Al-Khwarizmi Engineering Journal, Vol. 11, No. 2, P.P. 31-44 (2015)

Al-Khwarizmi Engineering Journal

Study on Flow Characteristics and Heat Transfer Behavior Around Different Geometrical Corrugated Extended Surfaces

Naseer Abdul Razzaq Mousa
Engineering Affaire Department/ University of Baghdad
Email: nasseer64@gmail.com

(Received 16 March 2014; accepted 15 December 2014)

Abstract

The current study presents numerical investigation of the fluid (air) flow characteristics and convection heat transfer around different corrugated surfaces geometry in the low Reynolds number region (Re<1000). The geometries are included wavy, triangle, and rectangular. The effect of different geometry parameters such as aspect ratio and number of cycles per unit length on flow field characteristics and heat transfer was estimated and compared with each other. The computerized fluid dynamics package (ANSYS 14) is used to simulate the flow field and heat transfer, solve the governing equations, and extract the results. It is found that the turbulence intensity for rectangular extended surface was larger than that of triangle and wavy extended surfaces at the same aspect ratio and number of cycles per unit length. Also, the increasing of turbulence intensity leads to enhance the heat transfer coefficient and consequently the amount of heat transfer. According to previous results, if the pressure head losses along the upstream are not important, the using of rectangular extended surface is better than the triangle which is also better than wavy extended surface.

Keywords: *Flow Characteristics, Heat Transfer, Corrugated Surfaces, Numerical Analysis by ANSYS.*

1. Introduction

The enhancement of heat transfer is required in wide engineering applications. Two groups of heat transfer enhancements techniques have been identified: “passive” and “active” techniques. Passive techniques use special surface geometries, or fluid additives for heat transfer enhancement, such as coated surfaces, extended surfaces, rough surfaces, swirl flow devices, surface tension devices, and additives for liquids or gases. The active techniques require external power, such as electric or acoustic fields and surface vibration [1]. Extended surfaces (fins) are widely used for many engineering applications such as heat exchangers. The extended surface improves the heat transfer coefficient by altering the flow field besides increasing the effective heat transfer surface area [2]. Examples of such fins include corrugated or wavy fins shown in figure 1.

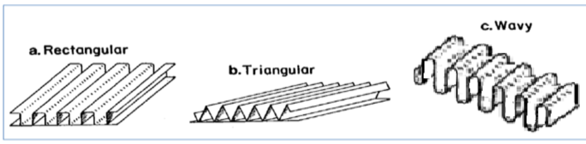

Fig. 1. Examples of corrugated extended surface [1].

The use of specially configured surfaces (such as fins) can provide the improved heat transfer coefficient and increased effective heat transfer surface area, and reduce the gas-side thermal resistance [.[3Due to small characteristic length (wave length) of extended surfaces and the low density of gas, special surface geometries must be effective in the low Reynolds number region [Re <1000].

Focke et al. [4] conducted flow visualization experimental studies in channels with wavy walls formed in a plate heat exchanger. In their study, the complex flow patterns in flow rate of Re = 10 to 1000 were reported. For the channel with corrugation 90°, they found that the main flow was undulating in the direction of channel axis, and at low flow rate no flow separation was observed; flow separation first took place at Re = 20, and with the increase of Re, the separated region increases in size, the flow became unsteady after Re ~ 260. Nishimura et al. [5] have investigated experimentally the flow pattern and mass transfer characteristics in symmetric wavy-walled channels at moderate Reynolds numbers (Re = 20–300). They concluded that the characteristics of mass transfer for wavy-walled channels differ from those of a straight-walled channel when flow separation takes place. Nishimura et al. (6) also experimentally investigated the longitudinal vortices generation, growth, and destruction in various wavy flow channels. Secondary flow characteristics were also reported. The authors believed that secondary flow played an important role to enhance the mass and heat transfer. Ali and Ramadhyani [7] conducted an experimental study on grooved (corrugated) channels of planner cross section in the steady and transitional Reynolds number regimes (150 < Re < 4,000). Their studies indicated the formation of longitudinal vortices which increased in size with increase in Reynolds number. In addition, spanwise vortices appeared from the shear layers to transfer near-wall fluid to the core area and enhance heat transfer rate. Performance evaluation indicated that wavy channels give better rates of heat transfer when operated at transitional Reynolds numbers. The fluid flow and heat transfer through a periodic array of sinusoidal-shaped channels were studied numerically by Wang and Vanka [8]. In their study, the flow was observed to be steady up to Re =180, after which self-sustained oscillatory flow was noticed. In the transitional flow regime, the heat transfer enhancement ratios were more than twice those of a parallel-plate channel, but were accompanied by a higher friction factor. Stone and Vanka [9] studied the developing flow and heat transfer in a wavy passage using a numerical scheme that solves the two-dimensional unsteady flow and energy equations. Their calculations were presented for a wavy channel consisting of 14 modules with a fixed set of geometric parameters. Consideration was given to sinusoidal channels only. Niceno and Nobile [10] studied the flow characteristics in both arc-shaped and sinusoidal wavy channels, employing an unstructured co-volume method. Their results were limited to one set of geometric parameters.

The current study numerically investigates the fluid flow and enhanced convection heat transfer in different extended geometry in the low Reynolds number region [Re <1000]. The geometries are included wavy, triangle, and rectangular. The effect of different geometry parameters on flow field characteristics and heat transfer coefficient was estimated and compared with each other. The computerized fluid dynamics package (ANSYS 14) is used to simulate the flow field and heat transfer, solve the governing equations, and extract the results.

2. Geometrical Description

Three shapes of corrugated surfaces with related geometrical parameters are taken into account in current study. The effects of these parameters on flow field and heat transfer coefficient are estimated numerically. These geometries and their related parameters are:

2.1. Wave Extended Surface

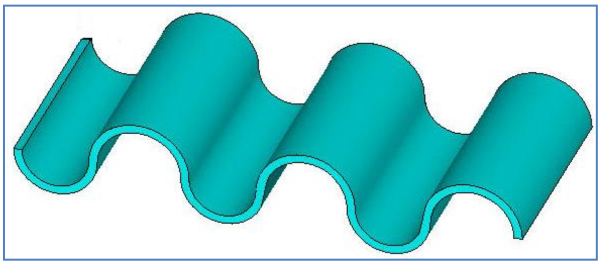

Fig. 2. Wave Extended Surface.

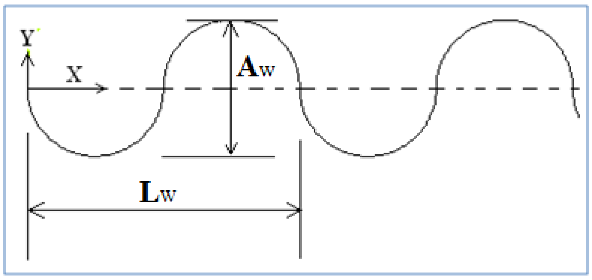

Fig. 3. Geometrical Description of Wave Extended Surface

Where (Aw) is the wave amplitude, (Lw) is the wave length of wavy surface, λw is the aspect ratio of the wavy surface (λw=Aw/Lw), and (nw) is the number of waves per unit length (nw=1/Lw).

2.2. Triangular Extended Surface

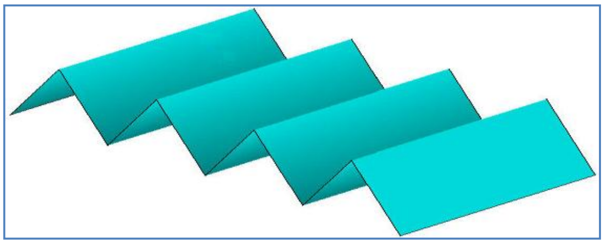

Fig. 4. Triangular Extended Surface.

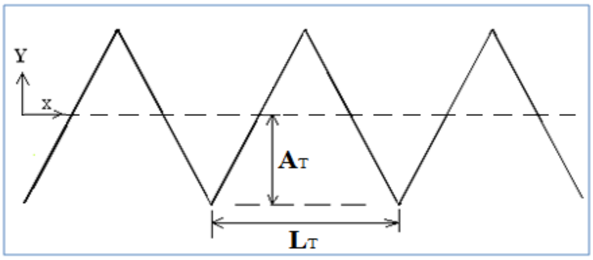

Fig. 5. Geometrical Description of Triangular Extended Surface.

Where (AT) is the amplitude, (LT) is the pitch of triangle, (λT) is the aspect ratio of the triangular extended surface (λT = AT/LT), and n is the number of triangles per unit length (nT=1/LT).

2.3. Rectangular Extended Surface

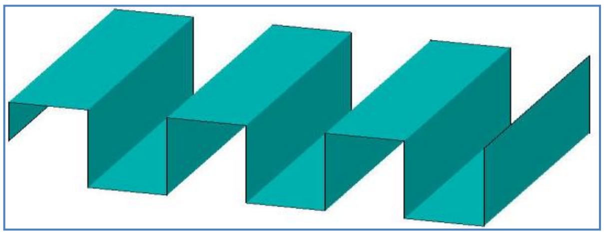

Fig. 6. Rectangular Extended Surface.

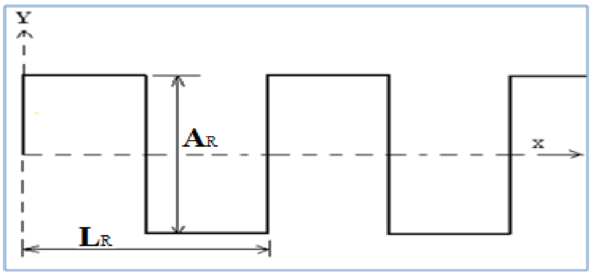

Fig. 7. Geometrical Description of Rectangular Extended Surface.

Where (AR) is the amplitude of rectangular circle, (LR) is the pitch of the rectangle, (λR) is the aspect ratio of rectangular extended surface (λR= AR/LR), and n is the number of rectangular circles per unit length (nR=1/LR).

3. Numerical Simulation

The aim of this simulation is to setting up and solving a conjugate heat transfer problem using ANSYS FLUENT 14. The geometry and flow domain consists of a corrugated surface with a heat generating source. Heat is conducted through the source and the corrugated surface. A laminar stream of air flows over the corrugated surface, causing simultaneous cooling of the corrugated surface and heating of the air stream due to convection. The numerical simulation steps consist of three steps as shown in Figure (8). The simulation began from preprocessing stage which included geometry setup, grid generation and boundary condition setup. The geometry of the model, the grid generation and boundary conditions setup was done by using software package (GAMBIT 2.4.6.). After that, the complete model (geometry and mesh) was exported from GAMBIT 2.4.6) to the (ANSYS FLUENT) software. The second stage was the computational simulation which done by using software package (ANSYS FLUENT) solver. Finally is the post-processing stage where the results were found.

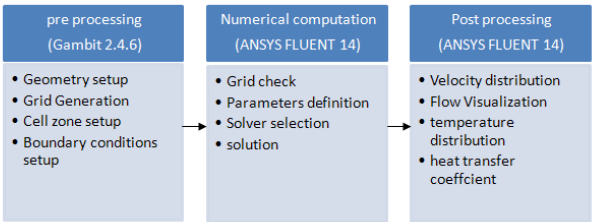

Fig. 8. Numerical Simulation Stages.

For flow simulating, the unstructured tetrahedral mesh was used. In unstructured approach, the integral form of governing equations is discretized and either a finite-volume or finite-element scheme is used. Unstructured grids are in general successful for complex geometries [11]. Volume mesh was created by using T-grid, Gambit scheme. Size functions are used to control the size of mesh interval for edges and mesh elements for faces or volumes and thus to keep smooth transition of mesh from fine mesh near the faces to coarse mesh far away at the undistributed boundaries [12]. After the meshing process, the mesh was checked. It was to check on the quality of the mesh by observing the skewness level and abrupt changes in cell sizes. Figures 9, 10, 11 show the mesh density around the wavy, triangular, and rectangular extended surfaces respectively. Also, figure 12 shows the mesh inspection around wavy, triangular, and rectangular extended surface.

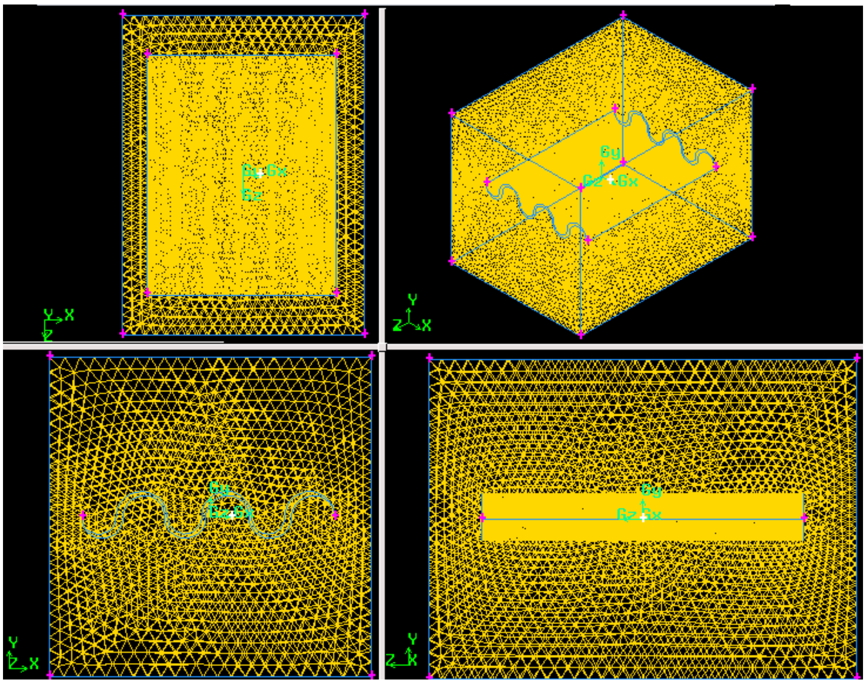

Fig. 9. Mesh Density around Wave Extended Surface.

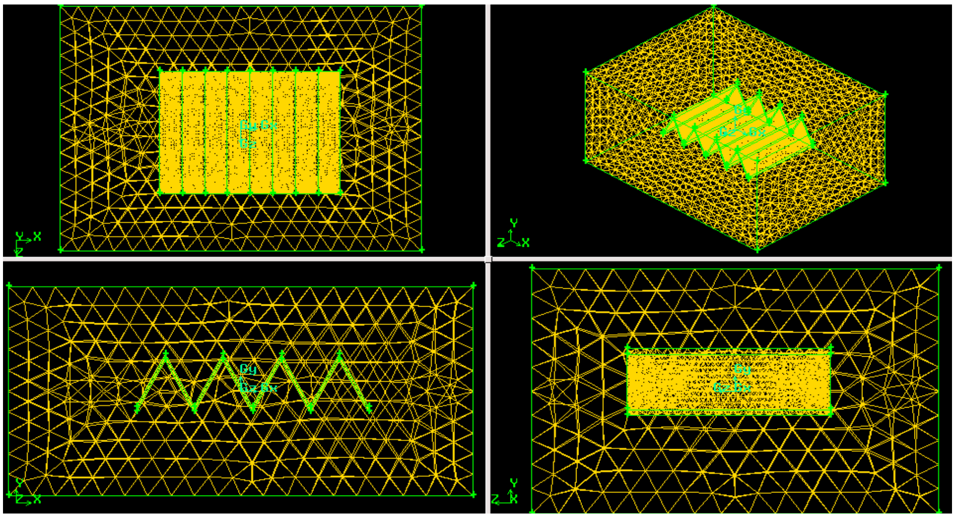

Fig. 10. Mesh Density around Triangular Extended Surface.

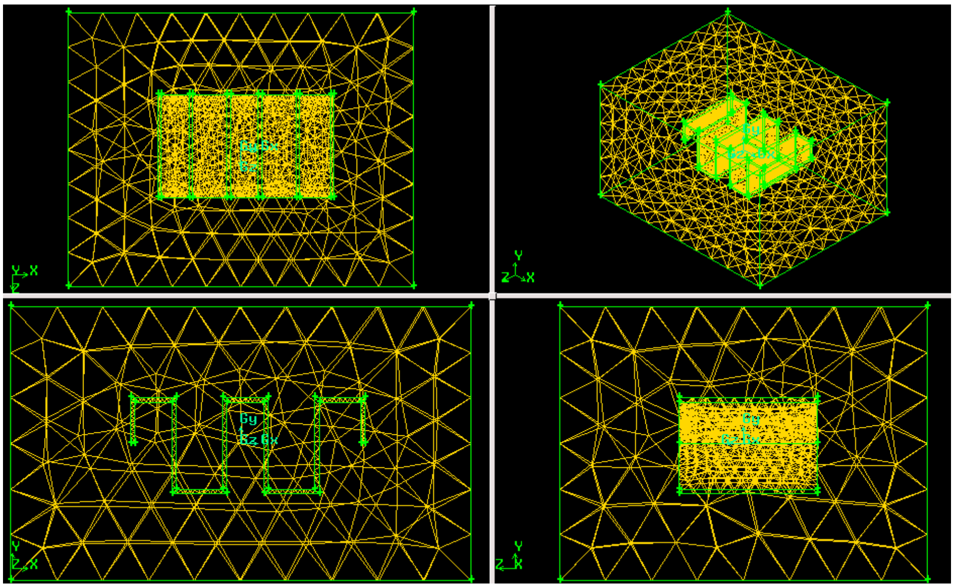

Fig. 11. Mesh Density around Triangular Extended Surface.

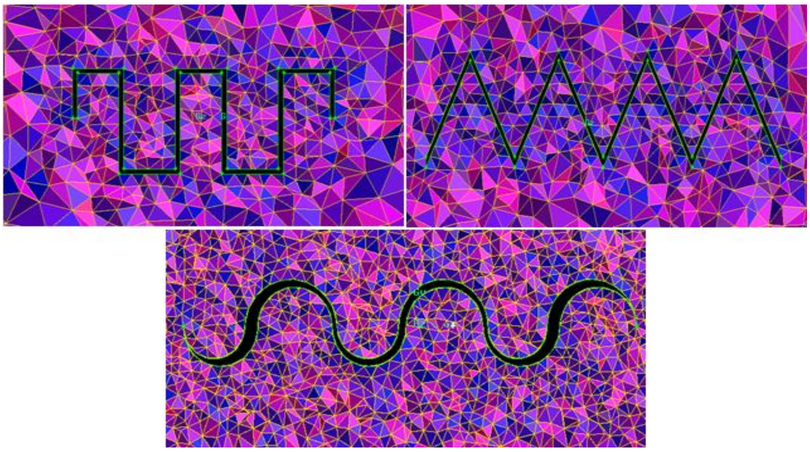

Fig. 12. Mesh Inspection Around Wavy, Triangular, and Rectangular Extended Surface.

4. Governing Equations

In current simulation, the governing equation for fluid flow and heat transfer are established. The governing equations to be considered are continuity, momentum and energy equation. Turbulent flow and convective heat transfer prevail. Steady state and constant thermo-physical proprieties are assumed. The three dimensional governing equations in Cartesian coordinates can be expressed as [1]:

4.1. Continuity Equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad \ldots(1)$$

4.2. Momentum Equation

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+V\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+V\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+V\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad \ldots(2)$$

4.3. Energy Equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{k}{\rho c_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad \ldots(3)$$

5. Boundary Conditions

Inlet. A uniform inlet velocity profile was assigned at the inlet boundary condition ($u = U_{in}$). The range of velocity (1-10m/s) is examined in current simulation because it mostly used in practical applications.

Wall. On the corrugated wall, the velocity fields meet no-slip wall boundary condition:

$$U = V = W = 0 \quad \ldots(4)$$

Outlet. Zero gage pressure at domain outlet was established. This means that the outlet open to the atmosphere.

Symmetry. Symmetry boundary conditions are used when the physical geometry of interest and the expected pattern of the flow solution have mirror symmetry. Also the symmetry boundaries are used to reduce the extent of computational model to a symmetric subsection of the overall physical system. All symmetric boundaries are assumed to be insulated. [13]

Thermal conditions. A constant inlet temperature (T = Tin = 24c̊) was assigned at the channel inlet.

6. Expressions for Dimensionles Parameters

Two significant dimensionless parameters help to quantify and evaluate the flow and heat transfer characteristics of the extended surface. These are: Reynolds number, Nusselt number and Prandtl number. [14]

6.1. Reynolds Number (Re)

For the developing flow simulations, a uniform velocity profile is prescribed at the domain inlet, so the Reynolds number is known in advance to be:

$$Re = \frac{\rho UL}{\mu} \quad \ldots(5)$$

Where (L) is the characteristic length of the wave length or pitch length to extended surfaces, the characteristic length is always very small especially when used in cooling circuits of electronic devices such as the cooling circuit of central processing unit of personal computers. In current study, this length is taken to be around 1.5 mm. Typical calculations of the range of Reynolds number used in current study are shown in appendix 1.

6.2. Nusselt Number (Nu)

The local Nusselt Number is defined as:

$$Nu = \frac{hL}{k} \quad \ldots(6)$$

6.3. Prandtl Number (Pr)

The Prandtl Number is defined as:

$$Pr = \frac{\mu\, c_p}{k} \quad \ldots(7)$$

7. Simulation Domain

Figure 13 shows the simulation domain. The heat generator (heat source) located at the corrugated surface. The heat transferred through the corrugated surface to the fluid (air) by forced convection. The stream of air enters the domain at specific temperature at entrance and leaves the domain from exit side. As a result of heat transfer, the temperature of air will increase. for simply, the wall of the domain and upper plane are assumed to be insulated to ensure that all heat generated by heat source will transferred to the air only and no heat losses from side wall and upper plane. The corrugated surfaces used are wavy, triangular, and rectangular. For this simulation, model air as an incompressible gas because there is air temperature rise but very little pressure change. The goal is to specify the effect of geometry of corrugated extended surface on heat transfer coefficient and heat transfer enhancement. The effects of geometrical parameters on flow characteristics, heat transfer behavior, and temperature distribution are investigated and analyzed. The flow characteristics included velocity distribution, pressure distribution, and turbulence intensity.

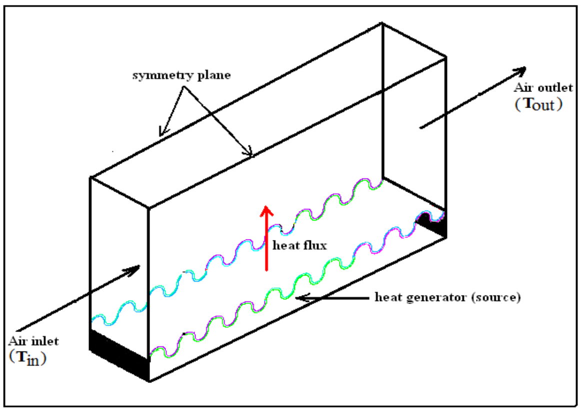

Fig. 13. Simulation Domain.

7.1. Simulation Validation

The present numerical method is validated by comparison the predicted results of simple case with the corresponding theoretical solutions provided by reference [14] of average heat transfer coefficient (h) The simple case included air moving and warmed by isothermal steam heated plate. The exact theoretical solution was provided by reference [14]. The similar simulation of current study is used to evaluate the heat transfer coefficient h. The detailed is comparison between the theoretical and numerical results are shown in appendix 2.

7.2. Calculation Average Convection Heat Transfer Coefficient ($\bar{\mathbf{h}}$)

The calculation of average heat transfer coefficient by convection is done by using the classical equation of heat transfer by convection.

$$q = \bar{h} A (T_w - T_{\text{ave}}) \qquad \ldots(8)$$

Where (A) is the area of corrugated surface, (T_{W}) is the temperature of the corrugated surface and (T_{ave}) is the average temperature of air through the domain. In current simulation, we suppose the same amount of heat transferred (q) and the same temperature (T_{W}) for all types of corrugated surfaces. The high value of average heat transfer coefficient leads to high leaving temperature (T_{out}) of air according to equation of heat transfer by convection. The average heat transfer coefficient by convection is calculated because this coefficient is changing locally due to changing the flow behavior along the corrugated surface.

7.3. Turbulence Model

The flow conditions were chosen to ensure laminar flow to distinguish the flow disturbance due to geometry of corrugated extended surfaces. This disturbance in flow near the surfaces will change the pattern of flow and increases the turbulence. So, the suitable turbulence model for current simulation should be chosen. The k-ε model is chosen for current simulation to estimate the turbulence intensity. The standard (k –ε) model is a semi-empirical model based on model transport equations for the turbulence kinetic energy k and its dissipation rate ε. This model is more suitable for complex geometries. [13]

8. Result and Discussion

The first part of results concerns the effect of geometrical parameters (λ and n) on flow characteristics. Figures 14, 15, and 16 are show the velocity distribution around the wavy, triangular, and rectangular extended surfaces respectively. These figures show complex flow pattern around these surfaces. The complex flow pattern is characterized by recirculation with periodic flow separation and reattachment. It is found that this flow pattern is very dependent to geometrical parameters (λ and n).

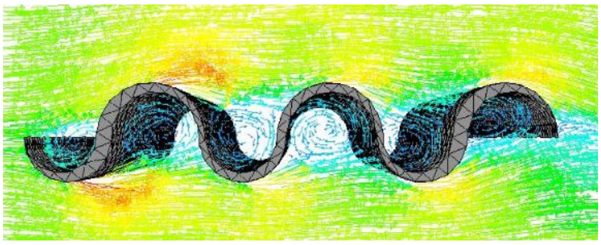

Fig. 14. Velocity Distribution Around Wavy Extended Surface.

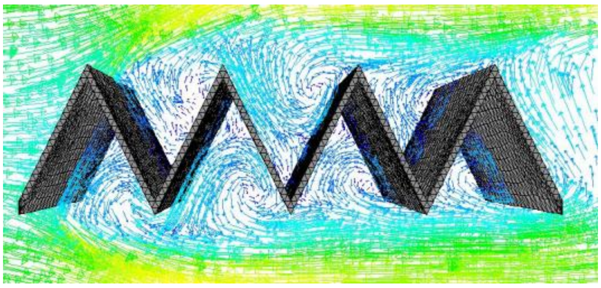

Fig. 15. Velocity Distribution Around Triangular Extended Surface.

Fig. 16. Velocity Distribution Around Rectangular Extended Surface.

The range of (λ) considered in current study is (0.1-1) with pitch 0.1 while the range of (n) is (3-10) with pitch 1. At λ = 0.1, the recirculation regions are small. For wavy surface, the increasing of (λ) leads to increasing the volume of the recirculation regions until the value of (λ) reaches 0.6 but beyond this value of (λ) the increasing of the volume of recirculation regions is not significant. This observation is noted also for triangular and rectangular extended surface but the value of (λ) beyond which the increasing of volume of recirculation regions become not remarkable is not same. This value is 3.5 for triangular surface and 5 for rectangular surface. The increasing of (n) has more effect on the recirculation pattern than the increasing of (λ). The increasing of the number of cycles of corrugated surfaces per unit length leads to increasing the number of recirculation regions and leads to periodic flow separation and reattachment. The increasing of the number of circulation regions and the volume of these regions leads to increase the turbulence intensity around the corrugated surface significantly. The turbulence intensity affects the heat transfer behavior. Figures 17, 18, and 19 are showing the turbulence intensity around the corrugated surfaces when the flow stream velocity was (1.5 m/s) and aspect ratio (λ = 1) for all types of corrugated surface. The first observation on these figures was that the values of turbulence intensity around the corrugated surfaces were close to each other but the difference was in the extension of the turbulence regions. The extension of turbulence regions around rectangular extended surface is larger than that of triangular extended surface which is also larger than that of the wavy extended surface.

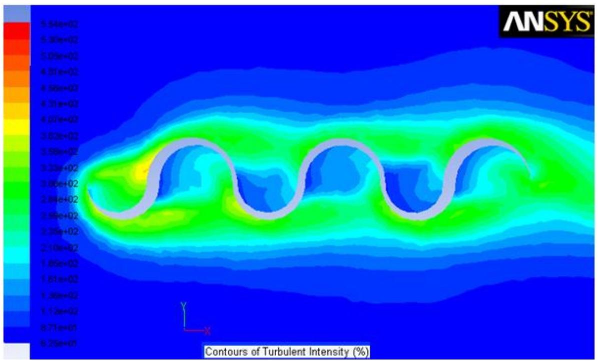

Fig. 17. The Turbulence Intensity Around the Wavy Extended Surface.

This result can be attributed to geometrical effect. The wavy surface is more streamlined body than the triangle and rectangle surface. The flow pattern around bluff body is more complex than that around streamlined body. The second observation from these figures is the presence of strong gradient in turbulence intensity values where the small value of the turbulence intensity noted near the corrugated surfaces and this value will increase away from surfaces to reach its maximum value and then decreases to lower value where the effect of corrugated surfaces will vanished. The changing of aspect ratio (λ) of any corrugated surface will affect the value of turbulence intensity around the corrugated surfaces. Figure 20 shows the value of turbulence intensity at specific point near the corrugated surfaces versus the aspect ratio (λ). It is clear that the turbulence intensity increases when λ increased. The increasing of turbulence intensity versus the aspect ratio (λ) is remarkable until the aspect ratio reaches to specific value beyond which the increasing of turbulence intensity becomes very small.

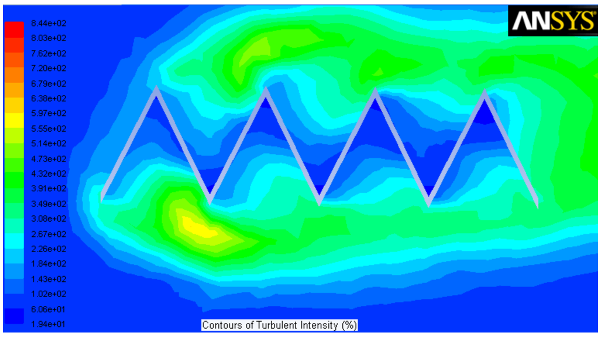

Fig. 18. The Turbulence Intensity Around the Triangular Extended Surface

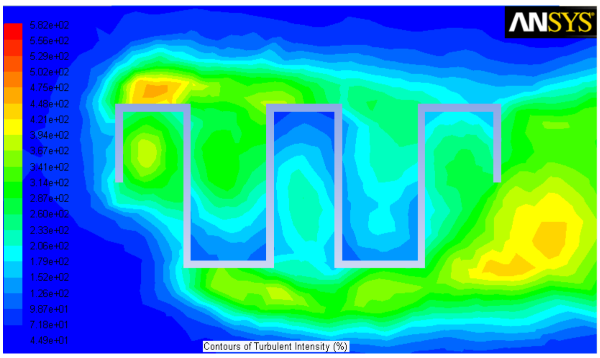

Fig. 19. The Turbulence Intensity Around the Rectangular Extended Surface.

These specific values are differing according to type of corrugated surface. These specific values are shown in figure 20. Also, it is clear from figure 20 that the values of turbulence intensity for rectangular extended surface are larger than the values of turbulence intensity of triangular and wavy extended surface at the same flow stream velocity and the number of pitch per unit length. This result confirms the previous obtained result. Figure 21 show the values of turbulence intensity versus the number of pitches per unit length (n) for all types of corrugated surfaces. It is clear from this figure that the turbulence intensity increases when the number of pitch per unit length increases. At specific point, the increasing in turbulence intensity due to increasing the number of pitch per unit length (n) is very small in comparison with the increasing of turbulence intensity due to increasing the aspect ratio as mentioned because the increasing the number of pitch per unit length (n) leads to increase the number of recirculation regions as first and then the turbulence intensity as whole. Also, it is obvious from figure 21 that at specific point, the behavior of increment in turbulence intensity was approximately identical for all types of corrugated surfaces.

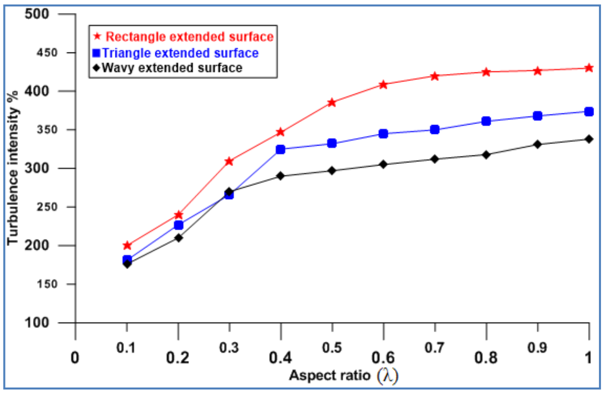

Fig. 20. Turbulence Intensity Versus Aspect Ratio (λ) for all Types of Corrugated Surface at Flow Stream Velocity 7 m/s and Pitch per Unit Length (n=3).

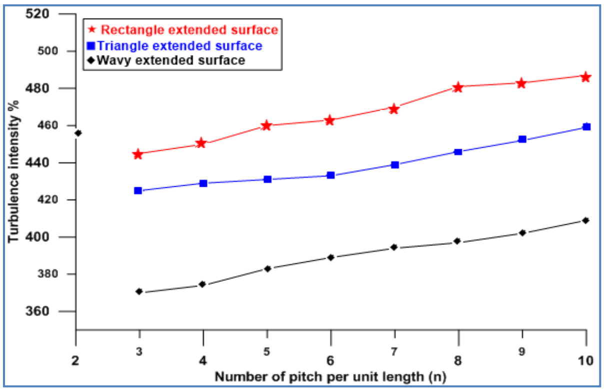

Fig. 21. Turbulence Intensity Versus Number of Pitch per Unit Length n at Flow Stream Velocity 7m/s and Aspect Ratio (λ=1).

It is clear that the flow pattern around the corrugated surface characterized by strong flow mixing, periodic generation of vortices or re-circulating regions. This flow pattern associated with pressure drop penalty. Figures 22, 23, and 24 are showing the pressure distribution around the corrugated extended surfaces when the flow velocity was 10 m/s and aspect ratio was 0.8 for all types of surfaces. It is shown from these figures that there is a significant decreasing in dynamic pressure along the corrugated surface. The decreasing in dynamic pressure was concentrated near the corrugated wall and continuous along the corrugated surface. The pressure increases gradually away from corrugated wall. The decreasing pressure extended to vicinity area of corrugated surface. These vicinity areas were differ in size according to shape of corrugated surface.

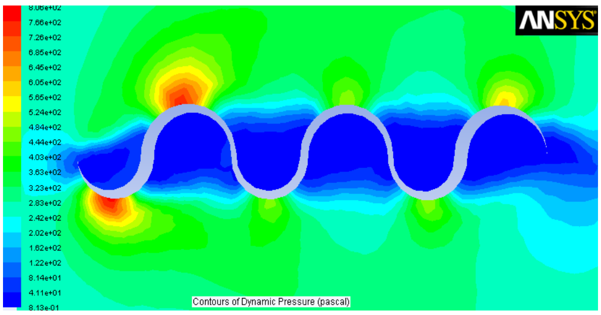

Fig. 22. Dynamic Pressure Distribution Around the Wavy Extended Surface at Flow Velocity 10m/s and Aspect Ratio 0.8.

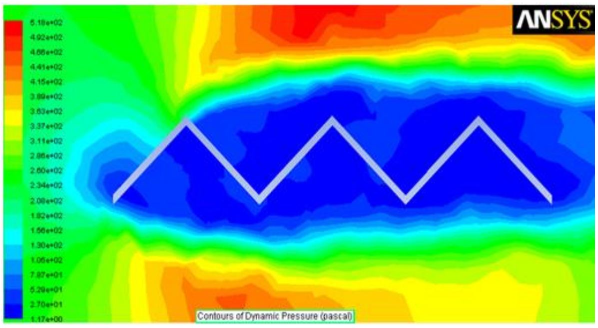

Fig. 23. Dynamic Pressure Distribution Around Triangle Extended Surface at Flow Velocity 10m/s and Aspect Ratio 0.8.

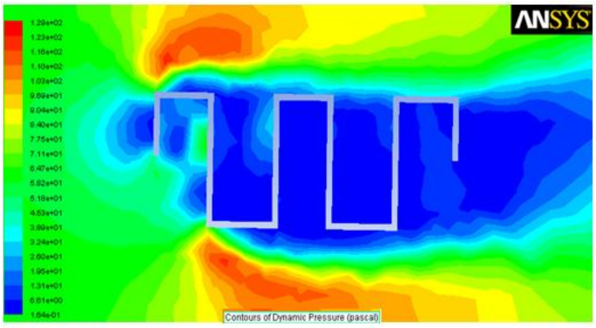

Fig. 24. Dynamic Pressure Distribution Around Rectangle Extended Surface at Flow Velocity 10m/s and Aspect Ratio 0.8.

The size of this area around the triangle extended surface was larger than this around the rectangle extended surface. Also, the size of this area around the rectangle extended surface was larger than this around the wavy extended surface. According to this result, if the pressure is an important issue, the sing of wavy extended surface is better than the rectangle corrugated surface and then the triangle surface.

8.1. Heat Transfer Result

The effect of flow characteristics around extended surfaces on heat transfer behavior are investigated numerically. All extended surface used in this simulation has the same number of cycles (3) and the same aspect ratio. The fluid domain has the same length for all types of extended surface. The depths of extended surface were changed to get the same area of all extended surfaces. Figures 25, 26, and 27 are showing the temperature distribution along the fluid domain above the extended surfaces at (Re = 964.577).

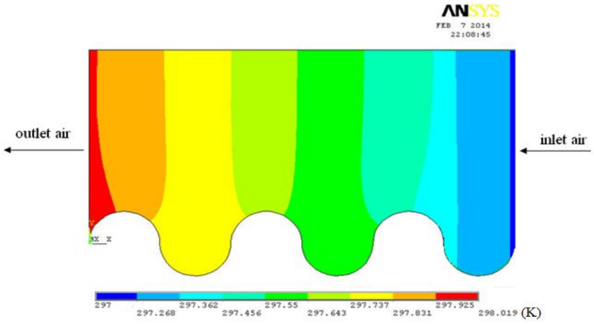

Fig. 25. Temperature Distribution Along Fluid Domain above Wavy Extended Surface at (Re=964.577).

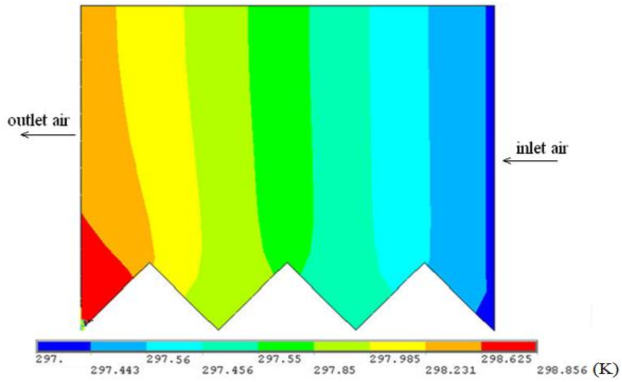

Fig. 26. Temperature Distribution Along Fluid Domain above Triangle Extended Surface at (Re=964.577).

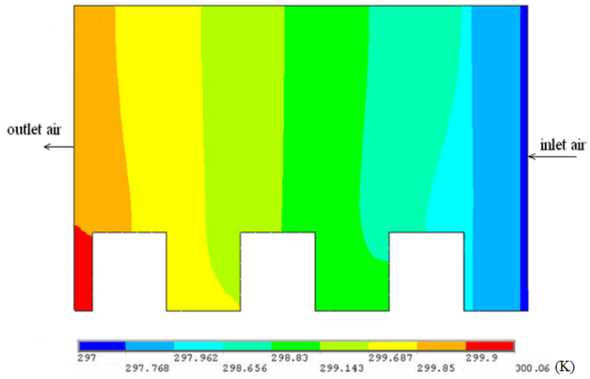

Fig. 27. Temperature Distribution along Fluid Domain above Rectangular Extended Surface at (Re=964.577).

It is clear from these figures that the amount of heat transferred to fluid domain across the rectangular extended surface was larger than that of triangle surface and the amount of heat transferred across the triangle extended surface was larger than that of wavy surface. This conclusion was established because the value of temperature of air at the outlet section above the rectangular extended surface was larger than that of triangle surface which also larger than that of wavy extended surface. This observation indicates that the flow behavior around the extended surface affected the heat transfer. Also it is clear that the value of heat transfer coefficient was affected by flow behavior because that in current simulation we established the same transferring area, the amount of heat transferred, the same temperature difference between the extended surface and inlet air.

Also, this observation indicates that increasing of turbulence intensity leads to increasing in the amount of heat transferred. In other words, the increasing of turbulence intensity leads to enhance the coefficient of heat transfer. According to this result, the changing in any geometrical parameters which leads to increase the turbulence intensity will leads to enhance the coefficient of heat transfer. Figure 28 show the values of average heat transfer coefficients versus the Reynolds number for all types of corrugated surface.

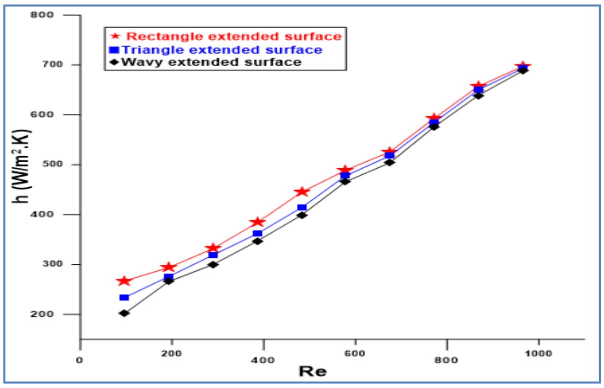

Fig. 28. Average Heat Transfer Coefficient Versus Re for all Corrugated Extended Surfaces

It is clear from this figure that the above result (the amount of heat transferred to fluid domain across the rectangular extended surface was larger than that of triangle surface and the amount of heat transferred across the triangle extended surface was larger than that of wavy surface) was established for all values of Reynolds number examined in current simulation. Also, it is clear from figure 28 that the values of heat transfer coefficients for all three corrugated surfaces becomes close to each other at high values of Reynolds. Figure 28 can easily transformed to illustrate the relation between Nusslet number versus Reynolds number for all corrugated surface examined in current study by using relation (6) as in figure 29.

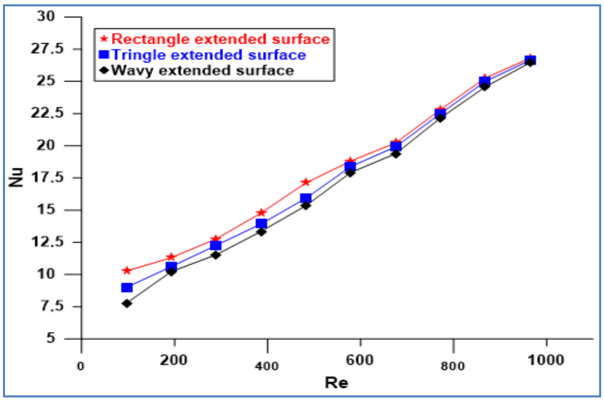

Fig. 29. Nusslet Number Versus Re for all Corrugated Extended Surfaces.

9. Conclusions

1- The increasing of aspect ratio (λ) leads to increasing the volume of the recirculation regions until the value of (λ) reaches to specific value for each type of corrugated surface and beyond this value of (λ) the increasing of the volume of recirculation regions is not significant. These specific values are 0.6 for wavy, 0.35 for triangle and 5 for rectangular extended surface.
2- The increasing of number of cycles per unit length (n) has more effect on recirculation pattern and turbulence intensity than the increasing of aspect ratio (λ).
3- The turbulence intensity for rectangular extended surface was larger than that of wavy and triangle extended surfaces at the same aspect ratio and number of cycles.
4- The increasing of turbulence intensity leads to enhance the heat transfer coefficient and consequently the amount of heat transfer.
5- The heat transfer enhancement for rectangular extended surface was larger than triangle extended surface which also larger than that of wavy extended surface.
6- According to previous results, if the pressure losses are not important, the using of rectangular extended surface is better than the triangle which is also better than wavy extended surface.

Notation

Aw, AT, AR	wave amplitude of wavy, triangular, and rectangular extended surfaces respectively. [m]
Lw, LT, LR	wave length of wavy, triangular, and rectangular extended surfaces respectively. [m]
n	generalized number of cycles per unit length. [cycle]
nw, nT, nR	number of cycles surface per unit length of wavy, triangular, and rectangular extended surfaces respectively. [dimensionless parameter]
u, v, w	the velocity components in coordinates directions x, y, z respectively. [m/s]
p	pressure [N/m2]
ρ	fluid density [kg/m3]
k	thermal conductivity [W/m.K]
h	heat transfer coefficient [W/m2.K]
cp, cv	specific heat of fluid at constant pressure and constant volume respectively. [J/kg.K]
T	Temperature. [K]
Re	generalized Reynolds number. [dimensionless parameter]
Nu	Nusselt number. [dimensionless parameter]
Pr	Prandtl number. [dimensionless parameter]

Greek letters

Λ	generalized aspect ratio [dimensionless parameter]
λw, λT, λR	aspect ratio of wavy, triangular, and rectangular extended surfaces respectively. [dimensionless parameter]
μ	dynamic viscosity. [kg/m.s]

10. References

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Appendix 1
Reynolds Number Calculations

In current simulation , the air enters to fluid domain at temperature 24c̊. The density of air at 24c̊ = 1.18 kg/m3 and dynamic viscosity at 24c̊ = 1.835*10-5 N.s/m2

Table 1
Reynolds number calculations.

Velocity(m/s)	Re
.	96.457
2	192.915
3	289.373
4	385.831
5	482.288
6	578.746
7	675.204
8	771.662
9	868.119
10	964.577

Appendix 2

The validation of numerical simulation is done by comparison the theoretical results of simple case of convection heat transfer between heated simple geometry plate and moving air. The exact solution of this problem is presented in pages 308-309 of reference [14]. The exact formula of average heat transfer coefficient of this problem is:

$$\bar{h} = 0.664\; Re^{\frac{1}{2}}\, Pr^{\frac{1}{3}}\, \frac{k}{l}$$

Where
Pr = 0.707, k = 0.02885 , L=0.5 (according to problem conditions) [14].
The similar simulation domain of current study is used to evaluate the Average heat transfer coefficient h with exception of geometry of plate as shown in figure (E1.1).

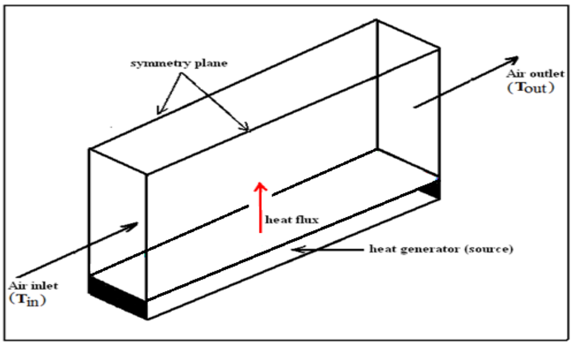

Fig. E1.1. Average heat transfer coefficient h with exception of geometry of plate.

The table 2 and figure E1.2 below show the values of average heat transfer coefficient obtained by numerical and theoretical solution at different values of Reynolds number.

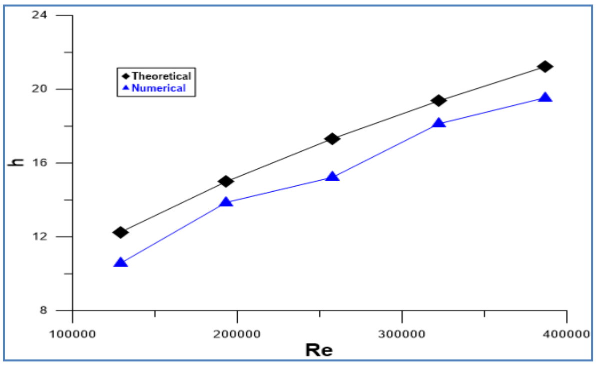

Fig. E1.2. Average Heat Transfer Coefficient h of Reynolds Number Re.

Table 2
Average Heat transfer coefficient h.

Velocity (m/s)	Re	h (theoretical solution)	h (numerical solution)	Error percentage %
5	128865	12.252	10.567	13.75
7.5	193299	15	13.87	7.533
10	257732	17.327	15.23	12.1
12.5	322165	19.372	18.13	6.4
15	386598	21.221	19.54	7.92

دراسة خصائص الجريان وانتقال الحرارة حول سطوح متموجة ذات اشكال هندسية مختلفة

نصير عبدالرزاق موسى
قسم الشؤون الهندسية / جامعة بغداد
البريد الالكتروني: nasseer64@gmail.com

الخلاصة

تتناول هذه الدراسة استقصاء عددي حول خصائص جريان المائع (الهواء) وانتقال الحرارة بالحمل حول سطوح متموجة ذات اشكال هندسية مختلفة لمدى عدد رينولد واطئ (Re<1000). الاشكال الهندسية تتضمن الشكل الموجي و المثلث والمستطيل. تم دراسة تاثير عوامل شكلية متغيرة مثل النسبة الباعية وعدد الدورات خلال وحدة الطول على خصائص الجريان وانتقال الحرارة وتمت مقارنتها مع بعضها. تم استخدام برنامج المحاكاة الحاسوبي (ANSYS 14) لمحاكاة مجال الجريان وانتقال الحرارة وحل المعادلات الحاكمة واستخراج النتائج. وجد ان شدة الاضطراب للسطح المتموج المستطيل اكبر منها للسطح المثلث ومن ثم السطح المتموج عند النسبة الباعية نفسها وعدد الدورات في وحدة الطول. كذلك وجد ان زيادة شدة الاضطراب حول السطوح المتموجة ادى الى تحسين معامل انتقال الحرارة ونتيجة لذلك