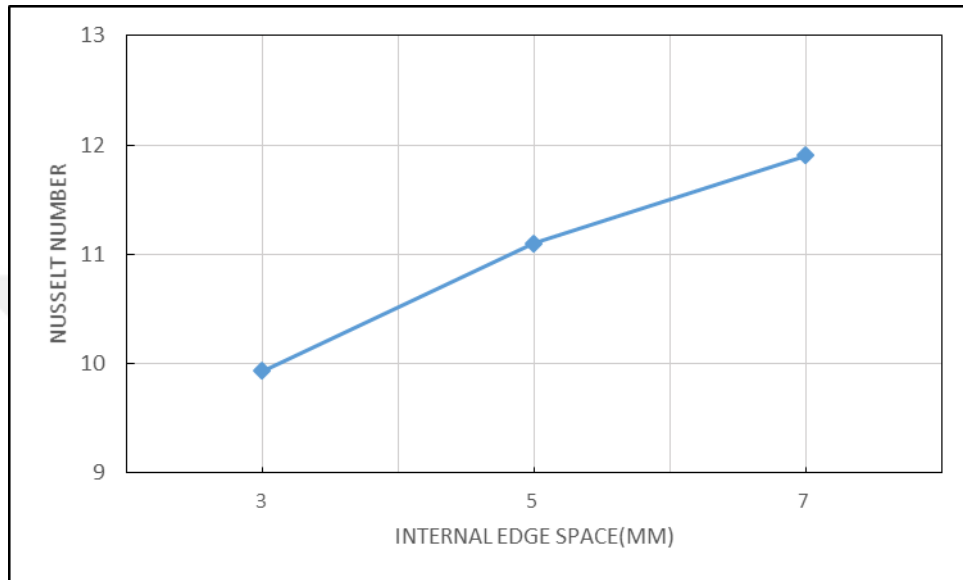


Figure 4.22. Nusselt numbers for different internal edges on 0.7 Wave amplitude

Figure 4.22 shows that the Nusselt number increases with internal edge space. The Nusselt number for 3mm edge space is 9.93, and for 7 mm, it raises 11.9. The results are compatible with respect to 0.3, 0.4, and 0.5 wave amplitudes.

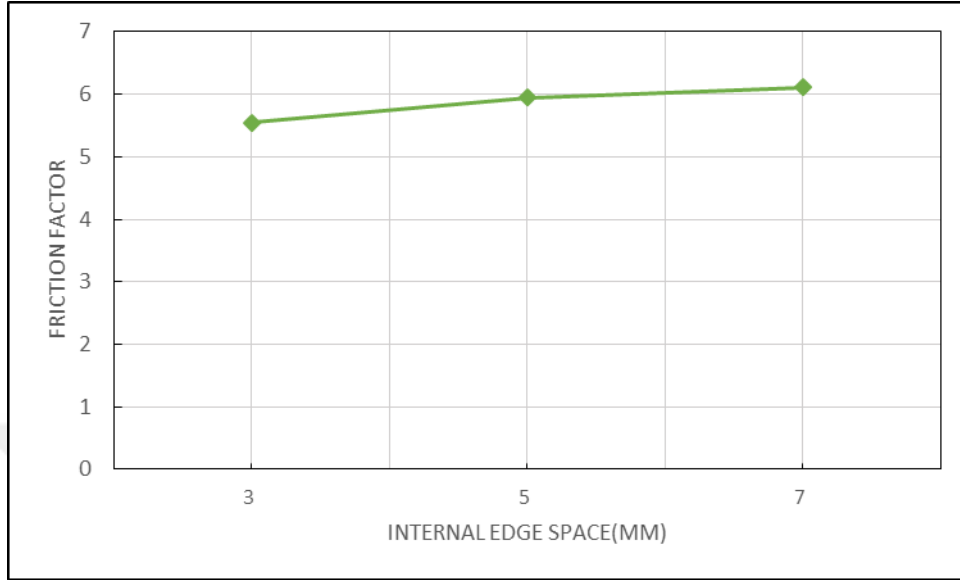


Figure 4.23. Friction factor for different internal edges on 0.7wave amplitude

As seen in figure 4.24, “ f ” increases with internal edge space. For 3mm internal edge space friction factor is 5.54, and for 7 mm internal edge space, the value raises 6.1. Assessment of figures 4.22 and 4.23 concludes that a thermal enhancement factor calculation is required, again. The data behavior both for Nu and f are compatible with the 0.3, 0.4, and 0.5 Wave amplitude assessment.

4.3.4. Wave Amplitude and Internal Edge Space Effect

The previous section demonstrated the internal edge space effect for each wave amplitude. In this section, a consolidation performed to show the main idea. At first, the Nu values contrasted. Again all values are generated with $Re=100$.

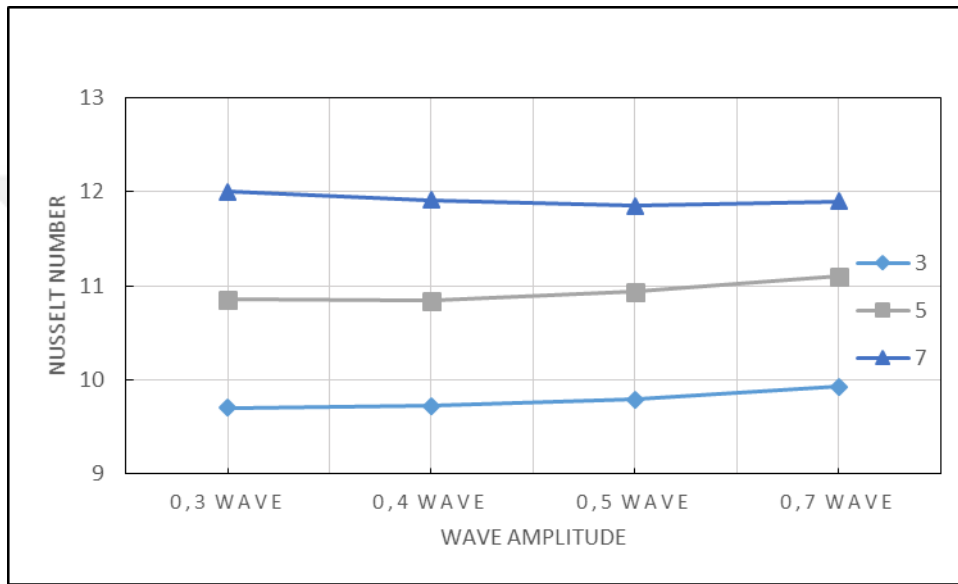


Figure 4.24. Nusselt numbers for Wave amplitude with respect to Internal Edge Space

Figure 4.24 shows that the Nusselt number increases with internal edge space, and wave amplitude directly affects the value as well. It is seen that the highest Nusselt number values are obtained for 7 mm edge space, and the lowest is obtained for 3mm. 3mm and 5mm edge spaces are in direct proportion to the wave amplitude effect on the Nusselt number, where 7mm has not. 7mm has an inverse ratio with wave amplitude, which means as the wave amplitude increases, the Nusselt number decreases. Nevertheless, 7mm edge space has the most significant values in every value of wave amplitude among all combinations.

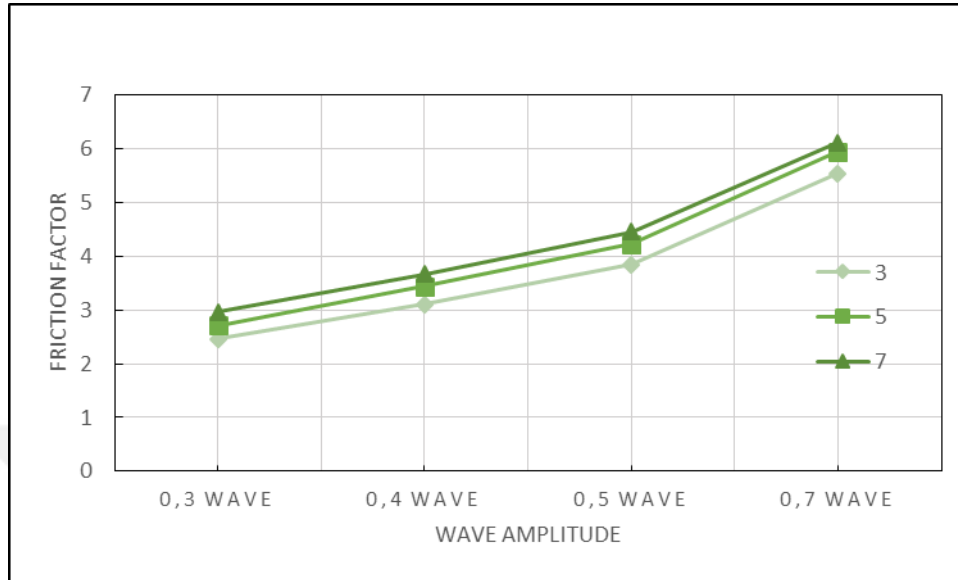


Figure 4.25. Friction factor for Wave amplitude with respect to Internal Edge Space

It can be concluded from figure 4.25 that friction factor is in direct proportion to wave amplitude and internal edge space, which means friction factor increases with internal edge space and wave amplitude. Therefore, the highest value obtained for 0.7 wave amplitude and 7mm edge space. From figures 4.24, and 4.25, one can conclude that the best geometry among them is the 0.3 wave and 0.7 internal edge space.

The last part of this section is the assessment of thermal enhancement factors for different wave amplitudes and internal edge spaces. The thermal enhancement factor had been defined in equation 4.3. It is an approach of the effect and relation of Nusselt number due to friction factor values. Obtained thermal enhancement factor results had been compared with a classical parallel plate, and the ratio for a more efficient geometry expected to be higher than 1.

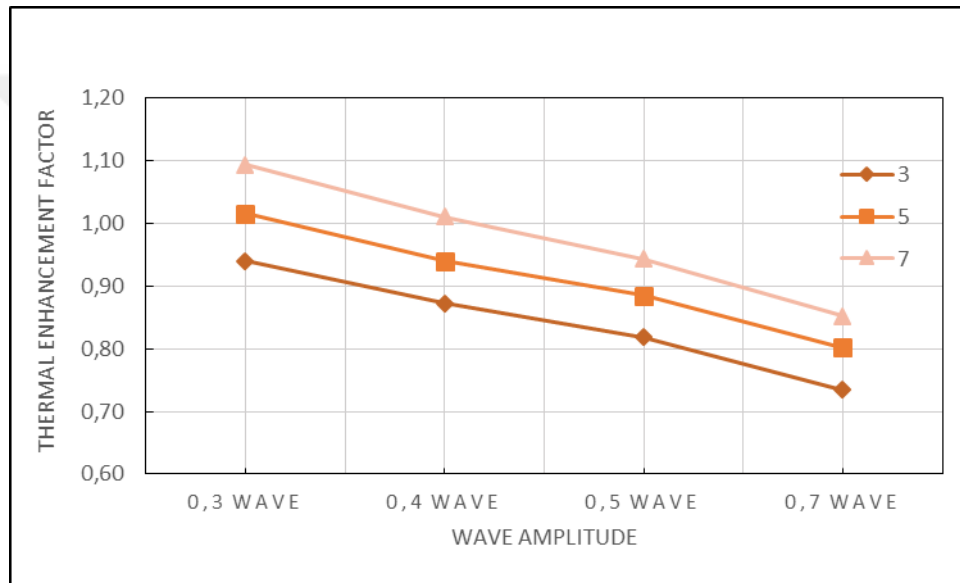


Figure 4.26. Efficiency values for Wave amplitude with respect to Internal Edge Space

Figure 4.26 represents these comparisons. Among these twelve values, three of them are higher than 1. It is seen from the figure that wave amplitude has an inverse ratio to thermal enhancement factor, which means as the amplitude increases, the thermal enhancement factor decreases. On the contrary, internal edge space has a direct proportion to this factor, and it raises as the edge space increases. To conclude, the best geometry has 0.3 wave amplitude and 7mm edge space.

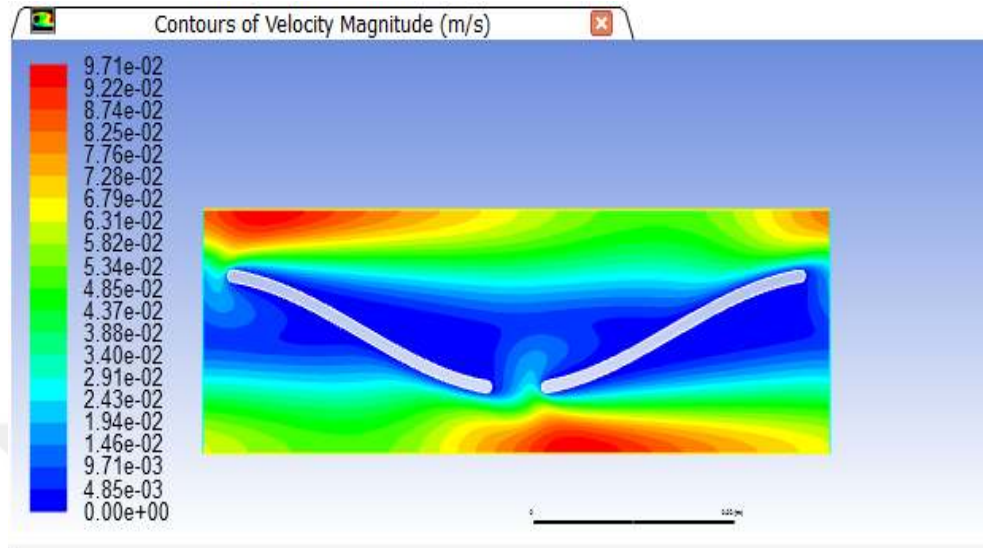


Figure 4.27. Velocity contour for $t=1$, radius=1, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

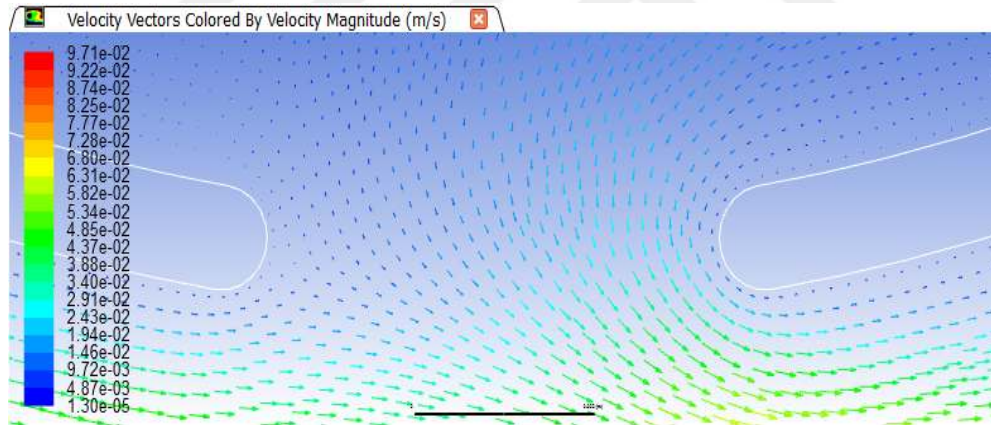


Figure 4.28. Velocity vectors for $t=1$, radius=1, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

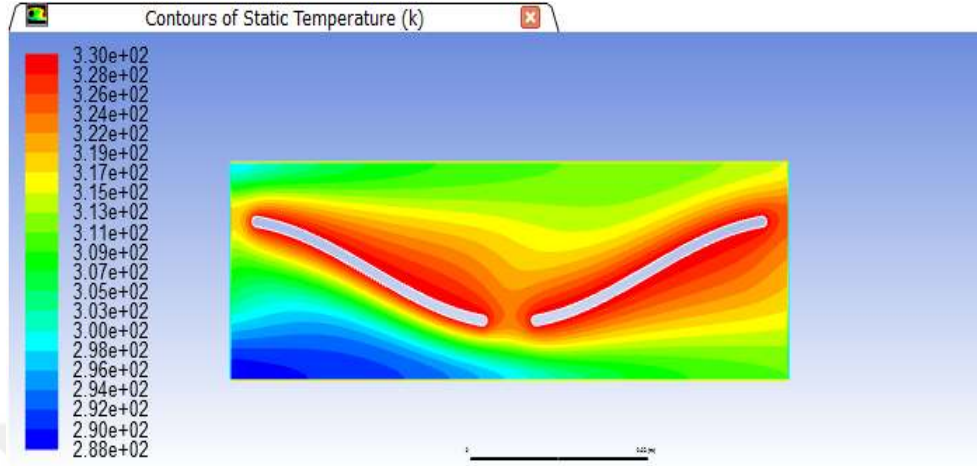


Figure 4.29. Temperature contour for $t=1$, radius=1, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

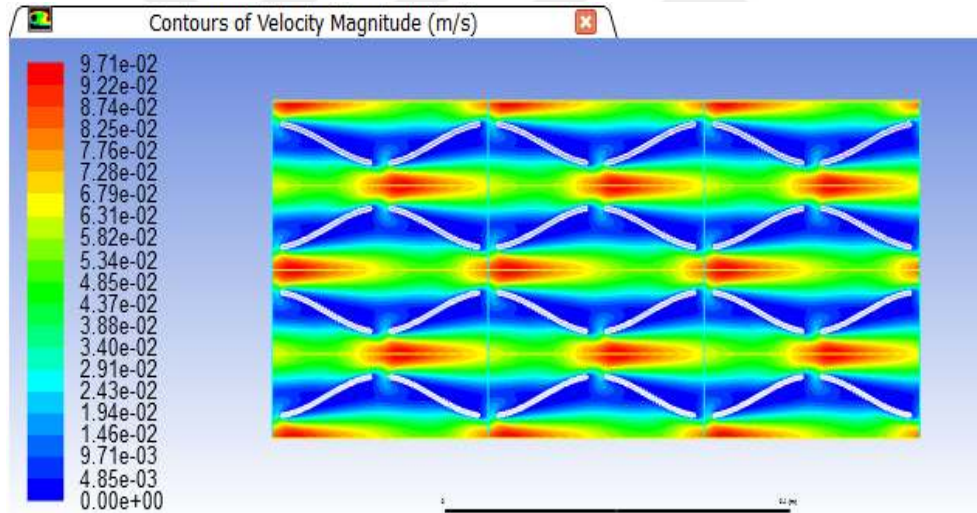


Figure 4.30. Periodic velocity contour for $t=1$, radius=1, $Re=100$, 0.5 Wave amplitude and 5 mm internal edge space

Velocity contours (figure 4.27, figure 4.28, and figure 4.30) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.29) displays the heat increment around the sinusoidal walls.

4.3.5. Vertically Interrupted Waves

In the following chapter, interrupted waves are examined. Vertically interrupted waves are shown in the figure.

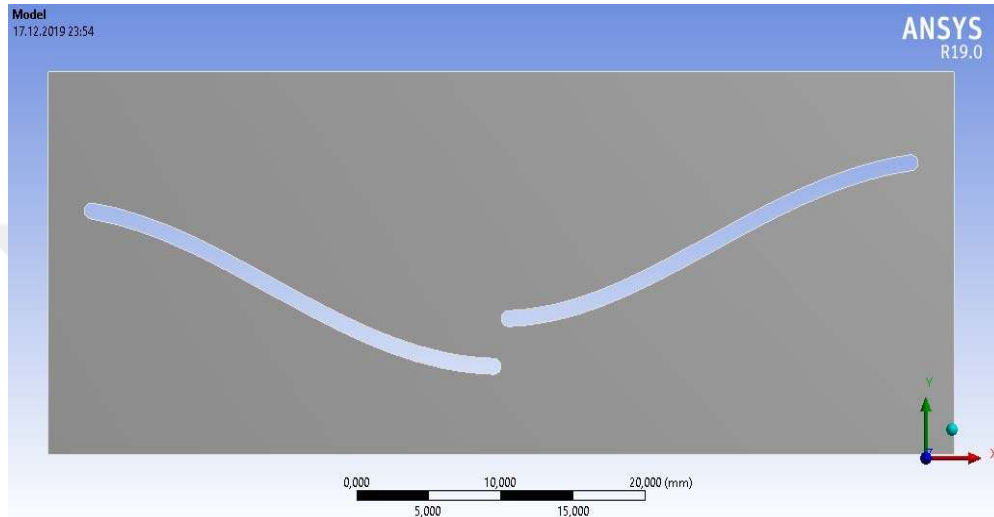


Figure 4.31. Vertically interrupted waves projection

Interrupted Wave investigation is performed for $Re=100$ and internal edge space 5. Figure 4.31 shows the wall, which is radius shaped, and the vertical distance between wavy walls differs from 2 to 5. Wave amplitude is again taken into consideration. At first, Nu and f values computed for different wave amplitudes and internal vertical spaces. After this process, the best geometry will be chosen, and the internal edge space value will be changed to reach the ultimate geometry. The last geometry is analyzed with different Re numbers to investigate Re effect on this kind of geometries.

4.3.5.1. Wave Amplitude of 0.3

The values for 0.3 Wave as follows. The calculations are done for $Re=100$.

Table 4.10 Vertical distance effect on Nusselt number for wave amplitude 0.3

0,3 Wave	Nu
2mm	10,06926
3mm	11,04027
4mm	11,91518
5mm	12,80134

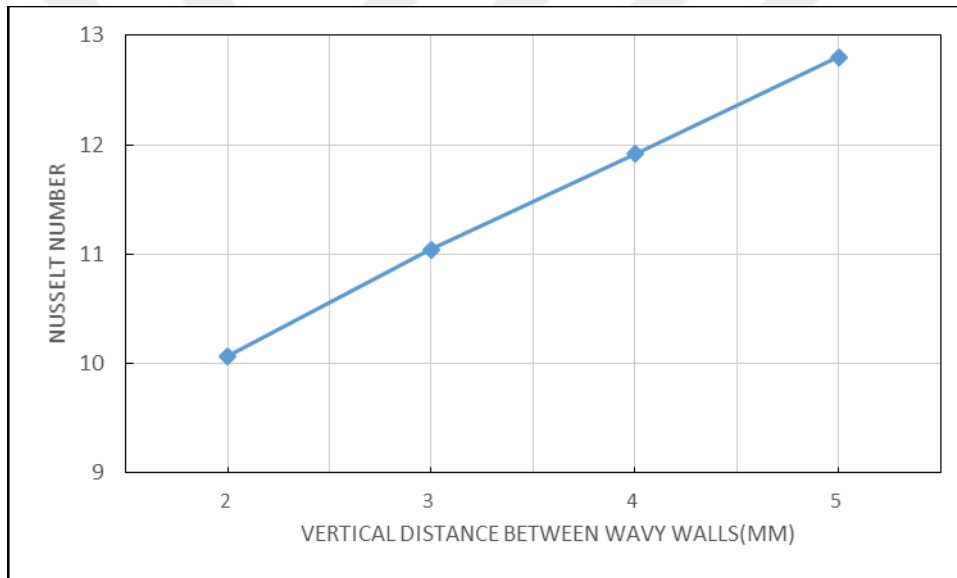


Figure 4.32. Vertical distance effect on Nusselt number for wave amplitude 0.3

Figure 4.32 and Table 4.10 present the vertical distance increment enhances Nusselt number. For the lowest vertical distance 2mm, the Nusselt number is 10.07, and for 5mm, the Nusselt number rises to 12.8.

Table 4.11 Vertical distance effect on Friction factor for wave amplitude 0.3

0,3 Wave	f
2mm	2,62093
3mm	2,78226
4mm	2,90843
5mm	3,02208

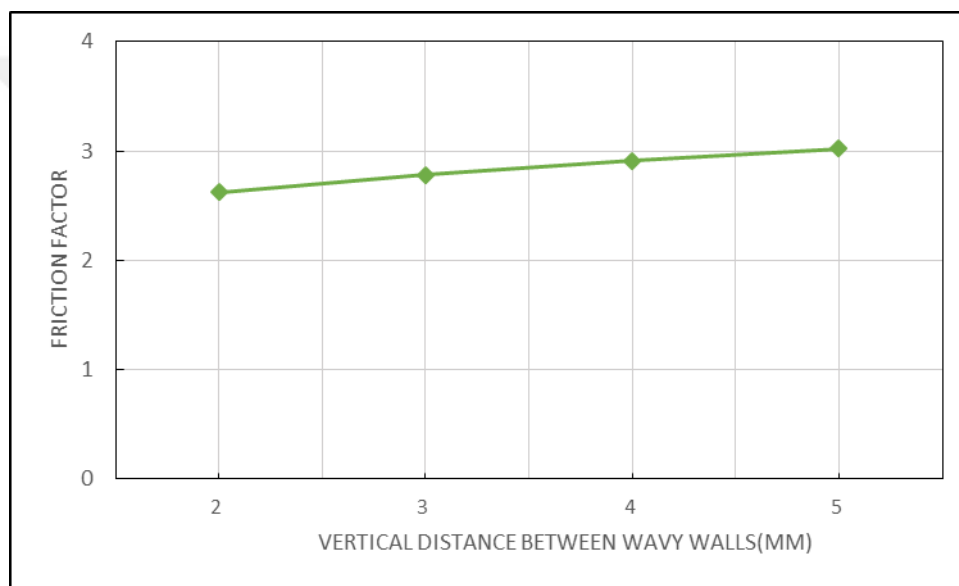


Figure 4.33. Vertical distance effect on friction factor for wave amplitude 0.3

Figure 4.33 and Table 4.11 present the vertical distance increment enhances friction factor. For the lowest vertical distance 2mm, the friction factor is 2.62, and for the vertical distance 5mm, the friction factor is 3.02.

4.3.5.2. Wave Amplitude of 0.4

The values for 0.4 Wave as follows. The calculations are done for $Re=100$.

Table 4.12 Vertical distance effect on Nusselt number for wave amplitude 0.4

0,4 Wave	Nu
2mm	9,85601
3mm	10,86416
4mm	11,73015
5mm	12,54498

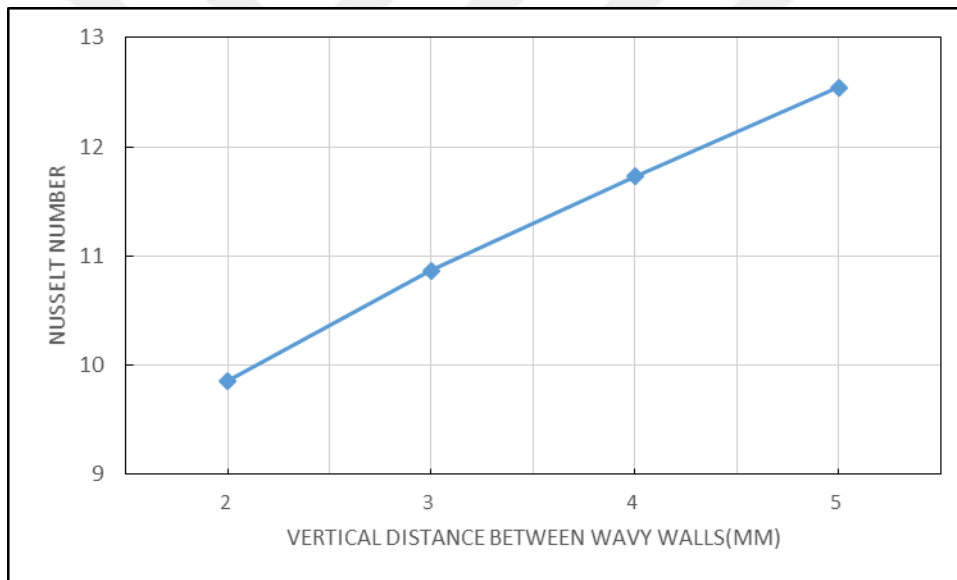


Figure 4.34. Vertical distance effect on Nusselt number for wave amplitude 0.4

Figure 4.34 and Table 4.12 present the vertical distance increment enhances Nusselt number. For the lowest vertical distance 2mm, the Nusselt number is 9.85, and for 5mm, the Nusselt number rises to 12.54.

Table 4.13 Vertical distance effect on friction factor for wave amplitude 0.4

0,4 Wave	f
2mm	3,19089
3mm	3,37470
4mm	3,52256
5mm	3,66006

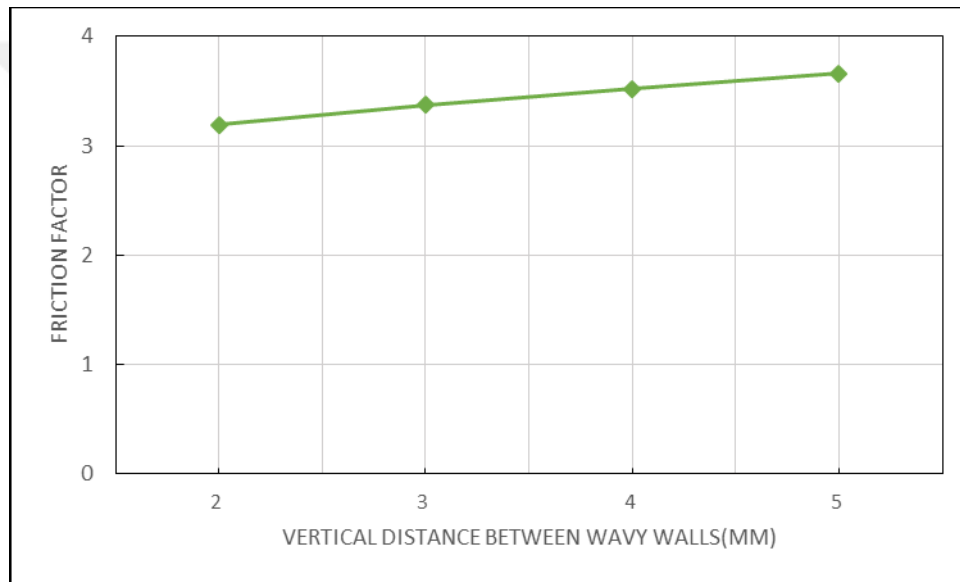


Figure 4.35. Vertical distance effect on friction factor for wave amplitude 0.4

Figure 4.35 and Table 4.13 present the vertical distance increment enhances friction factor. For the lowest vertical distance 2mm, the friction factor is 3.19, and for the vertical distance 5mm, the friction factor is 3.66.

4.3.5.3. Wave Amplitude of 0.5

The values for 0.5 Wave as follows. The calculations are done for $Re=100$.

Table 4.14 Vertical distance effect on Nusselt number for wave amplitude 0.5

0,5 Wave	Nu
2mm	9,89856
3mm	10,7777
4mm	11,65723
5mm	12,40209

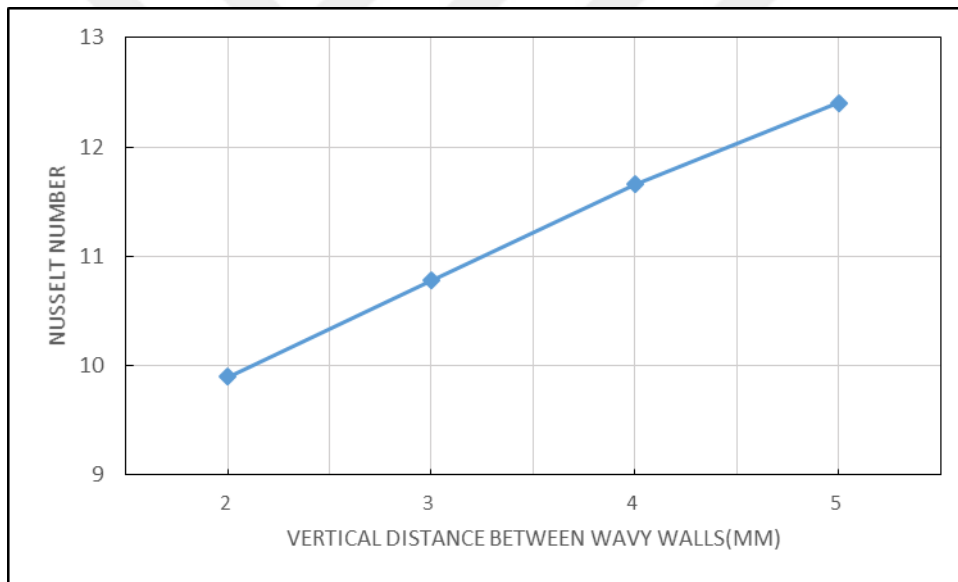


Figure 4.36. Vertical distance effect on Nusselt number for wave amplitude 0.5

Figure 4.36 and Table 4.14 present the vertical distance increment enhances Nusselt number. For the lowest vertical distance 2mm, the Nusselt number is 9.89, and for 5mm, the Nusselt number rises to 12.4.

Table 4.15 Vertical distance effect on friction factor for wave amplitude 0.5

0,5 Wave	f
2mm	3,85343
3mm	4,04055
4mm	4,23453
5mm	4,39567

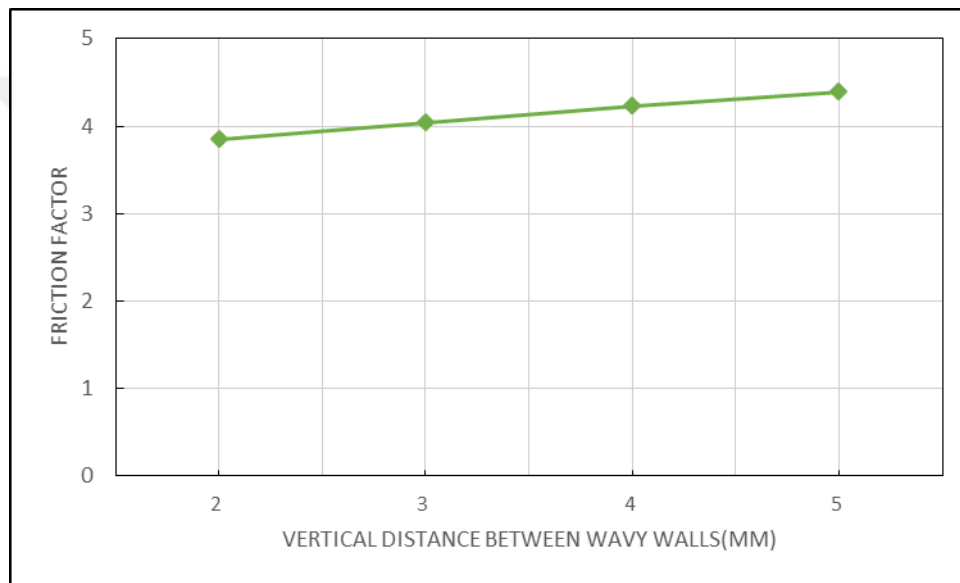


Figure 4.37. Vertical distance effect on friction factor for wave amplitude 0.5

Figure 4.37 and Table 4.15 present the vertical distance increment enhances friction factor. For the lowest vertical distance 2mm, the friction factor is 3.85, and for the vertical distance 5mm, the friction factor is 4.39.

4.3.5.4. Wave Amplitude of 0.7

The values for 0.7 Wave as follows. The calculations are done for $Re=100$.

Table 4.16 Vertical distance effect on Nusselt number for wave amplitude 0.7

0,7 Wave	Nu
2mm	10,33752
3mm	11,44170
4mm	12,12491
5mm	12,68970

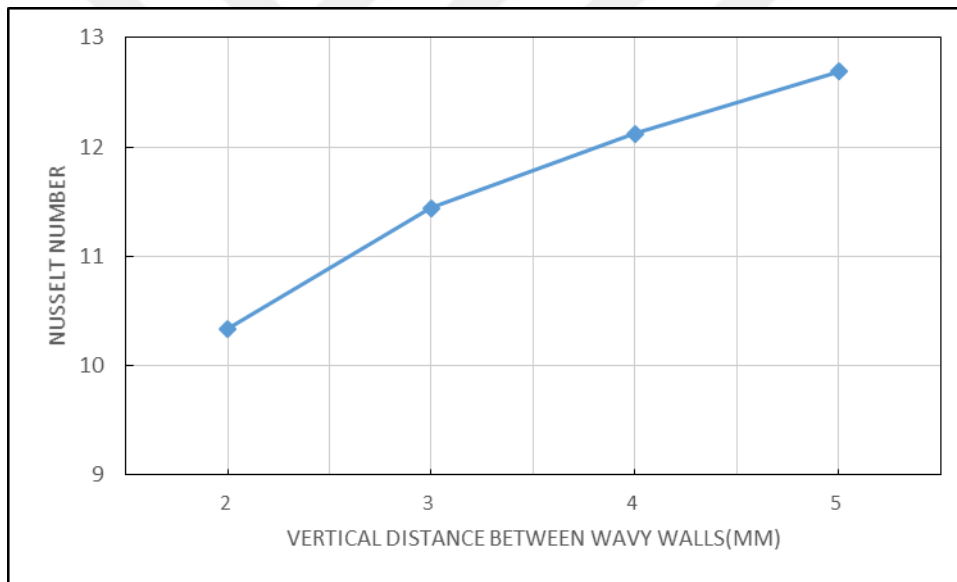


Figure 4.38. Vertical distance effect on Nusselt number for wave amplitude 0.7

Figure 4.38 and Table 4.16 present the vertical distance increment enhances Nusselt number. For the lowest vertical distance 2mm, the Nusselt number is 10.33, and for 5mm, the Nusselt number rises to 12.69.

Table 4.17 Vertical distance effect on friction factor for wave amplitude 0.7

0,7 Wave	f
2mm	5,37497
3mm	5,64677
4mm	5,84470
5mm	6,06079

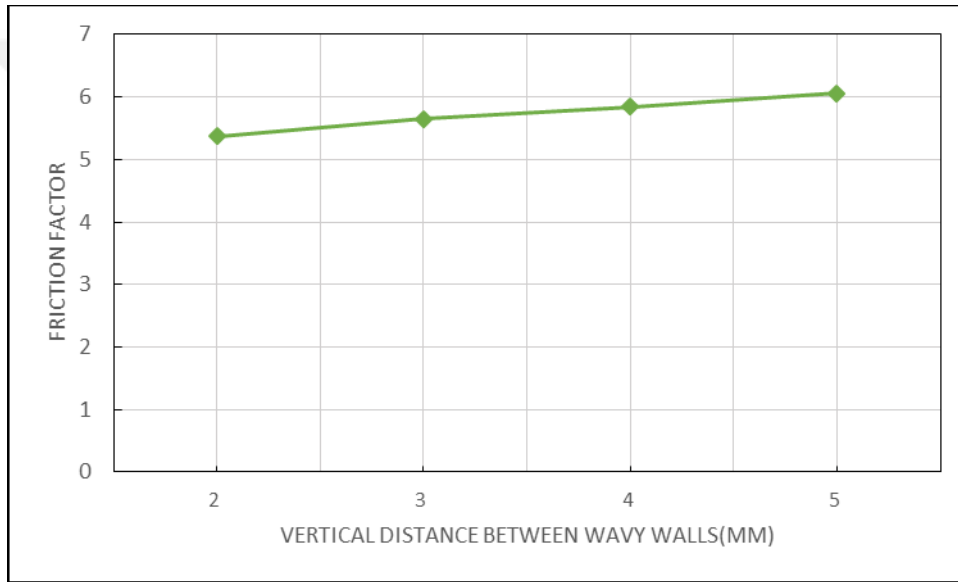


Figure 4.39. Vertical distance effect on friction factor for wave amplitude 0.7

Figure 4.39 and Table 4.17 present the vertical distance increment enhances friction factor. For the lowest vertical distance 2mm, the friction factor is 5.37, and for the vertical distance 5mm, the friction factor is 6.06. The following contours represent the behavior of the heat and flow characteristics in terms of temperature and velocity for inspecting data.

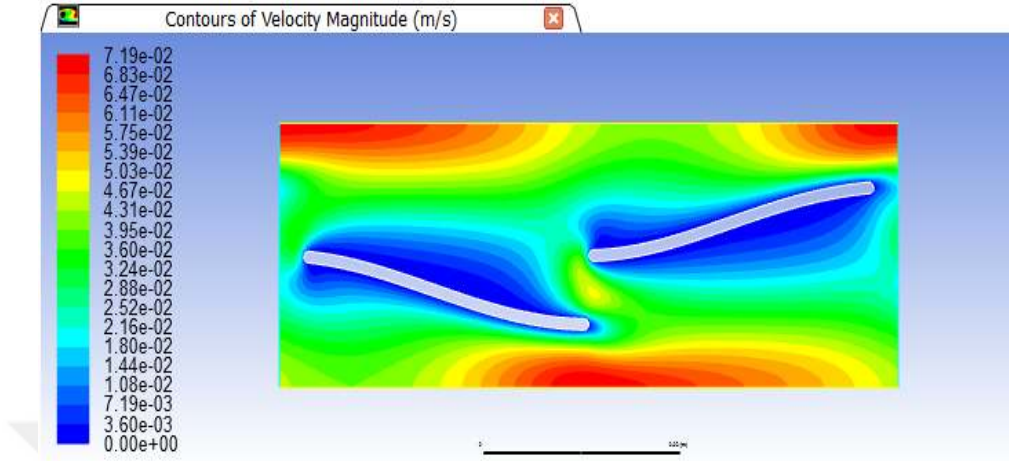


Figure 4.40. Velocity contour for $t=1$, radius=1, $Re=100$, 0.3 Wave amplitude, 5 mm internal edge space, 5mm vertical space

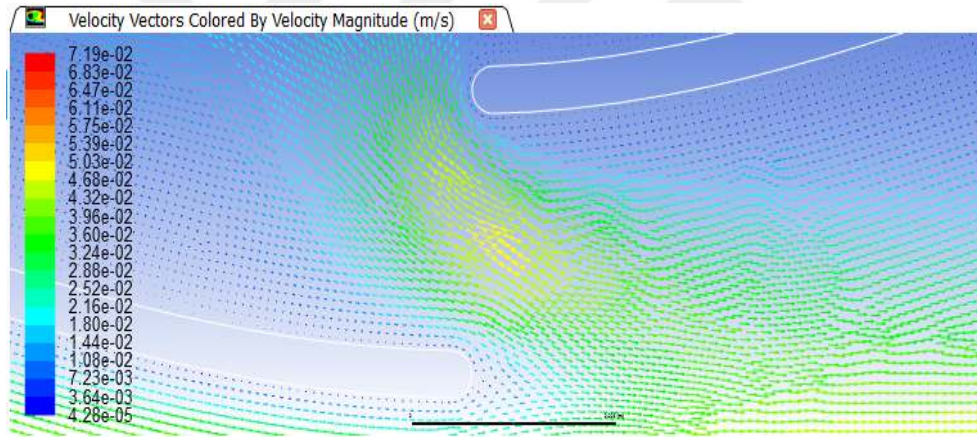


Figure 4.41. Velocity vectors for $t=1$, radius=1, $Re=100$, 0.3 Wave amplitude, 5 mm internal edge space, 5mm vertical space

Velocity contours (figure 4.40 and figure 4.41) show the flow separation, and this process provides the regeneration of the boundary layer, which increases the heat transfer. Temperature contour (figure 4.42) displays the heat increment around the sinusoidal walls.

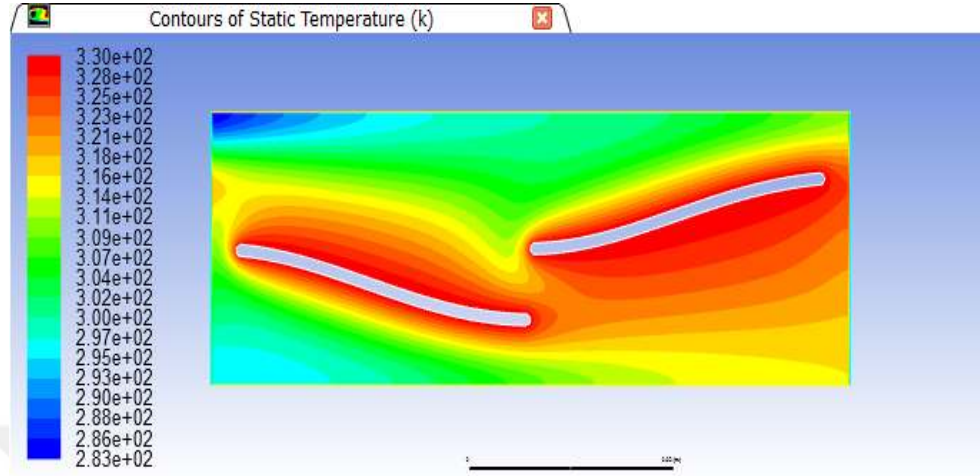


Figure 4.42. Temperature contour for $t=1$, radius=1, $Re=100$, 0.3 Wave amplitude, 5 mm internal edge space, 5mm vertical space

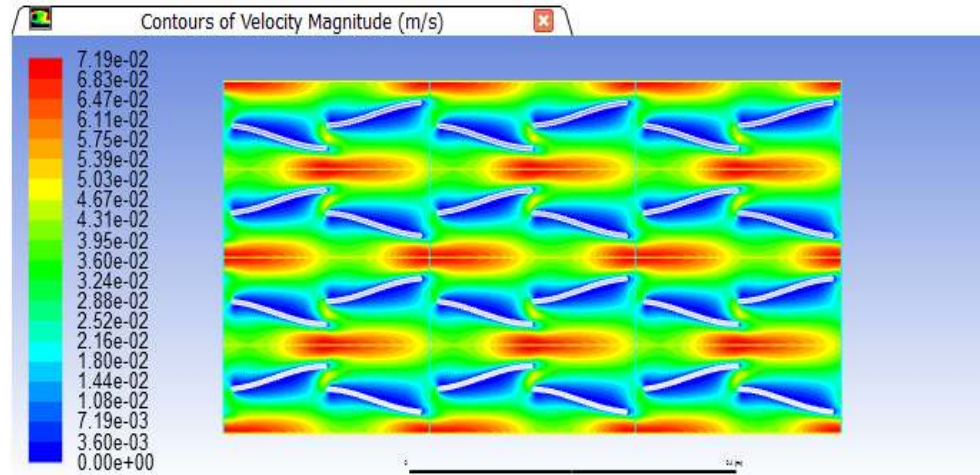


Figure 4.43. Periodic velocity contour for $t=1$, radius=1, $Re=100$, 0.3 Wave amplitude, 5 mm internal edge space, 5mm vertical space

As seen in the figure 4.43, the velocity is higher in vertically interrupted wave geometries than the normal pattern. This process highly affects the boundary layer regeneration and provides better heat transfer, which also numerically validated.

4.3.6. Wave Amplitude and Vertical Distance Effect

The previous chapter demonstrated the effect of the vertical distance between wavy walls for each wave amplitude. The data behavior both for Nu and f are compatible with the 0.3, 0.4, 0.5, and 0.7 Wave amplitude. In this chapter, a consolidation performed to show the behavior concerning each wave amplitude. At first, the Nu values contrasted. Again all values are generated with $Re=100$.

Table 4.18 Nusselt numbers for wave amplitude with respect to the vertical distance

Wave M. / Vertical Distance	2mm	3mm	4mm	5mm
0,3 Wave	10,06926	11,04027	11,91518	12,80134
0,4 Wave	9,85601	10,86416	11,73015	12,54498
0,5 Wave	9,89856	10,77777	11,65723	12,40209
0,7 Wave	10,33752	11,44170	12,12491	12,68970

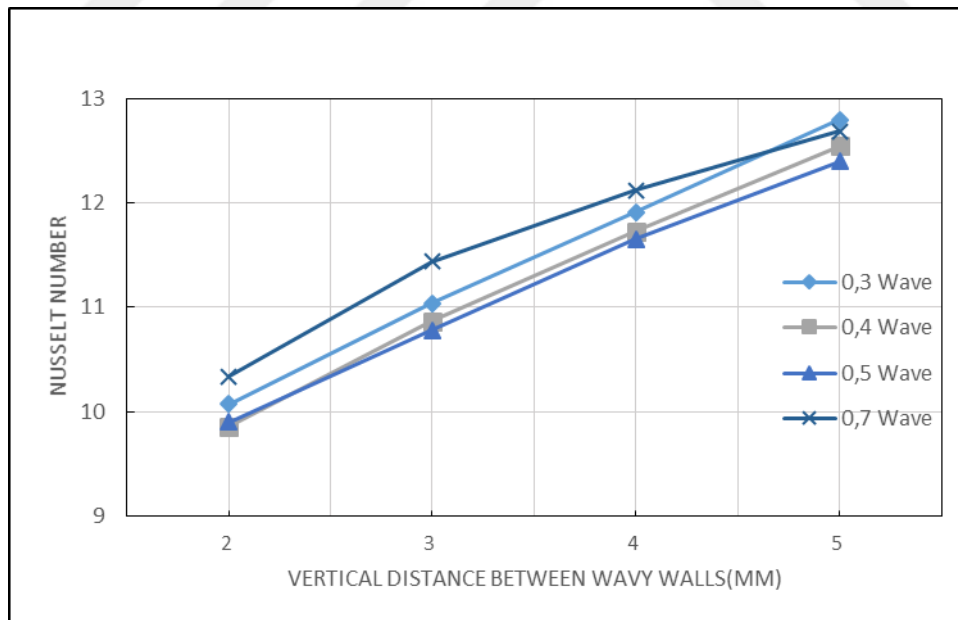


Figure 4.44. Nusselt numbers for wave amplitude with respect to vertical distance

The vertical Distance effect for the system is positive, and the best values generated for 0.7 wave and 0.3 wave, as shown in table 4.18 and figure 4.44.

Table 4.19 Friction factor for wave amplitude with respect to the vertical distance

Wave M. / Vertical Distance	2mm	3mm	4mm	5mm
0,3 Wave	2,62093	2,78226	2,90843	3,02208
0,4 Wave	3,19089	3,37470	3,52256	3,66006
0,5 Wave	3,85343	4,04055	4,23453	4,39567
0,7 Wave	5,37497	5,64677	5,84470	6,06079

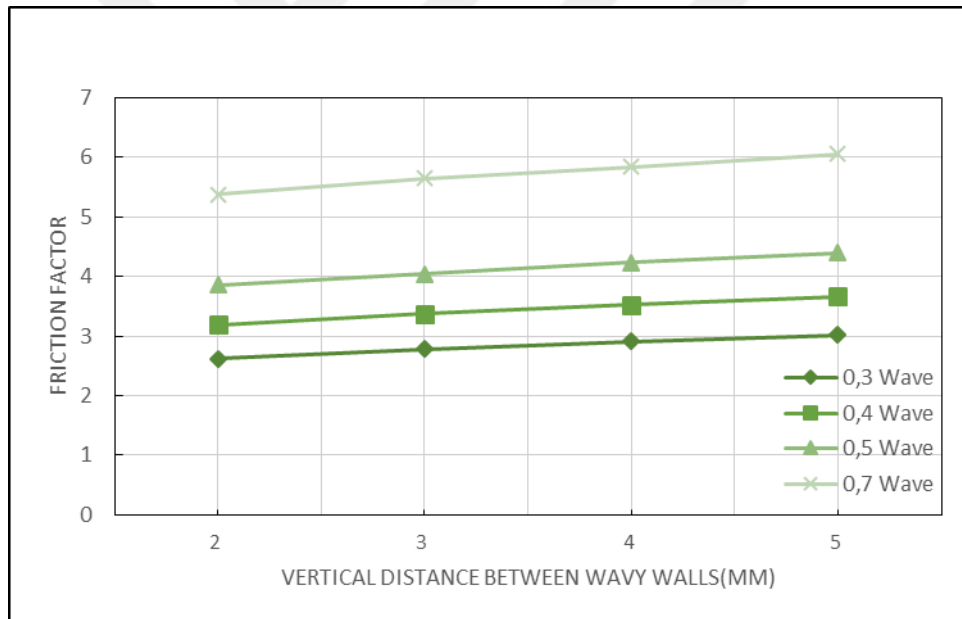


Figure 4.45. Friction factor for wave amplitude with respect to the vertical distance

Table 4.19 shows “f” values are directly proportional. Behaviors are compatible. From now on, 0.3 wave amplitude will be examined for different edge spaces 3mm - 5mm - 7mm.

4.3.7. Internal Edge Space Effect on Vertical Distanced Wavy Walls

This chapter covers the internal edge space effect on a specific vertical distanced wavy wall in terms of Nu , f , and η . The previous chapter concluded that the vertical distance increases the thermal enhancement factor and Nu . Since the most significant space examined for this study is 5, this distance will be used. From the previous chapters, one can conclude that lower wave amplitude performed more elegant outcomes. So the geometries will be 0.3 waved, 5mm vertical distanced, and different internal edge spaces of 3, 5 and 7mm. $Re=100$ again.

Table 4.20 Nusselt Number for 0.3 Wave and 5mm Vertical Distance

Nu	3mm	5mm	7mm
0,3 Wave – 5mm VD	12,33708	12,80134	13,24516

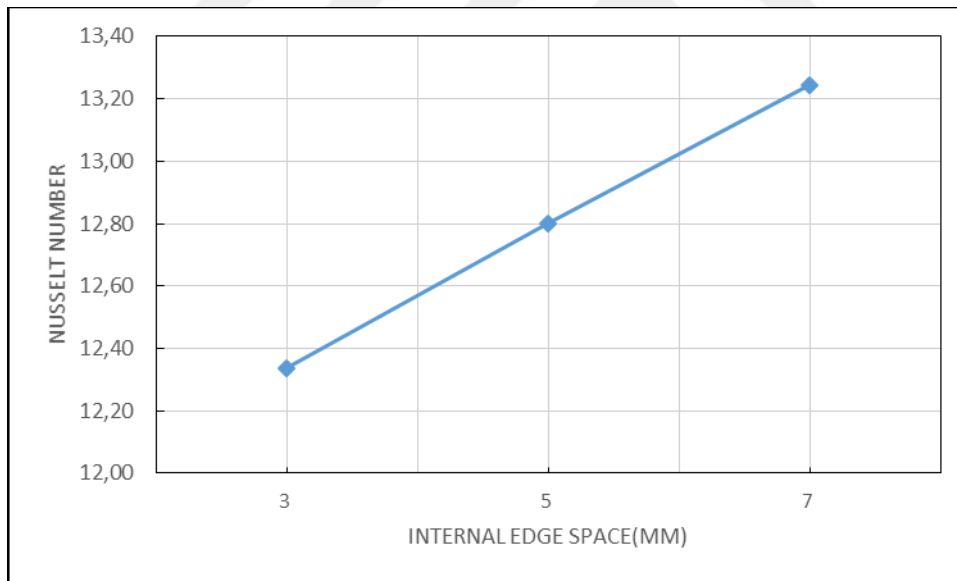


Figure 4.46. Nusselt Number for 0.3 Wave and 5mm Vertical Distance

The internal edge space effect on Nu is directly proportional, as seen in Figure 4.46.

Table 4.21 Friction factor for 0.3 Wave and 5mm Vertical Distance

f	3mm	5mm	7mm
0,3 Wave – 5mm VD	2,96253	3,02208	3,10730

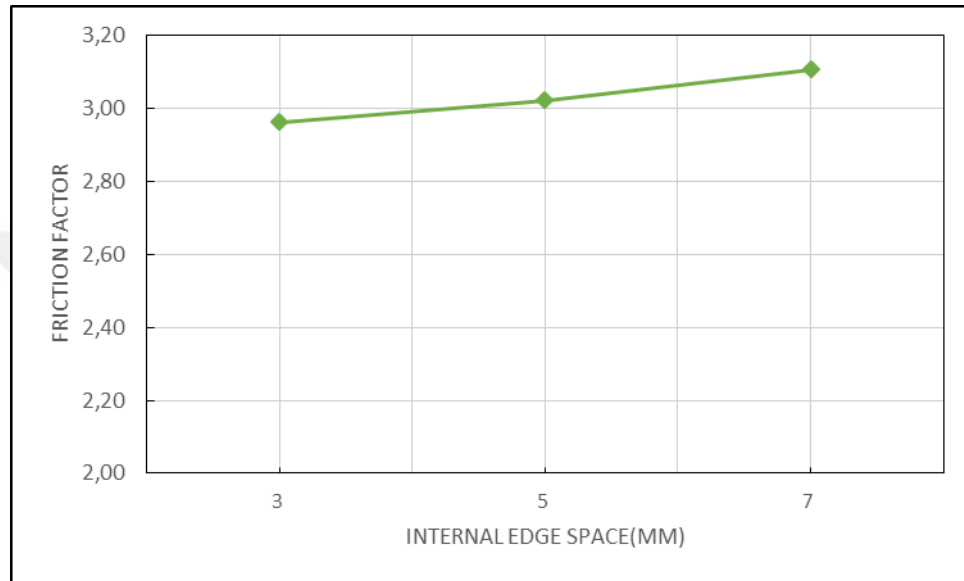


Figure 4.47. Friction factor for 0.3 Wave and 5mm Vertical Distance

The friction factor seems directly proportional to the internal edge space, as shown in figure 4.47. “ η ” thermal enhancement factor is reached as follows. From these values, the last part will be executed.

Table 4.229 Thermal enhancement factor for 0.3 Wave and 5mm Vertical Distance

η	3mm	5mm	7mm
0,3 Wave – 5mm VD	1,12386	1,15844	1,18754

All η values are higher than one, which means they are all more effective than a simple parallel plate.

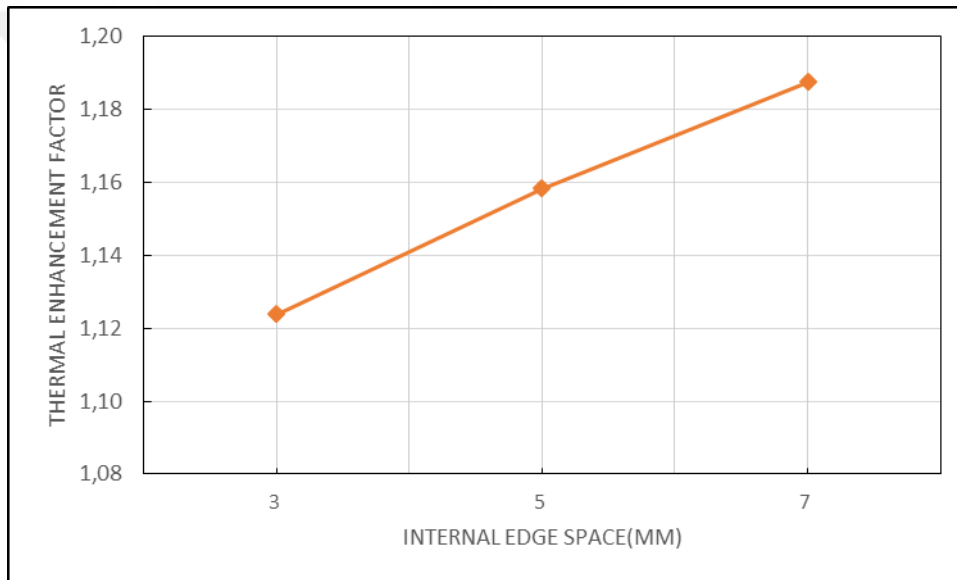


Figure 4.48. Thermal enhancement factor for 0.3 Wave and 5mm Vertical Distance

Prime geometry is constructed as 0.3 Wave amplitude, 5 mm vertical distance, and 7 mm internal edge space, which shown in figure 4.48. Table 4.20, table 4.21 and table 4.22 show the behaviors of Nusselt number, friction factor and thermal enhancement factor, respectively. After these conclusions, Reynolds number effect for this ultimate geometry will be investigated.

4.3.8. Reynolds Number Effect

Ultimate geometry among the researched interval is found as 0.3 Wave amplitude, 5 mm vertical distance, and 7 mm internal edge space. Figure 4.49 shows the increment of the Nusselt number with a raise in the Reynolds number. Table 4.23 is a consolidation for Nusselt number, friction factor and thermal enhancement factor versus Reynolds number increment.

Table 4.23 Nu,f and η values for different Re numbers

Re	Nu	f	η
50	12,34476	5,52999	1,15072
100	13,24516	3,10730	1,18754
150	13,72768	2,27792	1,19245
200	14,00952	1,85027	1,18501
250	14,14825	1,58564	1,16961

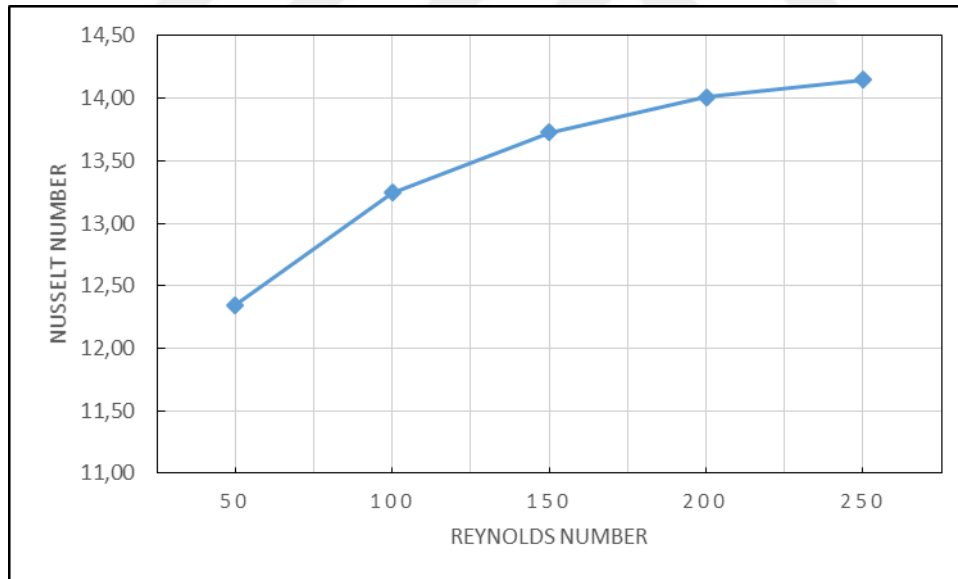


Figure 4.49. Nusselt number for different Re numbers

Figure 4.50 shows the behavior of the friction factor due to Reynolds number increment. The figure reveals that as the Reynolds number increases the friction factor decreases.

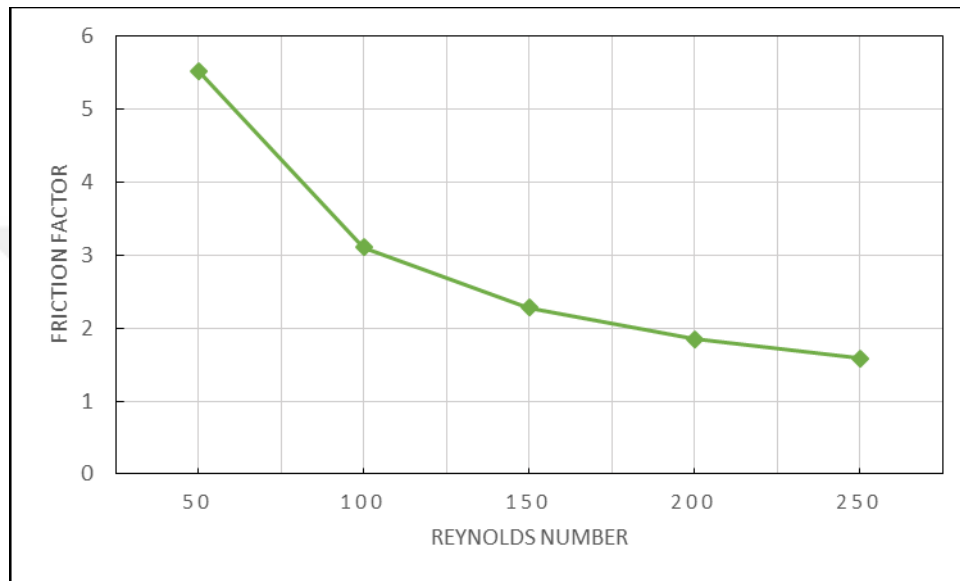


Figure 4.50. Friction factor for different Re numbers



5. CONCLUSION

In this study, an innovative plate geometry was investigated numerically. ANSYS Fluent software was used for these analyses. Seventy different calculations were performed, and all results consolidated by interrelation. Geometries were drawn in the Spaceclaim module of ANSYS; meshes were generated in the Meshing module in Workbench of ANSYS. The Workbench was provided with a compact and straightforward solution ambient and facilitation for analyses. All computations are based on the condition of hydrodynamically and thermally fully developed flow, which represented in ANSYS as the periodic boundary condition. The periodic condition was used based on literature because of its time advantage and convergence convenience. The thesis study comprises of 6 different sections.

In the first section, a validation process performed to assure the accuracy of the further results. That part covers a parallel plate calculation and 2 of interrupted plate computations in literature, which shows the proximity between early results and newly acquired values. All calculations were compatible with reference results.

In the second section, the thickness effect for sharp-edged wavy walls had been investigated. Just two different thickness values used for this topic. Thickness value of $t=1$ provided better results on Nu and f values; thus, further calculations executed due to this output.

In the third section, the radius intercalated edges effect investigated concerning sharp edges for different wave amplitudes under the circumstances of thickness $t=1\text{mm}$ and Reynolds number of 100. It has concluded that the friction factor is quite close to each other, yet Nusselt Number is 20% - 25% better for radius intercalated edges. Another output for that section, the wave amplitude effect for Nu is directly proportional. It means when the amplitude increases, the Nusselt number increases as well. However, the friction factor increases too. Therefore when Thermal enhancement factor analyzed, lower wave amplitude is better.

In the fourth section, internal space edge variety effect on performance, investigated for three different values, which are 3, 5 and 7. These calculations had been done for different wave amplitudes and radius intercalated edges. Thickness is $t=1\text{mm}$, and Reynolds number is $Re=100$. Analyses showed that higher values of internal edge space resulted in better in terms of Nusselt number. The friction factor is increased with internal edge space as well. Thus the thermal enhancement factor investigated, which was highest at edge space 7mm. The best outcome is obtained for 0.3 wave and 7mm edge space.

In the fifth section, interrupted waves have been investigated. A vertical distance is implemented. 4 different values used, and all wave amplitudes were compared. Radius edges were used again, and the thickness= 1mm , $Re=100$ values were available for these calculations as well. Nusselt number has resulted higher with higher vertical distance. The friction factor was increased while the vertical distance increased, as well. 0.3 wave amplitude and 5mm vertical distance performed best for this investigation among the others.

In the last section, the best geometry had been chosen among 70 analyses. At first, the last part of the induction is executed. Nu , f , and η values compared for 0.3 wave amplitude 5mm vertical distance for internal edge spaces of 3, 5 and 7 mm. Thickness was $t=1\text{mm}$ radius edges are available and $Re=100$. Nusselt was increased with internal edge space, and friction factor was increased as well. Higher internal edge space achieved the best thermal enhancement factor among all products. The ultimate geometry had been tested for different Reynolds numbers, which are 50, 100, 150, 200, and 250. This last part proved that the Nusselt number increases with Reynolds number, and Thermal enhancement factor had a peak in $Re=150$. The 3D version of this study might be a further research idea. Also, the results can be validated empirically and an experimental study can be performed. In addition to that, the turbulent version of the analyses can be conducted to reach more precise results. These are the future ideas which this work will lead.

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- Anonymous3: <https://www.alfalaval.se/microsites/packningsforsesdda-plattvarmevaxlare/verktyg/hur-packningsforsesdda-plattvarmevaxlare-fungerar/>
- Anonymous4: <https://www.sciencedirect.com/science/article/abs/pii/S1359431106004121>
- Anonymous5: <http://www.ierjournal.org/pupload/mit/HP1-5.pdf>

CURRICULUM VITAE

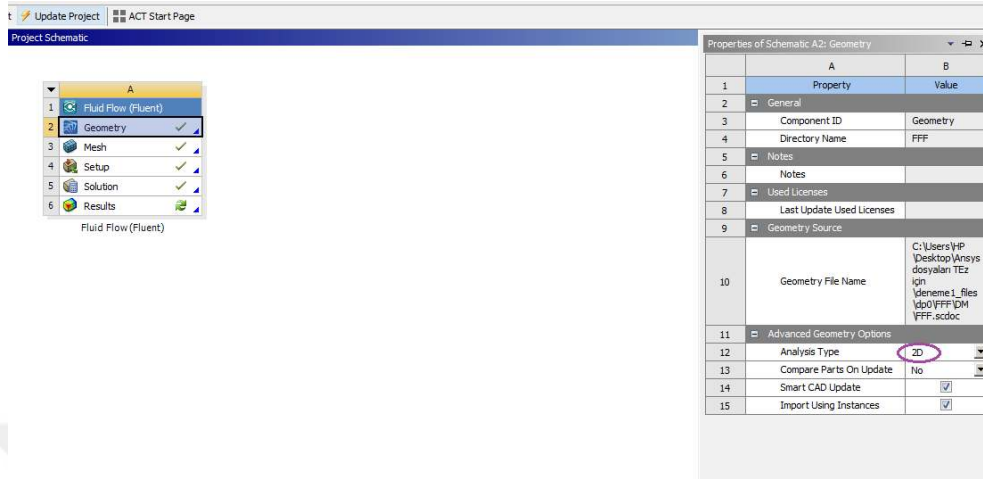
Cenk Büyükaşık was born in Kuwait in September 1989. He graduated from ATO Anatolian High School in 2007. In 2008 he started his Bachelor's education within the Çukurova University Automotive Engineering department and graduated in 2013. For an MSc degree, he participated in Çukurova University in February 2017. This Project is performed incorporate of the Çukurova University Automotive Engineering Department.





ATTACHMENTS





A.1 Analysis type selection

The analysis type must be selected at the beginning; in this study, the analysis type was 2D, and figure A.1 represents this.

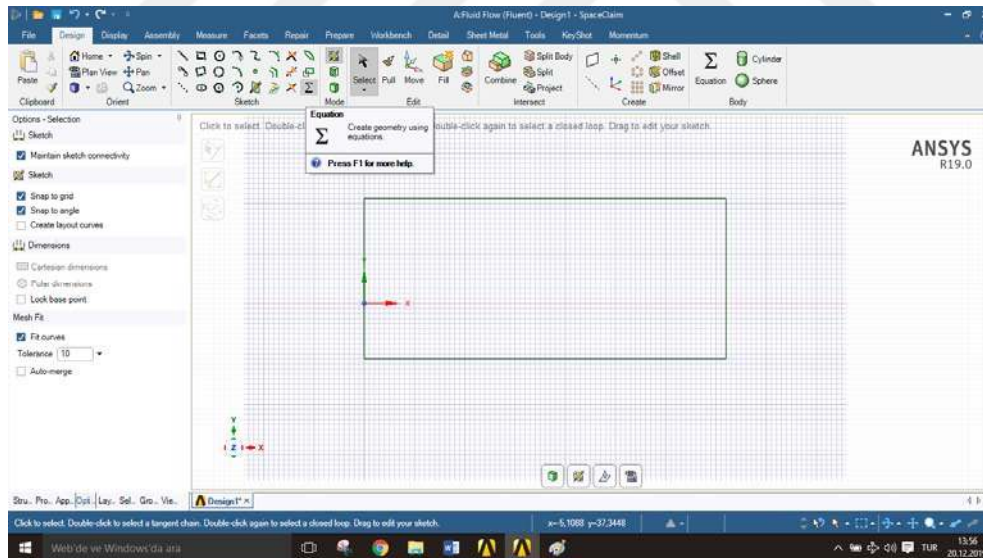


Figure A.2 Equation interface shown

Figure A.2 shows the Spaceclaim interface and equation command, which was the first step in this study to create sinusoidal waves.

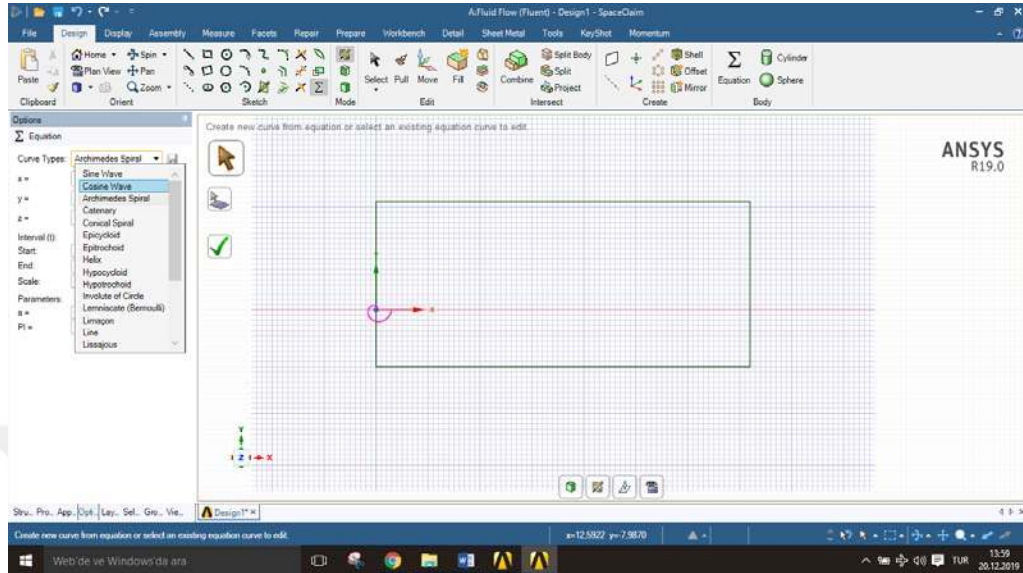


Figure A.3 Equation list on SpaceClaim

Figure A.3 represents the equation list one can generate in Spaceclaim. In this study, as shown, cosine wave selected.

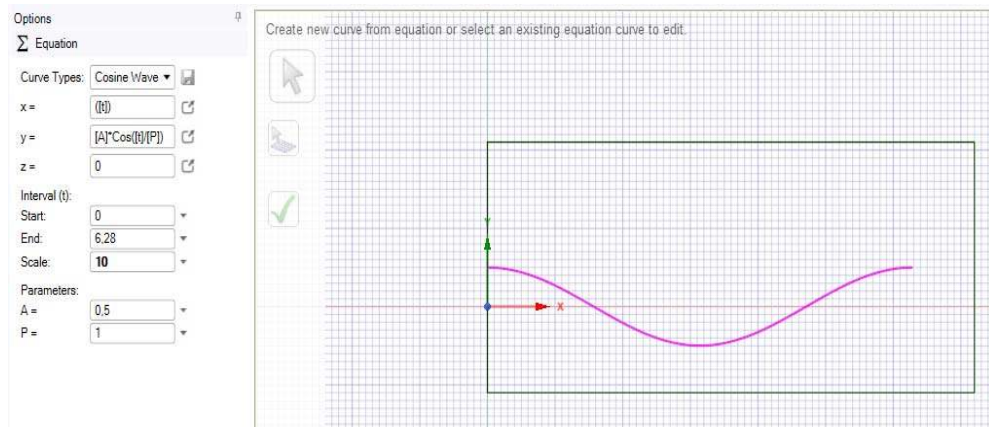


Figure A.4 Wave creating parameters

Parameters are changeable, as seen in Figure A.4. amplitudes differ with “A” Values. This was the last step to create the aforementioned sinusoidal waves.

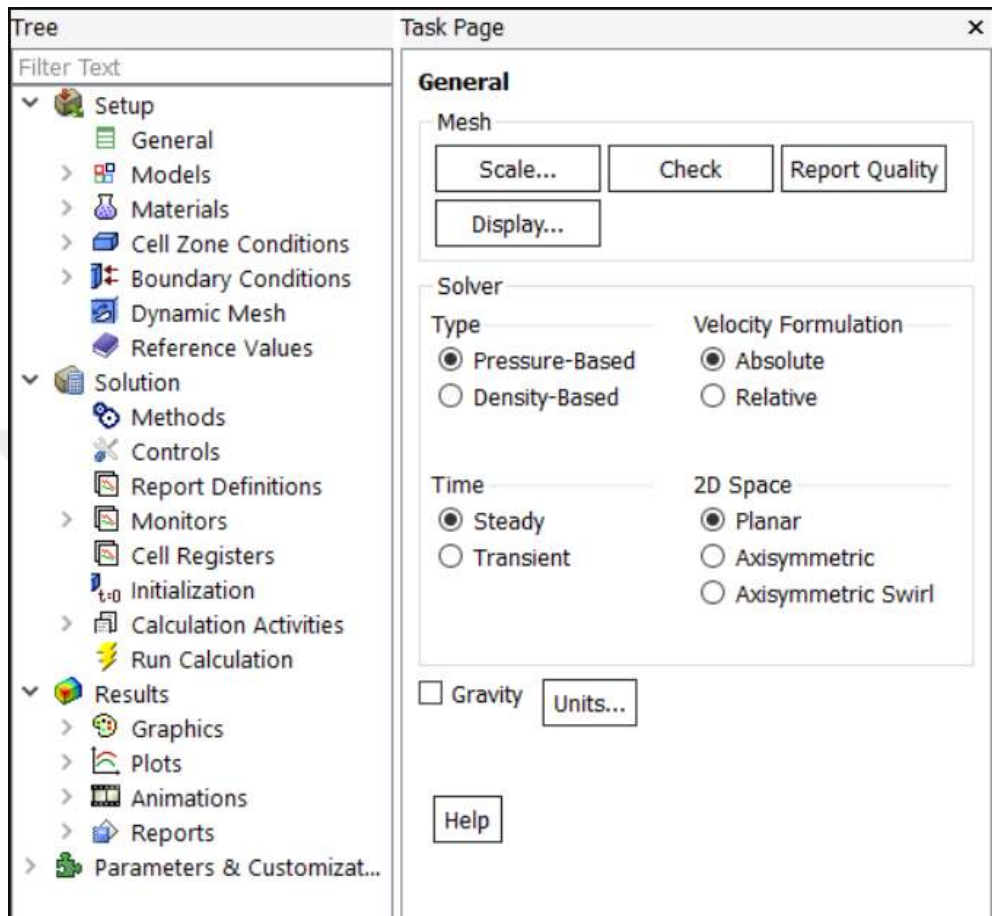


Figure A.5 The solution assumptions on the solution transaction on fluent

In figure A.5. the solution assumptions for this study have been shown. In solution transaction in FLUENT, the solver conditions were selected as seen.

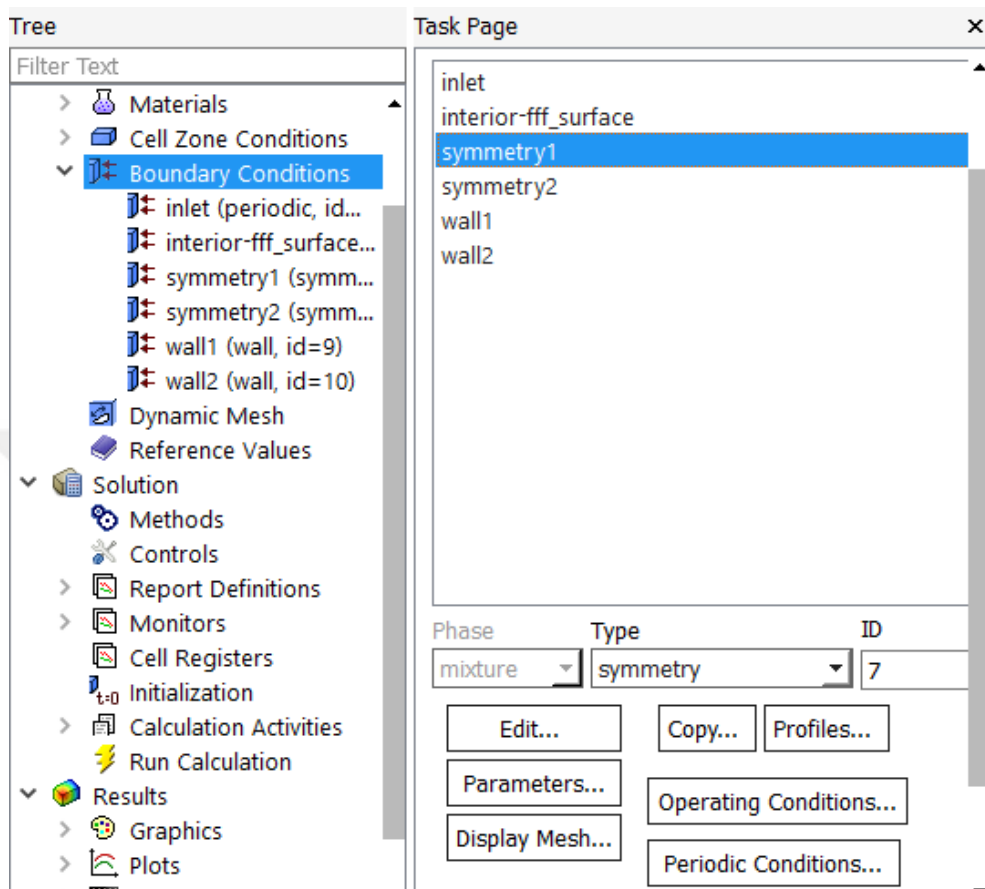


Figure A.6 Symmetry condition represented in the figure

Symmetry condition was defined with the edges before the solution, and this condition then represented and verified in the boundary condition interface, as seen in figure.A.6. Following figures A.7. and A.8. represents shear condition selection and temperature definition section. In this study, the no-slip condition selected, and the temperature defined as 330 K.

Wall dialog box showing the following settings:

- Zone Name: wall
- Adjacent Cell Zone: surface_body
- Thermal tab is selected.
- Wall Motion:
 - ☒ Stationary Wall
 - ☐ Moving Wall
- Motion:
 - ☒ Relative to Adjacent Cell Zone
- Shear Condition:
 - ☒ No Slip
 - ☐ Specified Shear
 - ☐ Specularity Coefficient
 - ☐ Marangoni Stress
- Wall Roughness:
 - Roughness Height (m): 0, constant
 - Roughness Constant: 0.5, constant

Buttons: OK, Cancel, Help

Figure A.7 No-slip condition on the sinusoidal walls

Wall dialog box showing the following settings:

- Zone Name: wall1
- Adjacent Cell Zone: fff_surface
- Thermal tab is selected.
- Thermal Conditions:
 - ☒ Heat Flux
 - ☒ Temperature
 - ☐ Convection
 - ☐ Radiation
 - ☐ Mixed
 - ☐ via System Coupling
 - ☐ via Mapped Interface
- Temperature (K): 330, constant
- Wall Thickness (m): 0
- Heat Generation Rate (W/m3): 0, constant
- Material Name: aluminum, Edit...

Buttons: OK, Cancel, Help

Figure A.8 Constant Temperature on wall

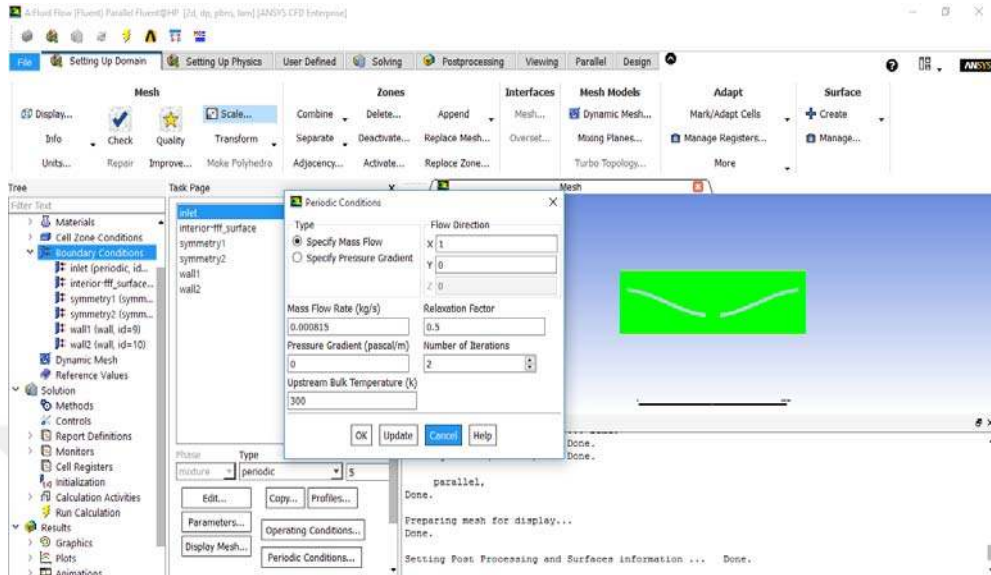


Figure A.9 Application of periodic condition in Fluent

Figure A.9 shows the application of periodic condition, which was performed in the boundary conditions section.

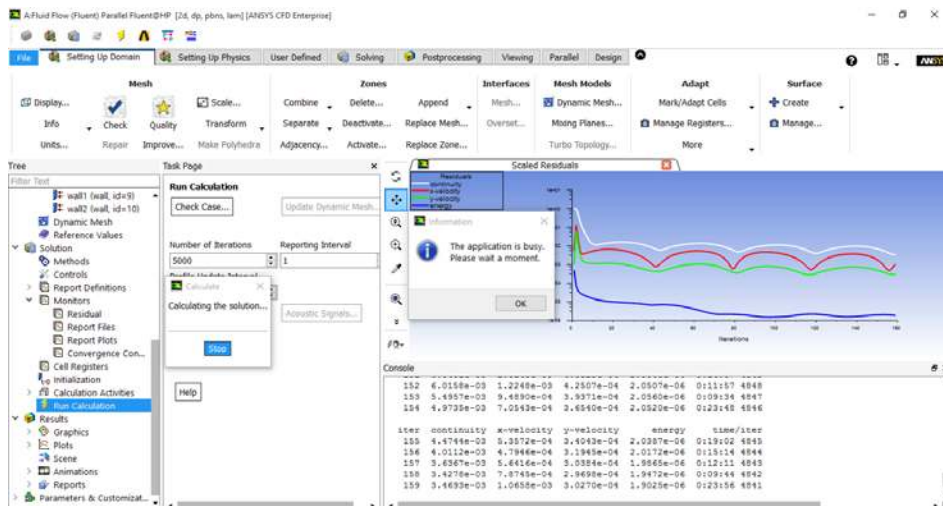


Figure A.10 Convergence Beginning

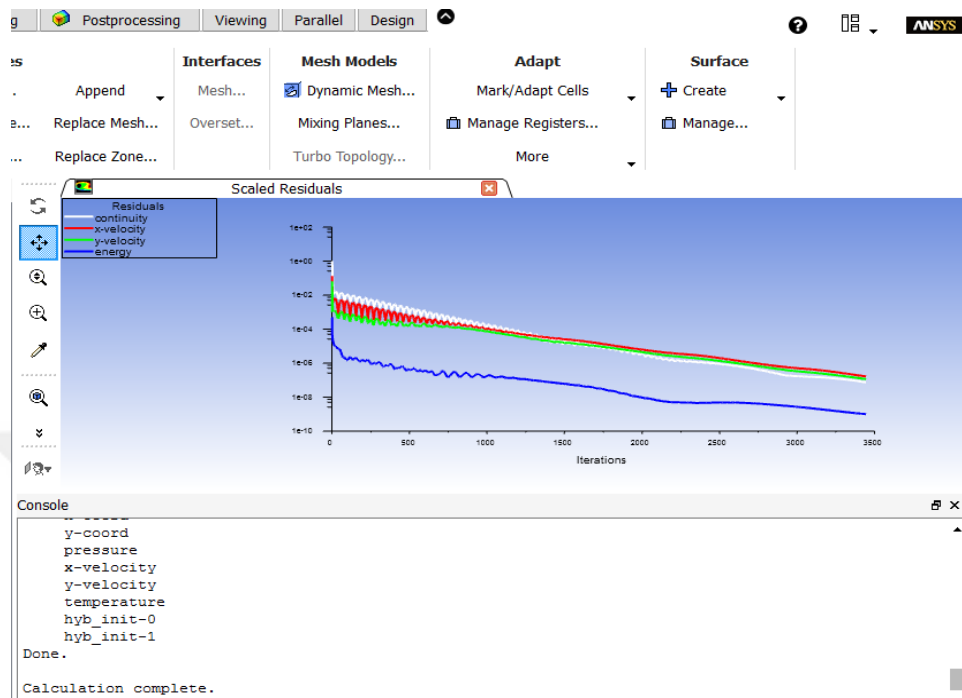


Figure A.11 Convergence Ending

Figures A.10 and A.11 represent the convergence.

DOSYA	GİRİŞ	EKLE	SAYFA DÜZENİ	FORMÜLLER	VERİ	GÖZDEN GEÇİR	GÖRÜNÜM																	
Yapıştır	Calibri	11	A		Metni Kaydır	Genel																		
Pano	K	T	A		Birleştir ve Ortala																			
	Yazı Tipi				Hizalama																			
					Sayı																			
U18																								
8	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U			
9																								
10		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
11	c2(0.5wave+5space)	100	1,79E-05	1,225	0,04752	0,03074	0,025	0,00087	0,05856	0,96	4,80764	5,00796	1,16E+02	300	315,893	2,11E+01	5,4857	10,7715	10,7715	1,42858	0,83499			
12																								
13																								
14		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
15	c18(0.7wave+5space+1thick)	100	1,79E-05	1,225	0,05074	0,02879	0,025	0,00088	0,06007	0,96	6,00458	6,25477	1,13E+02	300	314,9	2,17E+01	5,2114	10,9267	10,9267	1,44917	0,78653			
16																								
17		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
18	c19(0.7wave+5space+1thick+6)	100	1,79E-05	1,225	0,05151	0,02836	0,025	0,00087	0,0569	0,96	5,95074	6,19688	1,13E+02	300	314,967	2,17E+01	5,21596	11,1022	11,1022	1,47244	m			
19																								
20																								
21		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
22	c12(0.5wave+5space+1thick)	100	1,79E-05	1,225	0,04505	0,03242	0,021	0,00083	0,06026	0,96	4,2163	4,39198	1,19E+02	300	316,396	2,07E+01	5,75749	10,7185	10,7185	1,42155	0,86804			
23																								
24		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
25	c13(0.5wave+5space+1thick+6)	100	1,79E-05	1,225	0,04575	0,03193	0,021	0,00082	0,05764	0,96	4,22387	4,39988	1,20E+02	300	316,487	2,07E+01	5,78417	10,9352	10,9352	1,45029	0,88506			
26																								
27																								
28		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
29	c14(0.4wave+5space+1thick)	100	1,79E-05	1,225	0,04169	0,03503	0,019	0,00082	0,06159	0,96	3,41551	3,55782	1,24E+02	300	317,326	2,01E+01	6,15529	10,6049	10,6049	1,40649	0,92131			
30																								
31																								
32		Re	viskosite rho	d	u	A	m	Dp	f ₉₆ /Re	f _{ref}	oran	Q	Ti	To	DTm	h	Nu		oran	n				
33	c15(0.4wave+5space+1thick+6)	100	1,79E-05	1,225	0,04235	0,0345	0,019	0,0008	0,05922	0,96	3,44073	3,5841	1,24E+02	300	317,416	2,00E+01	6,19788	10,845	10,845	1,43833	0,93986			
34																								
35																								

Figure A.12 Some calculations used during the study.